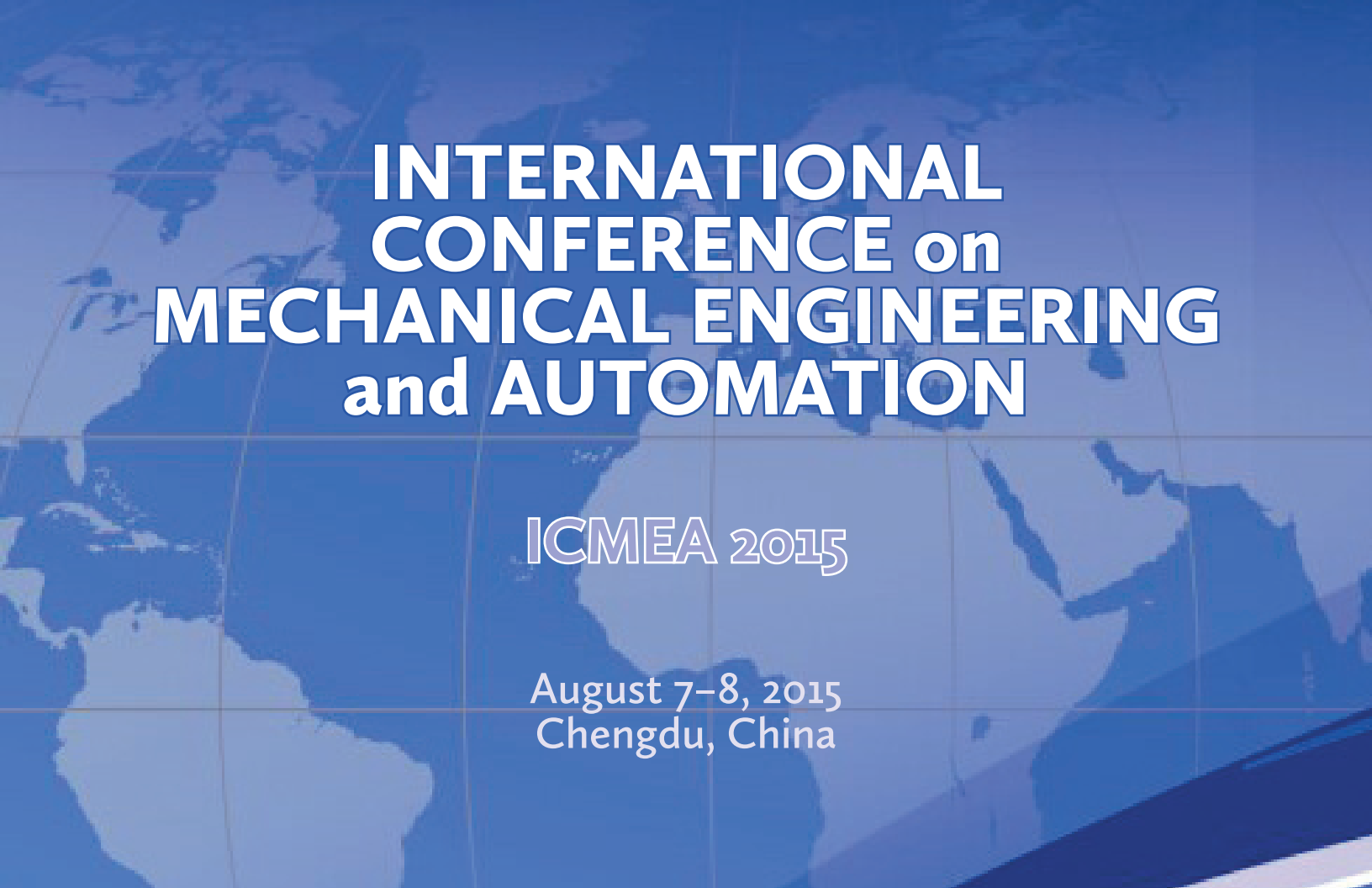


A stylized world map in shades of blue, showing the continents and major lines of latitude and longitude. The map is centered on the Atlantic Ocean, with North and South America on the left and Europe and Africa on the right.

INTERNATIONAL CONFERENCE on MECHANICAL ENGINEERING and AUTOMATION

ICMEA 2015

**August 7–8, 2015
Chengdu, China**

2015 International Conference on Mechanical Engineering and Automation

DEStech Publications, Inc.
439 North Duke Street
Lancaster, Pennsylvania 17602 U.S.A.

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Main entry under title:
2015 International Conference on Mechanical Engineering and Automation (ICMEA 2015)

ISBN: 978-1-60595-281-9

Permission for publication outside of this Conference Proceeding
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Preface

2015 International Conference on Mechanical Engineering and Automation [ICMEA 2015] was held in Chengdu, China on August 7–8, 2015. The ICMEA 2015 is co-sponsored by Advanced Information Science Research Center (AISRC). The conference proceeding is published by DEStech Publications, Inc. ICMEA 2015 provides an excellent international forum for sharing knowledge and results in theory, methodology and applications of Mechanical Engineering and Automation. The conference looks for significant contributions in theoretical and application aspects. The aim of the conference is to provide a platform to the global researchers and practitioners from both academia as well as industry to meet and share cutting-edge development in the fields.

This ICMEA 2015 proceeding tends to collect the up-to-date, comprehensive and worldwide state-of-art knowledge on Mechanical Engineering and Automation, including Control Theory and Applications, Electrical Engineering and Automation Intelligent Mechatronics and Robotics, etc. All of accepted papers were subjected to strict peer-reviewing by 2–4 expert referees. The papers have been selected for this volume because of quality and the relevance to the conference. We hope this book will not only provide the readers a broad overview of the latest research results, but also provide the readers a valuable summary and reference in these fields.

ICMEA 2015 organizing committee would like to express our sincere appreciations to all authors of this conference for their contributions to this book. We would like to extend our thanks to all the referees for their constructive comments on all papers; especially, we would like to thank organizing committee for their hard working.

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Condition Monitoring System of Diesel Engine Fuel Injection System Based on LabVIEW

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Keywords: Labview, Diesel engine, Fuel injection performance, Condition detection.

Abstract. This paper researched the relationship between elastic deformation of injection line and oil pressures of the diesel engine fuel system, got the oil pressure curve, and then test the dynamic performance of diesel engine, built the fuel performance collection system of diesel engine based on virtual instrument, established the neural network which can show the characteristic parameter, normal-failure determination of fuel system performance and accurately positioning failure type. The test and failure determination system is developed by LabVIEW.

Introduction

Fuel injection system, consisting of low-pressure diesel pump, high-pressure fuel injection line and fuel injector, is the essential part of diesel engine, which directly affects the combustion of diesel engine and its performance. However, the structure of this system is complicated and its failure rate is high. Failures provided by British Diesel Engineering and Users Associations, showed there was 27% failure of all related to fuel injection system. It's highly important to take real-time detection and failure determination of diesel fuel injection system. This article concentrated on failure detecting system of fuel injection performance of diesel engine based on LabVIEW, briefly introduced the mechanism of diesel engine fuel injection performance detecting, and particularly explained the structure components of this system and its technical problems and solutions in developing process.

Mechanism of Diesel Engine Fuel Injection Failure Detecting

There is a close relationship between fuel injection failure and diesel engine fuel system structure and fuel pattern. Oil pressure curve can well reflect fuel injection performance. It's essential for determining fuel injection system performance to measure oil pressure curve changing. High-pressure fuel line is made of thick-walled seamless steel tube, regarded as thick-walled vessel. Under interior pressure P_1 and external pressure P_2 , according to theory of materials of mechanism, thick-walled vessel deforming under pressure and radical displacement heft (u) of oil external surface is:

$$u = \frac{2}{E} \frac{a^2 b}{b^2 - a^2} P_1 - \left[\frac{b}{b^2 - a^2} \left(\frac{a^2 + b^2}{E} - \frac{a^2 - b^2}{E} \mu \right) \right] P_2 \quad (1)$$

In which, a is inner diameter of injection line, b is external diameter, E is elasticity modulus of injection line material, μ is Poisson's ratio. Actual inner pressure born by high-pressure injection line is P_1 , and external pressure is the summation of air-pressure and chucking power of transducer. Because $P_1 > P_2$, the effect of P_2 can be ignored. And in injection process, it presents linear relationship between radical dilatancy of high-pressure injection line and its inner oil pressure. And as there are failures such as oil leak and carbon deposit in oil injection nozzle, it's necessary to cause

the change of oil pressure in high-pressure injection line. So it is feasible to detect failure of fuel injection system through measuring swell increment of injection line.

Fuel Injection Performance Detecting System of Diesel Engine Based on LabVIEW

Structure & Composition of Fuel Injection Performance Detecting System

Detecting system based on LabVIEW adapts structural pattern of sequential motion and data transmission to realize the judgment on fuel injection system performance. And its structure is shown as the following Fig. 1:

The whole system is consisting of three parts: Part 1 is data collection and pre-processing. Oil pressure curve is obtained through measuring dilatancy value of high-pressure injection line in this system with clamp-on pressure transducer. Then use A/D acquisition card to take time domain sampling and input computer to get further process.

Part 2 is signal de noise and judgment of system performance. Because there are many noise signal in the oil pressure signal obtained in Part 1, which is not benefit for the judgment of system performance and related characteristic parameter obtaining of fuel injection system, wavelet process will be taken on obtained speed signals. It is to estimate whether there are failures existing in fuel injection system through average and threshold method on signals after de noising. Part 3 is fuel injection system failure type identification and subsequence. If detecting system determines the fuel injection system works well, then detection result will be reserved in database through conclusion saving program for backup. If the fuel injection system doesn't work well, then in this part, first obtain characteristic parameter and estimate with well-trained neural network to get accurate type of failure and backup with saving program.

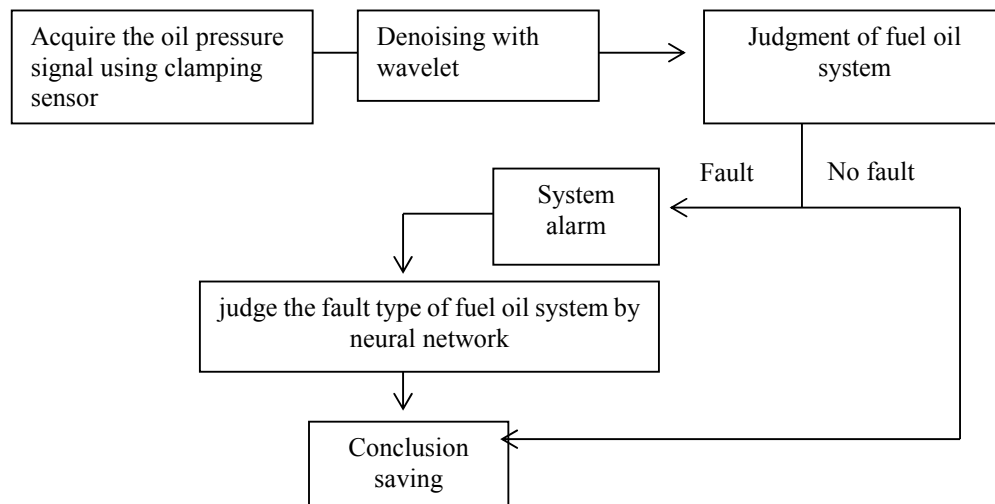


Figure 1. system structure diagram.

Major Technical Problems of System and Solutions

Widely viewing this system, there are major four technical problems:

- 1) Wavelet de noise on fuel pressure signal
- 2) Neural network construction
- 3) Failure type identification
- 4) System alarm and conclusion saving

These problems and related will be discussed in the following.

A. Wavelet-De noise Fuel Pressure

In real testing, there exist problems in fuel pressure wave by simply using clamp-on pressure transducer. Its wave is shown as Fig. 2.

From Fig. 2, there are many glitches in the wave because of noise effect. It's often to filter wave by de noise. But general filter techniques broaden wave when decreasing noise, and smooth sharp edge part in signals, which loses important information carried by this mutation and high frequency characteristic information in original information.

Figure 2. oil pressure chart without .

Figure 3. Denoising program of oil pressure.

layer and hidden layer to output layer, to drive the whole network overall error meeting the requirement. And overall error changing curve can show the change trend of this system.

C. Failure Type Judgment

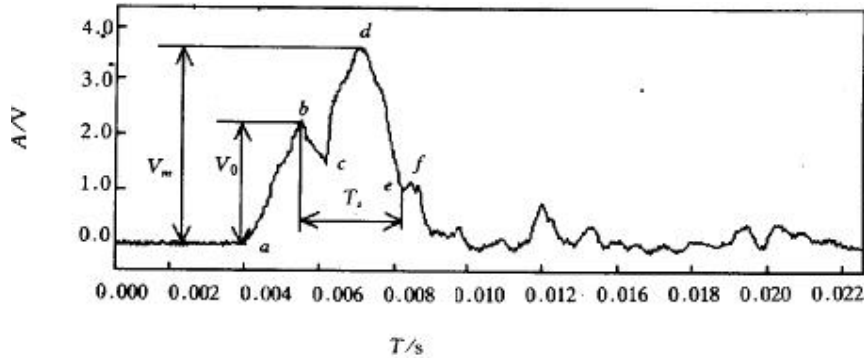


Figure 4. Oil pressure waveform of normal state.

Failure type judgment is failure identification after constructing network. It is realized through relative LabVIEW algorithm. First of all, measure and calculate values of following characteristic vectors through fuel pressure curve, shown in Fig. 4:

V_m : difference between fuel pressure wave crest value and residual fuel pressure; V_0 : difference between fuel injector open pressure and residual fuel pressure; T_d : maximum oil-feed time of fuel pressure; T_a : time from which high pressure fuel injection pump starting to feed; T_s : injection time, $T_s = T_e - T_b$; ΔV_m : difference between two wave crests; I_f : pulse factor, it is the ratio between wave crest and mean; C_f : Wave crest factor, it is the ratio between wave crest value and mean square value.

Then input above obtained characteristic vectors to trained neural network to identify signal, obtain failure type.

This part is an important one to achieve fuel injection system failure detecting. And it is reading-in real-time failure signal characteristic parameter, and making use of trained BP neural network to identify, and achieve last failure type. This procedure is standard dynamic calculation process of BP neural network.

D. System Alarm and Conclusion Storage

System alarm and conclusion storage are to accomplish the processing of gearbox status detecting. System alarm part is to prompt detectors through sound light signals of gearbox failure when gearbox status detecting conclusion shows it is in broke-down status. Conclusion storage part is to conserve gearbox status detecting conclusion in access database by the way of database insertion, convenient for conclusion administration and later inquiry.

Application of Diesel Engine Fuel Injection Performance Detecting System

We take a 12150L diesel engine which exists carbon deposition and didn't work well in fuel injection system as the detecting object. Nip a clamp-on pressure transducer on high pressure fuel injection line developed by Austrian AVL Company, concatenate charge amplifier and NI6013 data acquisition card of NI Company and apply this system to achieve the diesel engine fuel injection system performance detecting. Oil pressure curve is obtained as Fig. 5:

First of all, the system preliminarily detected there was certain failure in this diesel engine. Then bring obtained characteristic parameters, $V_m=0.151$; $V_0=0.121$; $T_d=0.020$; $T_a=0.005$; $T_s=0.550$; $\Delta V_m=0.570$; $I_f=0.995$; $C_f=0.998$, into trained neural network to identify there existed failure of carbon deposit. This conclusion was same as proposal, indicating that the detecting system was right and applicable.

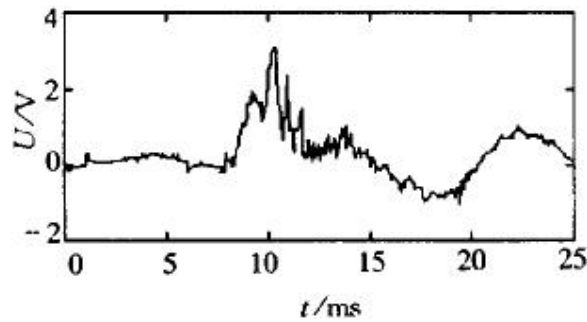


Figure 5. Oil pressure chart.

Conclusion

LabVIEW is the developing platform of this fuel injection performance detecting system, applying clamp-on transducer to measure injection line dilatancy under the circumstance that injection pump is under certain rotation speed and manual oil feed shaft is on the maximum oil-feed position to detect fuel pressure. Then de noise the pressure curve by wavelet transform. Judgment whether the fuel injection system works normally through threshold. Identify the accurate failure type by neural network and accomplish the performance detecting at last. The whole system applies signal collecting techniques of LabVIEW, theory of de noise by mean, fuel pressure detecting technique, neural network techniques to achieve interface program between LabVIEW and database and virtual instrument simulation alarm techniques, and achieves the whole working procedure of diesel engine fuel injection performance detecting system without disintegration under the condition of the overall process, from data real-time collection to status and failure identification.

However, there exists some error in de noising with wavelet in this detecting system, and has some difference with ideal result. This error could be acceptable from the viewpoint of engineering application. If need accuracy, there should be more improvement.

References

- [1] Yang Lepin, Li Haitao, Zhao Yong, Yang Lei, An Xueying, LabVIEW program design [M]. Beijing: Tsinghua University press. 2003.
- [2] Fang Liqing, Qu Zhangsui, Mi dong. Application of Artificial Neural Network to fault diagnosis of fuel oil system of self-propelled gun, journal of ordnance engineering college, 1999 (3).
- [3] Zhang Zhongming, Zhang Yintang, Zhang Peilin. Study of the Fault Diagnosis Method for Diesel Engines fuel oil system Chinese Internal Combustion Engine Engineering, 1998.
- [4] Sheng Shoulin, Zheng Haiqi, Zhang Yintang. Study on fault diagnosis for fuel injection system of diesel engine, Journal of ordnance engineering college, 1999.

Survey of Full Flight Envelope Control System Design Methods and Applications

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Keywords: Full flight envelope, Robust control, Gain-scheduling, QFT, Fuzzy control, Multiple model

Abstract. The dynamics of modern aircraft might vary widely throughout the operational envelope. Thus, a nominal plant based robust controller is not capable to meet the requirements of full envelope flight control. To this end, some novel techniques and methods are integrated with robust control theory to design the control system for full envelope, including gain scheduling, quantitative feedback theory, fuzzy logic and multiple model method. In this paper, full envelope control system design methods are summarized from three aspects. The advantages and disadvantages of different methods are discussed. At the same time, the applications of newly developed methods such as predictive control and neural network control in full envelope flight control are introduced. Finally, the basic rules of full envelope control system design are concluded and some research trends are given.

Introduction

In order to meet the requirements of future scientific research and war, aircraft with good maneuverability and multiple trajectories have caught a lot of attentions. With the expansion of flight envelope, design of full envelope flight autopilots becomes a hot and difficult issue for engineers. Since aircraft flight in a large envelope, unmodeled dynamics, nonlinearity and perturbations often introduce uncertainty to the mathematical model of aircraft. Thus the classical control theory based constructions of autopilots cannot satisfy the stability and performance specifications, designers of the full envelope control system encounter great challenges.

Robust control theory was gradually formed to deal with plant model uncertainty and disturbance. Up to now H_∞ control, μ synthesis, Loop-shaping and other techniques derived from robust control are employed to design the controllers of aircraft in a large flight envelope. The simulation results show that the robust controllers satisfy performance specifications and simultaneously guarantee stability when the system deviates from its nominal design condition and/or is subject to exogenous perturbations. However, even if robust control techniques are applied, the “local” controller designed on isolated operating point fails to achieve stability and performance specifications over the entire envelope. Considering this, some novel and effective methods, such as gain-scheduling, quantitative feedback theory, fuzzy logic and multiple model method, are incorporated to tackle the uncertainties and disturbances resulting from the dynamics variations in full envelope.

In this paper, our goal is to survey the methods and techniques that are integrated with robust control to design the control system of full flight envelope. Meanwhile, the merits and demerits of different methods are analyzed and compared. The main focus is to show how the different methods cope with the nonlinear design task, so full envelope control system design methods are summarized from three aspects according to their characteristics in applications.

The remainder of this paper is organized as follows. The gain-scheduled robust control examples are discussed in section 2. In section 3, the applications of quantitative feedback theory (QFT) which is viewed as a robust control synthesis technique in frequency domain, are showed. In section 4, the way how fuzzy logic and multiple model method are combined with robust control is briefly discussed. Some novel intelligent control algorithms examples which conquer the full envelope

design are introduced in section 5. Finally, a summary is concluded and some research inspirations are given in section 6.

Gain-scheduled Robust Control

Gain-scheduling is widely and successfully applied in dynamic plants with large-scale deviation of mathematical models in full envelope. Generally, many notions can be viewed as gain-scheduling. For example, switching gain values according to operating conditions or even according to preset time [1]. Gain-scheduling is popular since it enables well established linear design tools to be applied to nonlinear problems and does not require complete, analytical plant models. Meanwhile, gain-scheduling makes a controller respond rapidly to changing operating conditions. Linearization gain-scheduling is the most common design method in the field of flight control in a large envelope. The basic rules are solely designing controller on chosen isolated operating points or operating domains and then an interpolation or switching is performed to obtain the full flight envelope control ability.

Many researchers combine gain-scheduling with robust control to design the full flight envelope control systems. Harald Buschek [2] introduces uncertainty models for variations in the aerodynamic coefficients and in the first flexible-body model. Then the flight envelope is divided into three different dynamic pressure regions and μ synthesis is applied to design the corresponding controllers. The controllers are scheduled as a function of dynamic pressure using a conditioning/blending technique to ensure a smooth transition between controllers.

Considering the requirements of disturbance attenuation and robust stability, Avanzini & Minisci [3] applies the mixed sensitivity method to design the controller of a longitudinal control system for an unstable jet aircraft. In order to obtain satisfactory performance in full envelope, gain-scheduling is introduced, which adopts dynamic pressure as the scheduling variable. Unlike Bushchek, the variations of dynamic behavior when flying at the same dynamic pressure but different altitudes are taken into consideration, and different feasible controllers are synthesized at different altitudes for a given equivalent airspeed. Finally a multi-objective search is performed to determine the best suitable controller.

A control signal interpolation technique is presented by Lawrence, Kelly & Evers for dealing with the hidden coupling term arising in linearization gain-scheduling [4]. This method does not impose any constraints on the linear point designs compared with other interpolation schemes and may be of great practical interest.

Traditional linearization gain-scheduling has the following disadvantages: (1) Switching may lead to discontinuities (jumps) in the controller coefficients and possible chattering behavior. (2) Stability property of closed-loop system can be only guaranteed in a neighborhood of equilibrium manifold and under an assumption of slow variation of signals. (3) Extensive simulations need to be done for evaluation of stability and performance.

LPV (*linear parameter varying*) gain-scheduling, which has greatly expanded the connotation of gain-scheduling, is a hot subject of current research. An LPV plant description is

$$\begin{aligned}\dot{\mathbf{x}}(t) &= \mathbf{A}[\rho(t)]\mathbf{x}(t) + \mathbf{B}[\rho(t)]\mathbf{u}(t) \\ \mathbf{y}(t) &= \mathbf{C}[\rho(t)]\mathbf{x}(t) + \mathbf{D}[\rho(t)]\mathbf{u}(t),\end{aligned}\tag{1}$$

where $\rho(t)$ represents the scheduling variable which required to be measurable, captures the plant nonlinearity and is often taken as Mach number and dynamic pressure. The LPV gain-scheduling based controllers are time variant and controller coefficients depend on scheduling variables. The LPV gain-scheduling has many advantages over linearization gain-scheduling since the stability and performance are guaranteed potentially and theoretically and there is no need for interpolation schemes. The LPV gain-scheduling controller synthesis mainly includes two typical approaches: Lyapunov function/quadratic H_∞ approach and linear fractional transformation/small gain theorem approach. Both will convert to *linear inequality matrix* (LMI) based constructions of H_∞ optimal or sub-optimal controllers that guarantee stability and performance.

Apkarian, Biannic & Gahinet [5], Apkarian, Gahinet & Becker [6] consider a class of affine parameter dependent LPV plant. Based on the notion of quadratic H_∞ performance, the gain-scheduled controller is synthesized via solving a set of LMIs. Similarly, a longitudinal LPV gain-scheduling controller is synthesized via mixed sensitivity method which minimizes weighted combination of the infinite norm of the sensitivity function and complementary sensitivity function [7].

Seeing that the dynamic LPV gain-scheduling controllers generally have high orders and consume more time in implementation and testing, a static output feedback loop-shaping LPV controller is put forward by Natesan [8]. The static LPV controller is synthesized by solving two linear inequalities and simulation results show that the controller performs well even in the presence of extreme variations in the scheduling parameter. However, the controller may be conservative since the rate of parameter variation is neglected in the design procedure.

Li views the model variations on different operating points as model uncertainty deviating from the nominal model. Afterwards, the LFT description is obtained by introducing uncertainty in Mach number and altitudes. Finally, a controller which can guarantee the stability in a large envelope is synthesized by μ synthesis [9]. Unfortunately, the orders are very high and it's difficult to implement in practice. To this end, a multivariable reduction procedure based on the generalized grammians is proposed to reduce the size of LFT [10].

Still, LPV gain-scheduling has the following drawbacks: (1) The high orders generated by solving LMIs are difficult to implement and often need the extra reduction procedure. (2) LPV gain-scheduling requires fitting the time variant coefficients of state equation whereas low order polynomials fitting cannot achieve enough accuracy.

Quantitative Feedback Theory

Quantitative feedback theory (QFT), which takes advantage of feedback information to design a satisfying controller with the existence of uncertainty and perturbation, is a strong robust control theory developed by Dr. Horowitz [11]. Although H_∞ control and μ synthesis can guarantee robust performance and robust stability under some conditions, the description of performance specifications is in norm form and it's hard to convert to the specifications in frequency or time domain. This problem lead to the so called iterative design since the performance specifications often need to be evaluated afterwards through extensive simulations. On the contrary, QFT provides a direct way to address the model uncertainty and performance specifications simultaneously. QFT generates the boundaries in Nichols chart which reflect the scope of plant uncertainty and performance specifications by a quantitative way. Then the so called loop-shaping is performed to achieve the desired open loop frequency response. Obviously, QFT is more intuitive and easy to apply in practice, thus lots of researchers apply it to full flight envelope controller synthesis.

The SISO quantitative feedback theory is applied to design the controller for the longitudinal model of airplane and several flight states is chosen to test the performance of QFT controller in full flight envelope [12]. Furthermore, the MIMO quantitative feedback theory is employed for the synthesis of a full envelope flight control system for an F-16 derivative. In order to meet flying qualities, compensators and prefilters are designed [13]. Simulations validate that the QFT theory could address the plant's variability and uncertainty very well.

For simplifying the complex design procedure that MIMO systems is required to transform to equivalent MISO systems, eigenstructure assignment (EA) is proposed to decouple the controlled plant model [14]. However, the method is limited because only the nominal plant can be decoupled by eigenstructure assignment, not the case for other plants. For this reason, H_∞ /loop-shaping is applied to decouple the different channels of models for other plants [15]. Once again, the MIMO QFT design is transformed to a simpler SISO case. Nevertheless, this simplification comes at the cost of the existence of steady-state error in simulations, so there is much room for improvement.

To reduce the conservatism of treating the nonlinearity as additional uncertainty, dynamic inversion is incorporated to linearize the pitch-rate dynamics and to cancel known but nonlinear

dynamics before applying QFT [16]. Simulation results demonstrate that the resulting controller achieves considerable robustness in the presence of variations in model parameters. The inclusion of dynamic inversion effectively enhances the performance of QFT.

Since it is flexible and subjective to design a controller through QFT, there must be a tradeoff between the level of achieved performance specifications and the complexity of the controller to be synthesized. Compared with other robust design methods, the process of controller synthesis is simple and intuitive and it is valuable in practice. Often, a simple, low order stationary controller is easily obtained, so it is widely applied in the full flight envelope control system design.

Fuzzy Logic and Multiple Model Method

Fuzzy logic and multiple model method are novelly integrated with robust control to address the full envelope control issues from another point of view. Fuzzy logic control is a type of rules based control. It takes the experience of engineers and expert knowledge as the starting point and uses fuzzy inference to obtain the final controller. Fuzzy logic does not require accurate mathematical models of the plant, performs well in disturbances rejection and achieves considerable robustness for nonlinear, time varying and time-delay systems. In addition, multiple model method is regarded as another effective idea to address the plants with strong nonlinearity, uncertainty and large flight envelope. The basic ideas are as follows: the operation region of the system is divided into several overlapping sub regions, then sub models are established and the corresponding controllers are synthesized. In the sequel all sub models and sub controllers are combined together in some way. Finally some switching strategies are performed to select the closest model and the corresponding controller is obtained according to current operating conditions [17]. Multiple model method is often combined with fuzzy logic control to design the control system of full flight envelope.

The fuzzy set theory is used to partition the space of the nonlinear model to some sub-spaces of the multiple linearized models and the local H_∞ controllers is synthesized for each linearized model. The model of full envelope can be viewed as a weight sum of those local linearized models. To achieve smoothly switching of controllers, fuzzy logic strategy is employed [18]. Since linearized models at different equilibrium points are used, this modeling technique shows a big discrepancy with the actual plant. Whereas in [19] the model of full envelope is established by nonlinear sub-models. What's more, fuzzy neural network is used to switch the controllers. To its credit, the uniform neural network controller is given by training the neural network through sampling the input and output of the controller of each model.

In [20] multiple model method is combined with H_∞ theory to synthesize the controller for a BTT missile in large flight envelope. The flight envelope is partitioned into several overlapping subspaces and corresponding controllers are designed offline for each subspace via the mixed sensitivity method. Then the global controller output is selected as a weight sum of controllers designed for each subspace to achieve smoothly switching. The resulting controller demonstrates desired robustness and the decoupling ability. However, the weight sum strategy of different sub controllers brings chattering behavior to the output of system.

Considering the conservatism and chattering behavior brought by traditional gain-scheduling, in [21] a novel fuzzy robust control technique is proposed. Firstly the fuzzy T-S model is constructed to approximate the dynamics of UAV in full envelope by fuzzy c-means clustering. Then according to the guaranteed cost theory, the constraint conditions of robust controller are given in terms of LMIs via fuzzy Lyapunov function approach. Finally the control parameters are optimized by minimizing disturbance attenuation. The proposed method with less conservatism and reduced computation complexity shows strong robustness to parameter uncertainty and disturbance.

Some Novel Methods

Besides the widely used methods above, some other novel theories such as adaptive control, model based predictive control and neural network, etc., that compensate and approximate the

nonlinearity and uncertainty of plants in large envelope, inject new life into the full flight envelope control problem.

A new framework of modular adaptive control is provided in [22] which separate the design of controller and the online identifier. The entire flight envelope is partitioned into hyperboxes and for each hyperbox the linearized equations of motion based locally valid model is established. Then these local models are identified and interpolated to obtain a smooth valid model for the complete flight envelope. A backstepping controller is synthesized to resist parameter estimation errors and unknown bounded perturbations.

An active model based predictive control scheme is proposed for the full flight envelope control problem [23-25]. In these papers, the model errors due to flight conditions and its boundary is online estimated using an adaptive set-membership filter method. Then an active modeling based stationary increment predictive control is developed to optimally compensate model errors and aerodynamics time delay. The resulting controller is test in full envelope on UAV platform experimentally, and the proposed method shows its effectiveness.

In [26] a three-layer BP network gain-scheduling for multi-parameters in full flight envelope is proposed. What is more enlightening, a modified global PSO algorithm based on immune clone principle is used to optimize network weights to improve approximation for nonlinear functions. The neural network gain-scheduling controller demonstrates good performance at the designed flight conditions and between the operating points even with the existence of modeling errors.

Conclusion and Outlook

Since nonlinearity, uncertainty and perturbation have evident influence on the dynamics of aircraft in full flight envelope, the local, stationary controller designed at the equilibrium points for the “frozen parameters” linearization model is only valid in the vicinity of a specific trajectory or equilibrium point and is not capable to meet the performance specifications in full flight envelope. Typically, two types of solutions are developed to address such troublesome problems. One is to increase the allowable operating envelope of local controllers with the existence of uncertainty and disturbance. As a representative, robust control theory is developed to meet the control requirements to withstand a certain degree uncertainty and perturbation. Nevertheless, the output of controllers designed via the robust theory is usually very large to guarantee the state of the controlled plant converged, and the output of controllers is confined in practical use. What is worse, possible chattering behavior of system appears when the output of controllers is oversize. Given the drawbacks above, it is unrealistic for the robust theory based controller which is designed at single equilibrium point to meet the requirements of full flight envelope control. Another is switching or interpolating scheme of these controllers synthesized at different operating points or regions according to current operating conditions. Gain-scheduling and fuzzy logic are the most common methods. Although the methods turn to be valid in the full flight envelope control, the disadvantages of discontinuities (jumps) in the controller coefficients and chattering behavior confine their applications in practice. How to switch these controllers in a smoother way has become the key and difficult problem to be solved. The two strategies mentioned above are often combined together to design the controller of full flight envelope, producing the so called gain-scheduling robust control or fuzzy logic robust control, and a lot of design examples can be found in the relevant literature. Besides, the quantitative feedback theory is also popular in fields of full flight envelope control of aircraft due to its inherence of a simple, intuitive design procedure and its great value in practice.

At present, most of the advanced control methods are still in theoretical research stage, and the corresponding designed controllers are difficult to implement in practice. Different techniques need to be combined with each other to adopt one’s strong points while overcoming one’s weak points. There must be a compromise between the complexity and performance of the controllers.

The difficulty of full flight envelope control is mainly brought by the nonlinearity and uncertainty of mathematical models in full envelope. To this end, in the research community there has been a considerable upsurge in interest in many new and interesting approaches and the research trends are as follows:

(1) Absorb the latest achievements of some novel and advanced control theories. Note that at present, the nonlinear plant is decomposed into a number of linear sub-problems to synthesize the controller. However, the linear plants based controller might be difficult to achieve the desired performance due to the discrepancy between the nonlinear plant and the corresponding linear plant. Some novel control theories, for example, intelligent control, which has strong ability to approximate the nonlinearity of plant, can be employed to address some complex control problems.

(2) Try to obtain some more accurate and more reasonable descriptions for the uncertainty and external disturbance. In current studies, a mathematical model for aircraft is mainly modeled by mechanical laws combined with the aerodynamic parameters based on wind tunnel experiments. The unknown bounded uncertainty assumption of robust control theory might introduce certain degree of conservatism to the synthesis of controllers. Therefore, suitable plant information and, in particular, system representations which provide specific support for control design are of great value for the full flight envelope control.

References

- [1] W.J. Rugh, J.S. Shamma, Research on gain scheduling, *Automatica* 36 (2000) 1401-1425.
- [2] H. Buschek, Full envelope missile autopilot design using gain scheduled robust control, *J. Guid. Control Dynam.* 22 (1999) 115-122.
- [3] G. Avanzini, E.A. Minisci, Evolutionary design of a full-envelope full-authority flight control system for an unstable high-performance aircraft, *P.I. Mech. Eng. G-J. Aer.* 225 (2011) 1065-1080.
- [4] D. Lawrence, J. Kelly, J Evers, et al, Gain scheduled missile autopilot design using a control signal interpolation technique, In: *Proceedings of the AIAA Guidance, Navigation, and Control Conference*. New Orleans: LA. AIAA. 1998: A98-37143.
- [5] P. Apkarian, J.M. Biannic, P. Gahinet, Self-scheduled H-infinity control of missile via linear matrix inequalities, *J. Guid. Control Dynam.* 18 (1995) 532-538.
- [6] P. Apkarian, P. Gahinet, G. Becker, Self-scheduled H-infinity control of linear parameter-varying systems: A design example, *Automatica* 31 (1995) 1251-1261.
- [7] Zh. J. Li, J.W. An, Research on full envelope flight control system synthesis, *Flight Dynamics* 19 (2001) 5-9. [Chinese]
- [8] K. Natesan, D.W. Gu, I. Postlethwaite, Design of static H-infinity linear parameter varying controllers for unmanned aircraft, *J. Guid. Control Dynam.* 30 (2007) 1829-1835.
- [9] Zh. J. Li, W.K. Zhong, J. Du, Aircraft large envelope robust flight control law design, *Journal of Projectiles, Rockets, Missiles and Guidance* 30 (2010) 23-26. [Chinese]
- [10] A. Hired, G. Duc, J.P. Friang, et al, Linear-parameter-varying/loop-shaping H-infinity synthesis for a missile autopilot, *J. Guid. Control Dynam.* 24 (2001) 879-886.
- [11] I. Horowitz, Invited paper: Survey of quantitative feedback theory (QFT), *Int. J Control* 53 (1991) 255-291.
- [12] J. A. Zhao, H.G. Sun, Application of QFT in design of full envelope flight control system, *Journal of Projectiles, Rockets, Missiles and Guidance* 24 (2004) 130-133. [Chinese]
- [13] O.R. Reynolds, M. Pachter, C.H. Houppis, Full envelope flight control system design using QFT, In: *American Control Conference, IEEE*, Jan 1994, pp. 350-354.
- [14] S.F. Wu, W. Wei, M.J. Grimble, Robust MIMO control-system design using eigenstructure assignment and QFT, *IEEE Proc. -Control Theory Appl.* 151 (2004) 198-209.
- [15] E.M. Hua, Y. Wan, Q. Liang, Helicopter flight control system design using H_∞ Loop Shaping and quantitative feedback theory, *Electronics Optics and Control* 14 (2007) 87-90. [Chinese]

- [16]S.A. Snell, P.W. Stout, Full envelope longitudinal flight control law using a nonlinear controller combined with quantitative feedback theory, In: Dynamic Systems and Control: Proceedings. American Society of Mechanical Engineers, 1994, pp. 265-173.
- [17]G.R. Duan, H.Q. Wan, Multi-model switching control and Its application to BTT missile design, *Acta Aeronaut. Astronaut. Sin.* 26 (2005): 144-147. [Chinese]
- [18]J. Dai, C. Tan, J. Ying, et al, Fuzzy multi-model switching H-Infinity control for helicopters in a full Envelope, *Circuits Syst. Signal Process.* 32 (2013) 2185-2197.
- [19]M. Chen, C.S. Jiang, Q.X. Wu, Design of full envelope robust flight controller based on multiple model method, *Acta Aeronaut. Astronaut. Sin.* 27 (2006) 486-492. [Chinese]
- [20]Q. Sun, J. Zhou, W.J. Hu, Big envelope flight BTT missiles control system design based on multiple model H_{∞} robust control theory, *Computer Simulation* 27 (2010) 97-101. [Chinese]
- [21]Z. Liu, Y. Wang, Fuzzy robust tracking control within full envelope for unmanned aerial vehicle, *Journal of Beijing University of Aeronautics and Astronautics* 40 (2014) 762-768. [Chinese]
- [22]E.R. Van Oort, L. Sonneveldt, Q.P. Chu, et al, Full-envelope modular adaptive control of a fighter aircraft using orthogonal least squares, *J. Guid. Control Dynam.* 33 (2010) 1461-1472.
- [23]D. Song, J. Qi, J. Han, Active modeling based model predictive control for yaw-heave coupling dynamics of unmanned helicopters in full flight envelope, In: AUVSI Unmanned Systems North America Conference, Aug. 2009, pp. 492-505.
- [24]D. Song, J. Qi, J. Han, et al, Active model based predictive control for unmanned helicopter in full flight envelope, In: IEEE/RSJ International Conference on Intelligent Robots and Systems (IROS), Oct. 2010, pp. 616-621.
- [25]D. Song, J. Han, G. Liu, Active model-based predictive control and experimental investigation on unmanned helicopters in full flight envelope, *IEEE Transactions on Control Systems Technology* 21 (2012) 1502-1509.
- [26]Y. Sun, W. Zhang, M. Zhang, et al, Design of neural network gain scheduling flight control law using a modified PSO algorithm based on immune clone principle, In: International Conference on Intelligent Computation Technology and Automation, IEEE, Oct. 2009, pp. 259-263.

Dual-Period-Control in a Semiconductor Laser by Modulating the Injection Incoherent Light

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Keywords: Control, Laser, Dual-Period.

Abstract. A novel dual-period-control method of a semiconductor laser is presented via an external injection. Control model of a modulated semiconductor laser is studied by modulating an external injection incoherent light. Results show that dual-period-control can be achieved successfully. By adjusting the frequency of injection incoherent light, the laser can be conducted to show a single-cycle or quasi-cycles. By adding the modulation depth, the laser can be controlled into a tri-period, a period-4 or other quasi-period. The result is the benefit to study of control-laser and photoelectric laser system.

Introduction

Period-2 phenomenon is a kind of nonlinear behavior existed in nature. Period-2 shows regularity while its characteristic is very sensitive to variety of external condition. Its long oscillation behavior shows stability. Since “OGY” chaos-control method was presented [1], many techniques of chaos-control have been presented to reduce chaos into a periodic state [2,3]. Chaotic variation is stabilized via a small perturbation or proportional feedback [4-6]. Both experimental and numerical results have recently shown that chaotic lasers can be suppressed via current modulation or optical feedback, and so on [7-9]. However, few studies of controlling period-3 lasers were carried out. We present a novel period-3-control method of a modulated laser. We present this system produce other novel laser system. The result will be the benefit to study of control techniques, lasers, photoelectric parts.

Model

Due to the modulation of the laser current, the laser can emits period-2 pulse or other shape pulses [7,8]. To control the laser, an external incoherent light is introduced to inject into the laser. We use the lang-kobayashi equations describe the modulated semiconductor laser under the case of considering an external incoherent light [7]

$$\frac{dP}{dt} = (G - \gamma_p)P + \frac{k}{\tau_L} P_m (1 + \rho \times \cos 2\pi f t) \quad (1a)$$

$$\frac{dN}{dt} = \frac{I}{q} (1 + \varepsilon \times \sin 2\pi f_i t) - \gamma_e N - GP \quad (1b)$$

where the nonlinear mode gain is

$$G = (\Gamma v_g a / V) (N - N_{th}) / \sqrt{1 + P / P_s},$$

and the total carrier loss rate is

$$\gamma_e = A_{nr} + B(N/V) + C(N/V)^2.$$

where P is the photon number and N is the carrier number. $\gamma_p = v_g(\alpha_m + \alpha_{int})$ for the photon loss rate where its parameters are listed in table 1. $\tau_L = 2n_g L / c$ is the round-trip time in the cavity where its parameters are listed in table 1. Parameters related to γ_e are listed in table 1. q is the unit charge. The

other parameters are listed table 1. P_m is the photon number of the injection light. k is the injection factor of the injection light. ρ is the injection modulation depth. f is the injection modulation frequency. ε is the current modulation depth and f_i is the modulation frequency. When the injection is performed on the laser, we can adjust the modulation frequency or factor to realize to control the laser. Because period-2 being sensitive to the parameter varying, we will find that the control of laser be valid realized when the injection presents. When the current modulation frequency is taken as $f_i=1\text{GHz}$ and the modulation depth is taken as $\varepsilon=0.33$, a period-2 trajectory of the modulated laser and its output are illustrated by Fig.1.

Table 1. Laser Parameters[7].

Parameter	Symbol	Value and Unit
Active cavity length	L	$350[\mu\text{m}]$
Active cavity width	w	$2[\mu\text{m}]$
Active cavity thickness	d	$0.15[\mu\text{m}]$
Mode confinement factor	Γ	0.29
Group refractive index	n_g	3.8
Cavity loss	α_m	$29[\text{cm}^{-1}]$
Internal loss	α_{int}	$20[\text{cm}^{-1}]$
Carrier density at transparency	n_{th}	$1.2 \times 10^{18}[\text{cm}^{-3}]$
Nonradiative recombination rate	A_{nr}	$1.0 \times 10^8[\text{s}^{-1}]$
Radiative recombination coefficient	B	$1.2 \times 10^{-10}[\text{cm}^3/\text{s}]$
Auger recombination coefficient	C	$3.5 \times 10^{-29}[\text{cm}^6/\text{s}]$
Gain constant	a	$2.3 \times 10^{-16}[\text{cm}^2]$
The drive current	I	$25[\text{mA}]$
Photon number in the saturation	P_s	$1 \times 10^7[\text{m}^{-3}]$

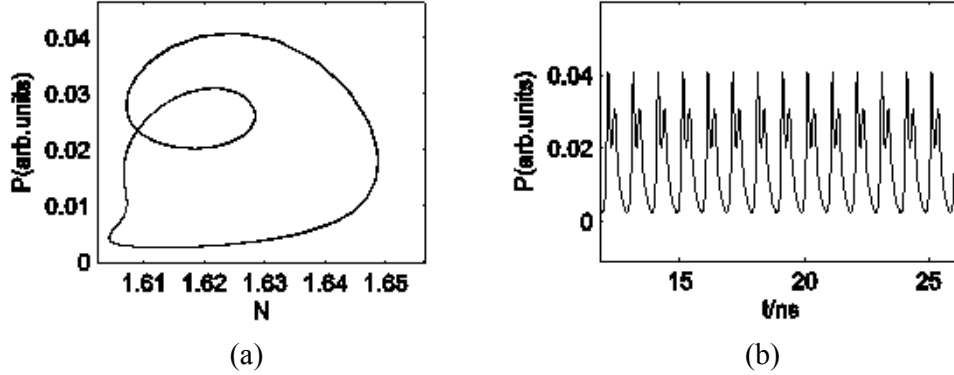


Figure 1. Period-2 trajectory of the laser and its output. Period-2 trajectory in (a) the laser output in (b).

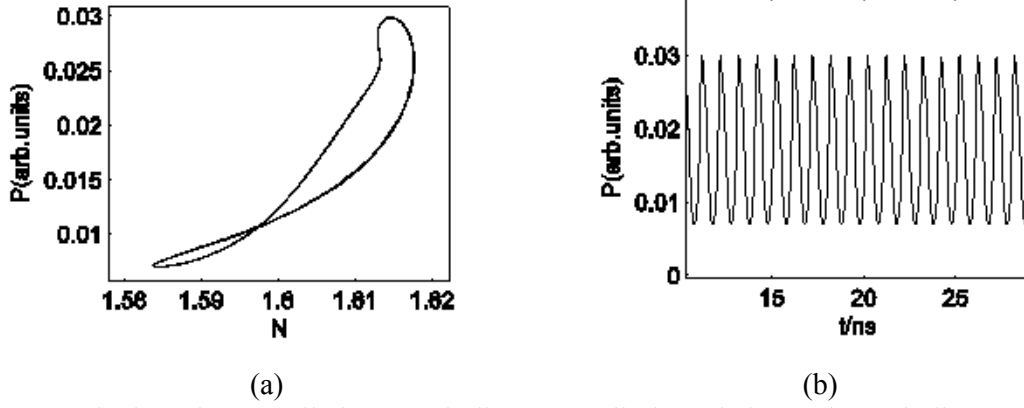


Figure 2. The laser is controlled to a periodic state. A limit cycle in (a) the periodic output in (b).

When an external incoherent injection light are used to perform period-2-control, we adjust the modulation frequency and factor to achieve period-2-control of the laser. When we take the incoherent light $P_m=0.1$, the injection factor $k=0.01$, the injection modulation depth $\rho=0.5$, the modulation frequency $f=1\text{GHz}$, laser dynamical behavior becomes of a limit orbit as shown in Fig.2, namely, the period-2 state is reduced to a single-period while period-2-control is successfully achieved.

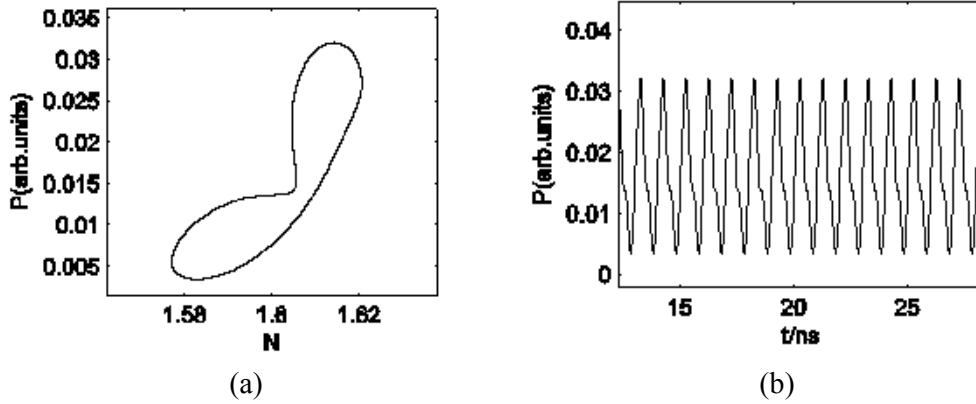


Figure 3. The laser is controlled to another periodic state. A limit cycle in (a) the periodic output in (b).

When the modulation frequency is taken as $f=3\text{GHz}$, dynamics state of the laser is conducted to another single-period shown in Fig.3.

When the modulation frequency $f=4.5\text{GHz}$, dynamics state of the laser is conducted to a period-8 shown in Fig.4.

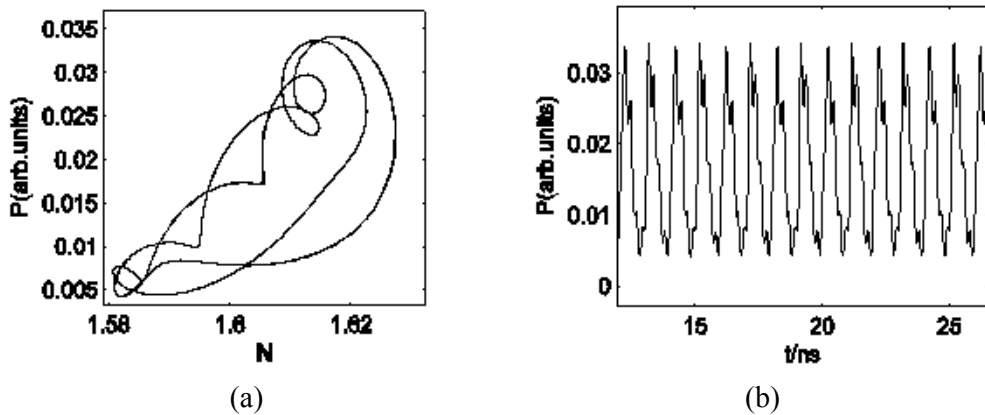


Figure 4. The laser is controlled to a period-8 state. A period-8 trajectory in (a) the laser output in (b).

When the modulation frequency $f=5.5\text{GHz}$, Fig.5 shows that dynamics behavior of the laser is conducted to a many-period state.

When the injection factor is added at $k=0.05$ and the modulation frequency is taken as $f=3\text{GHz}$, Fig.6 shows that dynamics behavior of the laser is altered to an period-3 state.

When the modulation frequency is taken as $f=4\text{GHz}$, Fig.7 shows that the laser dynamics shows an period-4 behavior.

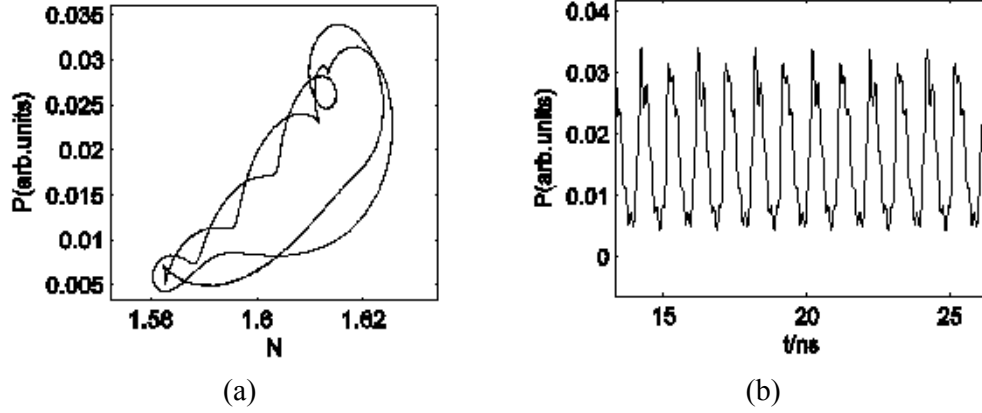


Figure 5. The laser at a many period state. A many-period trajectory in (a) the laser output in (b).

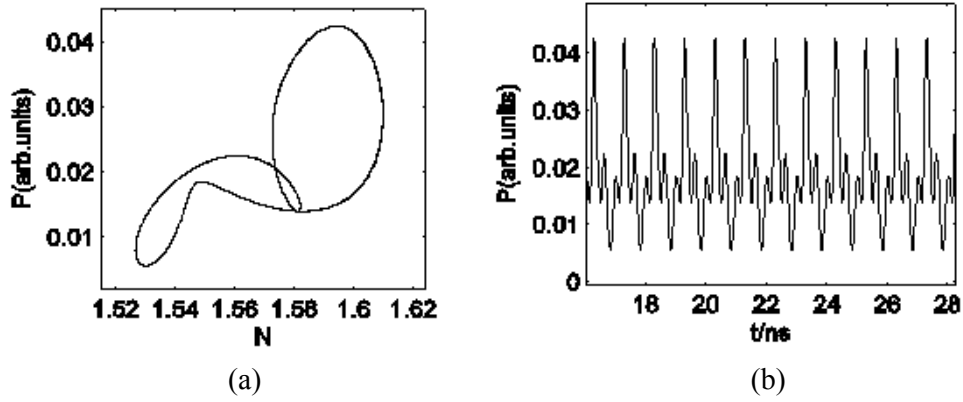


Figure 6. The laser at another period-3 state. A period-3 trajectory in (a) the laser output in (b).

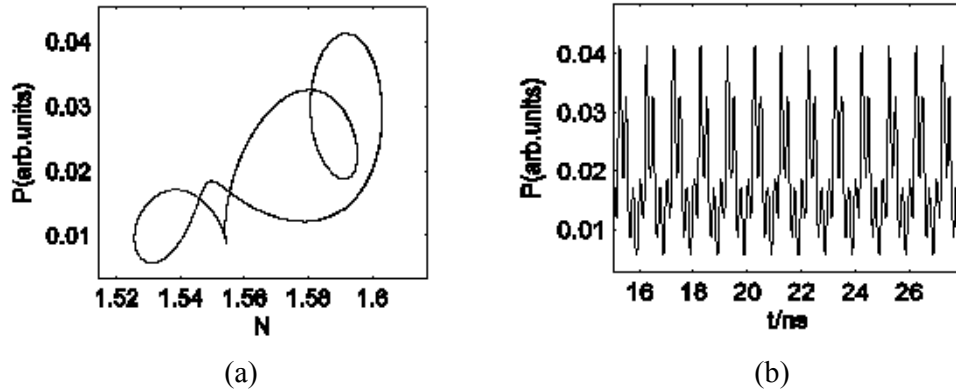


Figure 7. The laser at another period-4 state. A period-4 trajectory in (a) the laser output in (b).

When the modulation frequency is taken as $f=5\text{GHz}$, Fig.8 shows that the laser dynamics shows an period-5 behavior.

We find also that the laser can be controlled to other dynamics state by altering the other control parameters. So period-3-control can be realized via adjusting the control parameters of the external incoherent injection light.

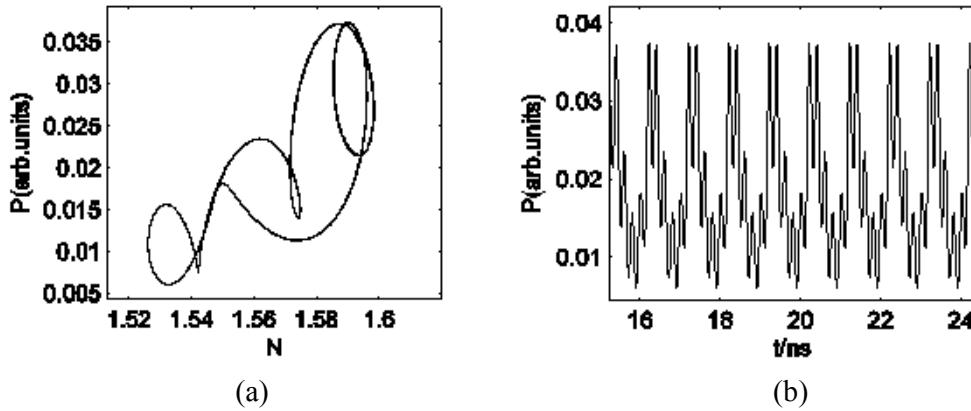


Figure 8. The laser at another period-5 state. A period-5 trajectory in (a) the laser output in (b).

Summary

We present the period-2-control method for the laser system. We find that the laser can be controlled to many dynamics states by altering the control parameters. Period-2-control can be achieved via adjusting the control parameters of the external incoherent injection light. The control method enriches the methodology of laser control, provides a new idea for the applications of the modulated laser system, and broadens the control method of other system. The result is the benefit to study of control-laser and photoelectric laser systems.

References

- [1] J.A. York, Controlling chaos, *Phys. Rev., Lett.*, 64 (1990) 1196-1199.
- [2] M.S. Torre, C. Masoller, P. Mandel and K.A. Shore, Transverse-mode dynamics in directly modulated vertical-cavity surface-emitting lasers with optical feedback, *IEEE Quantum Electron.*, 41(2004) 620-628.
- [3] T. Heil, I. Fisher and W. Elsässer, Dynamics of semiconductor lasers subject to delayed optical feedback: The short cavity regime, *Phys. Rev., Lett.*, 87 (2001) 243901-243905.
- [4] Y. Liu, N. Kikuchi and J. Ohtsubo, Controlling behavior of a semiconductor laser with external optical feedback, *Phys. Rev. E*, 51 (1995) 2697-2700.
- [5] G. Levy and A.A. Hardy, Control and suppression of chaos in flared laser systems: a numerical analysis, *IEEE J. QE*, 34 (1998) 1-6.
- [6] L.G. Luo and P.L. Chu, Suppression of self-pulsing in erbium-doped fiber lasers, *Opt. Lett.*, 22 (1997).2274-2276.
- [7] S.L. Yan, Controlling chaos in a semiconductor laser via weak optical positive feedback and modulating amplitude, *Comm. Theor. Phys.*, 47 (2007) 491-494.
- [8] S.L. Yan, Study on the method of controlling chaos in an Er-doped fiber dual-ring laser via external optical injection and shifting optical feedback light, *Chaos*, 1 (2007) 013106-1-6
- [9] A. Ahlborn and U. Parlitz, Chaos control using notch filter feedback, *Phys. Rev., Lett.*, 3 (2006) 34102-1-4

The Active/Passive Hybrid Control for Suppressing Milling Chatter with Dual Built-in Actuators

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Keywords: Built-in actuator, Chatter suppression, Milling stability, Motorized spindle, Rotor eccentricity.

Abstract. An active/passive hybrid control scheme with dual built-in actuators is proposed for suppressing chatter in milling system, which is to improve the performance of high-speed milling system. One built-in actuator is a set of windings, which are added to the stator of motorized spindle motor, the other is piezoelectric actuator which is embed into two bearings. The finite element model of motorized spindle-bearing-tool system and the dynamic milling are modeled. The milling stability is analyzed with/without damping coefficients of controller and bearings. The displacements of rotor and tool are analyzed. The results prove that the proposed active /passive hybrid control scheme have obvious effect on suppressing the milling chatter, the critical cutting depth is obvious improved by increasing the damping coefficients of bearings, the control current in active control scheme is decreased, the vibration displacement of rotor with piezoelectric actuator embed into bearings is reduced, which will reduce the vibration and noise of the motorized spindle motor.

Introduction

Chatter is much more detrimental to finished surfaces and cutting tools due to its unstable behavior which results in large relative displacements between the tool and work piece [1].

Many control techniques have been developed for chatter to improve the performance of the machining processes, For example, Radecki & Farinholt suppress the unwanted chatter with piezoelectric actuators [2]. Long & Balachandran suppressed chatter by spindle speed variation [3], etc. The external actuators above mentioned are relatively space, so a built-in actuator for suppressing chatter is investigated by Xiaoli & Changsheng in 2012[4]. The built-in actuator overcame the above disadvantages. However, the built-in actuator suppressed the chatter by applying magnetic control force on rotor of motorized spindle. When the rotor was applied on magnetic force, it is bound to bend which cause unequal air gap between stator and rotor. The electro-magnetic field in no uniform air-gap between stator and rotor in a rotating motor can produce unbalance of electro-magnetic forces (UMP). The UMP leads to rotor moving towards stator in the direction of minimal air gap and causes motor vibration and noise. So manufacturers always try to keep rotor eccentricity level even lower to minimize UMP and reduce vibration and noise and further aggravation.

In view of the above UMP caused by rotor eccentricity, the advantage of piezoelectric material and some excellent performances of permanent magnet synchronous motor (PMSM), an active / passive hybrid control scheme for suppressing chatter in milling system with a flexible permanent magnet (PM) motorize spindle is proposed, The built-in dual actuators control scheme can not only save space and real-time control chatter, but also reduce the motor electromagnetic vibration caused by rotor excessive eccentricity, and then to improve control performance.

Principle of Control Force Generation with Built-in Force Actuation

The principle of magnetic control force generation in PMSM with built-in actuator is shown in Fig. 1. Fig. 1(a) and Fig. 1(b) give the principle of radial control force generation in x indirection and negative x indirection. When current i_{2a} is applied in the actuator and the current direction is shown in the Fig. 1(a), a 2-pole magnetic flux ψ_{2a} is generated. The direction of the 4-pole fluxes is same as that of 2-pole fluxes, so the magnetic flux density is increased in the air-gap section 1. The flux density is decreased in the air-gap section 3, because the direction of these magnetic fluxes is opposite. The magnetic forces in the air-gap sections 1 and 3 are no longer equal, the force in the air-gap 1 is large than in the air-gap 3. A radial magnetic control force F in x -axis direction which depends on the magnetic flux is produced. The amplitude of radial force changes as the current i_{2a} value in the conductors 2a changes. When the current i_{2a} in actuator is reversed, the magnetic force in air-gap section 3 is larger than that in air-gap sections 1, which produces a radial control force in the negative x direction [4].

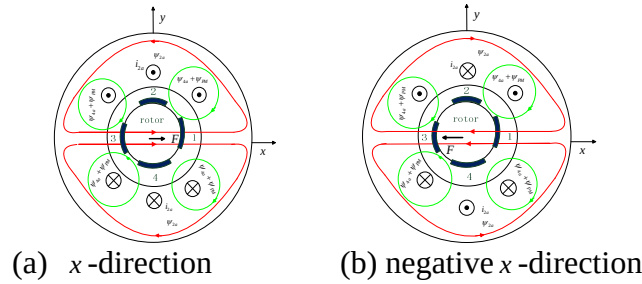


Figure 1. Radial control force.

Dynamic Model of a Flexible Spindle with Dual Built-in Actuators

The flexible motorized spindle with dual built-in actuators is composed of a shaft with distributed mass, bearing with built-in piezoelectric actuator, motor with built-in actuator, tool holder and tool and so on. The one of bearing with built-in piezoelectric actuator is shown in Fig. 2 (a). The model of the bearing with built-in piezoelectric actuator is shown in Fig. 2(b). The flexible motorized spindle with built-in actuators can be divided into different elements along the axis of spindle, such as discs, shafts, bearings, tool holder and tool. Each element is connected to each other in element node. The simplified finite element model of flexible motorized spindle with dual built-in actuators is shown in Fig. 2 (c). In Fig. 2 (c), 1, 2, 3...and 18 is element number. Each node contains the four degrees of freedom (DOF) of the bending motion, two lateral translations and two associated rotations. The differential equations of motion of the flexible motorized spindle with built-in actuator can be obtained by the finite element method[4],

$$M\{\ddot{\rho}\} + (D_s + \omega_s G_s)\{\dot{\rho}\} + K_s\{\rho\} = \{F_T\} + \{F_c\} + \{F_{um}\} \quad (1)$$

where $\rho = [x_1, \theta_{y1}, \dots, x_n, \theta_{yn}, y_1, -\theta_{x1}, \dots, y_n, -\theta_{xn}]^T$ in the general coordinates vector, M , K_s , D_s and G_s are respectively the mass, stiffness, damping and gyroscopic matrixes, F_T is milling force vector, F_c is control force vector, F_{um} is unbalanced magnetic pull force in the motor with built-in actuator, the subscript n is total nodes of motorized spindle, ω_s is rotation speed of motorized spindle, the superscript T is transpose.

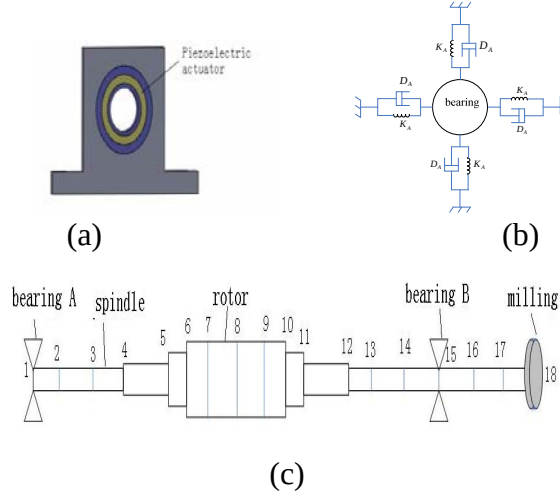


Figure 2. Motorized spindle with dual actuators ((a). Bearing with piezoelectric actuator, (b). The model of bearing with piezoelectric actuator, (c). The finite element model of built-in motorized spindle).

Dynamic Milling Model. A configuration representative of a work piece-tool system is illustrated for milling with a cylindrical end mill, the tool has 4 flutes. The dynamic cutting force model is expressed as follows [5].

$$\begin{pmatrix} F_{Ti,x} \\ F_{Ti,y} \end{pmatrix} = \frac{1}{2} b_c k_t \begin{pmatrix} F_{11} & F_{12} \\ F_{21} & F_{22} \end{pmatrix} \begin{pmatrix} \Delta x_i \\ \Delta y_i \end{pmatrix}, \quad (2)$$

where,

$$\begin{aligned} F_{11} &= \frac{1}{2} (\cos(2\phi) - 2k_n \phi + k_n \sin(2\phi)) \Big|_{\phi_s}^{\phi_e}; \\ F_{12} &= -\frac{1}{2} (\sin(2\phi) + 2\phi - k_n \cos(2\phi)) \Big|_{\phi_s}^{\phi_e}; \\ F_{21} &= -\frac{1}{2} (\sin(2\phi) - 2\phi - k_n \cos(2\phi)) \Big|_{\phi_s}^{\phi_e}; \\ F_{22} &= \frac{1}{2} (\cos(2\phi) + 2k_n \phi + k_n \sin(2\phi)) \Big|_{\phi_s}^{\phi_e}; \end{aligned}$$

here b_c is radial depth of cut, $\Delta x_i, \Delta y_i$ are dynamic depths of cut in the x - and y -directions, respectively, k_t is tangential stiffness of cutter, k_n is radial stiffness of cutter, i is the tool location of node number of finite element model of the motorized spindle.

Radial Electromagnetic Force. According to the Maxwell tensor, the components $F_{m,x}$ and $F_{m,y}$ of the radial tension F_m applied on rotor in x - and y direction can be obtained as follows [7],

$$\begin{bmatrix} F_{m,x} \\ F_{m,y} \end{bmatrix} = C_1 \begin{bmatrix} i_{2x} \\ i_{2y} \end{bmatrix} + C_2 \begin{bmatrix} x_r \\ y_r \end{bmatrix} \quad (3)$$

where, $C_1 = 9\mu_0 l r N_4 I_f / 8\pi \delta_0^2$, $C_2 = 9\mu_0 l r N_4^2 I_f^2 / 16\pi \delta_0^3$.

PD Controller. To make the vibration displacement zeros in x - and y direction ($x \rightarrow 0, y \rightarrow 0$), so PD controller can be expressed as,

$$\begin{aligned} F_{c,x}^*(t) &= -K_c x - D_c \dot{x} \\ F_{c,y}^*(t) &= -K_c y - D_c \dot{y} \end{aligned} \quad (4)$$

Here K_c is proportional coefficient, D_c is differential coefficient.

Chatter Stability and Vibration Analysis

Chatter Stability. The dynamic response is only considered in the analysis of the stability of motorized spindle –bearing-tool system [5], $\Delta\rho = (\Delta x, \Delta y)^T$, $\Delta\rho$ is the dynamic cutting depth of milling. Equation (5) can be derived from equation (1) and (2).

$$M(\Delta\ddot{\rho}) + (D_{1s} + \omega_s G_s)(\Delta\dot{\rho}) + K_{1s}(\Delta\rho) = \frac{1}{2} b_c k_t \frac{N_t}{2\pi} A \Delta\rho \quad (5)$$

where $D_{1s} = D_s + D_c + D_b$, $K_{1s} = K_s + K_c + K_b$, N_t refers to tool flutes, D_b and K_b represent damping and stiffness coefficients of bearing,

$$\begin{aligned} D_c &= (0 \ 0 \ \dots \ D_c^i \ \dots \ 0)^T, K_c = (0 \ 0 \ \dots \ K_c^i \ \dots \ 0)^T \\ D_b &= (0 \ D_A^j \ \dots \ D_B^j \ \dots \ 0)^T, K_b = (0 \ K_A^i \ \dots \ K_B^i \ \dots \ 0)^T. \end{aligned}$$

To analyze the influence caused by the controller parameters and the damping and stiffness coefficients of piezoelectric actuator for the milling chatter stability, the equation (5) can be described in frequency domain using harmonic function,

$$\begin{aligned} &(-M\omega_c^2 + i(D_{1s} + \omega_s G_s)\omega_c + K_{1s})(\Delta\rho(i\omega_c)) \\ &= \frac{N_t}{4\pi} b_c k_t A (1 - \exp(-i\omega_c T)) (\Delta\rho(i\omega_c)) \end{aligned} \quad (6)$$

Where the tooth passing periods are next expressed as $T = (\pi - 2 \tan^{-1} \xi + 2N\pi) / \omega_c$, $N = 0, 1, 2 \dots$ refers to the integer number of waves between teeth, ω_c represents chatter frequency.

The final expression for chatter axial depth b_{lim} of cut is found as,

$$b_{lim} = -\frac{2\pi}{N_t k_t} \bar{A}_{Re} (1 + \xi^2) \quad (7)$$

The Active/Passive Hybrid Chatter Control System for Motorized Spindle with Built-in Actuator. The active/passive hybrid chatter control scheme is divided into two parts, the active control and the passive control. The active vibration control scheme for the flexible PM motorized spindle with built-in actuator is designed as shown in Fig. 3. The drive control is applied to control spindle speed. The control currents i_{2sx}, i_{2sy} are used to suppressing the chatter of motorized spindle in milling system. The passive control scheme is to work by mainly the characteristics of the piezoelectric actuator itself and no special control, so the equivalent damping of the piezoelectric actuator is added to the two bearings during the simulation.

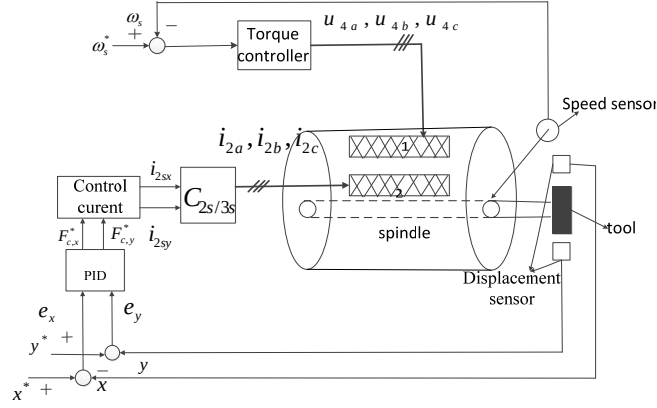


Figure 3. Active vibration control scheme for flexible PM motorized spindle with built-in actuator.

Results Analysis

The active vibration scheme is modeled in MATLAB. The chatter stability and vibration control effect are analyzed. The flexible motorized spindle used in simulation are given in Table1

Table1. Motorized spindle parameters.

m / kg	16.3	R_r / mm	35
L_1 / mm	200	R_{s1} / mm	15
L_2 / mm	544	R_{s2} / mm	30
L_3 / mm	705	R_{s3} / mm	12.5
L_4 / mm	950	R_t / mm	25

Stability Analysis. In the first part, the influences of different proportionality coefficients of controller and bearings on the milling stability are analyzed. The critical cutting depth lobes are shown in Fig. 4(a), in which there are three cases in the proportionality/ stiffness coefficients of controller and bearings ($K_{c0} = 0, K_{b0} = 0$, $K_{c0} = 0.5e7, K_{b0} = 0$,) $K_{c0} = 0.5e7, K_{b0} = 0.5e7$. The proportionality/stiffness coefficients of controller and bearings have scarcely influence on the critical cutting depth of the milling system from Fig. 4(a). There is only 0.0027mm different in different cases. In the second part, the influences of different damping coefficients of controller and bearings with the piezoelectric actuator on the milling stability are shown in Fig. 4(b). The same three cases are analyzed ($D_{c0} = 1e4, D_{b0} = 0$ and $D_{c0} = 1e4, D_{b0} = 1e4$). The critical cutting depth of milling system is larger raised with adding the damping coefficients of controller and bearings with the piezoelectric actuator to milling system. The about 0.01325mm is raised, especially the damping coefficient of bearings with the piezoelectric actuator is added to milling system. The result shows the damping coefficient of bearings with the piezoelectric actuator has larger influence on the critical cutting depth of milling system.

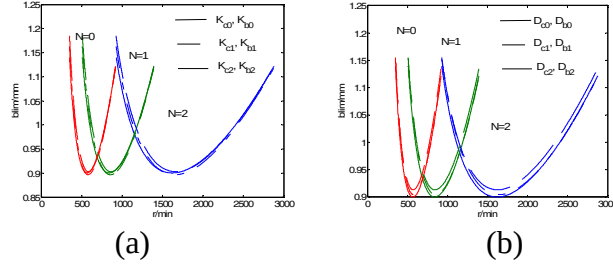


Figure 4. Stability lobe diagram (N=0 to 2)((a) Different proportionality coefficient ,(b) Different damping coefficient).

Vibration Analysis. The vibration effect and the displacements of the rotor and tool are analyzed in this section. The vibration displacement in x -direction is only given in this paper. The two cases used to analyze are the cutting depth $b = 2\text{mm}$ and $b = 2.24\text{mm}$, the displacements of rotor are shown in Fig. 5, that of tool are in Fig. 6 and the control currents are in Fig. 7. The Fig. 6 proved the active/passive hybrid control scheme has obvious control effect. According to Fig. 7, the control current applied to actuator is decreased with the damping coefficient adding to bearings. That is, the import energy is decreased. The Fig. 5 shows that the displacements of rotor are reduced with damping coefficients of bearings. The unbalanced magnetic pull (UMP) produced by the eccentricity between stator and rotor in a rotating electrical machine will be decreased, so the vibration and noise and further aggravation caused by UMP will be reduced to improve the performance of milling system.

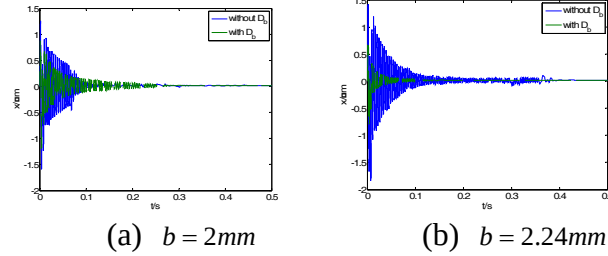


Figure 5. Vibration displacement of the rotor when different cutting depth.

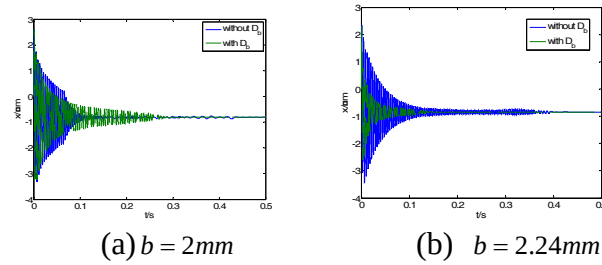


Figure 6. Vibration displacement of the Tool.

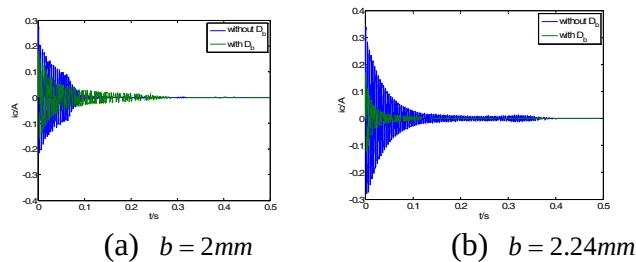


Figure 7. Control Current.

Conclusion

An active/passive hybrid control scheme for suppressing chatter in milling system is investigated, which is based on dual built-in actuators. The dual built-in actuators control scheme can not only save space and real-time control chatter but also reduce the import energy and raise the critical cutting depth, and then to improve control performance.

Acknowledgements

Zhejiang Provincial Natural Science Foundation of China (LY14E050004)

References

- [1] M. Siddhpura & R. Paurobally. A review of chatter vibration research in turning, *International Journal of Machine Tools & Manufacture*, 61 (2012) 27-47.
- [2] P. Radecki & K. Farinholt. Vibration suppression in cutting tools using a collocated piezoelectric sensor/actuator with an adaptive control algorithm, *Journal of Vibration and Acoustic*, 122 (2010) 1-8.
- [3] X. Long & B. Balachandran,. Stability of up-milling and down-milling operations with variable spindle speed, *Journal of Vibration and Control*, 16 (2010) 1151–1168.
- [4] X.L.Qiao & C.S. Zhu,. Active control of milling chatter based on the built-in force actuator, *Chinese Journal of Mechanical Engineering*, 48 (2012) 185-192
- [5] S. Gao, et al. Stability prediction in high-speed milling including the thermal preload effects of bearing, *Proceedings of the Institution of Mechanical Engineers Part E-Journal of Process Mechanical Engineering*, 224 (2010) 11-22.

Study on Critical Nodes in Social Networks

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Keywords: Social Networks, Community Structure, Critical Node Problem, Influence Maximization.

Abstract. Discovering critical nodes in social networks has many important applications and how to find out the K critical nodes with the most influence in a social network is a NP problem. Considering the widespread community structure in social networks, this paper presents an algorithm of discovering critical nodes based on two information diffusion models and obtains each node's marginal contribution by using Owen value and Monte-Carlo method. And the solution of the critical nodes problem is the K nodes with the highest marginal contributions.

Introduction

We may find out some individuals with influence and let them recommend the product to their friends so that such a cascade spreads by the greatest extent in the people. How to choose these influential individuals is called by critical node problem (CNP). An effective solution for the problem has an important practical value [1]. It's also a basic process happened in the network for the spread, diffusion and cascade behavior of information. This paper gives a solution for the CNP, which assigns a marginal contribution for every node in a community of social network using the solution concept and union concept of cooperative games. Then we sort all nodes by their contribution and obtain critical nodes according to some rules.

Backgrounds

The models for information propagation on networks have been widely studied [2]. We consider two models in this paper: independent cascade model (ICM) and linear threshold model (LTM) [3].

1) ICM Model. In this model, a propagation probability $p_{u,v}$ is given for each edge $(u,v) \in E$, that is, vertex v is activated with probability $p_{u,v}$ by u . When an initial set A_0 of active vertices is given, the diffusion process spreads up according to the following randomized rule. When a vertex u is activated at time-step i , it has a single chance for activating its neighbor v with $p_{u,v}$. If u succeeds, v will become active at time-step $i+1$. Here, if v has multiple parent vertices that become active at time-step i for the first time, then their activation attempts are sequenced in an arbitrary order. Whether or not u succeeds, it cannot make any further attempts to activate v in subsequent steps. The process runs until no more activation is possible.

2) LTM Model. In this model, vertex v is influenced by each neighbor u according to a weight $w_{v,u}$ such that $\sum_u w_{v,u} \leq 1$. Each vertex has a predefined threshold $\theta_v \in [0,1]$, which is chosen uniformly at random. When an initial set A_0 with active vertices is given, the diffusion process unfolds according to the following randomized rule. All activated vertices at time-step i still keep active at time-step $i+1$. Whether or not any inactivated vertex is activated is determined by its neighbors' weights such that $\sum_u w_{v,u} \geq \theta_v$. The process runs until no more activation is possible.

The difference between ICM and LTM is that each attempt of activation is independent of the attempts by all the other active individuals while in the later model each inactive individual is influenced by the aggregated weight of all its active neighbors.

A network is modeled as a graph $G=(V, E)$ with vertices in V modeling the individuals in the network and edges in E modeling the relationship between individuals. A vertex has two states: active and inactive, which means whether a product or idea is accepted by individuals or not. Assume that a vertex can only change from inactive to active and not vice versa; an inactive vertex can be activated by its active neighbor vertices and an active vertex can activate its inactive neighbors; the increment of activated vertices represents the dissemination of information.

The definition of critical node problem is as follow:

Input: $G=(V, E)$ and an integer k ;

Output: $A = \arg \max_{S \subseteq V, |S| \leq k} \sigma(S)$

For k -CNP, we hope to find out a set A with k elements and maximize $\sigma(A)$ which is a NP-hard. But, Nemhauser et al. has approved there is a greedy algorithm (see Algorithm 1) approximates the optimum within a factor of $(1-1/e)$ [4].

Algorithm 1. Greedy Algorithm

```

1:   $A = \Phi$ 
2:  for  $i = 1$  to  $k$  do
3:     $v_i = \arg \max_{u \in V \setminus A} (\sigma(A \cup \{u\}) - \sigma(A))$ 
4:     $A = A \cup \{v_i\}$ 
5:  end for

```

There is a key problem how to compute the value of $\sigma(A)$ in Algorithm 1. Currently, we haven't any efficient method to get its exact solution. However, we can use Monte-Carlo method to simulate the process of influence spread for obtaining approximate results by high probability.

The Algorithm

Introduction of Algorithm

Given a finite set of players N , cooperative game with transferable utility is a pair (N, v) , characteristic function $v: 2^N \rightarrow \mathbb{R}$ and $v(\emptyset) = 0$. For $\forall i \in N$, if payoff vector satisfies $x_i \geq v(\{i\})$ and $\sum_{i=1}^N x_i = v(N)$, then it is so-called an allocation of (N, v) . The solution of cooperative game is a kind of allocation rule and the allocated payoff for every player denotes a method to measure the negotiation strength of the players in the game. Shapely presented a solution concept which finds out the only allocation distribution scheme from the solutions with different property, i.e., it assigns the player's payoff according to the importance of every player for the game [5]. The Shapely value of the player i in the game (N, v) is

$$Sh_i(v) = \sum_{\{S \subseteq N | i \in S\}} \frac{(n-s)!(s-1)!}{n!} (v(S \cup \{i\}) - v(S)), \quad \forall i \in N \quad (1)$$

where $n = |N|$ and $s = |S|$.

However, the Shapely value doesn't consider the impact of coalition structure and Owen extends it [6]. Each union obtains its payoff from the game between the unions, and then the payoff is allocated by the internal game among the members of the union. All the payoffs are computed by the Shapely value.

Assumed that $N = \{1, 2, \dots, n\}$ and $M = \{1, \dots, m\}$, a partition $P = \{N_1, N_2, \dots, N_m\}$ is a coalition structure on N . Let N_k a union and $\bigcup_{1 \leq k \leq m} N_k = N$. When $l \neq k$, $N_l \cap N_k = \emptyset$. For $i \in N$, $k(i)$ denotes the index of the union containing player i , so $k(i)$ is defined by the relation $i \in N_{k(i)}$.

For $k \in M$ and $S \subseteq N_k$, the game $\hat{v}_S$ is defined by

$$\hat{v}_S(Q) = \begin{cases} v(\bigcup_{h \in Q} N_h), & k \notin Q \\ v(\bigcup_{h \in Q \setminus \{k\}} N_h \cup S), & k \in Q \end{cases} \quad (2)$$

where $Q \subseteq M$.

The game $\bar{v}_k$ is defined by $\bar{v}_k(S) = Sh_k(\hat{v}_S)$, then the Owen value of the player $i \in N$ in the game $\bar{v}_{k(i)}$ is $Ow_i(v, P) = Sh_i(\bar{v}_{k(i)})$.

A New Algorithm

Because the community structure is prevalent in the social networks [7], we respectively consider the community's influence on the information diffusion and every node's influence in the community. We take the nodes in the social network as the players in the cooperative game and information diffusion as coalition formulation. We can map the information diffusion into cooperative game with transferable utility which is formalized in the literature [8]. So, we can use the Owen value to obtain the marginal contribution of every node. Because the Owen value can be seen as a two-step procedure in which the Shapely value applies twice, we firstly compute a node's Shapely value.

Given a node $i \in N$ and a subset $S \subseteq N$ such that $i \notin S$, the marginal contribution of node i is $v(S \cup \{i\}) - v(S)$, $\forall S \subseteq N \setminus \{i\}$. Consider the set of all possible permutations Ψ on N , let $\psi \in \Psi$ and define $S_i(\psi)$ to be the set of all nodes appearing before node i in the permutation ψ . So, the average marginal contribution of node i to the given coalitional game is

$$\frac{1}{n!} \sum_{\psi \in \Psi} [v(S_i(\psi) \cup \{i\}) - v(S_i(\psi))]. \quad (3)$$

Note that, this method must work with $n!$ permutations and its computational complexity is $O((n/e)^n)$ [9]. Therefore, we give the approximate method for computing Shapely value. Randomly generate t sets Ψ_t with t permutations, see Algorithm 2, let $\psi \in \Psi_t$ and $\psi(i)$ denotes the i th node in the permutation. The number of activated nodes after running the diffusion model when the node $\psi(1)$ is activated is the contribution of $\psi(1)$. Next, we consider the node $\psi(2)$. If $\psi(2)$ becomes active after $\psi(1)$ is activated, then the contribution of $\psi(2)$ is 0. Otherwise, the contribution of $\psi(2)$ is the number of activated nodes by $\psi(2)$. Therefore, we can get the contributions of $\psi(3), \dots, \psi(n)$. For $\psi \in \Psi_t$, repeat the above process R times. Then the average contribution of each node in the diffusion process can be calculated. We can obtain the top-k nodes sorting by the greatest influence and ensure that they are not adjacent to each other. See Algorithm 3 and Algorithm 4.

Algorithm 2. CREAT_ Ψ_t (n)

```

t=random(n)
m=random(n)
Select_t (n)
  for i = 0 to m do
    arr[i] = random() % (n-i)
    for j=0 to i do
      if arr[j] <= arr[i] then
        arr[i]=arr[i]+1
      end
    end
    v = arr[i]
    for k = i-1 to j do

```

```

        arr[k+1] = arr[k]
        k=k-1
    end
    arr[j] = v
end
for h=2 to t do
     $|\Psi_{h-1}| = \text{arr}$ 
    n=n-m
    m=random(n)
    Select_t(n)
end
 $|\Psi_h| = \text{arr}$ 

```

Algorithm 3. ShapelyValue(v, R)

```

n= $|\Psi|$ 
t= $|\text{CREAT}_\Psi|$ 
tmp[1...n]=0
ShV[1...n]=0
for i=1 to t do
    for r=1 to R do
        for j=1 to n do
            tmp_b= $v(S_j(\psi_i) \cup \{j\}) - v(S_j(\psi_i))$ 
            tmp[j]= $\alpha \text{tmp}[j] + (1-\alpha) \text{tmp\_b}$ 
        end
    end
end
for i=1 to n do
    ShV[i]=tmp[i]/t
end

```

Algorithm 4. TOP_K(V)

```

TopK[1...k]=0
AsceSort(V);
TopK[1]= V[1]
i=1
j=2;
while i < k do
    if V[j] is not adjacent to TopK[1...i-1] then
        TopK[i]= V[j]
        i=i+1
    end
    j=j+1
end

```

According to the description in section 3.1, we give the calculation method of the Owen value. Roger Guimerà, et al study node roles in the community according to within-module degree z and participation coefficient P , which are divided into seven categories $Role = \{R1, R2, R3, R4, R5, R6, R7\}$ [10]. We consider two types of the roles: Non-hub connector node ($z < 2.5$ and $0.62 < P \leq 0.80$, R3) and Connector hub ($z \geq 2.5$ and $0.30 < P \leq 0.75$, R6). The most of the nodes which belong to these two roles connect to other communities.

We use the CNM algorithm [11] to divide the network $G=(V,E)$ into l communities $C=\{C_1,C_2,...,C_l \mid C_i=(V_i,E_i), i=1,...,l\}$ and assign the role $rv_i^h \in Role$ for every node $v_i^h \in V_i$ in the community. Let $Role'=\{R3,R6\}$, $i=1,...,l$, $j=1,...,l$,

$$V'=\{v_i^h, v_j^g \mid \exists(v_i^h, v_j^g), v_i^h \in V_i, rv_i^h \in Role', i \neq j\} \quad ,$$

$$E'=\{e_i^h, (v_i^h, v_j^g) \mid e_i^h=(v_i^{h_a}, v_i^{h_b}) \in E_i, v_i^{h_a} \in V_i, v_i^{h_b} \in V_i, rv_i^h \in Role', i \neq j\},$$

define $G'=(V',E')$ as community game network. So, we can obtain the Shapely value of every node in the network G' and take the sum of the Shapely values of all nodes in the same community as the community's payoff. Then we treat a community as a separate network and calculate the Shapely value of every node in the community. The node's Owen value is assigned according to the normalized Shapely value of the node in the community and the community's payoff. So, we can get the k critical nodes by Algorithm 4.

Conclusions

For solving the CNP, this paper presents a method based on the Owen value from cooperative game, which considers the widespread community structure in social networks. How to improve the time efficiency of the CNP-OV needs to be studied in the next step.

Acknowledgement

This work is supported by National Social Science Foundation of China (No.11BFX125), PuJiang Talent Project and Public Security Discipline Construction Foundation.

References

- [1] He N., Li D.Y., Gan W.Y., Zhu X. Mining vital nodes in complex networks. Computer Science, 2007, 34 (12): 1-5.
- [2] Even-Dar E., Shapira A. A note on maximizing the spread of influence in social networks. WINE 2007, LNCS 4858, 2007: 281-286.
- [3] Shetty J., Adibi J. Discovering Important Nodes through Graph Entropy The Case of Enron Email Database. The 3rd international workshop on Link discovery, Chicago, Illinois, 2005: 74-81.
- [4] Nemhauser G., Wolsey L., Fisher M. An analysis of the approximations for maximizing submodular set functions. Mathematical Programming, 1978, 14 (1): 265-294.
- [5] Shapley L.S. A value for n-person games. In: Kuhn H.W., Tucker A.W. (Eds.), Contributions to the Theory of Games II, Princeton University Press, 1953, 307-317.
- [6] Owen G. Values of games with a priori unions. In: Henn R., Moeschlin O. (Eds.), Essays in mathematical economics and game theory, Springer-Verlag, Berlin, 1977, 76-88.
- [7] Scott J. Social Network Analysis: A Handbook (2nd ed). Sage, London, 2000.
- [8] Jean-Francois Caulier. Network Games as TU Cooperative Games: The Core, the Shapley Value and Simple Network Games. 2009,
http://centres.fusl.ac.be/CEREC/document/seminars/caulier_cerec_feb2009.pdf.
- [9] Cormen T.H., Leiserson C.E., Rivest R.L., Stein C. Introduction to Algorithms, 2nd ed. Cambridge, MA: MIT Press, 2001.
- [10] Guimerà R., Amaral L.A.N. Functional cartography of complex metabolic networks. Nature, 2005, 433: 895-900.
- [11] Clauset A., Newman M.E.J., Moore C. Finding community structure in very large networks. PHYSICAL REVIEW E, 2004, 70 (6): 066111 (6).

Generalized Hooke's Law for Materials Between Orthotropic Materials and Isotropic Materials

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Keywords: Special metal, 8 to 3 Engineering Constant Materials, Flying Saucer Materials.

Abstract. In this paper, the generalized Hooke's law for the materials which are so-called materials of 8 parameter material to 3 engineering materials is discussed. So-called here from 8 parameter material to 3 parameter material are expressed and discussed. They include advanced composite materials and may include "Flying Saucer" material.

Introduction

In recent years, composite materials have been applied to various fields, such as aeronautics and civil areas [1]. In this paper, the generalized Hooke's law for the materials between isotropic and three dimensional orthotropic materials are discussed. First, the generalized Hooke's law for orthotropic materials, such as composites, is listed. Then, generalized Hooke's law for so-called here from 8 parameter material to 3 parameter material are expressed and discussed. Finally, Applications are also mentioned.

Generalized Hooke's Law

Orthotropic Materials of Nine Parameter Strength. Generalized Hooke's law for orthotropic materials [1], such as composite materials, is expressed below.

$$\begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ \tau_{yz} \\ \tau_{yz} \\ \tau_{yz} \end{Bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{13} & 0 & 0 & 0 \\ & C_{22} & C_{23} & 0 & 0 & 0 \\ & & C_{33} & 0 & 0 & 0 \\ & \text{symm.} & & C_{44} & 0 & 0 \\ & & & & C_{55} & 0 \\ & & & & & C_{66} \end{bmatrix} \begin{Bmatrix} \epsilon_x \\ \epsilon_y \\ \epsilon_z \\ \gamma_{yz} \\ \gamma_{zx} \\ \gamma_{xy} \end{Bmatrix} \quad (1)$$

It can also be expressed by engineering constant form.

The following materials are all estimated materials. Materials between anisotropic and orthotropic materials are discussed.

Materials of Eight Elastic Moduli Materials Estimated. Generalized Hooke's law for eight elastic modulus materials is expressed below. They may be semi-metal or semi-ceramics. These materials are estimated material. In compliance matrix form is

$$\begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ \tau_{yz} \\ \tau_{yz} \end{Bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{13} & 0 & 0 & 0 \\ & C_{22} & C_{23} & 0 & 0 & 0 \\ & & C_{33} & 0 & 0 & 0 \\ & \text{symm.} & & 0 & 0 & 0 \\ & & & & C_{55} & 0 \\ & & & & & C_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ \gamma_{zx} \\ \gamma_{xy} \end{Bmatrix} \quad (2)$$

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ \gamma_{zx} \\ \gamma_{xy} \end{Bmatrix} = \begin{bmatrix} S_{11} & S_{12} & S_{13} & 0 & 0 & 0 \\ & S_{22} & S_{23} & 0 & 0 & 0 \\ & & S_{33} & 0 & 0 & 0 \\ & \text{symm.} & & 0 & 0 & 0 \\ & & & & S_{55} & 0 \\ & & & & & S_{66} \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ \tau_{yz} \\ \tau_{yz} \end{Bmatrix} \quad (3)$$

One of this kind of materials can be also expressed, below.

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ \gamma_{zx} \\ \gamma_{xy} \end{Bmatrix} = \begin{bmatrix} \frac{1}{E_1} & -\frac{\mu_{21}}{E_2} & -\frac{\mu_{31}}{E_3} & 0 & 0 & 0 \\ -\frac{\mu_{12}}{E_1} & \frac{1}{E_2} & -\frac{\mu_{32}}{E_3} & 0 & 0 & 0 \\ -\frac{\mu_{13}}{E_1} & -\frac{\mu_{23}}{E_2} & \frac{1}{E_3} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{1}{G_{31}} & 0 \\ & & & & & \frac{1}{G_{12}} \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ \tau_{yz} \\ \tau_{yz} \end{Bmatrix} \quad (4)$$

Materials of Seven Elastic Moduli Estimated. Generalized Hooke's law for one of seven elastic modulus materials, is expressed below

$$\begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ \tau_{yz} \end{Bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{13} & 0 & 0 & 0 \\ & C_{22} & C_{23} & 0 & 0 & 0 \\ & & C_{33} & 0 & 0 & 0 \\ & \text{symm.} & & 0 & 0 & 0 \\ & & & & 0 & 0 \\ & & & & & C_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ \gamma_{xy} \end{Bmatrix} \quad (5)$$

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ \gamma_{xy} \end{Bmatrix} = \begin{bmatrix} S_{11} & S_{12} & S_{13} & 0 & 0 & 0 \\ & S_{22} & S_{23} & 0 & 0 & 0 \\ & & S_{33} & 0 & 0 & 0 \\ & \text{symm.} & & 0 & 0 & 0 \\ & & & & 0 & 0 \\ & & & & & S_{66} \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ \tau_{yz} \end{Bmatrix} \quad (6)$$

Compliance matrix and engineering constant forms are

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ \gamma_{xy} \end{Bmatrix} = \begin{bmatrix} \frac{1}{E_1} & -\frac{\mu_{21}}{E_2} & -\frac{\mu_{31}}{E_3} & 0 & 0 & 0 \\ -\frac{\mu_{12}}{E_1} & \frac{1}{E_2} & -\frac{\mu_{32}}{E_3} & 0 & 0 & 0 \\ -\frac{\mu_{13}}{E_1} & -\frac{\mu_{23}}{E_2} & \frac{1}{E_3} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{1}{G_{12}} \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ \tau_{yz} \end{Bmatrix} \quad (7)$$

Materials of Six Elastic Moduli Materials. This is six elastic modulus material.

$$\begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{13} & 0 & 0 & 0 \\ & C_{22} & C_{23} & 0 & 0 & 0 \\ & & C_{33} & 0 & 0 & 0 \\ & \text{symm.} & & 0 & 0 & 0 \\ & & & & 0 & 0 \\ & & & & & 0 \end{bmatrix} \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \quad (8)$$

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} = \begin{bmatrix} S_{11} & S_{12} & S_{13} & 0 & 0 & 0 \\ & S_{22} & S_{23} & 0 & 0 & 0 \\ & & S_{33} & 0 & 0 & 0 \\ & \text{symm.} & & 0 & 0 & 0 \\ & & & & 0 & 0 \\ & & & & & 0 \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \quad (9)$$

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} = \begin{bmatrix} \frac{1}{E_1} & -\frac{\mu_{21}}{E_2} & -\frac{\mu_{31}}{E_3} & 0 & 0 & 0 \\ -\frac{\mu_{12}}{E_1} & \frac{1}{E_2} & -\frac{\mu_{32}}{E_3} & 0 & 0 & 0 \\ -\frac{\mu_{13}}{E_1} & -\frac{\mu_{23}}{E_2} & \frac{1}{E_3} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \quad (10)$$

Materials of Five Elastic Moduli Estimated.

In compliance matrix form is

$$\begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{13} & 0 & 0 & 0 \\ & C_{22} & C_{23} & 0 & 0 & 0 \\ & & C_{22} & 0 & 0 & 0 \\ & & & 0 & 0 & 0 \\ & & & & 0 & 0 \\ & & & & & 0 \end{bmatrix} \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \quad (11)$$

symm.

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} = \begin{bmatrix} S_{11} & S_{12} & S_{13} & 0 & 0 & 0 \\ & S_{22} & S_{23} & 0 & 0 & 0 \\ & & S_{22} & 0 & 0 & 0 \\ & & & 0 & 0 & 0 \\ & & & & 0 & 0 \\ & & & & & 0 \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \quad (12)$$

symm.

Transverse orthotropic material is included in this kind of materials [1]. They are different from each other.

Expressing it by engineering constants, below.

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} = \begin{bmatrix} \frac{1}{E_1} & -\frac{\mu_{21}}{E_2} & -\frac{\mu_{31}}{E_2} & 0 & 0 & 0 \\ -\frac{\mu_{12}}{E_1} & \frac{1}{E_2} & -\frac{\mu_{32}}{E_2} & 0 & 0 & 0 \\ -\frac{\mu_{13}}{E_1} & -\frac{\mu_{23}}{E_2} & \frac{1}{E_2} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \quad (13)$$

Materials of Four Elastic Moduli Estimated.

In compliance matrix form is

$$\begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} = \begin{bmatrix} C_{11} & C_{12} & C_{12} & 0 & 0 & 0 \\ & C_{22} & C_{12} & 0 & 0 & 0 \\ & & C_{33} & 0 & 0 & 0 \\ & \text{symm.} & & 0 & 0 & 0 \\ & & & & 0 & 0 \\ & & & & & 0 \end{bmatrix} \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \quad (14)$$

Expressing it, as below.

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} = \begin{bmatrix} S_{11} & S_{12} & S_{12} & 0 & 0 & 0 \\ & S_{22} & S_{12} & 0 & 0 & 0 \\ & & S_{33} & 0 & 0 & 0 \\ & \text{symm.} & & 0 & 0 & 0 \\ & & & & 0 & 0 \\ & & & & & 0 \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \quad (15)$$

Materials of Three Elastic Moduli Estimated. Generalized Hooke's law for one of four elastic modulus materials, is expressed below. It is expressed by engineering constants.

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} = \begin{bmatrix} \frac{1}{E_1} & -\frac{\mu_{12}}{E_2} & -\frac{\mu_{12}}{E_2} & 0 & 0 & 0 \\ -\frac{\mu_{12}}{E_2} & \frac{1}{E_2} & -\frac{\mu_{12}}{E_2} & 0 & 0 & 0 \\ -\frac{\mu_{12}}{E_2} & -\frac{\mu_{12}}{E_2} & \frac{1}{E_3} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \quad (16)$$

Generalized Hooke's law for one of the three elastic modulus materials, is expressed below. They are in stiffness and compliance forms, respectively.

$$\begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} = \begin{bmatrix} C_{11} & 0 & 0 & 0 & 0 & 0 \\ & C_{22} & 0 & 0 & 0 & 0 \\ & & C_{33} & 0 & 0 & 0 \\ & \text{symm.} & & 0 & 0 & 0 \\ & & & & 0 & 0 \\ & & & & & 0 \end{bmatrix} \begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \quad (17)$$

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} = \begin{bmatrix} S_{11} & 0 & 0 & 0 & 0 & 0 \\ & S_{22} & 0 & 0 & 0 & 0 \\ & & S_{33} & 0 & 0 & 0 \\ & \text{symm.} & & 0 & 0 & 0 \\ & & & & 0 & 0 \\ & & & & & 0 \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \quad (18)$$

The one ply composite material is also three elastic modulus material. Expressing it, below.

$$\begin{Bmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} = \begin{bmatrix} \frac{1}{E_1} & 0 & 0 & 0 & 0 & 0 \\ & \frac{1}{E_2} & 0 & 0 & 0 & 0 \\ & & \frac{1}{E_3} & 0 & 0 & 0 \\ & 0 & 0 & 0 & 0 & 0 \\ & 0 & 0 & 0 & 0 & 0 \\ & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{Bmatrix} \sigma_x \\ \sigma_y \\ \sigma_z \\ 0 \\ 0 \\ 0 \end{Bmatrix} \quad (19)$$

Materials of Two Elastic Moduli Materials.

Whether the materials of 8 elastic constants to 3 elastic constants are so-called “Special metal” needs to be varied.

Conclusion

The following conclusions were obtained as below.

Materials of from 8 to 3 engineering constants were proposed. One kind of those stiffness matrice and compliance matrice expressed. That methodology was proposed to find new engineering materials. Applications were discussed.

References

- [1] L.M. Chen and B.N. Yang, Mechanical Analysis for Composite Materials, Beijing: China Science and technology publication, 2008, pp. 80-88.
- [2] W.X. Gou, Strength of Materials (I), 2nd ed., Beijing: Science publication, 2011, pp. 244-250.
- [3] M.H. Yu, New Systems of Strength Theory: Theory, Development and Application, 2nd ed., Xi'an: Xi'an Jiaotong University publication, 2011, pp. 63-65.
- [4] F. Sun, W.J. Tian, R. Yin and K.F. He, “Dynamic Mohr’s strength theory”, The 3rd International Conference on Mechanic Automation and Control Engineering, vol. 3, Baotou: IEEE, 2012, pp. 1457-1459.
- [5] F. Sun, W.J. Tian and R. Yin, “Dynamic Stiffness Criterion (1)”, The 3rd International Conference on Mechanic Automation and Control Engineering, vol. 2, Baotou: IEEE, 2012, pp. 806-809.
- [6] F. Sun, W.J. Tian and R. Yin, “Dynamic Stiffness Criterion (2)”, The 3rd International Conference on Mechanic Automation and Control Engineering, vol. 4, Baotou: IEEE, 2012, pp. 2261-2264.

Numerical Simulation of Quenching Residual Stress for Engine Crankshaft

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Keywords: Engine Crankshaft; Temperature field; Finite element analysis; Residual stress; Fatigue strength.

Abstract. The residual stress of engine crankshaft was generated through surface quenching so as to improve the fatigue strength of the crankshaft surface. For knowing the distribution of residual stress, the residual stress was analyzed through finite element numerical simulation. First, it needs to calculate the induction heating temperature field and simulate the cooling process, the cooling process will produce different residual stress. Since residual compressive stress can increase the fatigue strength of the crankshaft, residual stress of fillet portion and straight segments of crankshaft is based on that. The results show that: the crankshaft surface hardening by induction heating, the rounded part of the residual compressive stress to $-320\text{MPa} \sim -131\text{MPa}$ in the range, the straight sections of the residual compressive stress is about to $-616\text{MPa} \sim -454\text{MPa}$ range. The research method provides guidance for the quenching process.

Introduction

Crankshaft will undertake gas pressure and inertia forces in working state. The inertia force and inertia movement taken by the parts of crankshaft group will cause parts to suffer bending and torsion alternative loads. In a lot of theory and practice, the bending load plays a decisive role to influence crankshaft fatigue strength, because many engines have adopted damping device to decrease the impact of torsion load now [1~4.] From the statistics of crankshaft destruction, about 80% of the crankshaft destruction is caused by bending fatigue [5~6]. After the crankshaft surface is induction hardened, the fillet and the journal surface will produce residual compressive stress. Compressive stress will serve to improve the fatigue strength of the crankshaft and the journal will have better abrasion resistance, which could improve fatigue life of crankshaft. Therefore, the distribution of residual stress after quenching has a great impact on the performance of crankshaft which makes it a very important research issue [7]. Zhang and Shen made in-depth analysis on the numerical simulation of induction heating, they studied numerical simulation on axisymmetric workpiece by using boundary element method and finite element method respectively [8]. Fang, by using the electromagnetic coupling and hot - stress coupling function of ANSYS/Multiphysics finite element software, simulated the process of Induction hardening and the generation principle of residual stress, and the fatigue strength calculation of the 6110 crankshaft after being hardened is finished [9]. According to crankshaft on 16V240 diesel engine, Wang made the heating simulation under intermediate frequency quenching condition, and introduced the method to predict the depth of hardening layer by discussing the critical cooling rate [10].

On this basis, this paper established a model and simulated surface induction heating process and quenching process, the residual stress on the surface of crankshaft was carried out by using finite element simulation analysis. Then the residual stress state was analyzed, which could provide guidance for quenching process.

Finite Element Model and Boundary Conditions

In this article, the 1/2 of quenching partial section model was established in order to reduce calculation time and improve the result accuracy. In the meantime, the crankshaft induction hardening is operated sectionally, so it is logical to take one journal as the model in simulating calculation. Since the oilhole on the journal makes less influence to the residual stress distribution, it also can be simplified. Considering the magnetic flux leakage phenomenon, surrounding air also needs to be modeled. According to requirements of quenching process, the gap between the induction coil and the surface of journal is considered as 3 mm. Geometry model is shown in Fig. 1. A2 is the journal portion, A1 is the coil section, A3 is the air portion.

The Meshing model is shown in Fig. 2, mesh-intensive areas is located on outer surface of the journal, in order to simulate the rapid change of inducing eddy currents in this region due to surface skin effect, it is necessary to refine elements. During the Cooling temperature field calculation and the residual stress calculation process, the elements of coil portion and air need to be set as invalid (null) elements.

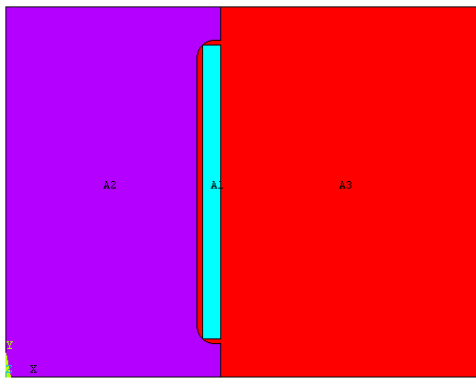


Figure 1. Schematic diagram of the crankshaft induction hardening.

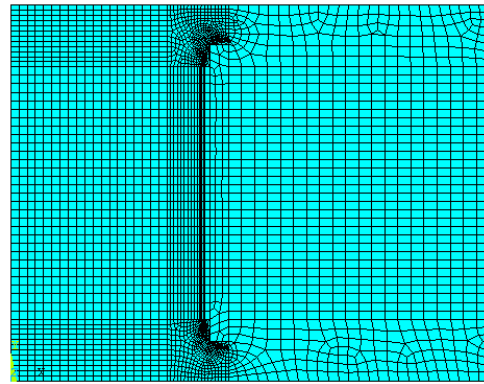


Figure 2. Physical field element mesh.

Electromagnetic boundary conditions: the magnetic potential is zero in the air and the outer boundary of the journal portion, and assuming the current through the coil cross section is uniform, the uniform current density load, i.e. electromagnetic excitation conditions.

Thermal boundary conditions: Set the temperature axisymmetrical about x axis, and distributed uniformly along the z-axis that the deflector of temperature on the y-axis is zero, i.e. $\frac{\partial T}{\partial y} = 0$. In the meantime, the eddy current generated by electromagnetic field is set as a heat source for heating the journal surface.

Calculation

Electromagnetic Induction Transient Temperature Field Calculation. When using sequential coupling method to calculate Electric-Magnetic-thermal coupling problem, the electromagnetic thermal coupling is complex because it has different type of physical field unit and interaction between thermal magnetic. In the meantime, the material properties with temperature changes, thus the thermal coupling electromagnetic calculations are very complex nonlinear problems, which need to conduct 2-D harmonic analysis and use analysis results for thermal analysis, then to fix the electromagnetic parameters according to thermal analysis results until we get the right quenching temperature. Fig. 3 shows the entire physical field distribution of magnetic field lines, Fig. 4 shows the crankshaft journal internal magnetic field lines distribution, Fig. 5 shows the temperature field distribution at the end of induction heating.

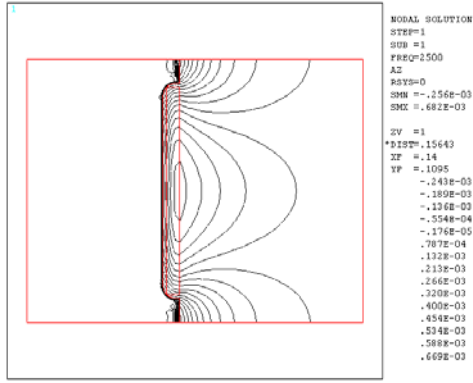


Figure 3. the entire physical field distribution of magnetic field lines.

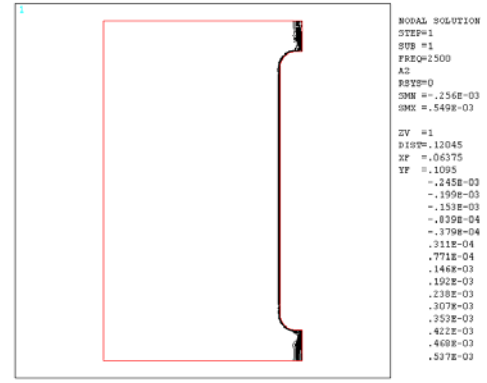


Figure 4. the crankshaft journal internal magnetic field lines distribution.

It can be obtained from the calculation results that the maximum value of the induction heating temperature appears on the surface of the connecting rod journal, while the heart is almost no heating, which is characterized by the induction heating temperature distribution. It can be seen that the surface temperature is 1034°C , the fillet surface temperature is 810°C , the heating thickness is about 15mm, heating surface temperature rose rapidly to austenite transition temperature, which leads the surface portion convert to austenite, while the heart portion has on metallographic transformation since it is not heated.

Temperature Calculation of Cooling Process. After the crankshaft surface is heated, the hole on the induction coil injects coolant to the heated surface in a certain pressure. The PAG is used as the coolant to conduct quench cooling in this article. In this paper, the quench cooling time is set as 40 seconds. The average convection heat transfer factor between the crankshaft surface and quenching coolant is took as $103000\text{W}/(\text{m}^2 \cdot \text{K})$, and it is $100\text{W}/(\text{m}^2 \cdot \text{K})$ when the crankshaft is being cooled in air. The temperature of coolant and air are both 25°C . The initial temperature field of journal in cooling condition is set according to the temperature results after being induction heated. The convection heat transfer coefficient above is applied to the journal surface as the boundary condition. The analysis type is transient thermal analysis. It can be seen from Fig. 9 that after coolant injection, the temperature on journal surface rapidly reduced 25.107°C , and the maximum temperature in the journal is 153.675°C . Because during quench cooling, the heat on the journal surface transmits to the inside part which leads the internal temperature rises. Since the surface temperature decreased rapidly, cooling speed is faster than the critical cooling rate, resulting in a conversion in the surface microstructure, namely the transformation of austenite to martensite, and internal microstructure change will not happen. After 2 hours cooling, the temperature of crankshaft turns equal to room temperature which is 25°C . Therefore, the residual stress at this time can be used as a journal transient stress after quenching.

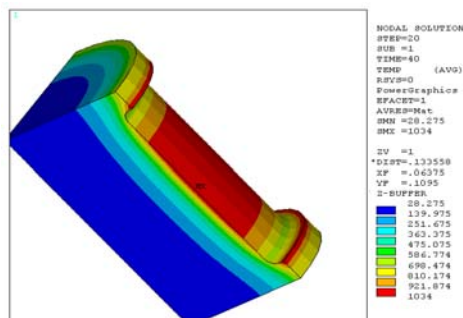


Figure 5. temperature field distribution at the end of induction heating.

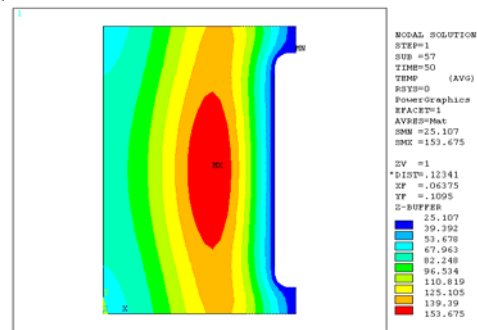


Figure 6. Temperature distribution at the end of coolant injection.

Calculation of Residual Stress. Calculation of residual stress in metal quenching process belongs to thermoelastoplastic problem, and thermoelastoplastic stress-strain relationship is nonlinear. To simplify calculation, it can be segmented calculated using the incremental loading method which deals with each segment as linear part. It can be described as Eq. (1):

$$\begin{cases} \Delta\{\sigma\} = [D](\Delta\{\sigma\} - \Delta\{\varepsilon\}_T) & (\text{In elastic region}) \\ \Delta\{\sigma\} = [D]_{ep}(\Delta\{\sigma\} - \Delta\{\varepsilon\}_T + \Delta\{\sigma\}_T) & (\text{In plastic region}) \end{cases} \quad (1)$$

Here is the analysis of the finite element method. Assuming we arranged n nodes in the region Ω that the Ω will be scattered into n ring discrete units. Taking nodal displacement increments $\Delta\{u\}^e$ as the basic unknown quantity, the unit displacement increment $\Delta\{u\}$ can be described as follows:

$$\Delta\{u\} = [N] \cdot \Delta\{u\}^e = \begin{bmatrix} N_i I & N_j I & \dots & N_k I \end{bmatrix} \begin{bmatrix} \Delta u_{ir} & \Delta u_{iz} & \dots & \Delta u_{kr} & \Delta u_{kz} \end{bmatrix}^T \quad (2)$$

The unit strain increment is:

$$\Delta\{\varepsilon\} = [B] \cdot \Delta\{u\} = \begin{bmatrix} B_i & B_j & \dots & B_k \end{bmatrix} \begin{bmatrix} \Delta u_{ir} & \Delta u_{iz} & \dots & \Delta u_{kr} & \Delta u_{kz} \end{bmatrix}^T \quad (3)$$

where $[B]$ is unit strain matrix, The basic equation of the workpiece is:

$$[K] \Delta\{u\} = \Delta\{R\} \quad (4)$$

Assuming the nodal displacement vector ${}^t\{u\}$ in t moment is known, the nonlinear transient basic balance Eq.s in $t + \Delta t$ can be established as:

$${}^t[K]^{t+\Delta t} \Delta\{u\} = {}^{t+\Delta t}\Delta\{R\} \quad (5)$$

where $[K]$ is the total stiffness matrix; $\Delta\{R\}$ is the incremental load vector. And the nodal displacement vector in moment t is:

$${}^{t+\Delta t}\{u\} = {}^t\{u\} + {}^{t+\Delta t}\Delta\{u\} \quad (6)$$

${}^{t+\Delta t}\Delta\{u\}$ can be obtained by using BFGS iterative method, which could be plug in formula (1) and formula (3) to calculate ${}^{t+\Delta t}\Delta\{\sigma\}$, the stress in moment $t + \Delta t$ is:

$${}^{t+\Delta t}\{\sigma\} = {}^t\{\sigma\} + {}^{t+\Delta t}\Delta\{\sigma\} \quad (7)$$

In the end of solving, the material phase-change has been completed, so the temperature of the workpiece basically reached the ambient temperature state. Therefore, in this case the instantaneous stress of the workpiece can be used as the residual stress of the workpiece. After meshing, the axisymmetric boundary condition is applied. The temperature results in above analysis as the load is applied to the model, the thermoelastoplastic model of the materials is used to solve this finite element problem. The result shows the surface residual stress in the journal reached -616 MPa, and the maximum residual stress on the fillet region reached -320 MPa, which as shown in Fig. 7. The fillet residual stress gradually increased by -320 MPa in compression to 2.58 MPa in tension, which as shown in Fig. 8. Fig. 9 shows the stress distribution in journal straight section, it can be seen that the residual compressive stress in the straight segment is significantly greater than at the corner, and gradually reduced to the internal until converted to tensile stress. Fig. 10 shows the

change of axial residual stress in journal straight section with depth, it described the residual stress trends in the journal which is basically same with the theoretical result.

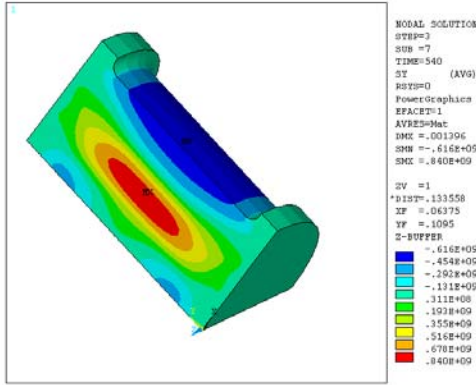


Figure 7. Axial residual stress distribution.

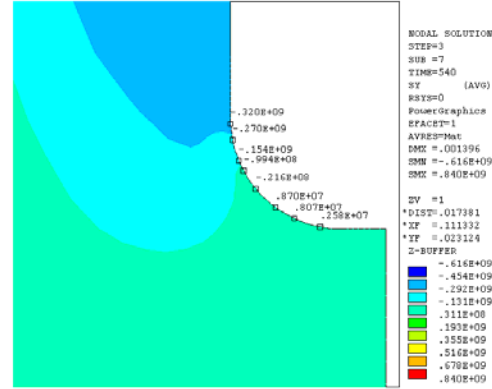


Figure 8. Fillet residual stress distribution.

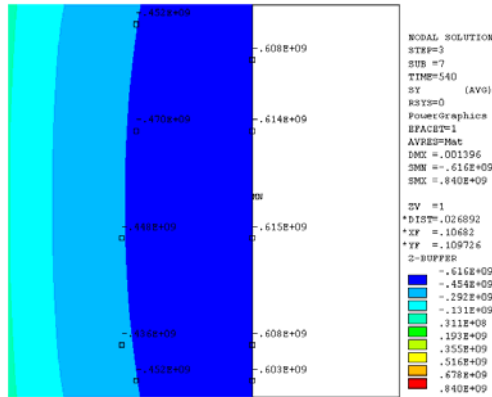


Figure 9. Residual stress distribution of straight sections of the journal.

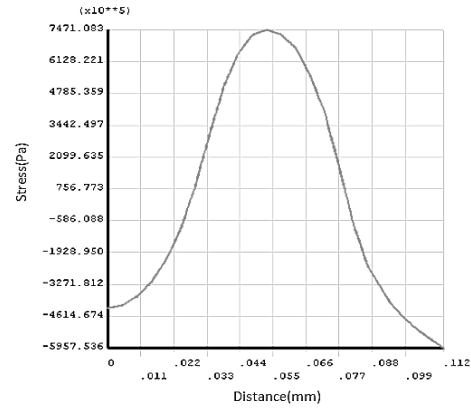


Figure 10. The change of axial residual stress in journal straight section with depth.

Conclusions

- (1) The maximum temperature on the surface of crankshaft is 1034°C , the temperature near the fillet reached 810°C . Since the temperature of the unheated part is close to room temperature, so that microstructure changes will take place within a certain depth under the journal surface.
- (2) After being induction heating quenched on surface, the residual compressive stress values of fillet parts is in the range of $-320\text{MPa} \sim -131\text{MPa}$, and the residual compressive stress values of the straight sections is about in the range of $-616\text{MPa} \sim -454\text{MPa}$. The results show that the surface residual compressive stress can significantly improve the fatigue strength and wear resistance of the crankshaft.
- (3) It is feasible using finite element analysis software to perform the crankshaft induction heating surface hardening process, and this method is provided with certain accuracy and reliability which could offer guidance for adjustment of the quenching process.

Acknowledgement

This research was financially supported by the National Science Foundation (Grant No. 51275428). Also this research was supported by the Fundamental Research Funds for the Central Universities (Grant No. 2682015CX034) and the Open Foundation of Traction Power State Key Laboratory (Grant No. TPL1213).

References

- [1] Z.M. Bulatovic, M.S. Stavljanin, M.V. Tomic. Measurement and analysis of angular velocity variations of twelve-cylinder diesel engine crankshaft. In *Mechanical Systems and Signal Processing*. 25 (2011) 3043-3061.
- [2] J.A. Becerra, F.J. Jimenez, M. Torres. Failure analysis of reciprocating compressor crankshafts. In *Engineering Failure Analysis*. 18 (2011) 735-746.
- [3] A. Ktari, N. Haddar, H.F. Ayedi. Fatigue fracture expertise of engine crankshafts. In *Engineering Failure Analysis*. 18 (2011) 1085-1093.
- [4] Zhi-wei Yu, Xiao-lei Xu. Failure investigation of a truck diesel engine gear consisting of crankshaft. In *Engineering Failure Analysis*. 17 (2010) 537-545.
- [5] Wu-jiao Xu, Kai-qing Wang, Jie Zhou. Experimental and computational failure analysis of TR upset-bending equipment for heavy crankshaft with continuous grain flow. In *Engineering Failure Analysis*. 17 (2010) 546-552.
- [6] X. Zhang, S.-D. Yu. Torsional vibration of crankshaft in an engine – propeller nonlinear dynamical system. In *Journal of Sound and Vibration*. 319 (2010) 491-514.
- [7] Yu-xia Gong. The crankshaft electromagnetic induction hardening of residual stress simulation of fillet stress and its round to research on the effects of stress. Master's thesis, North university of China (2012).
- [8] Yue-hua Zhang, Jin-fei Shen. Study on Numerical Simulation of Temperature Field for the Induction Heating Quenching Process. In *Journal of Huaiyin Institute of Technology*. 03 (2008) pp: 36-40.
- [9] Hua Fang, Zhen Gao, Zhao-cheng Yuan, et al. Numerical Simulation on Medium- Frequency Induction Hardening and 1. Calculation of Residual Stress of Crankshaft. In *Automotive Engineering*. 26 (2004) 354-358.
- [10] Wen-jing Wang, Qiang Li, Long-xiu Miao, et al. Numerical Simulation of Hardening Process in Intermediate Frequency of Combustion Engine Crankshaft. In *Transactions of CSICE*. 20 (2002) 278-282.

Method Research on Comparative Analysis of Structure Reliability and Parametric Sensitivity

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Keywords: First Order Second Moment method, Monte Carlo method, finite element method, reliability, parametric sensitivity

Abstract. With the application of the complex structures and the improvement of structure reliability requirement, the structure reliability becomes more important to ensure the structure safety. A box beam was taken as an object, Advanced First Order Second Moment (AFOSM) method and Monte Carlo method were applied respectively to solve the structure reliability and parametric sensitivity. The results show the way that Monte Carlo method combined with finite element method to solve the structure reliability and parametric sensitivity can meet the engineering accuracy requirements. Monte Carlo simulation calculates the parametric sensitivity by Spearman rank-order correlation coefficient, the interactions between those random variables are considered, the results of Monte Carlo simulation is consistent with the actual condition. Monte Carlo method is feasible to solve the structure reliability and parametric sensitivity of complex structure whose performance function cannot be expressed explicitly.

Introduction

In engineering analysis, the deterministic model is obtained by all kinds of assumptions and ideals. However, material properties, machining tolerances and loads of any actual mechanical product are uncertain, and it's difficult to completely eliminate the uncertainties of these input parameters [1]. In the traditional structure design, these variables are directly given by deterministic data, and safety factor is used to ensure the reliability of structure. But to the structure of higher reliability requirements, the structural redundancy caused by this design method is often unsatisfactory [2]. With the wide use of large complex structures and improving demand for the reliability of structure, how to determine the reliability and sensitivity of the structure accurately and efficiently becomes more and more important. Along with the continuous development of research, First Order Second Moment (FOSM) method is rapidly developed and widely used in practical engineering design [3-5], it has become the basic method of structural reliability analysis and it's also the simplest method to calculate the reliability approximately [6-7]. But FOSM method can't be widely used in the reliability analysis of multiple random variable parameters and the reliability analysis of large complicated structure with complex failure modes. Especially when the performance function of the structure cannot be expressed explicitly, the FOSM method is not available [8]. An effective way to solve this problem is Monte Carlo method developed in the last ten years, the convergence speed of this numerical simulation method is independent of the dimension of the variables and the complexity of the performance function. Monte Carlo method does not need to normalize the performance function or transform the random variable distribution to equivalent normal state, so this method can avoid the mathematical difficulties in reliability analysis and it's widely used in the reliability and parameter sensitivity analysis of complex structures [9-10]. In order to guarantee the feasibility of the Monte Carlo method in the reliability analysis of the large complex structure whose performance function cannot be expressed, FOSM method and Monte Carlo method will be used to analyze the reliability and parameter sensitivity of a box beam to verify the accuracy of Monte Carlo numerical simulation.

AFOSM Method

The basic idea of FOSM method is linearizing the nonlinear performance function by the Taylor series expansion, and then calculating the mean value and standard deviation of the performance function. The early FOSM method selects the linearization point at the mean value, causing that different forms of performance function for the same structure will obtain different reliability, to solve this problem, Advanced First Order Second Moment (AFOSM) method is developed, the AFOSM method selects the linearization point at the failure interface.

Reliability Analysis of Box Beam by AFOSM Method

The geometry size of the box beam is $800 \times 800 \times 5000\text{mm}$, the plate thickness is 6mm , the geometry model is shown in Fig.1. The box beam bears pressure at one end and the other end is constrained. The material strength S , load F and thickness t are defined as random variables, the distribution types and distribution parameters are just as Table1 shows.

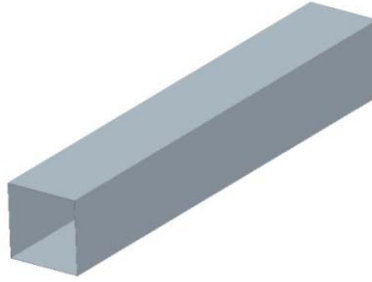


Figure 1. Geometry model of box beam.

Table 1. Distribution type and parameters of random variables.

random variables	distribution type	mean	standard deviation
$S(\text{MPa})$	normal	150	20
$F(\text{N})$	normal	1920000	384000
$t(\text{mm})$	normal	6	0.6

The performance function is

$$Z = G(S, F, t) = S - F / (3200t). \quad (1)$$

At the checking point (S^*, F^*, t^*) , the function is linearized by Taylor series expansion as follows

$$Z \approx G(S^*, F^*, t^*) + (S - S^*) \frac{\partial G}{\partial S} \Big|_{S^*, F^*, t^*} + (F - F^*) \frac{\partial G}{\partial F} \Big|_{S^*, F^*, t^*} + (t - t^*) \frac{\partial G}{\partial t} \Big|_{S^*, F^*, t^*} \quad (2)$$

For (S^*, F^*, t^*) is on the failure interface, so $G(S^*, F^*, t^*) = 0$, the mean value μ_z and standard deviation σ_z of performance function are as Eq. (3) and Eq. (4) show respectively.

$$\mu_z = (\mu_S - S^*) \frac{\partial G}{\partial S} \Big|_{S^*, F^*, t^*} + (\mu_F - F^*) \frac{\partial G}{\partial F} \Big|_{S^*, F^*, t^*} + (\mu_t - t^*) \frac{\partial G}{\partial t} \Big|_{S^*, F^*, t^*} \quad (3)$$

$$\sigma_z = \sqrt{\sigma_S^2 \left(\frac{\partial G}{\partial S} \right)^2 \Big|_{S^*, F^*, t^*} + \sigma_F^2 \left(\frac{\partial G}{\partial F} \right)^2 \Big|_{S^*, F^*, t^*} + \sigma_t^2 \left(\frac{\partial G}{\partial t} \right)^2 \Big|_{S^*, F^*, t^*}} \quad (4)$$

The solving process of Checking point (S^*, F^*, t^*) is just as follows:

$$\text{Making } \alpha_x = \sigma_x \frac{\partial G}{\partial i} \Big|_{x^*} / \sqrt{(\sigma_S \frac{\partial G}{\partial S} \Big|_{S^*, F^*, t^*})^2 + (\sigma_F \frac{\partial G}{\partial F} \Big|_{S^*, F^*, t^*})^2 + (\sigma_t \frac{\partial G}{\partial t} \Big|_{S^*, F^*, t^*})^2} \quad (x = S, F, t) \quad (5)$$

where α_x ($x = S, F, t$) is the relative influence of random variable on the standard deviation, which defined as sensitivity in this paper. Eq. (4) can be transformed to Eq. (6)

$$\sigma_z = \alpha_s \sigma_s \frac{\partial G}{\partial S} \Big|_{S^*, F^*, t^*} + \alpha_F \sigma_F \frac{\partial G}{\partial S} \Big|_{S^*, F^*, t^*} + \alpha_t \sigma_t \frac{\partial G}{\partial t} \Big|_{S^*, F^*, t^*}. \quad (6)$$

$$\text{Reliability index is } \beta = \frac{\mu_z}{\sigma_z} = \frac{(\mu_s - S^*) \frac{\partial G}{\partial S} \Big|_{S^*, F^*, t^*} + (\mu_F - F^*) \frac{\partial G}{\partial F} \Big|_{S^*, F^*, t^*} + (\mu_t - t^*) \frac{\partial G}{\partial t} \Big|_{S^*, F^*, t^*}}{\alpha_s \sigma_s \frac{\partial G}{\partial S} \Big|_{S^*, F^*, t^*} + \alpha_F \sigma_F \frac{\partial G}{\partial S} \Big|_{S^*, F^*, t^*} + \alpha_t \sigma_t \frac{\partial G}{\partial t} \Big|_{S^*, F^*, t^*}}. \quad (7)$$

Eq. (7) can be transformed to Eq. (8)

$$(\mu_s - S^* - \beta \alpha_s \sigma_s) \frac{\partial G}{\partial S} \Big|_{S^*, F^*, t^*} + (\mu_F - F^* - \beta \alpha_F \sigma_F) \frac{\partial G}{\partial F} \Big|_{S^*, F^*, t^*} + (\mu_t - t^* - \beta \alpha_t \sigma_t) \frac{\partial G}{\partial t} \Big|_{S^*, F^*, t^*} = 0 \quad (8)$$

$$\text{So } \mu_x - x^* - \beta \alpha_x \sigma_x = 0 \quad (x = S, F, t). \quad (9)$$

$$\text{The checking point is } x^* = \mu_x - \beta \alpha_x \sigma_x \quad (x = S, F, t). \quad (10)$$

There are three equalities and four unknown variables in Eq. (10), so iterative solution method is used to solve this Eq., the iterative process is referred to literature [3, 5]. After seven times iteration, the difference of (S^*, F^*, t^*) value between the sixth iteration result and the seventh iteration result is smaller than 2%, now it's considered that the iteration is convergent. Then the Checking point is obtained as (130.237, 2327700, 5.585) and the reliability index is 1.607 and the reliability is 0.9459. The calculation process is shown in Table2.

Table 2. Calculation process of AFOSM method.

Variables	First iteration	Second iteration	Third iteration	Fourth iteration	Fifth iteration	Sixth iteration	Seventh iteration
S^*	150	123.333	125.550	128.418	130.471	130.277	130.237
F^*	1920000	2432000	2423000	2374800	2328500	2327900	2327700
t^*	6	5.600	5.467	5.508	5.570	5.584	5.585
α_s	0.667	0.611	0.600	0.606	0.614	0.615	0.615
α_F	-0.667	-0.655	-0.658	-0.660	-0.661	-0.661	-0.661
α_t	0.333	0.444	0.456	0.445	0.432	0.430	0.430
β	2	2	2	1.80	1.612	1.607	1.607

Parameter Sensitivity Analysis of Box Beam by AFOSM Method

The gradient of the performance function at the checking point is defined as sensitivity, and the calculation of sensitivity is just as Eq. (5), sensitivity approximately reflects the change of failure when this parameter has some disturbance at the point where is most likely to occur failure. Table 2 shows that the sensitivity of strength, load and plate thickness is 0.615, -0.661 and 0.430 respectively. Positive sensitivity value expresses that the performance function value increases with the increase of the input variables, and structure tends to be safe. Negative sensitivity value expresses that the performance function value decreases with the increase of the input variables.

Monte Carlo Method

Based on probability theory and mathematical statistics theory, Monte Carlo method is a stochastic simulation method, and is widely used to deal with the problem of randomness.

Reliability Analysis by Monte Carlo Method

The principle of the Monte Carlo method is performing N times random sampling of all basic variables according to statistical characteristics, and the sampling values are used as the inputs of

the deterministic model for the finite element analysis, the response characteristics of the N times simulation results can be obtained after N times finite element calculation. Determining whether the structure is reliable or not through the performance function and the total times of structure reliability is recorded as n . Law of large number expresses that when sample number is large enough, the event occurring frequency converges to the occurring probability, so the reliability can be expressed as $R \approx n/N$. The accuracy of reliability depends on the size of the sample N , the result becomes more accurate with the increment of the sampling times. Latin Hypercube Sampling (LHS) is used in this paper, LHS can avoid the duplication of sampling.

Sensitivity Analysis by Monte Carlo Method

Monte Carlo method combined with ANSYS calculates the parametric sensitivity by Spearman rank-order correlation coefficient, Spearman rank-order correlation coefficient test is an important method to analyze the correlation between variables. According to N times random sampling and finite element analysis, input variables sample value $t_1, t_2, \dots, t_N, F_1, F_2, \dots, F_N$ and N time stress response $\sigma_1, \sigma_2, \dots, \sigma_N$ can be obtained. The rank-order of t_i in $t_1, t_2, \dots, t_N$ is defined as R_i , the rank-order of σ_i in $\sigma_1, \sigma_2, \dots, \sigma_N$ is defined as Q_i , the Spearman rank correlation coefficient between thickness and stress is as follows.

$$r = \frac{\sum_{i=1}^N (R_i - \bar{R})(Q_i - \bar{Q})}{\sqrt{\sum_{i=1}^N (R_i - \bar{R})^2} \sqrt{\sum_{i=1}^N (Q_i - \bar{Q})^2}} \quad (11)$$

where $\bar{R} = \sum_{i=1}^N R_i / N = (N+1)/2$, $\bar{Q} = \sum_{i=1}^N Q_i / N = (N+1)/2$.

Spearman rank correlation coefficient is in the range of $[-1,1]$. When the two variables are monotonically increasing, the correlation coefficient is positive, when the two variables are monotonically decreasing, the correlation coefficient is negative.

Analysis Result of Monte Carlo Method

The box beam above is taken as an object, and it's meshed by shell element, the finite element model is shown in Fig.2. The distribution types and distribution parameters are shown in Table1. Through Probability Design System of ANSYS, the parametric finite element model of the box beam is established. The performance function is determined as $Z = S - \sigma$, and LHS method is selected, the number of sampling times is determined as 2000, and then random finite element analysis is made and finally the reliability and parameter sensitivity of box beam is obtained: the reliability is 0.9458, the sensitivity of strength, load and plate thickness are 0.641, -0.649 and 0.326 respectively, just as Fig.3 shows.

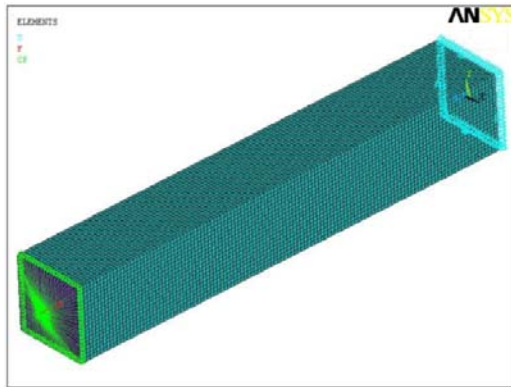


Figure 2. Finite element model of box beam.

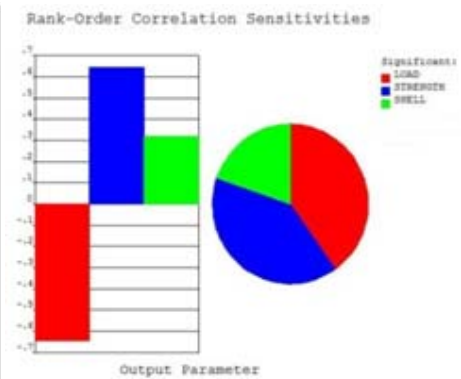


Figure 3. Parametric sensitivity chart.

Comparison Analysis of AFOSM Method and Monte Carlo Method

Analytical solution of AFOSM is selected as standard, comparison analysis between Monte Carlo method and analytic method is made. The reliability obtained by AFOSM method and Monte Carlo method are 0.9459 and 0.9458 respectively, the relative error is 0.01%, which can meet the engineering accuracy requirements, and calculation process shows that with the increase of the simulation times of the Monte Carlo method, the relative error decreases gradually. The parameters sensitivity obtained by those two methods is slightly different, but the overall trend is coincident, that is the load sensitivity is the biggest, strength sensitivity is the second and the thickness sensitivity is the lowest, as Table3 shows. When using AFOSM to analyze sensitivity, the basic idea is analyzing the change of response values while changing one of the variables and the other variables remaining unchanged, now sensitivity reflects the gradient of the performance function, and the relativity between input variables is ignored. When using Monte Carlo method, all input variables change at the same time and the response is obtained by finite element analysis, finally the parametric sensitivity is calculated by Spearman rank-order correlation coefficient, the relativity between input variables are taken into account, the sensitivity result is more consistent with the actual condition.

Table 3. Results comparison of AFOSM method and Monte Carlo method.

Reliability analysis method	Reliability	Parameter sensitivity		
		Strength	Load	Thickness
AFOSM method	0.9459	0.615	-0.661	0.430
Monte Carlo method	0.9458	0.641	-0.649	0.326

Conclusions

The FOSM method and Monte Carlo method were introduced in this paper. Then a box beam taken as an object, AFOSM method and Monte Carlo method were applied respectively to solve the reliability and parametric sensitivity, the following conclusions are obtained:

(1) The reliability and sensitivity results of those two methods are coincident and Monte Carlo method combined with finite element method to solve the structure reliability and parametric sensitivity can meet the engineering accuracy requirements. Monte Carlo method is viable to solve the structure reliability and parametric sensitivity for complex structure whose performance function cannot be expressed explicitly.

(2) Through Monte Carlo simulation method to calculate the parametric sensitivity by Spearman rank-order correlation coefficient, the interactions between those random variables are considered and the result of Monte Carlo simulation is consistent with the actual condition.

(3) In the design process of complex structure, it is significant to use Monte Carlo method to analyze parameter sensitivity so as to carry out structure optimization and reduce redundancy design.

Acknowledgement

This research was financially supported by the National Science Foundation (Grant No. 51275428) and the Open Foundation of Traction Power State Key Laboratory (Grant No. TPL1213).

References

[1] S. Reh, J.D. Beley, S. Mukherjee, et al, Probabilistic Finite Element Analysis using ANSYS, Structural Safety. 28 (2006) 17-43.

- [2] Zhou Yuan, Research on Reliability of Bogie Frame in High Speed Train based on ANSYS, Southwest jiaotong university, Chengdu, 2009.
- [3] Wu Xue-li, Jiang Jin, Zhang Le-min, et al. Application of the Improve First Order Second Moment Method (FOSM) in Tunnel Reliability, Industrial Construction. 38 (2008) 694-697.
- [4] Jin Xi. First-order Second moment Approximation Method for Solving Probabilistic Hydraulic Model of Water Supply Networks, China water & wastewater. 26 (2010) 78-83.
- [5] Zeng Zhao-hui, Jiao Guo-tai, Guo Yun-jian. A study of the circular arc gear reliability based on improved FOSM method, 36 (2009) 22-23.
- [6] Yang Jie. Research on Structure Reliability Calculation Method and Sensitivity Analysis. Dalian University of Technology, Dalian, 2012.
- [7] Hu Ming. Research of Structure Reliability Analysis Method, South China University of Technology, Guangzhou, 2012.
- [8] Lu Yao-hui. Study on fatigue reliability of welded bogie frame for railway vehicle. Southwest jiaotong university, Chengdu, 2011.
- [9] Lu Hao. Fatigue Reliability Sensitivity Analysis and Reliability Robust Design of High Speed Train Wheel. Northeastern University, Shenyang, 2009.
- [10] Lu Yao-hui, Zeng Jing, Wu Ping-bo, et al. Parametric Sensitivity Analysis for the Bogie Frame Reliability of Railway Vehicle, China Railway Science. 31 (2010) 111-115.

Heading Machine Rotary Hydraulic Cylinder Design and Its Hydraulic System Control Strategy

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Keywords: Road header, Stress analysis, Mathematical model, Simulation.

Abstract. For longitudinal excavators rotary cylinder stress analysis to calculate the rotary cylinder dimensions, analyzes the shortcomings of existing roadheader slewing hydraulic system, electro-hydraulic servo control system is proposed rotary cylinder constructed the hydraulic system mathematical model Model, simulation analysis using Matlab/Simulink software. Join PID controller were analyzed further to achieve the system optimization.

Introduction

Vertical axis type machine is the main production in coal mine of heavy machinery, digging into the machine lift and rotary hydraulic cylinder force is the largest, but now the machine rotary hydraulic cylinder mainly relying on experience in design, there is no specific formula design, this article take EBZ260 calculate its stress, select hydraulic cylinder type, Rotary hydraulic system mainly realize the cutting head direction and speed of change, to achieve automatic control hydraulic system has significant meaning for the whole machine work.

Rotary Hydraulic Cylinder Force Analysis

Roadheader slewing hydraulic cylinder is driven by a rotary turret rotation Move, rotary table and roadheader unit connection.

(1) The vertical axis type machine rotary part as shown in Fig.1 (1. Cutting head; 2. Cantilever; 3, 4. Lifting hydraulic cylinder rotary table; 5. Rotary hydraulic cylinder).

(2) Machine on the surface of the rotary cutting mechanism in horizontal stress is shown in Fig.2.

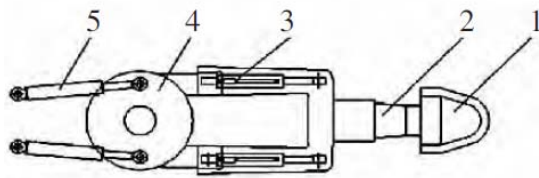


Figure 1. Cutting slewing mechanism diagram.

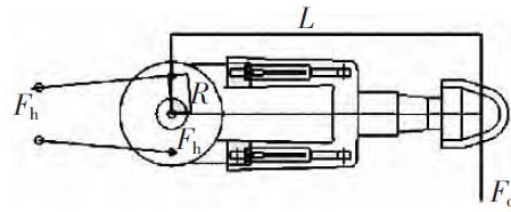


Figure 2. Horizontal cutting slewing mechanism.

(3) Machine on the surface of the rotary cutting mechanism in horizontal stress is shown in Fig. 2.

$$2F_h R = F_q L \quad (1)$$

$$F_h = F_q L / (2R) \quad (2)$$

$$F_q = (1.2 - 1.5) F_j \quad (3)$$

$$F_j = 9550 P_j / (n_j D_j) \quad (4)$$

F_q - Cutting head level traction,

F_j - Level cutting force,

N_j - Cutting power,

P_j - Cutting head rotation speed,

D_j - Cutting head average diameter.

Take EBZ260 generation into the corresponding numerical calculate the $F_h = 264800$ n, EBZ260

slewing hydraulic system pressure for $p = 25$ MPA. Take the load pressure force $p / 3 p_L = 2 = 16.67$ MPA. The relationship between the piston diameter d and piston rod diameter D is $d = 0.6 D$.

To make the effective area of $A = \pi (D_2^2 - D_1^2) / 4$,

$$p_L A = F_h \quad (5)$$

to calculate $D = 177.8$ mm, $D = 106.7$ mm.

Check manual roundness, $D = 180$ mm, $D = 110$ mm. Hydraulic cylinder range according to the pendulum Angle and the overall structure size is 880 mm.

Control Strategies

Cantilever machine work by cutting head rotation and cantilever oscillation, the implementation of cantilever oscillation controlled by hydraulic system, main control cantilever swing speed and direction. Traditional workers on-site control and meet the needs of the automatic extraction, and molding in the face of high demand for workers, the electro-hydraulic servo control can achieve automatic control, has high guiding significance to guide production.

(1) The basic equation for the four-way valve control hydraulic cylinder

With rotary hydraulic cylinder contraction namely $x_v > 0$ analysis under the condition of the mathematical model

$$q_L = A_p \frac{dx_p}{dt} + C_{tp} p_L + \frac{V_t}{4\beta_e} \frac{dp_L}{dt} \quad (6)$$

q_L — Load flow rate

A_p —The hydraulic cylinder piston area effectively

x_p —Piston displacement

C_{tp} —Total leakage coefficient of hydraulic cylinder

p_L —Loading pressure

V_t —Equivalents surface volume

β_e —Effective bulk modulus

(2) The hydraulic cylinder and the force balance equation of load

Load generally includes inertial force, viscous damping force, elastic force and load of accident.

$$A_p p_L = m_t \frac{d^2 x_p}{dt^2} + B_p \frac{dx_p}{dt} + K x_p + F_L \quad (7)$$

M_t - Piston machine total load conversion on the piston quality,

B_p - The viscous damping coefficient of piston machine load,

K - Load spring stiffness,

F_L - Any accidental load effect on the piston.

(3) The four-way valve controlled hydraulic cylinder transfer function

$$X_p = \frac{\frac{K_p}{A_p} - \frac{K_{ce}}{A_p^2} (1 + \frac{V_t}{4\beta_e K_{ce}} s) F_L}{\frac{m_t V_t}{4\beta_e A_p^2} s^3 + (\frac{m_t K_{ce}}{A_p^2} + \frac{B_p V_t}{4\beta_e A_p^2}) s^2 + (1 + \frac{B_p K_{ce}}{A_p^2} + \frac{K V_t}{4\beta_e A_p^2}) s + \frac{K K_{ce}}{A_p^2}} \quad (8)$$

A bit small, equation was simplified

$$x_p = \frac{\frac{k_q}{A_1} x_v - \frac{k_{ce}}{A_1} (1 + \frac{V_t}{4k_{ce}\beta_e} s) F_L}{s (\frac{s^2}{\omega_h^2} + \frac{2\zeta_h}{\omega_h} s + 1)} \quad (9)$$

(4) A mathematical model servo valve

1) Mathematical model of the servo amplifier

$$K_a = \frac{\Delta i}{U} \quad (10)$$

K_a - The servo amplifier gain
 Δi - Input current servo amplifier
 U - Servo amplifier output voltage.

2) Mathematical model for servo valve

$$G_s(s) = \frac{Q}{\Delta i} = \frac{K_s}{\frac{s^2}{\omega_s^2} + 2\frac{\zeta_s}{\omega_s}s + 1} \quad (11)$$

Because in the hydraulic servo valve frequency lenient inherent frequency, so the servo valve can be seen as 1 order inertial link

$$G_s(s) = \frac{Q}{\Delta i} = \frac{K_s}{T_s s + 1} \quad (12)$$

Q - Servo valve light flow,
 K_s - The servo valve gain,
 ζ_s - The damping coefficient of the servo valve,
 T_s - Time constant of the servo valve.

3) Mathematical model displacement sensor

$$K_f = \frac{U}{x_p} \quad (13)$$

K_f - The displacement sensor gain,
 U - The sensor output voltage,
 x_p - Rotary hydraulic cylinder displacement.

Through the above mathematical model of the structure, it is concluded that valve control hydraulic cylinder block diagram as shown in Fig. 3.

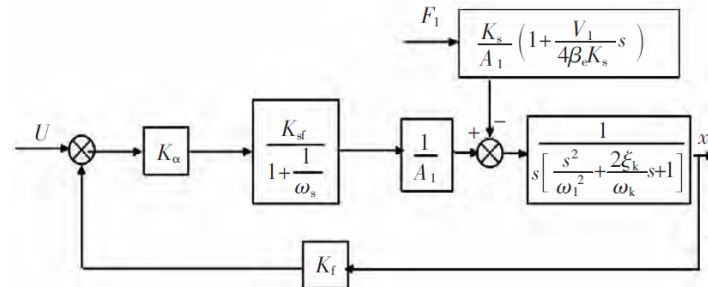


Figure 3. Control system block diagram.

Matlab/Simulink

Build the Simulink model, for example by EBZ260 entered step signal, the simulation time is set to 2s, system simulation diagram as shown in Fig.4, the simulation result is shown in Fig. 5.

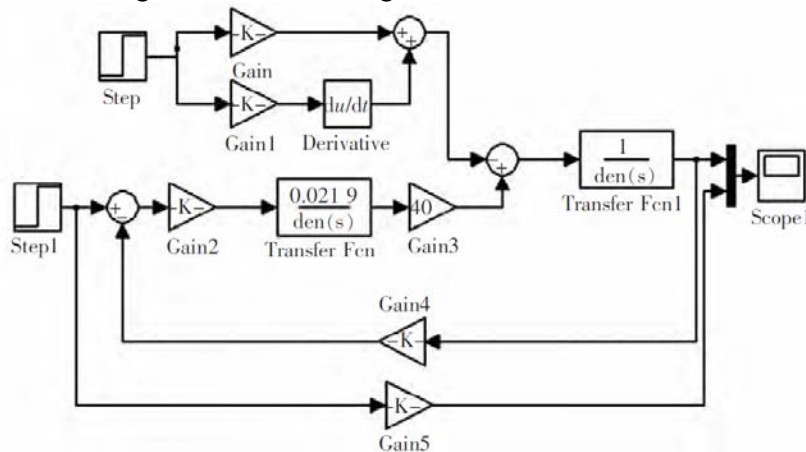


Figure 4. The simulink model.

It can be seen from the simulation results in not join controller system response before the result is bad, after joining PID controller, the simulation results as shown in Fig. 6. It can be seen clearly in the PID controller and its response is stable and fast.

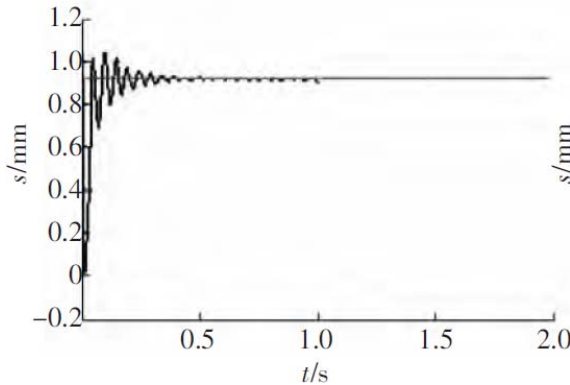


Figure 5. Simulink simulation results.

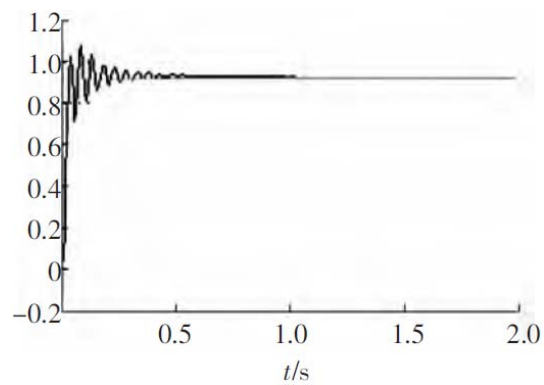


Figure 6. Simulation results after joining PID.

Conclusions

In this paper, the hydraulic system adopts servo valve controlled asymmetrical hydraulic pressure cylinder system, established the mathematical model of the use of space, greatly lift the quickness, accuracy and stability of the system response. After joining PID controller system has been further optimization, well meet the requirements of field use.

References

- [1]Zheng Chun. Cantilever machine hydraulic oil cylinder force analysis and computer aided design [J]. Journal of coal mine machinery, 1997, 18 (6): 12-14.
- [2]Xiang Yang. Cantilever machine cutting force and cutting force head [J]. The matching relation between coal mining machinery, 2003 (3): 5-7, 10.
- [3] Quan-zhong Ren. EBH - 132 type cantilever machine cutting head stability analysis [J]. Journal of coal mine machinery, 2008, 29 (1): 63-63.
- [4] Xiao-dong Tong. Horizontal shaft type tunneling [D]. Mechanical and electrical hydraulic control simulation of Huainan, Anhui University of Science and Technology, 2007.
- [5] Chuan-li Wang, Ding Fan, Qi-peng Li, etc. The symmetric four-way valve controlled asymmetric hydraulic cylinder servo system dynamic characteristic study [J]. China mechanical engineering, 2004, 15 (6): 471-471.
- [6] Song Long-tang. Limit mandrel rolling mill hydraulic servo control system for learning modeling study [J]. Journal of machine tools and hydraulic, 2007 (12): 104-106.

Stability Analysis of a Predator-prey Model with Impulsive Control

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Keywords: Predator-prey model, Stage structure, Impulsive control, Global attractive, Permanence.

Abstract. In this paper, we propose a stage-structured predator-prey model with impulsive stocking prey and continuous harvesting predator. Using the theories and methods of impulsive delayed differential equations, we analyze the global attractivity of the predator-extinction periodic solution, and obtain sufficient conditions for the permanence of the system.

Introduction

In recent years, Lotka-Volterra predator-prey models have been extensively investigated and many excellent results have been obtained [1-3]. But all of these works ignore the stage structure of species. However, in natural world, any species has a development process, some population is divided into two stages, that is, from childhood to adulthood, some species are divided into three stages, the youth, adulthood and agedness. As is common, the dynamics-eating habits, susceptibility to preys, etc. are often quite different in these sub-populations. Hence, it is of ecological importance to investigate the effects of such a subdivision on the interaction of species [4-7].

Since biological resources can reproduce themselves in a natural way, it is regarded as the most important part of the natural resources. How to get maximal profit and protect the sustainable development of biological system have been research problems. The results showed that harvesting and stocking prey or predator can influence ecological system, but overusing these methods sometimes can destroy the ecological balance of renewable resources.

In this paper, we study the following stage-structured predator-prey model with impulsive stocking prey and continuous harvesting predator

$$\left\{ \begin{array}{l} \dot{x}(t) = r_1 x(t) - ax^2(t) - k_1 x(t) y_2(t), \\ \dot{y}_1(t) = r_2 y_2(t) - d_1 y_1(t) - r_2 e^{-d_1 \tau} y_2(t - \tau), \\ \dot{y}_2(t) = k_2 x(t) y_2(t) - d_2 y_2(t) + r_2 e^{-d_1 \tau} y_2(t - \tau) - \nu y_2(t), \end{array} \right\} \quad t \neq nT, \quad n = 1, 2, 3, \dots \quad (1)$$

$$\left\{ \begin{array}{l} x(t^+) = x(t) + \mu, \\ y_1(t^+) = y_1(t), \\ y_2(t^+) = y_2(t), \end{array} \right\} \quad t = nT,$$

where $x(t)$ represents the density of the prey at time t ; $y_1(t)$ and $y_2(t)$ represent the densities of the immature and mature predator at time t , respectively; r_1 is the growth rate of the prey population; a is the self-regulation constant of the prey; r_2 is the birth rate of the predator population; d_1 and d_2 are the death rates of the immature and mature predator, respectively. Suppose that only the mature predator has the hunting capacity and its functional response functions is $k_1 x(t)$, k_1 is the capturing rate of the mature predator, k_2/k_1 is the rate of conversing prey into new mature predator; τ is the maturity of the predator; ν is the harvesting rate on the mature predator; μ is the stocking amount of the prey; T is the period of the impulsive stocking of the prey. All the parameters mentioned above are positive constants. The initial conditions for system (1) take the form

$$(\varphi_1(\theta), \varphi_2(\theta), \varphi_3(\theta)) \in C_+ = C([- \tau, 0], R_+^3), \quad \varphi_i(0) > 0, i = 1, 2, 3. \quad (2)$$

$$\varphi_2(0) = \int_{-\tau}^0 r_2 e^{d_1 s} \varphi_3(s) ds. \quad (3)$$

By the fundamental theory of functional differential equations [8], we know that system (1) has a unique solution $(x(t), y_1(t), y_2(t))$ satisfying initial conditions (2). Further, it is easy to show that all solutions of (1) with initial conditions (2) are defined on $[0, +\infty)$ and remain positive for all $t \geq 0$.

Lemma 1 All positive solutions of system (1) satisfying initial conditions (2) are ultimately bounded if $d_2 + v > \gamma_2$ holds, that is, there exist a constant $M > 0$ and a T such that $x(t) \leq M/k_2$, $y_1(t) \leq M/k_1$, $y_2(t) \leq M/k_1$ for $t > T$.

In the following, we suppose that $d_2 + v > \gamma_2$.

Lemma 2 The impulsive differential system

$$\begin{cases} \dot{v}(t) = v(t)(a - bv(t)), & t \neq nT, \\ v(t^+) = v(t) + \mu, & t = nT, \end{cases} \quad (4)$$

has a global asymptotic stable positive periodic solution

$$\tilde{v}(t) = \frac{av^* e^{a(t-nT)}}{bv^* e^{a(t-nT)} + a - bv^*}, \quad t \in (nT, (n+1)T], \quad n \in Z_+,$$

where $a, b, \mu > 0$ and $v^* = \frac{a + b\mu + \sqrt{(a + b\mu)^2 + 4ab\mu(e^{aT} - 1)}}{2b} > \frac{a}{b}$.

Existence and Global Attractivity of the Predator-Eradication Periodic Solution

Because the first and third equations of (1) do not contain $y_1(t)$, we can simplify model (1) and restrict our attention to the following model

$$\begin{cases} \dot{x}(t) = r_1 x(t) - ax^2(t) - k_1 x(t)y_2(t), \\ \dot{y}_2(t) = k_2 x(t)y_2(t) - d_2 y_2(t) + r_2 e^{-d_1 \tau} y_2(t - \tau) - v y_2(t), \end{cases} \quad t \neq nT, \\ \begin{cases} x(t^+) = x(t) + \mu, \\ y_2(t^+) = y_2(t), \end{cases} \quad t = nT, \end{cases} \quad (5)$$

the initial conditions for system (5) are

$$(\varphi_1(\theta), \varphi_3(\theta)) \in C_+ = C([- \tau, 0], R_+^2), \quad \varphi_i(0) > 0, i = 1, 3. \quad (6)$$

Theorem 1 System (5) has a mature predator-eradication periodic solution $(\tilde{x}(t), 0)$, where

$$\tilde{x}(t) = \frac{r_1 x^* e^{r_1(t-nT)}}{ax^* e^{r_1(t-nT)} + r_1 - ax^*}, \quad t \in (nT, (n+1)T],$$

here $x^* = \frac{r_1 + a\mu + \sqrt{(r_1 + a\mu)^2 + 4r_1 a\mu(e^{r_1 T} - 1)}}{2a} > \frac{r_1}{a}$.

Proof If $y_2(t) \equiv 0$, we have the following subsystem of (5)

$$\begin{cases} \dot{x}(t) = x(t)(r_1 - ax(t)), & t \neq nT, \\ x(t^+) = x(t) + \mu, & t = nT. \end{cases}$$

From Lemma 2, we know that the system has a global asymptotic stable positive periodic solution

$$\tilde{x}(t) = \frac{r_1 x^* e^{r_1(t-nT)}}{ax^* e^{r_1(t-nT)} + r_1 - ax^*}, \quad t \in (nT, (n+1)T],$$

here $x^* = \frac{r_1 + a\mu + \sqrt{(r_1 + a\mu)^2 + 4r_1 a\mu(e^{r_1 T} - 1)}}{2a} > \frac{r_1}{a}$, therefore system (5) has a mature predator-eradication periodic solution $(\tilde{x}(t), 0)$. This completes the proof.

Theorem 2 Let $(x(t), y_2(t))$ be any solution of system (5). If $v > r_2 e^{-d_1 \tau} + k_2 x^* e^{r_1 T} - d_2$ holds, then the mature predator-eradication periodic solution $(\tilde{x}(t), 0)$ of (5) is global attractive.

Proof If $v > r_2 e^{-d_1 \tau} + k_2 x^* e^{r_1 T} - d_2$, we may choose a sufficiently small $\varepsilon > 0$ such that

$$v > r_2 e^{-d_1 \tau} + k_2 x^* e^{r_1 T} - d_2 + k_2 \varepsilon. \quad (7)$$

With the first equation of (5), we know $\dot{x}(t) \leq r_1 x(t) - ax^2(t)$. Consider the following impulsive differential equation

$$\begin{cases} \dot{u}(t) = r_1 u(t) - au^2(t), & t \neq nT, \\ u(t^+) = u(t) + \mu, & t = nT, \\ u(0^+) = x(0^+). \end{cases}$$

From Lemma 2, we know that the system has a global asymptotic stable positive periodic solution

$$\tilde{u}(t) = \frac{r_1 u^* e^{r_1(t-nT)}}{au^* e^{r_1(t-nT)} + r_1 - au^*}, \quad t \in (nT, (n+1)T],$$

here $u^* = \frac{r_1 + a\mu + \sqrt{(r_1 + a\mu)^2 + 4r_1 a\mu(e^{r_1 T} - 1)}}{2a}$. By the comparison theorem of impulsive equation, we get that there exists a $k_1 \in \mathbb{Z}_+$ such that $x(t) \leq u(t) < \tilde{u}(t) + \varepsilon = \tilde{x}(t) + \varepsilon$, $nT < t \leq (n+1)T$, $n > k_1$, and then $x(t) < \tilde{x}(t) + \varepsilon \leq x^* e^{r_1 T} + \varepsilon$. With the second equation of (5), we know that for $t > k_1 T + \tau$

$$\dot{y}_2(t) \leq r_2 e^{-d_1 \tau} y_2(t - \tau) - [d_2 + v - k_2(x^* e^{r_1 T} + \varepsilon)]y_2(t).$$

Consider the following comparison equation

$$\dot{w}(t) = r_2 e^{-d_1 \tau} w(t - \tau) - [d_2 + v - k_2(x^* e^{r_1 T} + \varepsilon)]w(t).$$

With the condition (7) and Lemma 2.1 in [5] we know that $\lim_{t \rightarrow +\infty} w(t) = 0$. By the comparison theorem we get $\lim_{t \rightarrow +\infty} y_2(t) = 0$, Therefore for sufficiently small $\varepsilon_1 > 0$, there exists a $k_2 \in \mathbb{Z}_+$ ($k_2 > k_1$) such that $y_2(t) < \varepsilon_1$ for all $t > k_2 T + \tau$.

With the first equation of (5), we know that $\dot{x}(t) \geq (r_1 - \varepsilon_1 k_1)x(t) - ax^2(t)$. Similar to the previous, we get that there exists a $k_3 \in \mathbb{Z}_+$ ($k_3 > k_2$) such that $x(t) \geq z(t) > \tilde{z}(t) - \varepsilon$ for all $t > k_3 T$, where

$$\tilde{z}(t) = \frac{(r_1 - \varepsilon_1 k_1)z^* e^{(r_1 - \varepsilon_1 k_1)(t - nT)}}{az^* e^{(r_1 - \varepsilon_1 k_1)(t - nT)} + r_1 - \varepsilon_1 k_1 - az^*}, \quad t \in (nT, (n+1)T],$$

$$\text{here } z^* = \frac{r_1 - \varepsilon_1 k_1 + a\mu + \sqrt{(r_1 - \varepsilon_1 k_1 + a\mu)^2 + 4a\mu(r_1 - \varepsilon_1 k_1)(e^{(r_1 - \varepsilon_1 k_1)T} - 1)}}{2a}.$$

Let $\varepsilon_1 \rightarrow 0$, then we have $x(t) \geq \tilde{x}(t) - \varepsilon$, which implies that $\lim_{t \rightarrow +\infty} x(t) = \tilde{x}(t)$, that is, the mature predator-eradication periodic solution $(\tilde{x}(t), 0)$ of (5) is global attractive. This completes the proof.

Permanence

Theorem 3 If $v < \frac{r_1 k_2 x^*}{ax^* e^{r_1 T} + r_1 - ax^*} + r_2 e^{-d_1 \tau} - d_2$ holds, then system (5) is permanent.

Proof The second equation of (5) can be rewritten as

$$\dot{y}_2(t) = [r_2 e^{-d_1 \tau} + k_2 x(t) - d_2 - v]y_2(t) - r_2 e^{-d_1 \tau} \frac{d}{dt} \int_{t-\tau}^t y_2(u) du.$$

Define $V(t) = y_2(t) + r_2 e^{-d_1 \tau} \int_{t-\tau}^t y_2(u) du$. We calculate the derivative of $V(t)$ along the solution of (5)

$$\frac{dV(t)}{dt} = [r_2 e^{-d_1 \tau} + k_2 x_2(t) - d_2 - v]y_2(t).$$

Since $v < \frac{r_1 k_2 x^*}{ax^* e^{r_1 T} + r_1 - ax^*} + r_2 e^{-d_1 \tau} - d_2$, we choose $0 < m^* < \frac{r_1}{k_1}$ and sufficiently small $\varepsilon > 0$ such that

$$v < \frac{k_2(r_1 - k_1 m^*)W^*}{aW^* e^{(r_1 - k_1 m^*)T} + r_1 - k_1 m^* - aW^*} + r_2 e^{-d_1 \tau} - d_2 - k_2 \varepsilon,$$

$$\text{here } W^* = \frac{r_1 - k_1 m^* + a\mu + \sqrt{(r_1 - k_1 m^* + a\mu)^2 + 4a\mu(r_1 - k_1 m^*)(e^{(r_1 - k_1 m^*)T} - 1)}}{2a}.$$

We claim that for any $t_0 > 0$, it is impossible that $y_2(t) < m^*$ for all $t > t_0$. Otherwise there exists a $t_0 > 0$ such that $y_2(t) < m^*$ for all $t > t_0$. It follows from the first equation of (5) that $\dot{x}(t) \geq (r_1 - k_1 m^*)x(t) - ax^2(t)$. Consider the following impulsive differential equation

$$\begin{cases} \dot{W}(t) = (r_1 - k_1 m^*)W(t) - aW^2(t), & t \neq nT, \\ W(t^+) = W(t) + \mu, & t = nT, \\ W(0^+) = x(0^+). \end{cases}$$

From Lemma 2, we know that the system has a global asymptotic stable positive periodic solution

$$\tilde{W}(t) = \frac{(r_1 - k_1 m^*)W^* e^{(r_1 - k_1 m^*)(t - nT)}}{aW^* e^{(r_1 - k_1 m^*)(t - nT)} + r_1 - k_1 m^* - aW^*}, \quad t \in (nT, (n+1)T],$$

here $W^* = \frac{r_1 - k_1 m^* + a\mu + \sqrt{(r_1 - k_1 m^* + a\mu)^2 + 4a\mu(r_1 - k_1 m^*)(e^{(r_1 - k_1 m^*)T} - 1)}}{2a}$. Using the comparison theorem of impulsive equation, we know that there exists a $t_1 > t_0$ such that $x(t) > \tilde{W}(t) - \varepsilon$ for all $t \geq t_1$, then

$$x(t) > \frac{(r_1 - k_1 m^*)W^*}{aW^* e^{(r_1 - k_1 m^*)T} + r_1 - k_1 m^* - aW^*} - \varepsilon =: \eta.$$

Therefore

$$\frac{dV(t)}{dt} = (r_2 e^{-d_1 \tau} + k_2 \eta - d_2 - \nu)y_2(t), \quad t \geq t_1. \quad (8)$$

Let $y_2^l = \min_{t \in [t_1, t_1 + \tau]} y_2(t) > 0$. We will show that $y_2(t) \geq y_2^l$ for $t \geq t_1$. Otherwise there exists a $T_0 > 0$ such that $y_2(t) \geq y_2^l$, $y_2(t_1 + \tau + T_0) = y_2^l$ and $\dot{y}_2(t_1 + \tau + T_0) \leq 0$ for $t_1 \leq t \leq t_1 + \tau + T_0$. With the second equation of (5), we know

$$\begin{aligned} \dot{y}_2(t_1 + \tau + T_0) &= k_2 x(t_1 + \tau + T_0) y_2(t_1 + \tau + T_0) - (d_2 + \nu) y_2(t_1 + \tau + T_0) + r_2 e^{-d_1 \tau} y_2(t_1 + T_0) \\ &\geq (k_2 \eta - d_2 - \nu + r_2 e^{-d_1 \tau}) y_2^l > 0. \end{aligned}$$

This is a contradiction. Thus $y_2(t) \geq y_2^l$ for $t \geq t_1$. Therefore

$$\frac{dV(t)}{dt} \geq (r_2 e^{-d_1 \tau} + k_2 \eta - d_2 - \nu) y_2^l, \quad t \geq t_1,$$

which implies that $V(t) \rightarrow +\infty$ as $t \rightarrow +\infty$. This is a contradiction to $V(t) \leq \frac{M}{k_1} (1 + r_2 \tau e^{-d_1 \tau})$. Hence for any $t_0 > 0$, the inequality $y_2(t) < m^*$ can not hold for $t > t_0$. So there exist the following two possibilities.

Case 1. If $y_2(t) \geq m^*$ holds for all t large enough, then our aim is obtained.

Case 2. Assume that $y_2(t)$ is oscillatory about m^* for all t large enough.

Let $m_2 = \min \left\{ \frac{m^*}{2}, m^* e^{-(d_2 + \nu)\tau} \right\}$. We will prove that $y_2(t) \geq m_2$ for all t large enough. Suppose that

there exist two positive constants $\bar{t}$ and ψ such that $y_2(\bar{t}) = y_2(\bar{t} + \psi) = m^*$ and $y_2(t) < m^*$ for $\bar{t} < t < \bar{t} + \psi$, where $\bar{t}$ is sufficiently large such that $x(t) > \eta$. The positive solutions of (5) are ultimately bounded and $y_2(t)$ is not effected by impulses. Therefore $y_2(t)$ is uniformly continuous.

Hence there exists a T_1 ($0 < T_1 < \tau$ and T_1 is independent of the choice of $\bar{t}$) such that $y_2(t) > \frac{m^*}{2}$ for $\bar{t} < t < \bar{t} + T_1$. If $\psi \leq T_1$, our aim is met. If $T_1 < \psi < \tau$, because of $\dot{y}_2(t) \geq -(d_2 + \nu)y_2(t)$, $y_2(\bar{t}) = m^*$, then $y_2(t) \geq m^* e^{-(d_2 + \nu)\tau}$ for $\bar{t} < t < \bar{t} + \psi$. Clearly $y_2(t) \geq m^* e^{-(d_2 + \nu)\tau}$ for $\bar{t} < t < \bar{t} + \psi$ if $\psi \geq \tau$. Similarly, we can prove that $y_2(t) \geq m^* e^{-(d_2 + \nu)\tau}$ for $\bar{t} + \tau \leq t \leq \bar{t} + \psi$. Since the interval $[\bar{t}, \bar{t} + \psi]$ is arbitrarily chosen, we get that $y_2(t) \geq m^* e^{-(d_2 + \nu)\tau}$ for all t large enough. Therefore there exists a $m_2 > 0$ such that $y_2(t) \geq m_2$ for all t large enough.

With the first equation of (5), we know that $\dot{x}(t) \geq (r_1 - M)x(t) - ax^2(t)$. Consider the following impulsive differential equation

$$\begin{cases} \dot{Z}(t) = (r_1 - M)Z(t) - aZ^2(t), & t \neq nT, \\ Z(t^+) = Z(t) + \mu, & t = nT, \\ Z(0^+) = x(0^+). \end{cases}$$

From Lemma 2, we know that the system has a global asymptotic stable positive periodic solution

$$\tilde{Z}(t) = \frac{(r_1 - M)Z^* e^{(r_1 - M)(t - nT)}}{aZ^* e^{(r_1 - M)(t - nT)} + r_1 - M - aZ^*}, \quad t \in (nT, (n+1)T],$$

here $Z^* = \frac{r_1 - M + a\mu + \sqrt{(r_1 - M + a\mu)^2 + 4(r_1 - M)a\mu(e^{(r_1 - M)T} - 1)}}{2a}$, Using the comparison theorem of impulsive equation, we know that for all t large enough $x(t) \geq Z(t) > \tilde{Z}(t) - \varepsilon \geq \delta - \varepsilon$, here

$$\delta = \max \left\{ \frac{(r_1 - M)Z^*}{aZ^* e^{(r_1 - M)T} + r_1 - M - aZ^*}, Z^* e^{(r_1 - M)T} \right\}.$$

This completes the proof.

Discussion

In this paper, we have investigated a stage-structured predator-prey model with impulsive stocking prey and continuous harvesting predator. We obtain the conclusions: if $d_2 + v > \gamma_2$ and $v > r_2 e^{-d_1 \tau} + k_2 x^* e^{r_1 T} - d_2$ hold, the predator-eradication periodic solution of (1) is global attractive, which leads the predator species extinct; if $d_2 + v > \gamma_2$ and $v < \frac{r_1 k_2 x^*}{a x^* e^{r_1 T} + r_1 - a x^*} + r_2 e^{-d_1 \tau} - d_2$ holds, then system (1) is permanent, which leads both the prey and predator species to coexist.

Acknowledgement

This research was supported by the National Science Foundation of China (11271055).

References

- [1] L.S. Chen, Mathematical ecology modeling and research methods, Science Press, Beijing, 1988.
- [2] J. Murray, Mathematical biology, Springer-Verlag, 1989.
- [3] Y. Takeuchi, Global dynamical properties of Lotka-Volterra systems, World Scientific, Singapore, 1996.
- [4] W.G. Aiello, H.I. Freedman, A time delay model of single-species growth with stage-structure, Math Biosci, 101 (1990): 139-153.
- [5] R. Xu, Z.E. Ma, The effect of stage-structure on the permanence of a predator-prey system with time delay, Applied Mathematics and Computation, 189 (2007): 1164-1177.
- [6] S. Devi, Effects of prey refuge on a ratio-dependent predator-prey model with stage-structure of prey population, Appl. Math. Model., 37 (2013): 4337-4349.
- [7] X.K. Sun, H.F. Huo, C.C. Ma, Hopf bifurcation and stability in predator-prey model with a stage-structure for prey, J. Appl. Math. Comput., 219 (2013): 10313-10324.
- [8] J.K. Hale, Theory of Functional Differential Equations, Springer, New York, 1976.

The Influence on the Structure of the Continuous Grain Dryer Due to Grain Weight Change

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Keywords: Grain weight change, FEM, Structure stress, Continuous grain dryer.

Abstract. This paper took the corns in the north as the drying object. According to the change trend of the grain weight, we analyzed the pressure and the structure stress in the silo that could have effects on the structure of continuous grain drying machine by the finite element analysis software. The stress deformation and the change condition of diameter of the four upright columns and the oblique beams supporting the four upright columns were researched when the continuous grain drying was loaded the grains of the different weight. This method was the effective means to realize intelligent control of grain drying process. At the same time, it could provide certain reference for future testing.

Introduction

The main characteristics of continuous grain dryer is that food are dried in the process of continuous flow [1] and are finally expelled from dryer and sent into the storage. In this paper, with the help of computer technology to establish mathematical model of drying process, we selected 200t continuous grain dryer tower as the research object. Using the finite element analysis software ANSYS, we analyzed the deformation of the four supporting columns and the oblique beams on the setting when the dryer tower was loaded the grains of the different weight [2-3]. We also carried out the optimization analysis on the diameter deformation of the four supporting upright columns and the best optimization solution was obtained, which could realize the on-line measurement of moisture in drying process.

The Mathematical Model of Rectangular Silo

The 200t continuous grain drying tower was loaded every time 10t grains until 60t grains. The weight of the dryer tower was 20t. The 3D modeling was established for the setting of the dryer tower and guided into ANSYS analysis software [4]. It was shown in Fig. 1. We also carried out the detailed meshing. It was shown in Fig. 2.

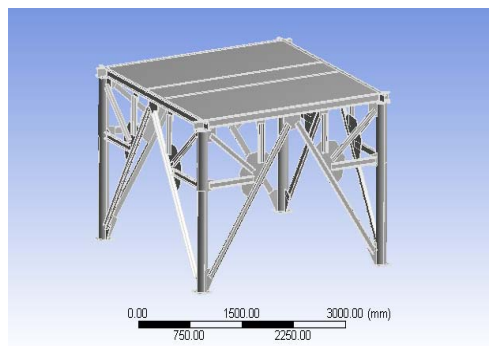


Figure 1. The 3D modeling for the setting of the dryer tower.

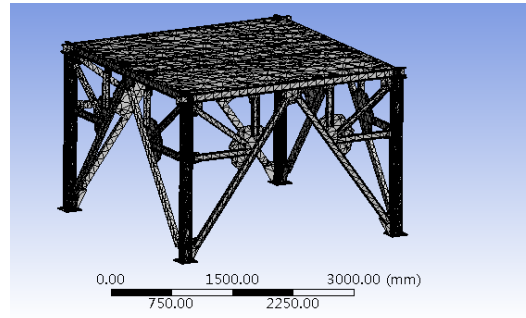


Figure 2. The mesh drawing of 3D modeling for the setting of the dryer tower.

The Analysis on Deformation of Grain Dryer Equipment due to Grain Weight Change

When 10t grains were loaded in dryer tower, the stress deformation was shown in Fig. 3 through the statics analysis [5-6].

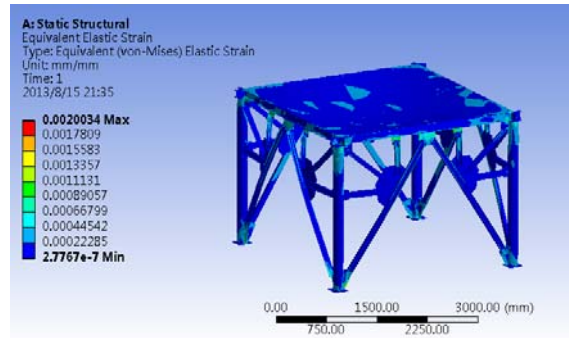


Figure 3. The deformation nephogram of the dryer tower.

Every time 10t grains was added, we could get maximum deformation of four upright columns and four oblique beams of the dryer tower, as were shown in Table 1 and Table 2, respectively.

Table 1. the maximum deformation of four upright columns for the setting of the dryer tower.

Condition	Force (N)	the maximum deformation(m)
The weight of the dryer tower+10t grains	3E+05	0.000665
The weight of the dryer tower+20t grains	4E+05	0.000887
The weight of the dryer tower+30t grains	5E+05	0.001109
The weight of the dryer tower+40t grains	6E+05	0.00133
The weight of the dryer tower+50t grains	7E+05	0.001552
The weight of the dryer tower+60t grains	8E+05	0.001774

Table 2. the maximum deformation of four oblique beams for the setting of the dryer tower.

Condition	Force(N)	the maximum deformation(m)
The weight of the dryer tower+10t grains	3E+05	0.000338
The weight of the dryer tower+20t grains	4E+05	0.000451
The weight of the dryer tower+30t grains	5E+05	0.000564
The weight of the dryer tower+40t grains	6E+05	0.000677
The weight of the dryer tower+50t grains	7E+05	0.00079
The weight of the dryer tower+60t grains	8E+05	0.000902

Through the fitted equation, we would draw the fitted curve of the upright columns and the oblique beams due to the stress deformation. They were shown in Fig. 4 and Fig. 5.

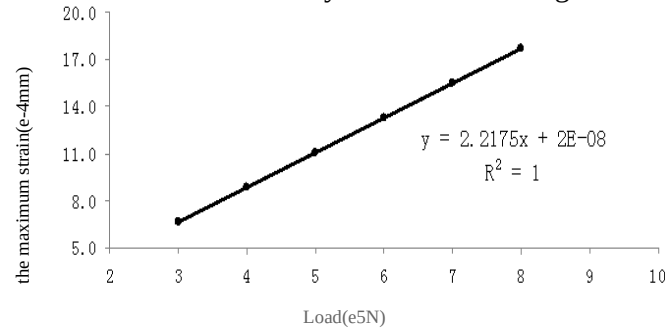


Figure 4. The maximum strain of the four upright columns for the setting of the dryer tower.

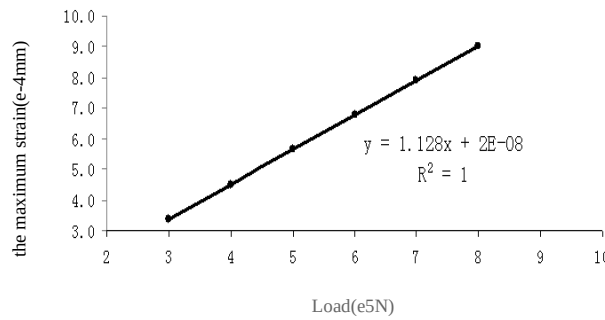


Figure 5. The maximum strain of the four oblique beams for the setting of the dryer tower.

From the fitted curve, we can observe that the deformation value of upright columns and oblique beam increased with the increase of grains. It was the linear change.

The Optimization of the Structure of Grain Dryer

Through the above analysis, the maximum deformation value of the upright column was 0.00174 m after adding the oblique beams. The pole length was 300mm and the force loading on the pole was 5kg. It could be concluded that the diameter of the pole was 72mm in Fig. 6 (a). The maximum deformation value of the upright column was 0.000276m. It could be concluded that the diameter of the pole was 116mm in Fig. 6 (b).

The weight of the whole of the dryer is supported by the four upright columns. It is very necessary to carry on the optimization analysis. The original diameter of four upright columns was 160mm. It was clear that the optimization analysis was feasible.

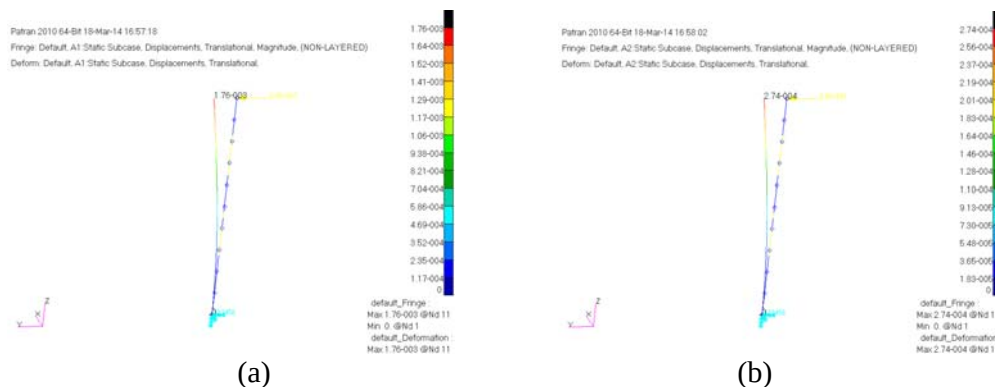


Figure 6. The deformation nephogram of the pole stress.

Conclusions

We have analyzed the working principle of continuous grain dryer machine and established the drying process model. This model was used to study the change correlation between the total weight of grains during the whole drying process and the moisture content of grains.

Using the finite element analysis software to establish the mathematical model of rectangular silo, we have analyzed the influence on the structure of the continuous grain dryer due to grain weight change. Simulation results showed that the deformation value of the four upright columns and oblique beams also would increase with the increase of grains in the dryer tower. It was the linear change. At the same time, the optimal diameter value of the four columns with oblique beams was determined. Therefore, the design of the rectangular dryer silo satisfied the requirement of mechanical structure design target in this paper.

Acknowledgement

This paper was supported by the sub-topic of the national science and technology support (2011BAD03B01).

References

- [1] F. Ayuga, M. Guaita. Discharge and the eccentricity of the hopper influence on the silo wall pressure [J]. 14th Engineering Mechanical conference, May 21-24, 2000, Austin.
- [2] Pablo Vidal, Manuel Guaita, Francisco Ayuga. Analysis of Dynamic Discharge Pressures in Cylindrical Slender Silos with a Flat Bottom or with a1-Iopper [J]. Comparison with Euro- code 1; Bio-systems Engineering, 2005: 335-348.
- [3] S.O. Nelson. Sensing Moisture Contentin [J]. Instrument and Measurement, 2000, 3 (1): 17-20.
- [4] Ki-Bok Kim, Jong-Heon Kim, Seung Seok Lee. Measurement of grain moisture content using microwave attenuation at 10.5GHz and moisture density [J]. Instrument and Measurement, 2002, 51 (1): 72-77.
- [5] Li, Y. and Puri, V.M. Finite element model prediction of cyclic thermally induced in a scaled bin using elastic-vis-co plastic constitutive equation [J]. ASAE.1991, 34 (5): 2207-2215.
- [6] H.A. Slight. The Measurement of Moisture Content [J]. Measurement and Control, 1989, 22 (2): 43-44.

The Optimization Design of the New Core Furrow Opener Based on the Physical Test

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Keywords: The core furrow opener, Discrete element, Optimization design, Simulated analysis.

Abstract. The core furrow opener was the important working part of the seeding machine. Its working process was the state of movement between the working parts and the soil particles. In this paper, the new core furrow opener was designed based on the physical object test and optimization analysis using the discrete element method (DEM). We also revised the discrete element simulation model using the small glass soil bin and the high-speed camera technology. The results of physical object test compared with discrete element simulation showed that both methods had the good correlation and the same laws. Finally, the new core furrow opener was designed. It would provide the effective method for the study and optimized design of the core furrow opener.

Introduction

In this paper, the new core core furrow opener of 8 different structure parameters was made of solid model by 3D printer and optimized design [1-2]. We would carry out the physical test by the small glass soil bin. By analyzing the test results, we obtained the optimal structural parameters of the new core furrow opener. At the same time, comparing the results of the physical test and the DEM simulation analysis, it was shown that the conclusion and the rule were all the same.

The Physical Test of the Core Furrow Opener

We used the two-towards force sensor and the dynamic strain meter to measure horizontal and vertical resistance of the opener in the experiment table. It was shown in Fig. 1.



Figure 1. The experiment table of the small glass soil bin.

We would have the soil bin tested for the new core furrow opener. We tested the opener of different parameters in the soil bin test. We also recorded the working resistance of the opener. It was shown in Fig. 2. We would record the resistance of the furrow opener in the working process by the two-dimensional sensor and the dynamic strain meter. The horizontal and vertical resistance were small that could be negligible.



Figure 2. The diagram of the furrowing resistance.

We selected the five numbers in relatively stable furrowing resistance area. The average was the test result[3-4]. At the same time, energy consumption was an important index to assess if the furrow opener was good. The energy consumption [5] is:

$$\eta = \frac{W}{V} = \frac{FL}{SL} = \frac{F}{S} \quad (1)$$

where, W was the energy consumption(J). V was the soil volume (cm^3). S was the furrowing area (cm^2). F was the traction (N). L was the working length (cm). The experimental results were shown in Table 1. We analyzed the test results in this experiment. The horizontal resistance of the furrow opener accounted for 70% of the results. The energy consumption accounted for 30% of the results.

Table 1. The test results of the core furrow opener.

test number	furrowing area(cm^2)	width(cm)	energy consumption	horizontal resistance(N)	result of weighted
1	20	11.5	1.74	21.94	15.88
2	20	11	1.82	21.7	15.74
3	19	10	1.9	22.48	16.31
4	19	10.5	1.81	7.94	6.1
5	18	14	1.29	18.34	13.23
6	25	15	1.67	70.9	50.13
7	30	15	2	38.66	27.66
8	30	14.5	2.07	56.5	40.17

The weighted analysis showed that the horizontal resistance and energy consumption were relatively small in the furrow opener of the number four. That was the best design scheme. We analyzed the correlation between above factors using DPS data analysis software [6-8]. the distribution was shown in Fig.3. x_1, x_2, x_3, x_4, x_5 were respectively the width of the opener, the grave angle, the slope angle, the operating speed and the soil moisture content. Y was the opener performance metrics.

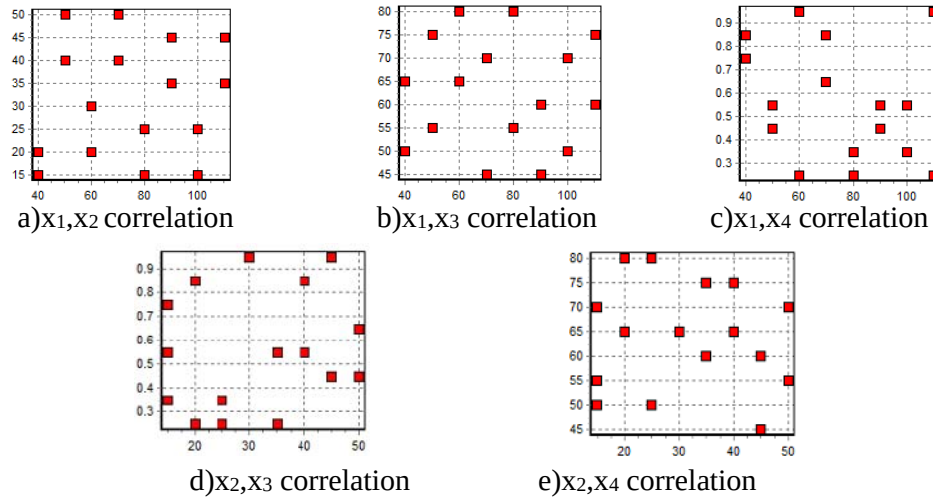


Figure 3. The correlation of the factors.

To clearly reflect the relationship between factors and indexes, we would analyze the influence of the single factor on the indexes. It was shown in Fig. 4.

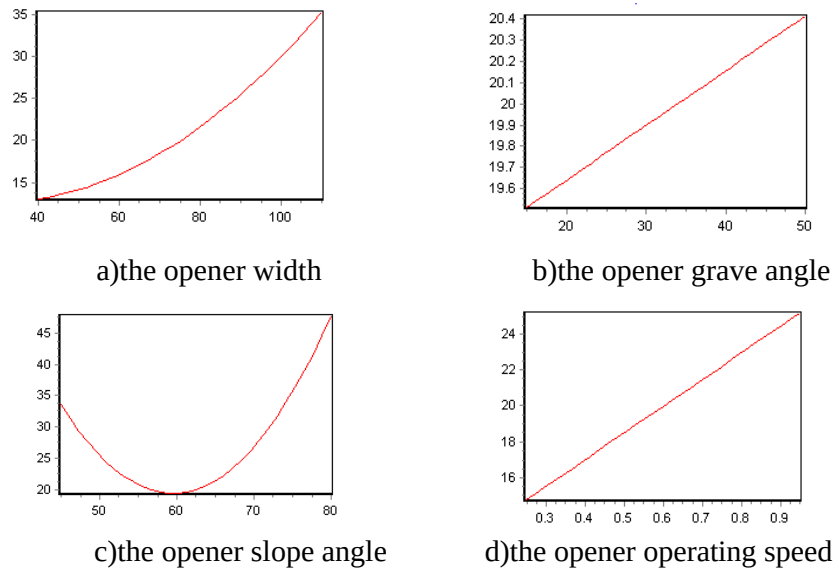


Figure 4. The date analysis of the single factor.

Because performance index of the opener was its working resistance and energy consumption. The parameter value was small, the work performance of the opener was good.

With the increase of the ditching width, opener performance decrease in Fig. 4(a). So, the width of the opener should be taken the small value. In Fig. 4(b), (d), the grave angle of the opener and the operating speed of the opener were the linear relationship basically. But the grave angle of the opener had very small influence on parameter values. The value of the operating speed of the opener was as small as possible, but it was associated with machine working efficiency. As was shown in Fig. 4(c), it was clear to see there was a turning point of performance of the opener influenced by the slope angle of the opener, the value should be controlled within 60° .

Conclusions

With glass soil bin and high-speed camera technology, we carried out the physical test to study the working process of opener through effective testing methods. We have measured the working resistance of opener, namely the horizontal working resistance, which was between 7.94~70.9N.

We established discrete element simulation models and gave the appropriate discrete element simulation parameter values.

The simulation results were compared with the physical test results. We have analyzed the influence of some factors such as the width of the opener, the grave angle, the slope angle and the operating speed on the new core furrow opener and have obtained the optimal parameter combination.

Acknowledgement

This paper was supported by the National high-tech research and development program of 863 project (2006AA04Z125) and the international natural science fund project (11172112 and 51175219).

References

- [1] Chinese Academy of Agricultural Mechanization Sciences Agricultural machinery design manual (volume one) Beijing: China Machine Press 2007.
- [2] Wang Guoqiang, Hao Wanjun, Wang Jixin. Discrete element method and its practice in EDEM. Northwestern Polytechnical University Press, 2010.
- [3] Tang Qiyi. DPS The DPS data processing system (Volume II). Modern statistics and data mining. Beijing: Science Press, 2013.
- [4] Zhao Xuanmin. Design of Experiment. Beijing: Science Press, 2006.
- [5] Gebresenbet G., Johnson H. Performance of seed drill coulters in relation to speed, depth and rake angles [J]. Journal of Agricultural Engineering Research, 1992, 52 (2): 121-145.
- [6] Wang Yanlong. Research on Opener Resistance Test and Ditching Performance Paddy Soil Conditions [D]. Nanjing Agricultural University. 2012.
- [7] Qian Liji. The Research of Opener Digital Design Methods based on Discrete element Method [D]. Changchun: Jilin University, 2008.
- [8] Tanaka H., Momozo M., Oida A., Yamazaki M., 2000a. Simulation of soil deformation and resistance at bar penetration by distinct element method [J] Terramech. 2010, 37, 41–56.

Modeling and Analysis of Human Lower Limb Reference Trajectory Based on Different Velocities

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Keywords: Human lower limb, Reference trajectory, Kinematics, Treadmill, Model analysis.

Abstract. In order to obtain kinematic reference trajectories for the control of lower limb rehabilitation robot and evaluation of patients with rehabilitation effects, Vicon three-dimensional motion capturing system was adopted to measure the three-dimensional movement information of seven healthy human with walking on the ground and walking on the treadmill at different speeds(1km/h, 2km/h, 3km/h). The difference between walking on the ground and walking on the treadmill at same velocity was analyzed. Based on these difference and Matlab software, three-dimensional joint angle data was fitted for specific velocities and regression model was created. The results showed that there are significant differences in joint angle of hip and knee for the sagittal plane ($p < 0.05$), while no significant difference was found for ankle in the sagittal plane ($P > 0.05$) and hip in the frontal plane ($P > 0.05$), Regression models can express change trend of joint angle well, the results of experiment and fitting provide reference for the control of lower limb rehabilitation robot and evaluation of patients with rehabilitation effects.

Introduction

Repetitive rehabilitation training can effectively promote patients with lower limb dysfunction recovery lower limb muscle strength and movement coordination. Rehabilitation trajectories are often obtained from data collection experiment with healthy human, some researchers adopt gait pattern adaptation algorithms to obtain rehabilitation trajectories. Clinicians who wish to provide rehabilitation trajectories often adopt to have two choices that walking on the ground and walking on the treadmill. Such as, Clinicians collect kinematic data from healthy subjects at their slow or normal self-selected speed during walking on the ground or at slow speed during walking on the treadmill. Some researchers have adopted kinematic data with healthy subjects during walking on the treadmill as rehabilitation trajectories [3,4]. Adopting kinematic data with healthy subjects for patients as rehabilitation trajectories is proved to have effective treatment [5]. However, a neurological or neuromuscular condition can often reduce an individual's gait speed by more than 50%. In order to ensure safety of patients, it is significant that different standard reference trajectory of the lower limbs to suitable for different patients and to achieve the goal of rehabilitation with different treadmill speed. Thus, selecting suitable rehabilitation trajectories for different patients has practical significance.

Several studies have investigated the influence of velocity on human lower limbs. Some researchers have studied the effects of treadmill velocity on spatio-temporal, kinematic with hip joint, knee joint and ankle joint of young healthy subjects and shown that regular variation in the sagittal plane [6,7,8]. Pépin A et al [9] have studied the effects of velocity on patients with spinal cord injury and analyzed the difference between walking on the ground and walking on the treadmill at same velocity with healthy subjects and patients with spinal cord injury. Alton F et al [10] and Riley Po et al [11] have studied the difference between walking on the ground and walking on the treadmill at

same velocity with young subjects. However, little literature about walking on the ground and walking on the treadmill at same velocity have been referred, which may create confusion on using rehabilitation trajectories during rehabilitation training. And previous work almost did not involve the study of kinematic data in the frontal plane and transversal plane. Suitable rehabilitation trajectories for rehabilitation training should not only consider detailed information of lower limb joint, also realize the influence of velocity on human lower limbs.

For ensure control of rehabilitation robotic accurately and avoid a larger number of data use inconveniently. It is important to establish model for discrete kinematic data, which can predict variation tendency of rehabilitation trajectories. Some researchers have reconstituted kinematic data based on regression and fitting method [12,13,14,15]. However, little literature involved the study of establishing model for kinematic data in the frontal plane and transversal plane.

In this paper, Vicon three-dimensional motion capturing system was adopted to measure the three-dimensional movement information of seven healthy human with walking on the ground and walking on the treadmill at different speeds(1km/h, 2km/h, 3km/h). The difference between walking on the ground and walking on the treadmill at same velocity was analyzed. Three-dimensional joint angle data was fitted for specific velocities and regression model was created.

Materials and Methods

Seven young healthy volunteer (7 male, aged: 23-26, weight: 58-90kg, height: 1.70-1.82m, average leg length: 964.35mm, average knee width: 86.46mm, average ankle width: 73.56mm) with no reported orthopedic disorders performed gait on the ground and treadmill at different speeds (1km/h, 2km/h, 3km/h, same treadmill speed with walking) without using the treadmill handrails. Leg length is the distance from anterior superior iliac spine to sole, knee width is the distance from medial point on tibial plateau to lateral point on tibial plateau, ankle width is the distance from tip of the medial malleolus on ankle to tip of the lateral malleolus on ankle. The locations of the Markers are presented in Fig. 1.



Figure 1. Actual location of the markers.

Results and Discussion

Gait Analysis. Three dimensional motion data was obtained for hip, knee and ankle joints of both lower limbs. The raw data were exported from the system software and each velocity one gait cycle was selected. The X, Y, Z in diagram and Table 1 indicates sagittal, frontal and transversal plane respectively and positive value of angles of hip (Fig. 2), knee (Fig. 3) joints indicates flexion, adduction and internal rotation. For ankle joint (Fig. 4), positive value of angles indicates dorsiflexion, adduction and internal rotation. Positive/ negative value of curves indicates direction. Due to the periodicity of human gait, the curve of hip, knee and ankle are presented for left lower limb only at same speed (3.8km/h) during two gaits with average joint angles data. The t-test will be implemented with two kinds of gait data, $P < 0.05$ illustrates significant differences of two groups of data.

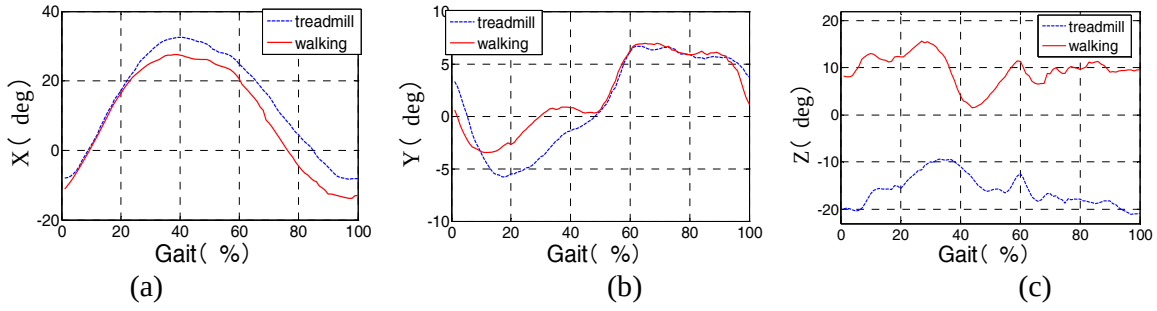


Figure 2. Curves of hip angles in the sagittal plane, frontal plane and horizontal plane at same velocity: (a) sagittal plane (b) frontal plane (c) horizontal plane.

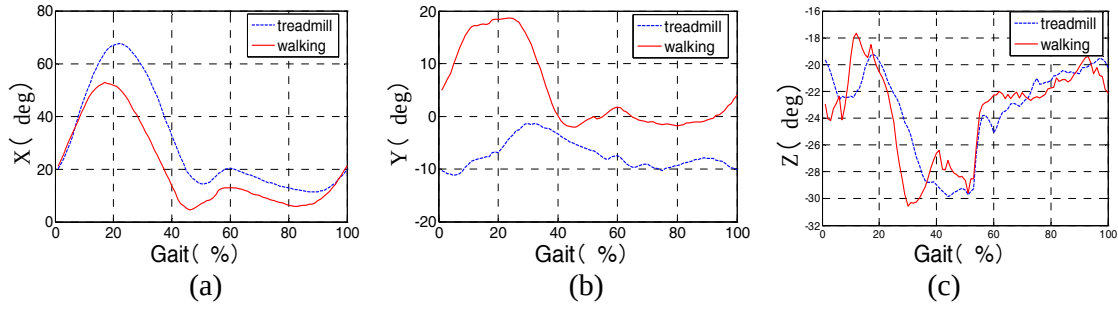


Figure 3. Curves of knee angles in the sagittal plane, frontal plane and horizontal plane at same velocity: (a) sagittal plane (b) frontal plane (c) horizontal plane.

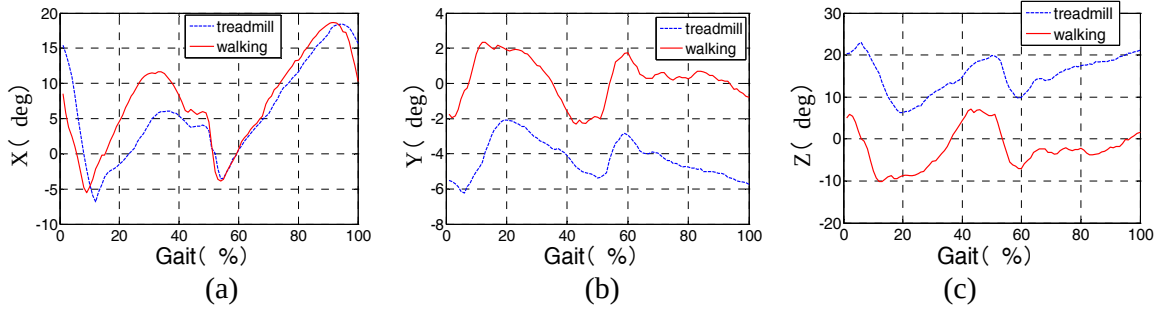


Figure 4. Curves of ankle angles in the sagittal plane, frontal plane and horizontal plane at same velocity: (a) sagittal plane (b) frontal plane (c) horizontal plane.

Angle graphs of lower limb joint movement at same speed are presented in Fig. 2, Fig. 3 and Fig. 4. Fig. 2(a-c) illustrates the variation of average angle in hip joint for two gaits at same speed. In Fig. 2(a), The maximum value of flexion angle during walking on the ground is less than the value of walking on the treadmill (difference close to 5° , $P < 0.05$). In the frontal plane ($P > 0.05$), The maximum value of adduction angle during walking on the ground is less than the value of walking on the treadmill (difference close to 2.3°). The arising time of maximum adduction angles during walking on the ground appeared ahead of time during walking on the treadmill. In the transversal plane ($P < 0.05$), curves of hip joint during walking on the treadmill were under the zero line, while curves of hip joint during walking on the ground were above the zero line, which means last longer in outward rotation of hip joint. In general, the change trend and variation range of joint angle at two gaits has large difference.

Fig. 3 (a-c) illustrates the variation of average angle in knee joint for two gaits at same speed. In the sigittal plane ($P < 0.05$), The maximum value of flexion angle during walking on the ground is much less than the value of walking on the treadmill (difference close to 15°). The variation range of joint angle during walking on the ground is less than walking on the treadmill. In the frontal plane

($P < 0.05$), there is a similar curve change with variation of hip joint in the transversal plane for knee joint. And The variation range of joint angle during walking on the ground is large than walking on the treadmill. In the transversal plane ($P > 0.05$), The arising time of maximum outward rotation angles during walking on the ground appeared ahead of time during walking on the treadmill.

Fig. 4 (a-c) illustrates the variation of average angle in ankle joint for two gaits at same speed. In the sigittal plane ($P > 0.05$), Curve trend of ankle joint during walking on the ground is closest to curve during walking on the treadmill. The arising time of first peak value and first trough value during walking on the ground appeared ahead of time during walking on the treadmill. The first peak value is greater than the treadmill gait. In the frontal plane ($P < 0.05$) and transversal plane ($P < 0.05$), there are obvious differences between walking on the ground and treadmill. Curves fluctuate up and down near zero line with data from walking on the ground, which indicates ankle motion change alternately. While curves are all above and below zero line during walking on the treadmill, however, the variation range of joint angle during two gaits seems same.

Mathematic Model Analysis. Based on the previous section, two gaits have obvious difference at same speed. In the rehabilitation training, rehabilitation trajectories derived from data during walking on the treadmill instead of walking on the ground. In this section, the mathematic model was established for data of treadmill (1km/h, 2km/h, 3km/h) by curve fitting, which make discrete data into a continuous trajectory and control rehabilitation robot conveniently.

Combined with the tendency and characteristics of curves and sinusoidal and gaussian function, curve with variation tendency relatively simple were fitted by global curve fitting. curve with variation tendency relatively complex were fitted by piecewise fitting for improve the precision. The location of segment point for piecewise fitting was adopted difference method (Fig.5) We used the following fitting formula.

$$Y = a_0 * \sin(b_0 x + c_0) \quad (1)$$

$$Y_1 = a_1 * e^{-\frac{(x_1 - b_1)^2}{c_1}} \quad (2)$$

where x , x_1 are undetermined function arguments and represents the percentage of gait cycle. a_0, b_0, c_0 is constant and represents amplitude, frequency and initial phase respectively in formula (1). b_1 represents the center of function, c_1 represents the width of function in formula (2). For the nonstandard of joint angle, the form of sum of sinusoidal and gaussian function was adopted. As a reference for the mathematic model.

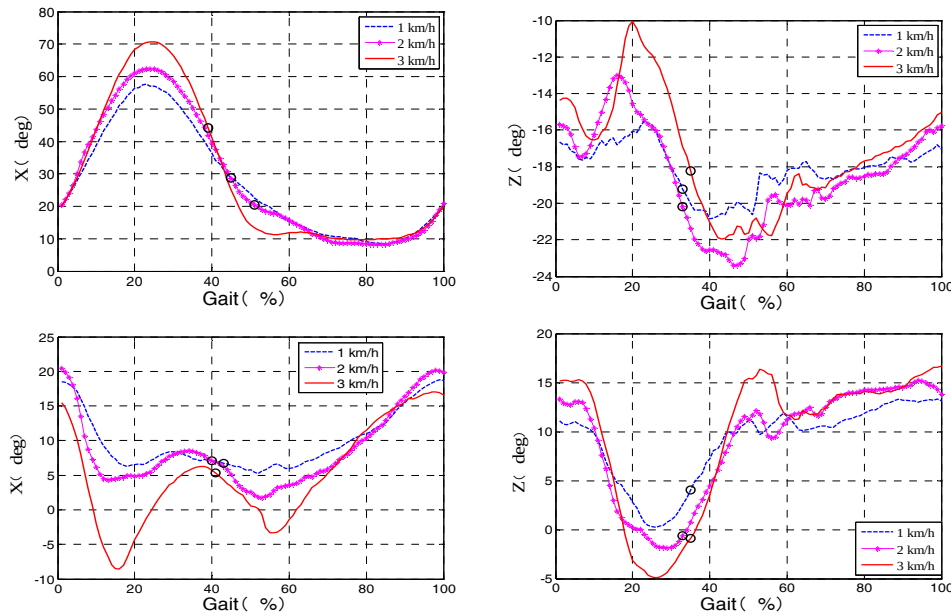


Figure 5. Location of segment point.

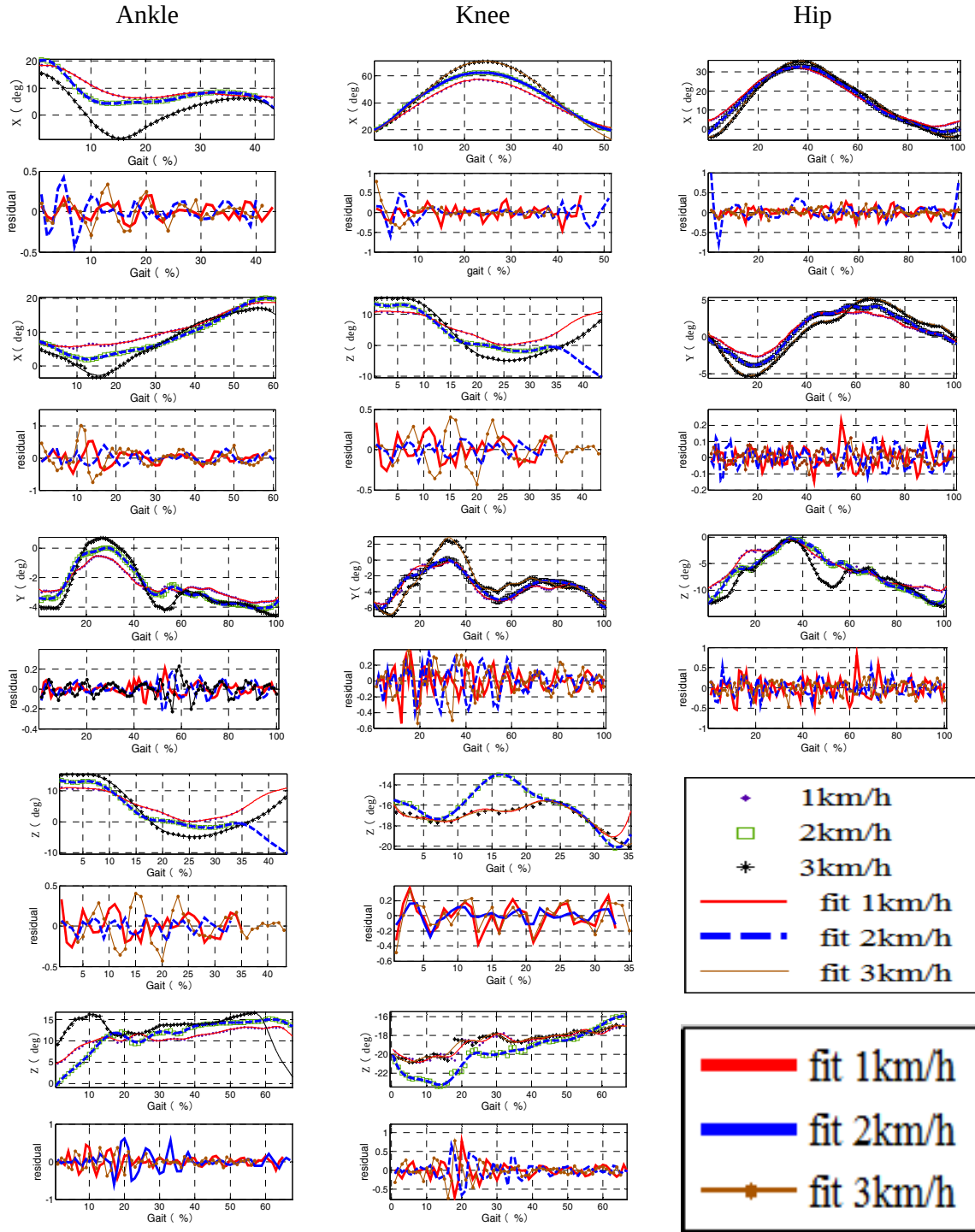


Figure 6. Curve fitting of reference trajectories.

Fig. 6 illustrates curves fitting of reference trajectories using by Gauss and Sinusoidal function. In Fig. 6, The reconstructed trajectories matched the measured trajectories very well. Fitting residual of hip in the sagittal plane with low speed (1km/h) is larger than $(\pm 1^\circ)$ other at the initial and end of the gait cycle. Fitting residual of hip in the transversal plane with high speed (3km/h) is larger than $(\pm 0.8^\circ)$ other at the 65 percent of the gait cycle. For knee joint, Fitting residual of knee in the sagittal plane with high speed (3km/h) is larger than $(\pm 0.8^\circ)$ other at the 65 percent of the gait cycle. For knee joint angle curve in the transversal plane, Fitting residual are $\pm 0.6^\circ$, $\pm 0.8^\circ$ and $\pm 0.8^\circ$ respectively. For ankle joint, Fitting residual of knee in the sagittal plane with high speed (3km/h) is larger than

($\pm 1^\circ$) other at the 10 percent-20 percent of the gait cycle. In general, Fitting residual of fitting is smaller and range approach to 1.5° .

Conclusions

In this study, Vicon three-dimensional motion capturing system was adopted to measure the three-dimensional movement information of seven healthy human with walking on the ground and walking on the treadmill at different speeds (1km/h, 2km/h, 3km/h). The difference between walking on the ground and walking on the treadmill at same velocity was analyzed. Results showed that there were significant differences in joint angle of hip and knee for the sagittal plane ($p < 0.05$), while no significant difference was found for ankle in the sagittal plane ($p > 0.05$) and hip in the frontal plane ($p > 0.05$). We draw inspiration from difference of two gaits that rehabilitation trajectories should be obtained from data collection experiment with healthy human during walking on the treadmill not walking on the ground at same speed. The fitting results show that the reconstructed trajectories matched the measured trajectories very well, which shows mathematic model can express measured data very well and provide reference for rehabilitation robot controlling.

Acknowledgement

This research was financially supported by the National Natural Science Foundation of China under Grant No. 61273342, the Beijing Natural Science Foundation under Grant No. 3132005.

References

- [1] Jezernik S., Colombo G., Morari M., Automatic Gait-Pattern Adaptation Algorithms for Rehabilitation With a 4-DOF Robotic Orthosis. *IEEE Transactions on Robotics and Automation*. 20 (2004) 574-582.
- [2] Michael H., Ross A., Prediction methods to account for the effect of gait speed on lower limb angular kinematics. *Gait & Posture*. 24 (2006) 280-287.
- [3] Chia-Yu Wang, James E. Bobrow, David J. Reinkensmeyer, Dynamic Motion Planning for the Design of Robotic Gait Rehabilitation, *Journal of Biomechanical Engineering*, 127 (2004) 672-679.
- [4] Qiyuan Wang, Jinwu Qian, Guozhi Feng, Physiological Gait Planning and Testing for a Lower Limb Gait Orthosis, *China Mechanical Engineering*. 20 (2009) 928-932.
- [5] Vallery H., van Asseldonk E.H.F, Buss M., et al. Reference Trajectory Generation for Rehabilitation Robots: Complementary Limb Motion Estimation. *IEEE Transactions on Neural Systems and Rehabilitation Engineering*. 17 (2009) 23-30.
- [6] Stoia D.I., Toth-Tascau M., Influence of treadmill velocity on joint angles of lower limbs during human gait. *Proceedings of the 3rd International Conference on E-Health and Bioengineering*, Iasi, Romania. (2011) 1-4.
- [7] Weijun Tian, Qian Cong, Zhen Yan, et al. Spatio-temporal Characteristics of Human Gaits based on Joint Angle Analysis. *Proceedings of the 3rd International Conference on Computer Science and Information Technology*. (2010) 439-442.
- [8] Han Ya-li, Wang Xing-song, The biomechanical study of lower limb during human walking. *Sci China Tech Sci*. 41 (2011) 592-601.
- [9] Pépin A., Norman K.E., Barbeau H., Treadmill walking in incomplete spinal-cord-injured subjects: 1. Adaptation to changes in speed. *Spinal Cord*. 41 (2003) 257-270.
- [10] Alton F., Baldey L., Caplan S., et al. A kinematic comparison of overground and treadmill walking. *Clinical Biomechanics*, 1998, 13 (1998) 434-440.

- [11] Riley Po, Paolini G., Della C.U., *et al.* A kinematic and kinetic comparison of overground and treadmill walking in healthy subjects. *Gait & Posture*. 26 (2007) 17-24.
- [12] Tufekciler N., van Asseldonk E.H., van der Kooij H., Velocity-dependent reference trajectory generation for the LOPES gait training robot, *IEEE International Conference on Rehabilitation Robotics (ICORR)*. (2011) 1-5.
- [13] Koopman B., van Asseldonk E.H.F., van der Kooij H., Speed-dependent reference joint trajectory generation for robotic gait support. *Journal of Biomechanics*. 47 (2014) 1447-1458.
- [14] Chen Gui-liang, Zhao Yue, Li Chang-peng, *et al.* Research on the Gait Planning Method of Lower Limbs Exoskeletons Rehabilitation Robot. *Science Technology and Engineering*. 13 (2013) 1671-1815.
- [15] Chew Yean Yam, Mark S. Nixon, John N. Carter, Automated person recognition by walking and running via model-based approaches. *Pattern Recognition*. 37 (2004) 1057-1072.

Kinematical Performance Analysis of a 3-/RRS Spherical Parallel Mechanism for Ankle Rehabilitation

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Keywords: Rehabilitation mechanism, Transmission performances, Ankle rehabilitation.

Abstract. There exist some input errors in rehabilitation training of rehabilitation mechanism, different design parameters can lead to different errors in mechanism. As an evaluation index of mechanism performances, global conditioning index did the analysis work of mechanism performances through defining global angular velocity precision transmission evaluation index and global torque accuracy transmission evaluation index. According to 3-RRS mechanism Kinematics constraint, it solved the corresponding Jacobian matrix, though which it analyzed the performances index of rehabilitation mechanism transmission and built the relations of every design parameter and performances index. In addition, it is verified that there is no singular posture in rehabilitation exercising of mechanism thus can meet demands of ankle rehabilitation.

Introduction

The 3-RRS ankle rehabilitation mechanism is the new-type one with the remote center of rotation which can keep coincides with the center of ankle joint in the rehabilitation. It has good flexibility and mechanical properties benefiting from three spherical joints S connected with the moving platform. It can achieve 3-dimentional rotation, and also meet the requirement of ankle rehabilitation motions for eversion /inversion, dorsiflexion /plantarflexion, adduction/abduction. A good mechanism must have good movements and mechanical properties, so the mechanical properties analysis of the mechanism is very importance as same as the forward Kinematics and the inverse kinematics of mechanism. Until now many achievements about the properties of the Parallel Mechanism have been made [1-2]. Feng GAO and Ling-jing ZHENG studied the acceleration performance index of the 6-SPS micromanipulator and the deficient DOF Parallel Robot [3-4]. Juan-guo XI used a dummy body effect coefficient made by Zheng HUANG to discuss the acceleration performance indicators theoretically [5-6]. Gosselin and Angeles Completed 3-RRR planar parallel mechanism optimization design for the purpose of maximizing the workspace area of performance indicators [7]. Pursuing of maximizing workspace volume performance index, Miller optimized the arrangement of Delta rotation axis [8-9]. Li, along with Gosselin and other people, completed the optimization design of 3 - RPR parallel mechanism [11-12] in pursuit of maximizing singularity-free workspace as the performance index. Based on the discrete points of montecarlo method. Stamper and Tsai defined the number of jacobian matrix of discrete points in the work-space, and evaluated the quality of the work space adopting the method of Inverse integral in the workspace [13, 14]. Donghua University and Jiaotong University carried out an analysis of kinematics transmission performance towards a kind of spherical body mechanics. The article analyzed the performance through defining the angular velocity precision transmission index and the index of torque precision performance evaluation, which provided more accurate basis for optimal design of institutions.

Kinematics Transmission Performance Analysis

The analysis of the kinematics transmission performance is the revelation of its nature and also the foundation for other institutions for performance analysis and design. The ankle rehabilitation institution is shown in Fig. 1

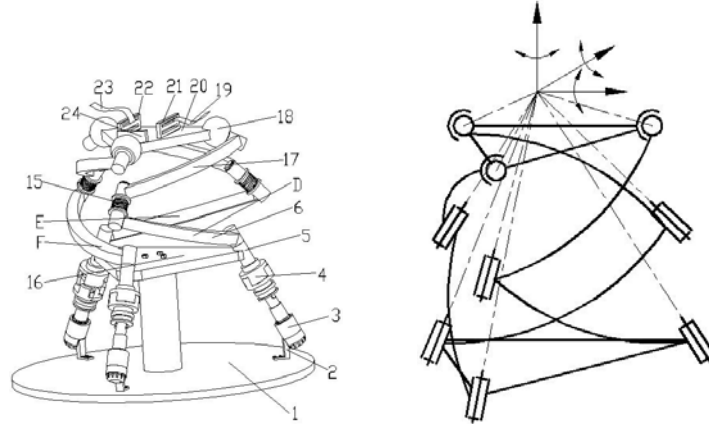


Figure 1. the mechanical structure prototype.

Solving of Jacobian Matrix

The inverse-solution of constraint equations based on kinematics:

$$S_i \cdot W_i = \cos \alpha_2 \quad (1)$$

The input and output of the constraint equations can be expressed as a function of time t , so do the derivative of the function t towards both sides of the constraint equations at the same time, then we can get below equation.

$$d_i \dot{\phi}_1 + e_i \dot{\phi}_2 + f_i \dot{\phi}_3 + l_i \dot{\alpha} + m_i \dot{\beta} + n_i \dot{\gamma} = 0 \quad (2)$$

Based on Eq. 2 we can get:

$$M\dot{\phi} + N\dot{\psi} = 0 \quad (3)$$

In the equation

$$\begin{aligned} \dot{\phi} &= (\dot{\phi}_1 \ \dot{\phi}_2 \ \dot{\phi}_3)^T, & \dot{\psi} &= (\dot{\alpha} \ \dot{\beta} \ \dot{\gamma})^T \\ M &= \begin{bmatrix} d_1 & e_1 & f_1 \\ d_2 & e_2 & f_2 \\ d_3 & e_3 & f_3 \end{bmatrix}, & N &= \begin{bmatrix} l_1 & m_1 & n_1 \\ l_2 & m_2 & n_2 \\ l_3 & m_3 & n_3 \end{bmatrix} \end{aligned}$$

In the Eq. 3, $\dot{\phi}$ is the input angular velocity vector for the spherical 3-dof parallel mechanism; $\dot{\psi}$ is for YRP angular speed vector; M is for the first kind of jacobian matrix, as well as the diagonal matrix, N is for the second category of Jacobian matrix. When institutions is non-singular configuration, M and N are full rank matrix $rank(M)=3$, $rank(N)=3$.

Trim the Eq. 3, we can get below equation:

$$\dot{\psi} = J_1 \dot{\phi} \quad (4)$$

where

$$J_1 = -N^{-1}M \quad (5)$$

Matrix J_1 is the jacobian matrix by through the input angular velocity vector and succession of angular speed of the vector transformation. However, it should be pointed out that Matrix J_1 is the

Jacobian matrix, only through the input angular velocity vector and succession of angular speed of the vector transformation, not the way the required input angular velocity vector to the output angular velocity vector transformation. The output angular velocity vector is that of the end of the actuator in the fixed coordinate system. In order to calculate the angular velocity vector of the fixed coordinate system, it is a must to get the Jacobi matrix transformed by succession of angular speed vector to the angular velocity vector of the fixed coordinate system. The formula is the relationship between the rotation transformation matrix of robotics and the angular velocity of operator.

$$\dot{T} = S(\omega)T \quad (6)$$

In the formula $S(\omega)$ is the angular velocity of operator and $S(\omega) = \begin{pmatrix} 0 & -\omega_z & \omega_y \\ \omega_z & 0 & -\omega_x \\ -\omega_y & \omega_x & 0 \end{pmatrix}$

T is YPR rotation transformation matrix. Sort out the above three formulas to get the equation:

$$\omega = C\dot{\psi} \quad (7)$$

In the equation:

$$\omega = (\omega_x \quad \omega_y \quad \omega_z)^T \quad (8)$$

and

$$C = \begin{pmatrix} \sin \beta & 0 & 1 \\ -\sin \gamma \cos \beta & \cos \gamma & 0 \\ \cos \gamma \cos \beta & \sin \gamma & 0 \end{pmatrix} \quad (9)$$

in which ω is the output angular velocity vector at the end actuator; C is The third type of Jacobi matrix, Matrix C is got through the rotation Angle vector of Z- Y- X to the output angular velocity vector of the fixed coordinate system, is about the function of α , β , γ . Through Eq. 4 and Eq. 7, we can get the equation:

$$\omega = J\dot{\phi} \quad (10)$$

where

$$J = CJ_1 = -CN^{-1}M \quad (11)$$

J is Jacobi matrix of the Spherical parallel mechanism, it will be full rank when institutions are not in a singular configuration.

The Evaluation Index of the Kinematics Transmission Performance

Three degrees of freedom parallel mechanism only transfer angular velocity in the process of motion transmission, so the input and output are all angular velocity. At the end of the actuator, ω is the output angular velocity vector, $\dot{\theta}$ is the input angular velocity, so the equation is $\omega = J\dot{\theta}$, that is, when there is a deviation in the input speed, it will cause certain deviation in the output angular velocity. Set the input speed deviation is $\Delta\dot{\theta}$, and the output angular velocity deviation is $\Delta\omega$, in this case, the equation is transformed into below, see Eq. 12.

$$\omega + \Delta\omega = J(\dot{\theta} + \Delta\dot{\theta}) \quad (12)$$

Combined with equation formula 12, we can get below equation :

$$\Delta\omega = J\Delta\dot{\theta} \quad (13)$$

According to the theory of matrix norm theory get below equation :

$$\|\Delta\omega\| = \|J\Delta\dot{\theta}\| \leq \|J\| \cdot \|\Delta\dot{\theta}\| \quad (14)$$

$$\|\Delta\dot{\theta}\| = \|J^{-1}\Delta\omega\| \leq \|J^{-1}\| \cdot \|\Delta\omega\| \quad (15)$$

By Eq. 14 and Eq. 15, we can get the equation

$$\frac{\|\Delta\omega\|}{\|\omega\|} \leq \|J\| \cdot \|J^{-1}\| \frac{\|\Delta\dot{\theta}\|}{\|\dot{\theta}\|} \quad (16)$$

Set $k(J) = \|J\| \cdot \|J^{-1}\|$, then $k(J)$ is The condition number of Jacobi matrix, which is a measure of the precision passed between the input Angle and output Angle seen from formula 16. Adopting the method of spectral norm of the matrix to calculate the condition number of Jacobian matrix, then get Eq. 17.

$$\|J\| = \left[\max \lambda(J^T J) \right]^{\frac{1}{2}} \quad (17)$$

$\max \lambda(J^T J)$ in the formula is the the maximum characteristic value by Jacobi matrix transpose times Jacques. If Jacobi matrix J is non-singular, then the $J^T J$ is the positive definite matrix and the characteristic value will be positive number. In the regard, J matrix is the extraction of the maximum characteristic value for $J^T J$, then we can get the equation:

$$\|J\| = \sigma_{\max}(J) \quad (18)$$

Similarly, we can get Eq.19 since the $J^T J$ is the positive definite matrix.

$$\|J^{-1}\| = \frac{1}{\sigma_{\min}(J)} \quad (19)$$

Then can get conditional number from Eq. 18&19, shown as below equation .

$$k(J) = \frac{\sigma_{\max}(J)}{\sigma_{\min}(J)} \quad (20)$$

In the formula 20, the $\sigma_{\max}(J)$ is the maximum singular value of Jacobian matrix .and the $\sigma_{\min}(J)$ is the minimum one. Because the condition number is the ratio of the largest singular value and the minimum singular value, so that the scope of condition number will be $1 \leq k(J) \leq \infty$. By analysis of the scope, we can know that when the value of condition number is becoming greater, the smaller input deviation may lead to larger output deviation, that is, the value is bigger, so the lower of transmission precision and vice versa. When the conditional number $k(J)=1$, the transition condition will be the best, as the scope of condition number is on a large scale, it is not advantageous for analysis of mechanism, so in order to simplify the analysis, we define the inverse of Jacobi matrix as performance evaluation index of the angular velocity, shown as equation :

$$\rho(J) = \frac{1}{k(J)} \quad (21)$$

In the process of movement, the moving platform position differs by time, so is the $\rho(J)$, which means the analysis cannot be done easily, so we defined $\rho(J)$ in the workspace average value as the

transfer performance evaluation index of the angular velocity precision for the agency's full domain. Its expression is shown in the following type equation:.

$$\overline{\rho(J)} = \frac{\int_V \rho(J) dV}{\int_V dV} \quad (22)$$

In the formula 22, the V is the workspace of Spherical parallel mechanisms, whose value range is $0 < \overline{\rho(J)} \leq 1$, so the larger the value is, the better the precision transited performance of the angular velocity is.

Suppose the structure parameters of the 3-RRS separately $\alpha_1 = 30^\circ$ $\alpha_2 = 30^\circ$ $\beta_1 = 30^\circ$ $\beta_2 = 60^\circ$ $\varepsilon_i' = 0$, $\frac{2}{3}\pi, \frac{4}{3}\pi$, and the range of motion is $\alpha = -30^\circ \sim 30^\circ$ $\beta = -30^\circ \sim 40^\circ$, $\gamma = -30^\circ \sim 30^\circ$. Fig. 2 to Fig. 4 is the performance evaluation index of global angular velocity transmission when α , β and γ get different values.

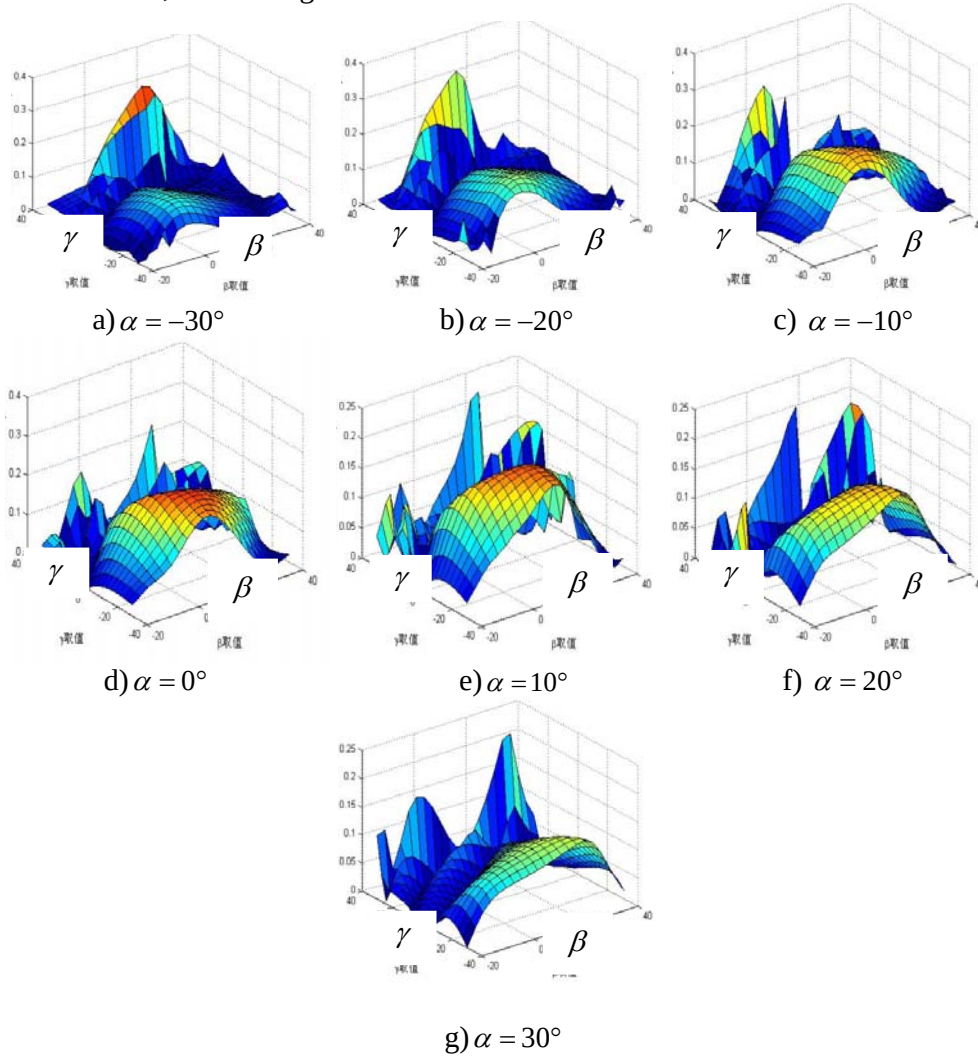
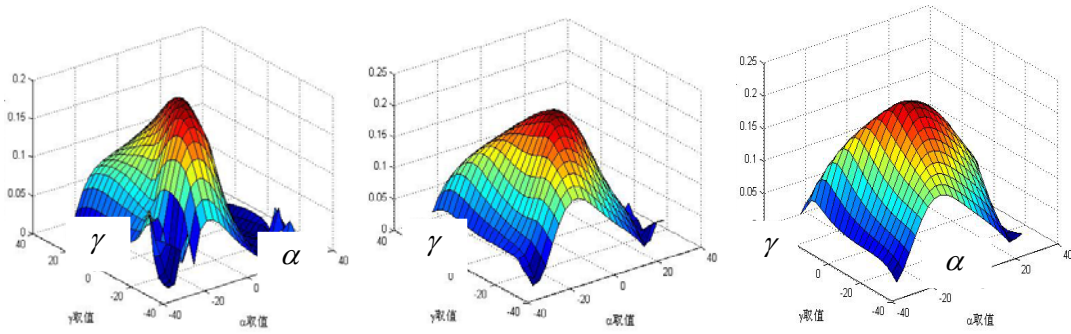


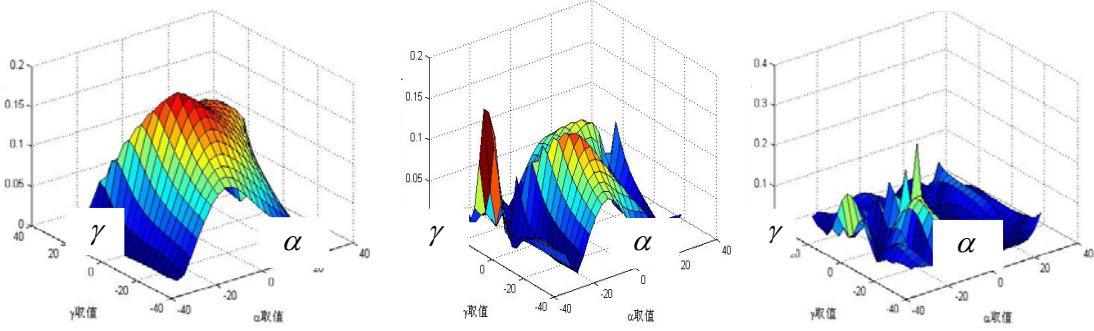
Figure 2. The relations between $\overline{\rho(J)}$ and α .



a) $\beta = -20^\circ$

b) $\beta = -10^\circ$

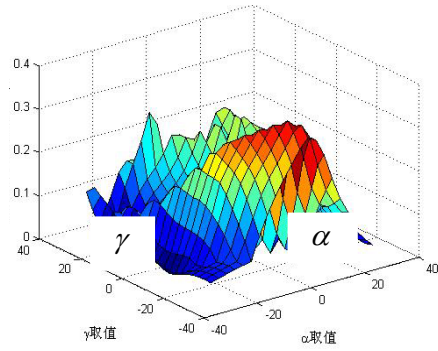
c) $\beta = 0^\circ$



d) $\beta = 10^\circ$

e) $\beta = 20^\circ$

f) $\beta = 30^\circ$



g) $\beta = 40^\circ$

Figure 3. The relations between $\overline{\rho(J)}$ and β .

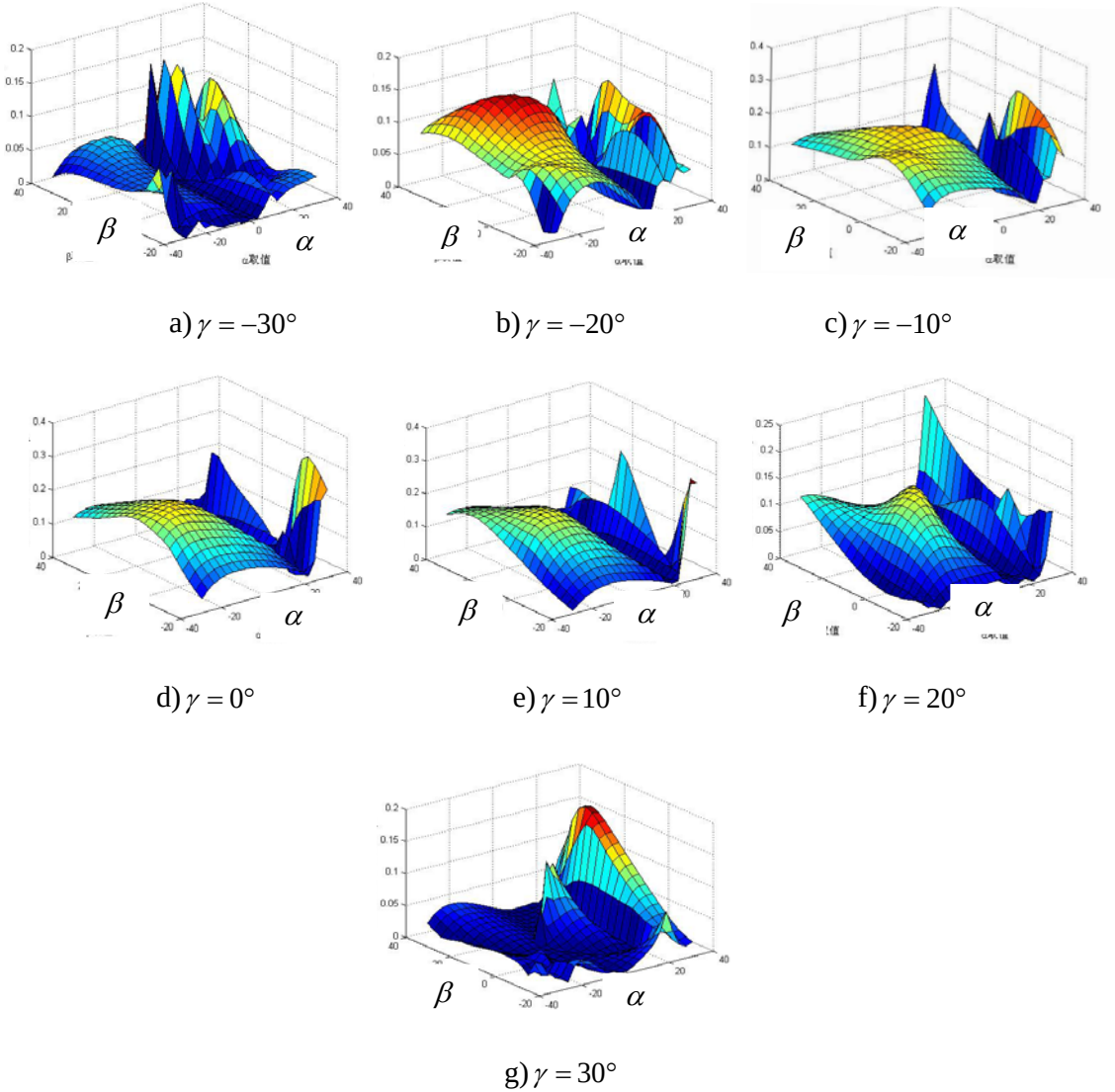


Figure 4. The relations between $\overline{\rho(J)}$ and γ .

Conclusion

In this paper, the mechanism of Jacobi matrix was solved through the motion equation of mechanism, and also we studied the kinematics transmission performance by defining these performance indexes, as well as the relationship between evaluation indexes of angular velocity & force precision and the motion dynamic posture. Through a numerical example analysis, it can be concluded from the picture that as the mechanism motion is in smaller posture Angle, the transfer performance change will be quite gentle, and the reverse is also true. However, there is no singular position type appearing within the scope of movement. This paper provided an important theory for parameter optimization and reasonable design method by establishing the relationship between the performance and position.

Acknowledgement

This research was financially supported by the National Natural Science Foundation of China under Grant No. 61273342, the Beijing Natural Science Foundation under Grant No. 3132005.

References

- [1] Shi Zhi-xin, Luo Yu-feng. On global performance Indices of Robotic Mechanisms. *ROBOT*, 2005, 27 (5): 420-422.
- [2] F. Gao, X.J. Liu. The relationships between the shapes of the work-spaces and the link lengths of 3-DOF symmetrical planar parallel manipulators, *Mechanism and Machine Theory*, 2001, 36 (2): 205-220.
- [3] X.J. Liu, Z.L. Jin, Gao F. Optimum design of 3-DOF spherical parallel manipulators with respect to the conditioning and stiffness indices, *Mechanism and Machine Theory*, 2000, 35 (9): 1257-1267.
- [4] Jin Zheng-lin, Gao Feng. The analysis of acceleration index for a novel parallel 6-PSS micromotion Manipulator. *Machinery Design & Manufacture*, 2000, (3): 29-30.
- [5] Guo Xi-juan. Basic theory of dynamical on parallel robot mechanism. [YANSHAN UNIVERSITY PHD Thesis]. 2002: 1-77.
- [6] Huang Zhen. Parallel robot mechanism theory and control. Beijing: CHINA MACHINE PRESS, 1997: 5-100.
- [7] Gosselin C.M., Angeles J., The optimum kinematic design of a planar three-degree-of-freedom parallel manipulator, *Journal of Mechanisms, Transmissions, and Automation in Design*, 1988, 110: 35-41.
- [8] Miller K., Maximization of workspace volume of 3-DOF spatial parallel manipulators, *ASME Journal of Mechanical Design*, 2002, 124 (2): 347-350.
- [9] Stock M., Miller K., Optimal kinematic design of spatial parallel manipulators: application to linear Delta robot, *ASME Journal of Mechanical Design*, 2001, 125 (2): 392-301.
- [10] Miller K., Optimal design and modeling of spatial parallel manipulators, 2004, 23 (2): 127- 140.
- [11] Li H.D., Gosselin C., Richarda J., Determination of maximal singularity-free zones in the workspace of planar three-degree-of-freedom parallel mechanisms, *Mechanism and Machine Theory*, 2006, 41 (10): 1157-1167.
- [12] Gallant M., Boudreau R., The synthesis of planar parallel manipulators with prismatic joints for an optimal, singularity -free workspace, *Journal of Robotic System*, 2002, 19 (1): 13-24.
- [13] Stamper R.E., A three degree of freedom parallel manipulator with only translational degrees of freedom, Ph.D thesis, University of Maryland, 1997.
- [14] Stamper R.E., Tsai L.-W., Walsh G.C., Optimization of a three DOF translational platform for well-conditioned workspace, In: *Proceedings of the 1997 IEEE International Conference on Robotics and Autom-ation*, Albuquerque, New Mexico, 1997: 3250-3255.
- [15] Sheng Ji. Analysis and Simulation of Dynamic Performance of a Spherical Parallel Mechanism. [EAST CHINA JIAO TONG UNIVERSRRY Master Thesis]. 2013: 27-36.

Disruptions Optimization Model of Supply Chain Coordination

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Abstract. As business environment is more and more volatile, there are usually more than one factor disruptions in the supply chain system, which makes the extant research outdated. With consideration of both market demand scale and retailers purchase costs disruptions in supply chain disruption management, this article finds the manufacturers production cost is a convex of retailers deviate if demand is a linear function of the price. Based on the assumption of Stackelberg game leader, this article formulates a disruption strategy and coordination mechanism to optimize the supply chain profit.

Introduction

Since science and technology has achieved a great development, the competition among business becomes fiercer and fiercer in the world, which results in an increasingly stress on the survival and development of companies. Unfortunately, the extant management model could not satisfy the volatile market demand. Meanwhile, it has been gradually realized that supply chain management is an important way to enhance companies' competitiveness by increasing their adaptability to global competition. As one of integral part of supply chain management research, the supply chain coordination research has a great practical significance in improving supply chain performance.

Extant researches on supply chain disruption management have been done usually under the condition that only one factor was disrupted. Unfortunately, the recent environment uncertainty always results in a lot of factors disruptions in the supply chain system. This article takes both market demand scale and retailer's purchase costs disruptions in supply chain disruption management into account and find that when demand is a linear function of the price, the manufacturer's production cost is a convex of retailers deviate at the same time. In order to optimize the supply chain profit, this article put forward a disruption strategy and coordination mechanism to coordinate the supply chain in the case that the manufacture is Stackelberg game leader.

Supply Chain Coordination with Market Scale and Costs Disruptions

Once the manufacturer makes the production plan, the emergency event can change not only market requirement but also retailer's purchase costs. Suppose $a > 0$ represents market demand scale, $k > 0$ represents price elasticity coefficient, and $c > 0$ represents retailer's purchase costs. Let Δa be variable of the requirement, Δc be variable of retailer's purchase costs, and c_r be retailer's purchase assistant costs. If $a + \Delta a > 0, c_r + \Delta c > 0$, the assumption has the signification of fact.

Suppose $\bar{q}$ be the requirement of market after emergency event, $\bar{q} = a + \Delta a - k\bar{p}$. Let the difference between requirement $\bar{q}$ and the manufacturer's form production project q^* be $\Delta q = \bar{q} - q^*$, when $\Delta q = \bar{q} - q^* < 0$, production has some surplus, which need to be sold in low price of secondary market; when $\Delta q = \bar{q} - q^* > 0$, the more product need to be produced to meet the new requirement. Since new resource needs to be used and manufacturer's production plan has to be changed, the increase and surplus lead to the incidental expenses.

After the emergency event, for the requirement $\bar{q}$, consider the profit function of supply chain on production disruption cost:

$$\bar{f}^{sc} = \bar{q} \left(\frac{a + \Delta a - \bar{q}}{k} - c_r - \Delta c \right) - C(\bar{q}) - \lambda_1 (\bar{q} - q^*)^+ - \lambda_2 (q^* - \bar{q})^+ \quad (1)$$

$\lambda_1, \lambda_2 > 0, (x)^+ = \max(0, x)$, λ_1 represents the unit incidental cost of exceeding the former production; λ_2 represents the unit incidental cost of reducing the former production. Since the surplus production should be sold in the lower price of marginal cost $C'(\bar{q})$ in the secondary market, $0 < \lambda_2 < C'(\bar{q})$.

To emphasize the influence of market scale and retailer's purchase costs disruptions, theory 2.1 can be set up:

Theory 2.1 In the case of market scale and retailer's purchase costs disruptions of supply chain, suppose $\bar{q}^*$ represents the optimal order quantity of the maximal profit function $\bar{f}^{sc}$ of supply chain, (a) when $\Delta a \geq k\Delta c$, $\bar{q}^* \geq \bar{q}$; (b) when $\Delta a \leq k\Delta c$, $\bar{q}^* \leq \bar{q}$.

Prove (reduction to absurdity): suppose when $\Delta a \geq k\Delta c$, $\bar{q}^* < \bar{q}$, $\bar{q}^*$ is the optimization of (1), in another words, $\bar{q}^*$ is the optimal solution of (2):

$$\bar{f}_0^{sc}(\bar{q}) = \bar{q} \left(\frac{a + \Delta a - \bar{q}}{k} - c_r - \Delta c \right) - C(\bar{q}) - \lambda_2 (q^* - \bar{q}) \quad (2)$$

When q^* meets

$$2\bar{q}^* = a + \Delta a - k(c_r - \Delta c + C'(\bar{q}^*) - \lambda_2) \quad (3)$$

$\bar{f}_0^{sc}(\bar{q})$ reaches the maximal value.

$$\bar{q}^* - q^* = \frac{1}{2}(\Delta a - k\Delta c - k(C'(\bar{q}^*) - C'(q^*)) + k\lambda_2) \quad (4)$$

$C(q)$ is the convex function, $C''(q) > 0$, so $C'(q)$ is incremental function. If $\bar{q}^* < q^*$, $C'(\bar{q}^*) < C'(q^*)$; Since $\Delta a \geq k\Delta c$, $(k + \Delta k)\lambda_2 > 0$,

$$\bar{q}^* - q^* = \frac{1}{2}(\Delta a - k\Delta c - k(C'(\bar{q}^*) - C'(q^*)) + k\lambda_2) \geq 0 \quad (5)$$

Hence, $\bar{q}^* \geq q^*$. This is contrary to the assumption, which means the assumption is not correct.

Therefore, if $\Delta a \geq k\Delta c$, $\bar{q}^* \geq \bar{q}$. Likewise, it can be calculated that, if $\Delta a \leq k\Delta c$, $\bar{q}^* \leq \bar{q}$.

Theory 2.1 indicates when market scale and retailer's purchase costs change simultaneously, if the market demand increase but retailer's purchase costs decrease, $\Delta a \geq k\Delta c$, the manufacturer should increase production level to meet market demand. Therefore, the manufacturer orders more production to achieve the maximal profit; if the market demand decrease but retailer's purchase costs increase, $\Delta a \leq k\Delta c$, the manufacturer should decrease production level. When market scale and retailer's purchase costs increase or decrease together, the market demand is not constant. The manufacturer should make production plan and decide the optimal order quantity according theory 2.1.

Based on theory 2.1, when $\Delta a \geq k\Delta c$, $\bar{q}^* \geq \bar{q}$ and market scale and retailer's purchase costs disrupt simultaneously, the object function (1) can be predigested to:

$$\bar{f}_1^{sc}(\bar{q}) = \bar{q} \left(\frac{a + \Delta a - \bar{q}}{k} - c_r - \Delta c \right) - C(\bar{q}) - \lambda_1 (\bar{q} - q^*) \quad (6)$$

It can be proved easily that $\bar{f}_1^{sc}(\bar{q})$ is concave function about $\bar{q}$. So $\bar{f}_1^{sc}(\bar{q})$ need to have optimal solutions. If neglecting the condition $\bar{q}^* \geq \bar{q}$, the optimal solution is:

$$2\bar{q}^* = a + \Delta a - k(c_r + \Delta c + C'(\bar{q}_1) + \lambda_1) \quad (7)$$

(2-7) subtracts (2-3):

$$2(\bar{q} - q^*) + k(C'(\bar{q}_1) - C'(q^*)) = \Delta a - k\Delta c - k\lambda_1 \quad (8)$$

$C(q)$ is the convex function, $C''(q) > 0$, so $C'(q)$ is incremental function, and

if $\bar{q}_1 \geq q^*$, $2(\bar{q} - q^*) + k(C'(\bar{q}_1) - C'(q^*)) \geq 0$;

if $\bar{q}_1 \leq q^*$, $2(\bar{q} - q^*) + k(C'(\bar{q}_1) - C'(q^*)) \leq 0$.

Disruption Strategy Proposed to Optimize the Supply Chain Profit

The case 1: $\Delta a - k\Delta c > k\lambda_1$

In this situation, $\bar{q}_1$ meets the condition $\bar{q}^* \geq q^*$, so $\bar{q}_1$ is the optimal solution

$$\bar{q}_1^* = \bar{q}_1 = \frac{a + \Delta a - k(c_r + \Delta c + C'(\bar{q}_1^*) + \lambda_1)}{2} \quad (9)$$

The corresponding optimal retail price is

$$\bar{p}_1^* = \frac{a + \Delta a - \bar{q}_1^*}{k} \quad (10)$$

And the maximal profit of supply chain is

$$\begin{aligned} \bar{f}_1^{sc*} &= (\bar{p}_1^* - c_r - \Delta c) \bar{q}_1^* - \lambda_1 (\bar{q}_1^* - q^*) \\ &= \bar{f}^{sc*} + \left(\frac{\Delta a - k\Delta c}{k} \right) q^* - (C(\bar{q}_1^*) - C(q^*)) + \frac{(\Delta a - k\Delta c - k(\lambda_1 - C(q^*)))^2 - (kC(\bar{q}_1^*))^2}{4k} \end{aligned} \quad (11)$$

Based on (9) to (11), if $\Delta a - k\Delta c > k\lambda_1$, all the optimal order quantity, optimal retail price and the maximal profit of supply chain are affected by market scale and retailer's purchase costs.

The case 2: $0 < \Delta a - k\Delta c \leq k\lambda_1$

In this situation, $\bar{q}_1 \leq q^*$, according to concave function property, in $[\bar{q}_1, \infty]$ the maximal point of $\bar{f}_1^{sc}(\bar{q})$ is

$$\bar{q}_2^* \leq q^* \quad (12)$$

The corresponding optimal retail price is

$$\bar{p}_2^* = \frac{a}{2k} + \frac{1}{2}(c_r + C'(q^*)) + \frac{\Delta a}{k} = p^* + \frac{\Delta a}{k} \quad (13)$$

The maximal profit of supply chain is

$$\bar{f}_2^{sc*} = f^{sc*} + \frac{\Delta a - k\Delta c}{k} q^* \quad (14)$$

Based on (12) to (14), if $0 < \Delta a - k\Delta c \leq k\lambda_1$, the optimal order quantity are not affected by market scale and retailer's purchase costs. In another words, manufacturer doesn't need to change production plan. As long as market scale disrupts, the retail price should be adjusted, and the price has positive linear relationship with market scale. It can be concluded from (14) that the profit is larger than the former.

If $\Delta a \leq k\Delta c$, the optimal order quantity is less than q^* , hence,

$$f_2^{sc*}(\bar{q}) = \bar{q} \left(\frac{a + \Delta a - \bar{q}}{k} - c_r - \Delta c \right) - C(\bar{q}) - \lambda_2 (q^* - \bar{q}) \quad (15)$$

If $\bar{q}_2$ meets

$$2\bar{q}_2 = a + \Delta a - k(c_r + \Delta c + C'(\bar{q}_2) - \lambda_2) \quad (16)$$

(15) achieves the optimal solution.

As a result,

$$2(\bar{q}_2 - q^*) + k(C'(\bar{q}_2) - C'(q^*)) = \Delta a - k\Delta c + k\lambda_2 \quad (17)$$

Below is the discussion of the optimal order quantity of (15) satisfying with $\bar{q}^* \leq q^*$.

The case 3: $k\lambda_2 \leq \Delta a - k\Delta c \leq 0$

The case 4: $\Delta a - k\Delta c \leq k\lambda_2$

Similar to the above analysis, the optimal order quantity of the two situations:

$$q^* = \begin{cases} \bar{q}_3^* = q^*, & k\lambda_2 \leq \Delta a - k\Delta c \leq 0 \\ \bar{q}_4^* = \frac{a + \Delta a - k(c_r + \Delta c + C'(\bar{q}_4) - \lambda_2)}{2}, & \Delta a - k\Delta c \leq k\lambda_2 \end{cases}$$

The corresponding optimal retail price is:

$$\bar{p}^* = \begin{cases} \bar{p}_3^* = p^* + \frac{\Delta a}{k}, & k\lambda_2 \leq \Delta a - k\Delta c \leq 0 \\ \bar{p}_4^* = \frac{a + \Delta a + k(c_r + \Delta c + C'(\bar{q}_4) - \lambda_2)}{2k}, & \Delta a - k\Delta c \leq k\lambda_2 \end{cases}$$

The maximal profit of supply chain in the two situations is:

$$\begin{aligned} \bar{f}_3^{sc*} &= f^{sc*} + \frac{\Delta a - k\Delta c}{k} q^* \\ \bar{f}_4^{sc*} &= (\bar{p}_4^* - c_r - \Delta c) \bar{q}_4^* - \lambda_1 (\bar{q}_4^* - q^*) \\ &= \bar{f}^{sc*} + \left(\frac{\Delta a - k\Delta c}{k} \right) q^* - (C(\bar{q}_4^*) - C(q^*)) + \frac{(\Delta a - k\Delta c - k(\lambda_2 - C(q^*)))^2 - (kC(\bar{q}_4^*))^2}{4k} \end{aligned} \quad (18)$$

Conclusion

When market scale and retailer's purchase costs disruptions happen simultaneously, if the transformation of $|\Delta a - k\Delta c|$ is small (satisfying $k\lambda_2 \leq \Delta a - k\Delta c \leq 0$), the original production plan should be maintained, and the demand disruption cost resulting from retail price adjustment should be compensated. In this circumstance, retail price is only affected by market scale and will increase with the development of market scale. If the transformation of $|\Delta a - k\Delta c|$ is large, the original production plan should be changed. For retailer price, in spite of magnitude of the transformation of $|\Delta a - k\Delta c|$, as long as market demand changes, retail price should be adjusted. As a result, if market demand scale develops, the retail price increases; if market demand scale shrinks, the retail price decreases.

References

- [1] Yu H., Chen J., Yu G. How to coordinate supply chain under disruption. System Engineering Theory and Practice, 7 (2005), p. 9-16.
- [2] Tomlin B. On the value of mitigation and contingency strategies for managing supply chain disruption risks. Management Science, 5 (2006), p. 639-657.
- [3] Z Guo, F. Fang, A.B Whiston, Supply chain information sharing in a macro prediction market, Decision Support System, 5 (2006), p. 427-438.
- [4] C. Schneeweiss, K. Zimmer, M. Zimmermann. The design of contracts to coordinate operational interdependencies within the supply chain [J]. International Journal of Production Economics, 1 (2004), p. 43-59.

Study on Modeling and Operating Characteristics of the DC Transformer

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Keywords: DC transformer, Magnetization curve, MATLAB simulation model.

Abstract. Although there is an increasing interest in the DC current measurement, the study for modeling and parametrizing the DC transformer is still inadequate. In this paper, by analyzing the sensing mechanism of DC transformer, we build up the mathematical model for the magnetization curve and set up the DC transformer MATLAB platform based on the principle of magnetic modulation. This MATLAB model can reflect the physical mechanism of the DC transformer such that the transmission characteristics of DC transformers can be studied by changing the model parameters. The simulation results verified the operational characteristics of the DC transformer under low load conditions, which provides a comparable significance to further study of DC transformer and products design.

Introduction

The measurement of DC current has become an essential part in magnetic measurement technology [1]. Many research institutions are undertaking research on DC current measurement instruments, in particular the primary transducer. However, till now, existing methods and instruments still cannot fully meet the research requirements. At the meanwhile, the testing for high-current measuring instruments has not been fully resolved. In fact, existing DC sensors all have their own defects. Hence, there is a big developing space for the sensing technology. And the studies on the mechanism and analysis of DC current sensing have drawn a lot of attentions.

Research on the transfer characteristics of the DC transformer consists of two aspects [2]: one is to conduct the theoretical research based on the ideal model and analytical expressions; the other is to conduct research by setting up the DC transformer prototype experiment to acquiring testing data, wherein using the computer software for modeling and analyzing the DC transformer is still not well established. In this paper, based on Matlab (2013Rb) software, we model the DC transformers and study the magnetic modulation principle for the DC transformer under different parameters. Specifically, we focus on the working mechanism of the DC transformer and the transformer characteristics were also studied.

The Basic Principle of DC Current Transformer

Fig. 1 shows the basic principle of DC current transformers, where a sine exciting current i_0 periodically drives two magnetic cores changing from positive to negative. The external DC current, which is to be measured, generates the iron core an external magnetic field and we denote the magnetic intensity to be H_{ext} . Due to the external magnetic field H_{ext} , the relative permeability magnetoconductivity, which makes the iron core change from positive to negative, is different. In particular, the voltage generated in the magnetic test coil reflects the intensity of the external magnetic field, i.e., H_{ext} , thereby reflecting the magnitude of the DC current to be measured.

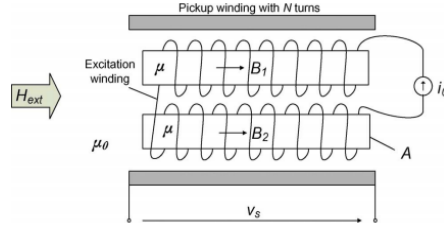


Figure 1. The basic principle of DC current transformer.

The principle for measuring the DC current transformer lies in the nonlinear relationship between the magnetic field strength H of the ferromagnetic material and magnetic flux density B [3]. The excitation coil twines the two cores parallelly. The sinusoidal excitation current in the coil makes the magnetic field intensity in the core periodically changed from positive to negative. More importantly, the excitation currents in both the iron cores change in the opposite direction. [4] Accordingly, the voltage in the magnetic test coil does not include the potential generated by the magnetic field induced by the excitation current in the magnetic test coil. In fact, the detected voltage in the magnetic test coil essentially reflects the synergistic effect of the magnetic flux density in the two cores, i.e.,:

$$V_s = -2NA\left(\frac{dB_1}{dt} + \frac{dB_2}{dt}\right) \quad (1)$$

where N is the number of turns of magnetic test coil and A is the area of magnetic flux passing through the core region.

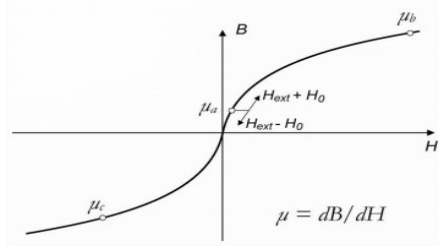


Figure 2. Magnetization curve diagram of the DC transformer.

In Fig. 2, considering the excitation magnetic field and the external DC magnetic field strength generated by the excitation current, the relative permeability of the magnetic flux density in the iron core can be expressed as:

$$\mu = \frac{dB_{H_{ext} \pm H_0}}{d(H_{ext} \pm H_0)} \quad (2)$$

and the detected voltage can be expressed as:

$$V_s = -2NA\left(\mu_1 \frac{d(H_{ext} + H_0)}{dt} + \mu_2 \frac{d(H_{ext} - H_0)}{dt}\right) \quad (3)$$

The external magnetic field strength generated by the measured DC current, i.e., H_{ext} , can be considered to be constant. Thus, the above detected voltage can be expressed as:

$$V_s = -2NA \frac{dH_0}{dt} (\mu_1 - \mu_2) \quad (4)$$

Then, we can define the differential relative magnetic permeability as:

$$\mu_d = \mu_1 - \mu_2 = \frac{dB_{H_{ext}+H_0}}{d(H_{ext}+H_0)} - \frac{dB_{H_{ext}-H_0}}{d(H_{ext}-H_0)} \quad (5)$$

In this way, we have

$$V_s = -2NA\mu_d \frac{dH_0}{dt} \quad (6)$$

It is worth mentioning that the hysteresis effect is ignored in the analysis. This effect can be realized by considering the phase offset of H_0 and μ_d .

From the above analysis, we see that for the DC current transformer, the current to be measured can be reflected by the detected voltage V_s , while the detected voltage V_s is determined by three factors, i.e., the magnetic field strength H_{ext} generated by the measured current, the intensity of magnetic field H_0 which is generated by the exciting current, and the specific shape of the magnetization curve. [5] Therefore, by changing the parameters of these three factors, one can see how different parameters affect the DC current transformer. On the other hand, the external magnetic field strength H_{ext} and the excitation field intensity H_0 can be directly represented by the measured current and the exciting current. When the magnetization curve model is decided, the detected voltage waveform is completely determined by the measured current and the exciting current. Moreover, when exciting current is determined, the waveform of the detected voltage directly reflects the current to be measured.

The Model of the DC Current Transformer

In this section, we set up the simulations for the DC transformer based on the Matlab platform. Specifically, the simulation consists of two parts: [6] in the first part, we build up the DC transformer simulation module on Matlab platform by controlling the system model, system circuit, and the magnetic circuit equation constraints; in the second part, based on the signal processing module, we write up the signal processing algorithms on Matlab to demodulate the test voltage V_s and display the demodulated results.

Fig. 3 establishes a DC current transformer In order to simplify the DC transformer model, we ignore the hysteresis characteristics and core eddy current effects in the simulation, and choose low residual magnetism magnetic as a reference to model the magnetization curve.

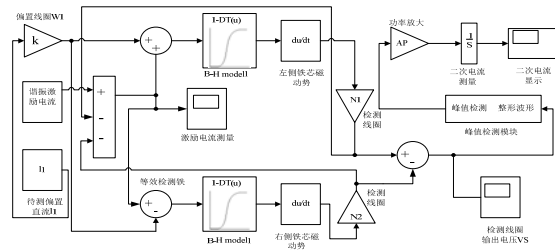


Figure 3. MATLAB simulation of the DC transformer.

Simulation Results and Analysis

DC Transformer Mechanism. The first part of the simulation is to verify the DC transformer working mechanism. The key is to use the look-up table to set B-H model. [7] Once the B-H model is determined, the ferromagnetic material properties for the DC current transformer are also

identified. Different B-H models will affect the exciting power supply module, the excitation coil and the detected coil module parameters.

The exciting voltage waveform is shown in Fig. 4, where the frequency is 50Hz and the unit of time is $\frac{1}{100\pi}$ S.

The resulting detected voltage V_s waveform is shown in Fig. 5. Herein, we set the simulation parameters to be 500A, which is the standard current for the primary measured current. The working mechanism of the DC current transformer is reflected by the waveform of V_s .

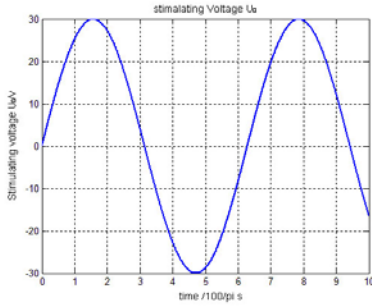


Figure 4. Waveform of the excitation power.

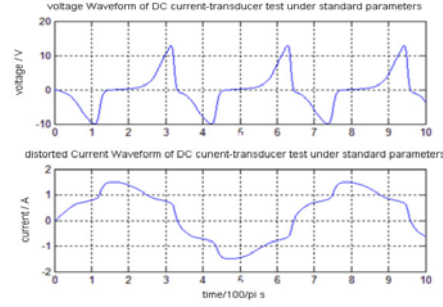


Figure 5. Waveforms of the detected voltage and the distortion of exciting power.

We see from the simulation waveform that when the measured DC current is not zero, the output signal frequency of the detected voltage is twice of that for the source voltage, and the phase of the induced voltage is also shifted, which is consistent with the principle of the DC current transformer. The distortion of the signal reflects the working mechanism of magnetic modulation.

The Model Parameters for the DC Transformer. The second part of the simulation is to study how the parameters affect the characteristics of the DC transformer. According to the results of the signals, the output secondary voltage is linear with the measured DC offset primal voltage and such a linear relationship is within a certain range. The magnetization curve obtained by Method I is shown in Fig. 6, which is in contrast to the forward part of this method.

In the following Fig. 7, we show how the magnetization curves change with the transmission characteristics of transformers. Herein, we let the parameters remain the same as previous and set the same primal measured currents for Method I and Method II, which reflect different magnetization curves for different materials.

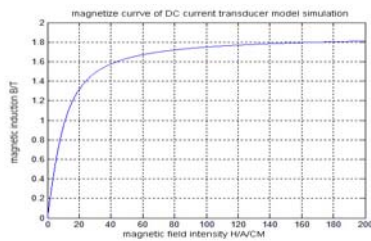


Figure 6. Magnetization curves obtained by Method I.

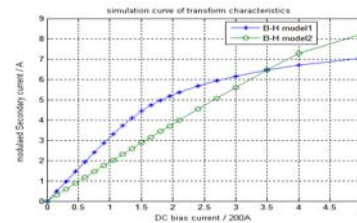


Figure 7. Magnetization curve changes with the transmission characteristics of transformers.

This group of simulations shows how the magnetization curve simulation model affects the simulation of the DC transformer. Specifically, it implies that different magnetization curve models have different linear ranges, which also affect the output secondary modulated voltage.

Simulation for the DC Current Transformer. For the DC current transformer model, the detected voltage waveform directly includes the information for the measured current. In other words, the detected voltage waveform changes with the DC current. As for the DC transformer models, once the magnetization curve that representing the physical characteristics of the core is

decided, it is the primal current and the exciting current that decided the detected results. These two parameters affect the DC current transformer in the sense that when they are working on different sections in the magnetization curves, they generate detected voltage waveforms with different times and different pulse peak, and thus affect the results of the demodulated DC current transformer waveform. Moreover, they even affect the metrological characteristics of the DC current transformer [8]. We simulate this phenomenon in three aspects and study the impact of these two parameters on the detected voltage waveform of the DC current transformer.

(1). The detected voltage under a low load.

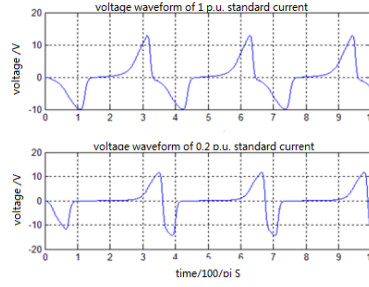


Figure 8. Detected voltage waveform under a low load.

Fig. 8 shows that when at a low load, the detected voltage waveform has significant changes. Firstly, under low load conditions, the time for the voltage starting to rise is significantly lag to the first positive half-peak period, e.g., at 3 seconds before, with a standard current load, the detected voltage has rapidly saturated, while under low load conditions, the detected voltage has just entered the initial stage of saturation. This result reflects that a low load current does affect the working conditions for the core saturation. Hence, the difference between positive and negative peak varies with the measured current, and their relationship is no longer ideally linear. Considering error characteristic of the actual modulated circuit, the uncertainty of the detected results for the DC current transformer under low load conditions will increase

(2). The influence on the detected voltage when increasing the load.

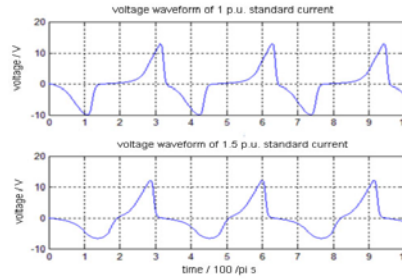


Figure 9. Detected voltage waveform under a high load.

On the other hand, increasing the load current also plays an impact on the detected voltage waveform of the DC current transformer. Since the load current increases, it will make the voltage saturated for a long time, while withdrawing from saturation in a relatively short time.

(3). The effect of the exciting power under low load conditions.

In this section, we further explore the output waveform of the detected voltage under low loads in the DC current transformer model.

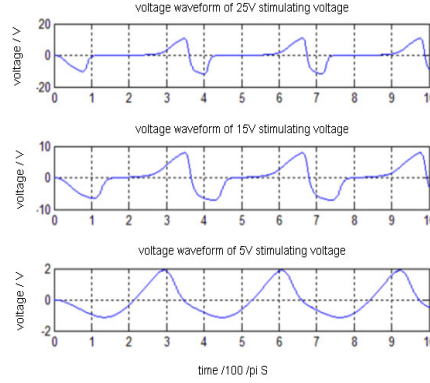


Figure 10. The impact of the exciting voltage on the detected voltage waveform under low load conditions.

We can see from the waveforms in Fig. 10 that under low current loads, if we gradually reduce the exciting voltage, the detected voltage waveform will change significantly. Under low current loads, the iron core is easier to work on the linear region of the core magnetization curve during the positive and negative alternating time. This lead to a very smooth waveform of the detected voltage and the waveform changes according to the exciting power. Therefore, under low load conditions, the exciting power becomes the key [9]: if the power supply voltage is too small, the DC current transformer may not work, and thus we may not measure the DC current.

Conclusion

In the power system, with the development of the UHV power grid as the backbone of a strong smart grid, HVDC technology has a very broad application prospect [10]. The research on modeling the DC transformer is still relatively adequate, and the on-site application is usually the current comparator which is based on the magnetic modulator, magnetic amplifiers and the Hall-effect principle. There are many opening questions on the mechanism for measuring the DC current. Therein, using the computer technology to determine and optimize the DC current transformer is indeed meaningful. In this paper, we compare and adjust the modeling parameters and make three contributions:

1. We set up simulations based on the magnetization curve model, which combines the control system and signal processing system to result in a configured DC transformer model for verifying the DC transformer working mechanism. We show that the model is reliable.
2. We studied the transfer characteristics of the DC transformer, and explore how different parameters affect the transfer characteristics of transformers. This provides a significant impact on simulations for the DC current transformer.
3. We studied why the measurement error will increases when the DC current transformer is working under low load conditions. By setting a reasonable exciting voltage, we let the DC transformer work in a detection period from the unsaturated state to the saturated state, which helps improve the accuracy of the current mutual inductance metrological.

With the continuous development of the electricity market, in order to meet the actual needs of production and accommodate higher accuracy requirements, more efforts should be devoted to improving the measurements of the DC current, the DC huge current, as well as the stability and accuracy of the measurements.

References

- [1]Wu Bo. Design of DC transducer with magnetic auto-balance compensation [D]. Wuhan, Huazhong University of Science and Technology, 2006.

- [2]Huang Hao. The Research on Heavy DC Transformer [D]. Wuhan, Huazhong University of Science and Technology, 2004.
- [3] Silvio Ziegler, Robert C. Woodward, Herbert Ho-Ching Iu, Senior Member, IEEE, etc. Current Sensing Techniques: A Review [J]. IEEE SENSOR JOURNALS, VOL. 9, NO. 4, APRIL 2009: 472-479.
- [4]Hu Xiangyong, Xiang Xiaomin, Huang Nanshan. DC detection method based on detection offset of magnetic modulator [J]. Eletrical Measuremnt & Instrumentation, 1999, 36 (7): 20-22.
- [5] Li He, Li Qian, Zhang Shuhan, Lei Min, Li Dengyun, Hu Haoliang. Development of a Zero-flux DC Current Transformer for HVDC Projects [J]. High Voltage Engineering, 2012, 38 (11): 2981-2985.
- [6]Yu Guo-bin. Simulation of current transformer based on gauge solution [J]. Relay, 2007, 35 (15): 54-58.
- [7] Li Qian, Mao Cheng-xiong, Lu Ji-ming, Li Wei-bo. Study of modeling and simulation of magnetic modulator [J]. Journal of Transducer Technology, 2005, 24 (2): 29-34.
- [8] Manuel Román, Guillermo Velasco, etc. “Low Consumption Flux-gate Transducer for AC and DC high-current Measurement” in Proc. 39th IEEE Power Electron. Specialists Conf., PESC’ 08, GREECE, 2008, PP. 535-560.
- [9]Ren Shi-yan, Wu Bo,Wang Xiao-wei, Yang Hua-yun. Research on Design of DC Sensor Based on Magnetic Potential Self-balance and Feedback Compensation [J]. ACTA METROLOGICA SINICA, 2007, 28 (3): 202-205.
- [10] Cai Donghui. Research on test method and related uncertainty calculation of Metering Standard Device of Heavy Direct Current [D]. Wuhan, Huazhong University of Science and Technology, 2012.

Research on Order Cepstrum Method for Gear Cracking Fault Diagnosis

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Keywords: Fault diagnosis, Gear cracking, Order cepstrum, Frequency mesh.

Abstract. A new method called order cepstrum analysis which is used to analyze the instantaneously signals in the rotate mechanism is introduced. First, the vibration signals at start-up in the gearbox are sampled at constant time increments in time-domain and then the data are resampled with software at constant angle increments in angle-domain. so, the time domain instantaneously signals is changed into angle domain stationary ones. Second, the stationary ones are analyzed with order cepstrum. The outcome shows it can avoid effectively the “frequency mesh” with order tracking technology, which cannot be solved with the traditional frequency spectrum method and the non-stationary signals are translated into stationary signals.

Introduction

Many fault such as gear wearing, crack and gear broken are created easily in gearbox because the gear always works in high speed, heavy load and strong impulse conditions. In the end, the other mechanical parts inactivations are induced. So the failure-free operation of the system is influenced by the operation status of gearbox. It is more difficulty to confirm the frequency components of vibration signals in gearbox because the components in gearbox is excessive, the frequency components is complexity, and the rotate frequency of the axis is changing along with the add-and-subtract of drive ratio. At the same time, the signals during rising or reduction of speed are belonged to non-stationary signals. Under this condition, the request of stationary about signals can't be suited to fast Fourier transform. If the signals are assumed with stationary ones to analyze, the graveness “frequency smear” phenomena are brought in the outcome [1-3]. In order to solve the problem, the theory of angle sample and theory of order tracking analysis, which based on the theory of angle sample are brought forward.

In the paper, the vibration signals during start-up of the gearbox are resampled in angle-domain, and the non-stationary signals in time-domain are changed into stationary signals in angle-domain. The signals in angle-domain are analyzed with order cepstrum and the fault condition of gearbox is diagnosed exactly.

Order Tracking Analysis

Excellence of Using Order Tracking

The sampling at constant angle increments in angle-domain is employed in order tracking analysis; some character points are avoided to lose, which can't be reserved in time-domain sampling [2-6]. The advantages of using the order tracking analysis are as following:

- (1) The fatal “leakage effect” is solved. The sampling order is used to cycle continuation in the signals after resampled in angle-domain, so the leakage, which had happened during DFT changing in

time-domain, could be avoided. The analysis precision in system can be improved greatly, and the most important is that the phasic information of signals is distilled accurately.

(2) The comparable quality of the historical data is assured. In order to pledge the sampling signals at the same of initial point in each of sampling courses, the rising edge of keyphasor is introduced to sample signals in system, so the phasic of all historical vibration data have lengthwise direction comparable quality.

(3) The changing line-frequency components are isolated in the sampling sequential. The fault information, which is connected with the line-frequency of facility and harmonic component are captured with order spectrum analysis.

Computed Order Tracking

To determine the resample times, it will be assumed that the shaft is undergoing constant angular acceleration. With this basis, the shaft angle, θ , can be described by a quadratic equation of the following form:

$$\theta(t) = b_0 + b_1 t + b_2 t^2 \quad (1)$$

Where b_0, b_1, b_2 is the unknown coefficients, t is time.

The arrival times, t_1 through t_3 are known from the sampling of the keyphasor pulse signal. Substituting these conditions into Eq. 1), and arranging in a matrix format gives,

$$\theta(t_1) = 0 \quad \theta(t_2) = \Delta\phi \quad \theta(t_3) = 2\Delta\phi \quad (2)$$

This set of equations is then solved for the unknown b_i components. Once these values are known, Eq. 1 may be solved for t , yielding

$$t = \frac{1}{2b_2} \left[\sqrt{4b_2(k\Delta\theta - b_0) + b_1^2} - b_1 \right] \quad (3)$$

Where k is interpolation coefficient and it can be solved by Eq. 4,

$$\theta = k\Delta\theta \quad (4)$$

The signals after resampled in angle-domain are obtained through interpolation method. The precision of computed order tracking is decided by the following factors: such as the distinguishability and precision of the estimation time of the keyphasor impulse; the choice of sample frequency in digit filter; the changing of rotate is satisfied the assumed conditions; the precision of interpolation and fitting of digital signals, etc. But the sampling theorem is must be satisfied during the sampling.

Realization of Order Cepstrum

The cepstrum is another spectrum analysis to frequency spectrum. It is more efficiency for spectrum analysis of the sideband in multi-components and spectrum wave of cognition or different cognition, and it has the power for deconvolution. The original signals and the transfer system character can be separated and extracted [7-8]. Assumed the power spectrum of signal $x(t)$ is $S_x(f)$, then the cepstrum

$$C_x(\tau) \text{ is } C_x(\tau) = F^{-1}[\log S_x(f)] \quad (5)$$

Where $F^{-1}[\]$ is Fourier inverse transformation, τ is the time variable quantity of cepstrum, it is gerfuency. The advantage with the order cepstrum is as follows,

(1) The import effect of the signal sources and transfer channels effect are divided among the output signals, and then the source of trouble is easily finding.

(2) The cycle components in power spectrum are simplified to single spectrum line, and the outcome is more distinguish.

(3) The false hood peaks are reduced in spectrum analysis.

The signals after resampled in angle-domain are analyzed with cepstrum, and this method is the order cepstrum. It is an availability method for fault diagnosis in gearbox [8].

Example of Diagnosis

The examination is certificated in one kinds of single-stage gearbox. In the system, the input axis is driven by an electric motor; and the load is driven by the output axis. The number of teeth in drive gear is $Z_1=30$, the number of teeth in driven gear is $Z_2=50$. One wheel tooth in drive gear is burnished with abrasive paper, and then the wear fault in tooth surface is simulated. The start-up process in gearbox is analyzed; the rotate speed of input axis is speed-up from 0 to about 600 rpm. The vibration signals with B&K4508 acceleration transducer and speed signals with JN338A rotary rotational speed and torque-admeasuring apparatus are delivered to B&K3560 multi-analyser for data processing. Firstly, the signals of vibration and rotational speed are sampled at constant time increments in time-domain, the frequency of sampling is 16384Hz; the sampling time is 2s.

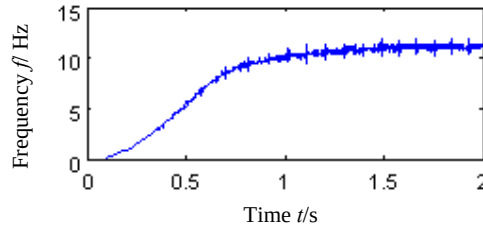


Figure 1. Instantaneous frequency of drive axis.

Fig. 1 is the instantaneous frequency of drive axis. The speed rotates of drive axis speed-up from 0 to about 10 Hz. It is a representative non-stationary course.

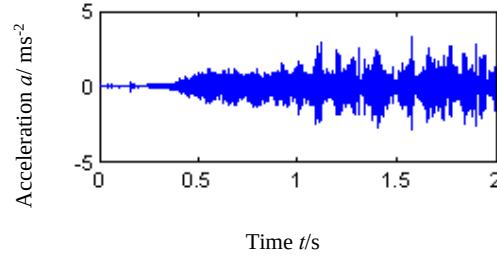


Figure 2. Time course of the original vibration signals.

Fig. 2 is the time course of the original vibration signals. It can be seen from the Fig. 2 the vibration signals is becoming more strengthen following the step up of the rotate speed in drive axis. It can be explained that there is direct connection between the vibration signals in gearbox and rotate speed of the drive axis. At the same time, the vibration signals are non-stationary signals and couldn't satisfy the request of stationarity signals in FFT. The data should be resampled in angle-domain. It can be drawn through the calculation.

Gear mesh order $x_m = 30$, Gear mesh frequency $f_m = 30f_{r1}$,

Gear wearing character order $x_{crack} = 1$, Gear mesh order cepstrum $\hat{x}_m = 360^\circ / x_m = 12^\circ$,

Tooth surface order cepstrum $\hat{x}_{crack} = 360^\circ / x_{crack} = 360^\circ$, Where f_{r1} is the frequency of drive axis.

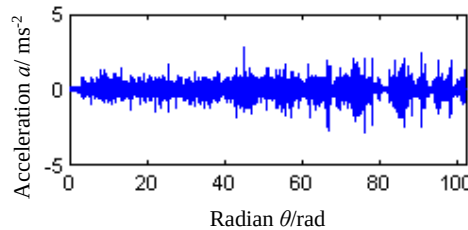


Figure 3. Vibration signals after resampled.

Fig. 3 is the vibration signals after resampled. Compared with the original signals, the stationarity of the signals in angle-domain is more improvement, and it can fit the request of the FFT.

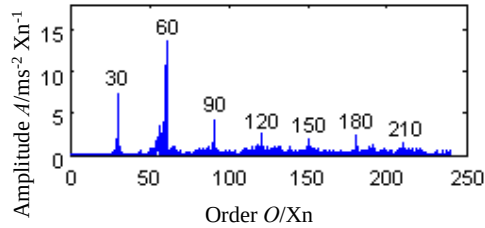


Figure 4. Order spectrum of the resampled vibration signals.

Fig. 4 is the order spectrum of the resampled vibration signals. It can be seen from the Fig. 4 that there are a series of apparent amplitudes in 30, 60, 90, 120, 150, 180, 210, and they difference corresponding the mesh order of drive gear 1,2,3,4,5,6,7 times. The mesh condition of the drive gear is reflected through the Fig. 4.

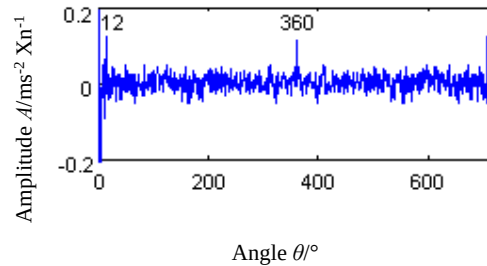


Figure 5. Order cepstrum of the resampled signals.

Fig. 5 is the order cepstrum of the resampled signals. It can be seen from the Fig. 5 that the order cepstrum of the gear is corresponded with 12° , and the tooth surface order cepstrum is corresponded with 360° . The fault in gear can be explained with the peak value in Fig. 5. It is according with the premise in the test, and the method can be diagnosed effectively in the gear fault.

Conclusions

- (1) During analyzing the speed-transforming transmission of gearbox, the “frequency smear” phenomenon can be avoided effectively with order tracking technology, which can’t be solved with the traditional frequency spectrum method and the non-stationary signals in time-domain are translated into stationary signals in angle-domain.
- (2) The signals after resampled in angle-domain are analyzed with order cepstrum; the working condition of the gearbox can be reflected by the outcome. It is trenchancy supplement for traditional spectrum analysis and has more amplitude appliance foreground.

References

- [1] Zhu Jimei. The analysis of Non-stationary vibration signals (serialize) [J]. Journal of vibration and shock, 2000,19 (2): 87-90.
- [2] Fyfe K.R., Munckeds. Analysis of computed order tracking [J]. Mechanical Systems and Signal Processing, 1997,11 (2): 187-205.
- [3] K.M. Bossley, R.J. Mckendrick. Hybrid computed order tracking [J]. Mechanical Systems and Signal Processing, 1999,13 (4): 627-641.
- [4] Jia Jide, Liu Zhigang, Zhang Ping. etc. Feature extraction and state monitoring of engine faults based on the order bi-spectrum estimation [J]. Transactions of CSICE, 2004, 22 (5): 462-469.

- [5] Guo Yu, Qin Shuren. Pseudo-speed tracking order analysis for non-stationary vibration signals of rotating machinery [J]. Journal of vibration and shock, 2004, 23 (1): 61-66.
- [6] Xu Minqiang, Huang Wenhui, Zhang Jiazhong. Application of Haar Wavelet on Analysis of Vibration Signal of Rotating Machinery in Fast Run-up State [J]. Journal of Vibration Engineering, 2000, 13 (2): 216-221.
- [7] Kang Haiying. Research on method of condition detection and fault diagnosis of self-propelled gun gearbox based on order tracking [D]. Shijiazhuang, Ordnance Engineering College, 2005.
- [8] Li Hui. Study on fault diagnosis of gearbox based on transient process [D]. Shijiazhuang, Ordnance Engineering College, 2005.

Research of Influence of Interface Roughness on the Stress Characteristics of Bolted Joints*

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Keywords: Bolted joint; Principal strain; Stress test; Joint surface roughness.

Abstract. Bolted joints stress is the important part of the assembly stress in machines, the uniformity of the assembly stress is the assurance of the precise assembly. This paper used an assembly stress test method based on strain test, besides, no need to damage the surface of the workpiece. aiming at the single bolted joint, the paper researched how joint surface roughness affects the bolted joint stress characteristics by the experiment. The result of the experiment indicates that the inside stress characteristics of the workpiece can be expressed accurately by the test points set in the side face of the workpiece, besides, the result validates that the roughness of the joint surface can make a big influence on the stress characteristics of the bolted joint, that with the increases of the roughness of the joint surface, the compression strain in the direction along the joint increases manifestly when preloaded by the same stress.

Introduction

Bolted joints is the very important part of the assembly in machines, the stress characteristics of the bolted joints makes a deep influence on the assembly precision of the machine. The homogeneity of the stress distribution of the bolt joints is the guarantee of the accuracy and reliability when the machine is under the interaction of vibration and heat. To make sure of the homogeneity of the stress distribution of the bolt joints, acquaintance of what is the main influence factor of the stress characteristics and how the factor makes influence is very important. Because of the difficulties of the accurate simulation and theoretical calculation, experimental test is the main research method of the stress characteristics of the bolted joints.

At present, stress testing method is mainly divided into the destructive and the non destructive. The destructive test methods in hole drilling method, cutting method, delamination method etc [1]. Because this kind of method is likely to destroy the original workpiece itself, so it does not apply to the situation when the workpiece can not be damaged. The nondestructive test method is also called the physical measurement, including X-ray method, neutron diffraction method, ultrasonic method, magnetic method, electrometric method [2]. Among the all methods, electrometric method has its unique advantages, such as cheapness, simpleness, accuracy, besides, it can be used in the working field and do not have too much demanding on the environment [3]. So it can meet the general need of the stress test.

The roughness of the interface is closely related with the abrasion resistance, fatigue strength, contact stiffness, vibration and noise of the machine, it has an important influence on the service life and reliability of mechanical products. This paper researches how the roughness of the interface influences the stress characteristics of the bolted joints. Using the surface roughness of single bolt connection as the only experimental variables, the paper researched how the stress characteristics of the bolted joints changes with the changes of the roughness of the interface. the result validates that the roughness of the joint surface can make a big influence on the stress characteristics of the bolted joint, that with the increases of the roughness of the joint surface, the compression strain in the direction along the joint increases manifestly when preloaded by the same stress.

*The National High Technology Research and Development Program (863 Program) (2012AA040703)

Test System and the Principle of Stress Testing

Test System

The test system as shown in Fig 1, the Strain gauges are pasted on the surface of the workpiece after treatment, strain changes are sampled in the bolt tightening process through the data acquisition system, and then stress state and distribution are analysed by computer through relevant procedures. The strain gauge used in the test is BE120-2BC(11), its resistance is 120 ohms, The sensitivity coefficient is 1.86-2.20, the fatigue life is 10000000, appropriate temperature range is -30°C--+80°C. The data acquisition system is DH3816N.

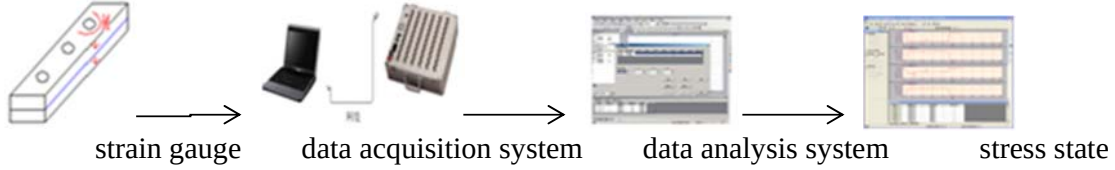


Figure 1. flow chart of test system.

Principle of Stress Testing

The points on the surface is in the two-dimensional state of stress, so after getting the strain of the 0°,45°,90° direction, a pair of vertical normal stress (ϵ_x, ϵ_y) and shear stress (γ_{xy}) can be figured out through Eq.1,

$$\begin{aligned}\epsilon_{\alpha 1} &= \frac{\epsilon_x + \epsilon_y}{2} + \frac{\epsilon_x - \epsilon_y}{2} \cos 2\alpha_1 - \frac{\gamma_{xy}}{2} \sin 2\alpha_1 \\ \epsilon_{\alpha 2} &= \frac{\epsilon_x + \epsilon_y}{2} + \frac{\epsilon_x - \epsilon_y}{2} \cos 2\alpha_2 - \frac{\gamma_{xy}}{2} \sin 2\alpha_2 \\ \epsilon_{\alpha 3} &= \frac{\epsilon_x + \epsilon_y}{2} + \frac{\epsilon_x - \epsilon_y}{2} \cos 2\alpha_3 - \frac{\gamma_{xy}}{2} \sin 2\alpha_3\end{aligned}\quad (1)$$

then the main strain (ϵ_1, ϵ_2) and its direction (α) of this point can be figured out through the Eq.2,3.

$$\tan 2\alpha = -\frac{\gamma_{xy}}{\epsilon_x - \epsilon_y} \quad (2)$$

$$\left. \begin{matrix} \epsilon_1 \\ \epsilon_2 \end{matrix} \right\} = \frac{\epsilon_x + \epsilon_y}{2} \pm \sqrt{\left(\frac{\epsilon_x - \epsilon_y}{2} \right)^2 + \left(\frac{\gamma_{xy}}{2} \right)^2} \quad (3)$$

So the main stress (δ_1, δ_2) can be figured out through the generalized Hooke's law.

$$\begin{aligned}\delta_1 &= \frac{E}{1 - \nu^2} (\epsilon_1 + \nu \epsilon_2) \\ \delta_2 &= \frac{E}{1 - \nu^2} (\epsilon_2 + \nu \epsilon_1)\end{aligned}\quad (4)$$

Research of Influence of Interface Roughness on the Stress Characteristics

The workpieces used in the test have the same size and material, but the roughness is different. The bolt model used in the test is M14X80 with full thread. The preload is loaded as 80NM-100NM.

Fig. 2 and Fig. 3 show the experiment site and the size of the workpiece.



Figure 2. experiment site.

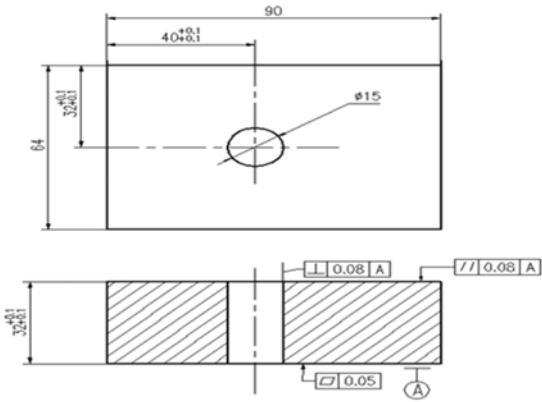


Figure 3. size of the workpiece.

The workpieces with different roughness were connected together, and five groups of tests were set as shown in the following chart.

Table 1. the roughness of each group.

Test group	Group 1	Group2	Group 3	Group 4	Group 5
Ra (μm)	0.21+0.21	0.21+0.43	0.43+0.43	0.43+3.14	3.14+3.14

The test points were shown in Fig.4, test points were set symmetrically and 5cm away from the combined surface.

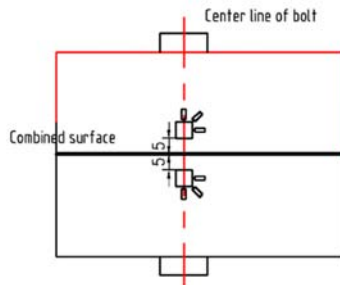


Figure 4. location of the test points.

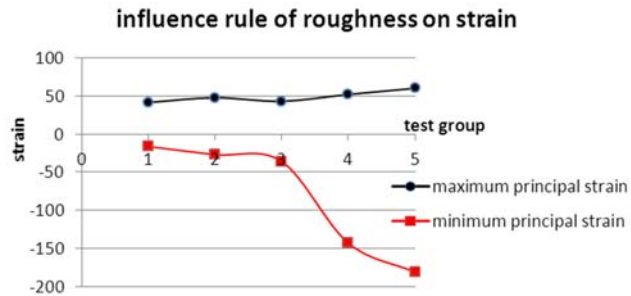


Figure 5. influence rule of roughness on strain.

The result of the tests is shown in Fig. 5, as shown in the Fig, the maximum principal strain changes little with the increase of roughness, while the minimum principal strain increases manifestly when preloaded by the same stress with the increase of the roughness. Besides, the direction of the maximum principal strain is perpendicular to the bolt and the direction of minimum principal strain is along with the bolt.

The contact state of the mechanical joints can be simplified as the contact of a rough surface and a rigid smooth one. On the base of Mandelbrot's research, Majumdar and Bhushan [4] assumed that the distribution density function of the contact points area is similar with that of the sea islands area. And on the base of the assumption, the contact area of the mechanical joints before loading calculated according to the fractal contact model is kept constant as the roughness changing.

$$A = \int_0^{a_1} n(a) da = \frac{D}{2-D} a_1 \quad (5)$$

where D is the fractal dimension, and $1 < D < 2$, a_1 is the maximum area of the hypothetical contact points, a is the contact area of the contact point, $n(a)$ is the distribution of contact point's area

dimensions. The contact surface contour curve can be represented the Weierstrass-Mandelbrot function (simplified as W-M function)[5,6] and the expression can be describe as:

$$z(x) = G^{(D-1)} \sum_{n=1}^{\infty} \gamma^{-(2-D)n} \cos(2\pi\gamma^n x) \quad (6)$$

where $z(x)$ is the height of the contact surface contour curve, x is the contour surface displacement coordinates, G is characterization of the surface roughness. G is larger as the surface is rougher, and the height of the surface contour is larger; γ is a constant and $\gamma > 1$. W-M function model suggested that the amplitude of the contact surface contour curve is different as the roughness changing. The greater the roughness, the greater the corresponding amplitude, and as a result, under the same pressure and the initial contact area, the peak of the contact point must produce a larger strain to make the contact point where the amplitude is lower contact. Therefore, as the pre-tightening force is the same, the roughness is large, the contact points are few, and the strain of the contact points becomes larger. Therefore, the strain of each point along the direction of the bolt increases as the roughness of the joint increasing.

Summary

The result validates that the roughness of the joint surface can make a big influence on the stress characteristics of the bolted joint, that with the increases of the roughness of the joint surface, the compression strain in the direction along the joint increases manifestly when preloaded by the same stress.

References

- [1] Wang Qingming, Sun yuan, Developments and trends of the test technique of residual stress, J. Mechanical & Electrical Engineering 28.1 (2011): 11-15.
- [2] Shen Jun, Lin Bo, Chi Yonggang, etc, The research status of physical measurement technology of residential stress, J. Materials Review, S1 (2012): 120-125. .
- [3]Yan Haokui, Ren Jian Guo,The working principle of Resistance strain gauge, J. Metrology & Measurement Technique, 4 (2013): 12-12.
- [4]Majumdar A. and B. Bhushan, Fractal model of elastic-plastic contact between rough surfaces, J. Journal of Tribology 113.1 (1991): 1-11.
- [5]Majumdar A. and C.L. Tien, Fractal characterization and simulation of rough surfaces, J. Wear 136.2 (1990): 313-327.
- [6]Ganti Suryaprakash and Bharat Bhushan, Generalized fractal analysis and its applications to engineering surfaces, J. Wear 180.1 (1995): 17-34.

Quadrotor Waypoint Navigation and Trajectory Tracking Using Integral Backstepping Technique

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Keywords: Quadrotor, Waypoint navigation, Trajectory tracking, Integral backstepping.

Abstract. Researchers have given much attention on quadrotor due to its potential military and civilian application. Quadrotor is a system which has complex dynamics, nonlinearities and high degree of coupling between control inputs and state variables. To confront these problems, this paper developed an integral backstepping controller for quadrotor waypoint navigation and trajectory tracking control, and the controller can set quadrotor track desired positions and yaw angle while stabilize pitch and roll angles. Simulations have been done on waypoint navigation and trajectory tracking and simulation results show that integral backstepping controller got good performance.

Introduction

The past several years has seen significant growth on controlling of quadrotors though there are many theoretical and technical challenges on the design of quadrotor controller. In terms of quadrotor controllers, it is universal between indoor and outdoor quadrotors, which has two main directions: linear controllers and nonlinear controllers. Proportional integral derivative (PID) and linear quadratic regulator (LQR) controllers are representatives of linear controllers [1]. Linear controller, especially PID controller, can get good results in experiments with less disturbances and small angles. Whereas linear controllers fail to finish aggressive maneuver experiments since linearized dynamics model is no longer suitable for big angles. To overcoming limitations and drawbacks of linear controllers, researchers begin to focus on nonlinear controllers. Feedback linearization, model predictive control, dynamics inversion, adaptive control, sliding mode, nested saturation-based nonlinear controller, and quaternion-based nonlinear controller have received much attention and showed great promising [2-5].

This paper is focusing on integral backstepping controller based on nonlinear dynamics. After dynamics modeling, integral backstepping control law is derived for waypoint navigation and trajectory tracking. In waypoint navigation simulation, a simple path planning algorithm is proposed to get a time parameterized polynomial between two points, which satisfies initial and final velocities and accelerations are zero. Trajectory generation of trajectory tracking was conducted in two steps. Firstly, the quadrotor flies to 20m high and then began a spiral trajectory tracking. Same path planning method was utilized in the first step to get desired position and velocities. In second step, adopted trajectory is a continuous curve. Therefore, desired velocities can get from first derivative of position instead of from polynomial path planning method. Good results are obtained from simulations.

In the next section quadrotor dynamics modeling is discussed. Then in Section III integral backstepping control law is developed. Simulation results are described in Section IV. Finally, in Section V, conclusion and discussion are presented.

Quadrotor Dynamics Modeling

Quadrotor dynamics model is usually obtained by two different approaches: Euler-Lagrange and Newton-Euler. And these two approaches can get same motion equations. This paper's dynamics model referred to project OS4's dynamics model [6] which considered many nonlinear effect factors

such as gyroscopic torque, angular acceleration of blades, and drag force on the vehicle but neglected blade flapping and angle attack effects. The equations of motion and system inputs are described as follows:

$$f(X, U) = \begin{bmatrix} x_2 \\ x_4 x_6 (I_y - I_z) / I_x - x_4 J_r \Omega / I_x + U_2 l / I_x \\ x_4 \\ x_2 x_6 (I_z - I_x) / I_y + x_2 J_r \Omega / I_y + U_3 l / I_y \\ x_6 \\ x_2 x_4 (I_x - I_y) / I_z + U_4 l / I_z \\ x_8 \\ -g + (\cos x_1 \cos x_3) \frac{1}{m} U_1 \\ x_{10} \\ \frac{U_1}{m} (\cos \phi \sin \theta \cos \psi + \sin \phi \sin \psi) \\ x_{12} \\ \frac{U_1}{m} (\cos \phi \sin \theta \sin \psi - \sin \phi \sin \psi) \end{bmatrix}, \text{ with } \begin{array}{l|l} x_1 = \phi & x_7 = z \\ x_2 = \dot{x}_1 = \dot{\phi} & x_8 = \dot{x}_7 = \dot{z} \\ x_3 = \theta & x_9 = x \\ x_4 = \dot{x}_3 = \dot{\theta} & x_{10} = \dot{x}_9 = \dot{x} \\ x_5 = \psi & x_{11} = y \\ x_6 = \dot{x}_5 = \dot{\psi} & x_{12} = \dot{x}_{11} = \dot{y} \end{array} \quad (1)$$

Integral Backstepping Controller

The main objective of this section is to design an integral backstepping controller that ensures a quadrotor to track desired trajectory and following desired point, asymptotically.

Integral Backstepping for Attitude Control. In this section, one example is presented to get control law of angular rotations. To start our design, roll angle tracking error is defined as $e_1 = \phi_{ref} - \phi$ and compute its dynamics as:

$$\frac{de_1}{dt} = \dot{\phi}_{ref} - w. \quad (2)$$

One can get exponential convergence for the system by choosing w as $w = q_1 e_1 + \dot{\phi}_{ref}$ if the angular velocity w were control input. It only can be treat as virtual input and the desirable behavior for this virtual control input is

$$w_{ref} = q_1 e_1 + \dot{\phi}_{ref} + \lambda_1 \chi_1, \quad (3)$$

where λ_1 is a positive constant at the disposal of the designer and $\chi_1 = \int_0^t e_1(\tau) d\tau$ is the integral of the roll angle tracking error. And its dynamics can be computed as:

$$\frac{de_2}{dt} = q_1 (\dot{\phi}_{ref} - w) + \ddot{\phi}_{ref} + \lambda_1 e_1 - \ddot{\phi}, \quad (4)$$

where e_2 , the roll angular velocity tracking error is defined as $e_2 = w_{ref} - w$. With the definition for e_2 , the roll tracking error can be rewritten as:

$$\frac{de_1}{dt} = -q_1 e_1 - \lambda_1 \chi_1 + e_2. \quad (5)$$

By replacing $\ddot{\phi}$ in Eq. 4 by its corresponding expression from dynamics model, the control input U_2 appears in the following equation Eq. 6.

$$\frac{de_2}{dt} = q_1 (\dot{\phi}_{ref} - w) + \ddot{\phi}_{ref} + \lambda_1 e_1 - \dot{\psi} \left(\frac{I_y - I_z}{I_x} \right) - \frac{J_r}{I_x} \dot{\Omega} + \frac{l}{I_x} U_2. \quad (6)$$

Then one can use Eq. 3 and Eq. 5 to combine the tracking errors of roll angle e_1 , the angular velocity tracking error e_2 and the integral tracking error λ_1 and obtain angular velocity tracking error as:

$$\frac{de_2}{dt} = q_1(-q_1 e_1 - \lambda_1 \chi_1 + e_2) + \ddot{\phi}_{ref} + \lambda_1 e_1 - \dot{\theta} \dot{\psi} \left(\frac{I_y - I_z}{I_x} \right) - \frac{J_r}{I_x} \dot{\Omega} + \frac{l}{I_x} U_2. \quad (7)$$

To cancel some undesirable dynamics and generate the desirable dynamics of the angular velocity tracking error $\frac{de_2}{dt} = -q_2 e_2 - e_1$, one can choose the control input U_2 as:

$$U_2 = \frac{1}{b_1} \left[(1 - q_1^2 + \lambda_1) e_1 + (q_1 + q_2) e_2 - q_1 \lambda_1 \chi_1 + \ddot{\phi}_{ref} - \dot{\theta} \dot{\psi} \left(\frac{I_y - I_z}{I_x} \right) - \frac{J_r}{I_x} \dot{\Omega} + \frac{l}{I_x} U_2 \right], \quad (8)$$

where q_2 is positive constant which determines the convergence speed of the angular velocity tracking loop.

So far we have synthesized the control law for control input U_2 . To analyze the stability and asymptotic convergence performance of the proposed control law, the following Lyapunov function is chose for the closed-loop system.

$$V = \frac{\lambda_1}{2} \chi_1^2 + \frac{1}{2} e_1^2 + \frac{1}{2} e_2^2, \quad (9)$$

where the tracking errors of roll angle e_1 , the angular velocity tracking error e_2 and the integral action λ_1 are all included. One can obtain the derivative of Lyapunov function as:

$$\dot{V} = -q_1 e_1^2 - q_2 e_2^2 \leq 0. \quad (10)$$

From Eq. 9 and Eq. 10, we can conclude that the definition of Lyapunov function guarantee the global boundedness of the tracking errors of roll angle e_1 , the angular velocity tracking error e_2 and the integral action λ_1 . Using the same method, one can obtain control law for inputs U_3 and U_4 :

$$\begin{cases} U_3 = \frac{1}{b_2} [(1 - q_3^2 + \lambda_2) e_3 + (q_3 + q_4) e_4 - q_3 \lambda_2 \chi_2 \\ \quad + \ddot{\theta}_d - \dot{\phi} \dot{\psi} a_3 + \dot{\phi} a_4 \Omega] \\ U_4 = \frac{1}{b_3} [(1 - q_5^2 + \lambda_3) e_5 + (q_5 + q_6) e_6 - q_5 \lambda_3 \chi_3] \end{cases} \quad \text{with} \quad \begin{cases} e_3 = \theta_{ref} - \theta \\ e_4 = c_3 e_3 + \dot{\theta}_{ref} + \lambda_2 \chi_2 - \dot{\theta} \\ e_5 = \psi_{ref} - \psi \\ e_6 = c_5 e_5 + \dot{\psi}_{ref} + \lambda_3 \chi_3 - \dot{\psi} \end{cases}. \quad (11)$$

Backstepping for Position Control. Position control law is derived using the same method but small angle approximation was applied to linearize the horizontal motion dynamics of quadrotor, which simplify horizontal motion dynamics to

$$\begin{bmatrix} m\ddot{x} \\ m\ddot{y} \end{bmatrix} = \begin{bmatrix} -\theta U_1 \\ \phi U_1 \end{bmatrix}. \quad (12)$$

With the new horizontal motion dynamics and integral backstepping technique, one can derive position control law as:

$$\begin{cases} U_1 = \frac{m}{\cos \phi \cos \theta} [g + (1 - q_7^2 + \lambda_4) e_7 + (q_7 + q_8) e_8 - q_7 \lambda_4 \chi_4] \\ U_x = \frac{m}{U_1} [(1 - q_9^2 + \lambda_5) e_9 + (q_9 + q_{10}) e_{10} - q_9 \lambda_5 \chi_5] \\ U_y = -\frac{m}{U_1} [(1 - q_{11}^2 + \lambda_6) e_{11} + (q_{11} + q_{12}) e_{12} - q_{11} \lambda_6 \chi_6] \end{cases} \quad \text{with} \quad \begin{cases} e_7 = z_{ref} - z \\ e_8 = c_7 e_7 + \dot{z}_{ref} + \lambda_4 \chi_4 - \dot{z} \\ e_9 = x_{ref} - x \\ e_{10} = c_9 e_9 + \dot{x}_{ref} + \lambda_5 \chi_5 - \dot{x} \\ e_{11} = y_{ref} - y \\ e_{12} = c_{11} e_{11} + \dot{y}_{ref} + \lambda_6 \chi_6 - \dot{y} \end{cases}. \quad (13)$$

Simulation Results

Waypoint Navigation Simulation. Fig. 1 describes waypoint navigation simulation. In this test, the quadrotor is commanded to perform automatic takeoff, waypoint navigation, and stationary flight at 20m high, and automatic landing. The flight path is defined by 5 points and the reference trajectories are generated in real time. The simulation trajectory exactly matches desired trajectory and tracking error is almost zero. Fig. 2 shows the attitude tracking errors in radian is small. These results demonstrate that the quadrotor passed successfully through all the points.

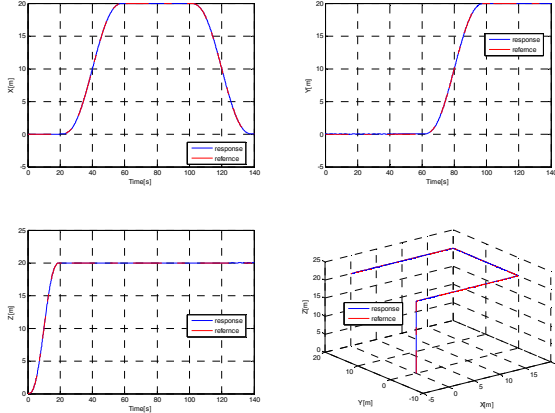


Figure 1. Autonomous waypoint navigation simulation.

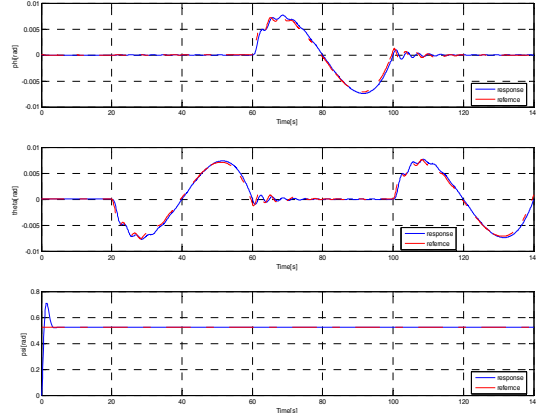


Figure 2. Angles response of waypoint navigation simulation. The yaw is $\pi/6$ given by the operator.

Trajectory Tracking Simulation. In this test, the quadrotor firstly fly to 20m high, and then autonomously execute a spiral trajectory tracking. As Fig. 3 shown, one can see that the quadrotor tracks successfully the reference trajectories including the height trajectory for automatic takeoff and altitude holding. Attitude control results, shown in Fig. 4, also confirm good performance of rotational controller.

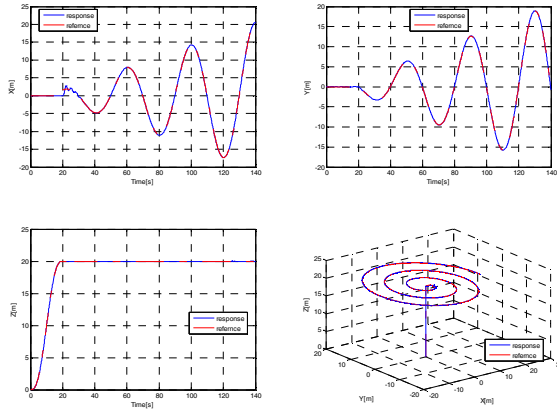


Figure 3. Autonomous trajectory tracking simulation. The trajectory is a spiral.

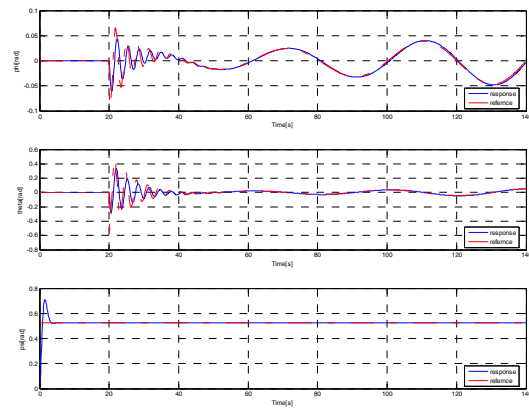


Figure 4. Angles response of trajectory tracking simulation.

Conclusion

In this paper, we have described the design of integral backstepping controller of a quadrotor that can fulfill waypoint navigation and trajectory tracking. Nonlinear dynamics model of quadrotor has been chosen and considered most nonlinear effect factors except for blade flapping. Simulations have been done to demonstrate the ability of integral backstepping controller to provide effective waypoint

navigation and accurate trajectory tracking. From the simulation results, we conclude that designed integral backstepping controller shows good and reliable performance.

Acknowledgement

This research was supported by Basic Science Research Program through the National Research Foundation of Korea (NRF) funded by the Ministry of Science, ICT & Future Planning (NRF-2013R1A2A2A01068127).

References

- [1] S. Bouabdallah, A. Noth and R. Siegwart, "PID vs LQ control techniques applied to an indoor micro quadrotor," in *Intelligent Robots and Systems, 2004.(IROS 2004). Proceedings. 2004 IEEE/RSJ International Conference on*, 2004, pp. 2451-2456.
- [2] D. Pebrianti, F. Kendoul, S. Azrad, W. Wang and K. Nonami, "Autonomous hovering and landing of a quad-rotor micro aerial vehicle by means of on ground stereo vision system," *Journal of System Design and Dynamics*, vol. 4, pp. 269-284, 2010.
- [3] S.L. Waslander, G.M. Hoffmann, J.S. Jang and C.J. Tomlin, "Multi-agent quadrotor testbed control design: Integral sliding mode vs. reinforcement learning," in *Intelligent Robots and Systems, 2005. (IROS 2005). 2005 IEEE/RSJ International Conference on*, 2005, pp. 3712-3717.
- [4] A. Das, K. Subbarao and F. Lewis, "Dynamic inversion of quadrotor with zero-dynamics stabilization," in *Control Applications, 2008. CCA 2008. IEEE International Conference on*, 2008, pp. 1189-1194.
- [5] R. Xu and Ü. Özgüner, "Sliding mode control of a class of underactuated systems," *Automatica*, vol. 44, pp. 233-241, 2008.
- [6] S. Bouabdallah and R. Siegwart, "Backstepping and sliding-mode techniques applied to an indoor micro quadrotor," in *Robotics and Automation, 2005. ICRA 2005. Proceedings of the 2005 IEEE International Conference on*, 2005, pp. 2247-2252.

Research on Cabin Safety Risk Evaluation System

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Keywords: Cabin Safety, Risk Evaluation System, Comprehensive Evaluation, Risk Management.

Abstract. Through theoretical analysis, expert survey and issuance of the questionnaire to cabin crew, factors relevant to cabin safety are summarized. According to the comprehensive analysis of the cabin safety influential factors, a cabin safety risk comprehensive evaluating system is established. Then, order relation analysis method (G1 method) is used to calculate the weight of the risk factors. By expert questionnaire and scoring by experts, we can calculate the possibility of risk factors. Thus, the cabin safety risk value can be calculated. Then we can develop measures to control safety risk.

Introduction

Cabin safety is one of the important parts of flight safety [1,2]. With the growth of civil aviation transportation, the eternal theme of the development of the civil aviation still is the issue of aviation safety. Aviation safety management system has also been developed from the initial mechanical period to the current system security management period. In the system security management period, the most significant feature is the introduction of safety management system (SMS) [3]. This "toolbox" engaged the safety management advances from open-loop to closed-loop, from the individual to the organization, from local to major changes.

Because of our cabin SMS still in its infancy, the research on this area is still limited and not much experience can learn from. Cabin SMS encountered a series of problems in the process of implementation. This article finds out the problems and analyzes them and presents the improving suggestion of cabin SMS. The purpose is to improve cabin safety management level and conduct the cabin SMS to better and faster development, and to provide a reference for airlines.

This article uses literature method, theoretical research method, questionnaire method, order relation analysis method and so on.

Cabin Safety Analysis

Aircraft cabin safety system is a complex system, including the crew members, maintenance personnel and passengers. Cabin safety relates to cabin equipment protection, operation procedures, Airlines cabin safety management etc.

Combing tasks and goals of the cabin safety management, analyzing work characteristics of cabin department and operation, and reviewing all kinds of unsafe events happened in the past, we can get a series of factors affecting cabin safety.

Risk Management Process

Risk management is an important part of safety management. The risk management of the cabin is to identify, analyze and eliminate the danger and threat of cabin safety risk. The risk management has 3 main components: the risk identification, risk assessment and control of the cabin risk, the process is shown in Fig. 1.

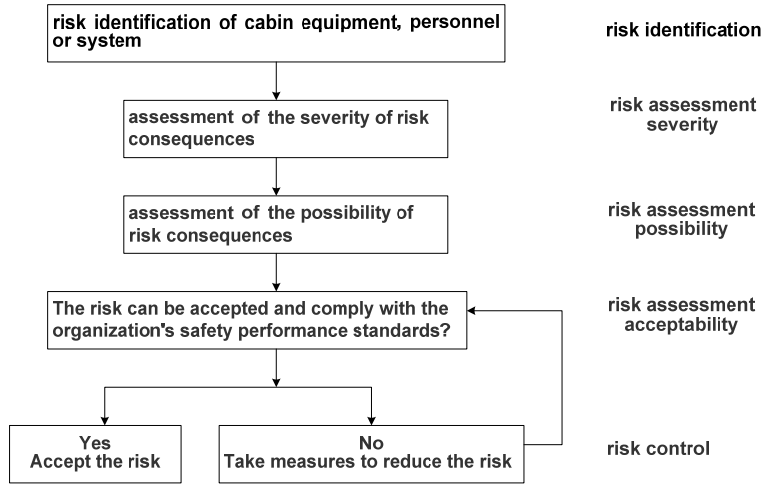


Figure 1. Risk management process.

Cabin Risk Identification. Cabin safety depends on a series of complex factors. According to expert and cabin crew questionnaire, also analyzing the past cabin incidents, there are nine factors affecting cabin safety. These nine factors are: (1) The qualified operation and management personnel; (2) The implementation condition of personnel; (3) The psychological and physiological state of personnel; (4) Route features; (5) The objective environment; (6) Equipment check; (7) The use of equipment; (8) Equipment management; (9) Manual and program management. According to the nine factors, further there are some detail risk factors. Table 2 lists all the factors.

Cabin Risk Assessment

Weight of Cabin Risk Factors. According to Table 1 a cabin risk comprehensive evaluating system is established. Then by expert and cabin crew questionnaire, we can use order relation analysis method (G1 method) to calculate the weight of the risk factors. How to determine the weight coefficient, is the core problem in comprehensive evaluation. Order relation analysis method (G1 method) is an improved method based on AHP method, which is proposed by Professor Guo Yajun from Northeastern University [4]. This method overcomes the defect of AHP method. This method can fully reflect every step of the experts, and it is a clear, simple and practical method. This method does not require the judgment matrix, and there is no need to do consistency check.

Step 1, to determine the order relationship: If the evaluation index relative to the degree of importance of an evaluation criteria is greater than (or less than), is recorded as. About an evaluation criteria, if the degree of importance of the evaluation index x_i is greater than (or less than) x_j , then we record as $x_i > x_j$.

If the evaluation index $x_1, x_2, \dots, x_m$, according to some evaluation criteria with relation equation $x_1^* > x_2^* > \dots > x_m^*$, then we call the evaluation index $x_1, x_2, \dots, x_m$, have established order relation based on “>”.

For the evaluation index set $\{x_1, x_2, \dots, x_m\}$, experts (or decision makers) select from it the most important index (based on an evaluation criteria) (choose only one index), and the selected indexes will be marked as $x_1^*, x_2^*, \dots, x_m^*$, then the only order relation is determined. For convenience, in the following we still record as $x_1^* > x_2^* > \dots > x_m^*$. Due to the influence of subjective factors, different experts will determine different order relations.

Step 2, to give the compared judgment for the relative importance of x_k and x_{k-1} . For some problems, it is not enough only to give the order relation. In order to determine the weight coefficient of the evaluation index with respect to an evaluation criteria (or target), we need to determine the

relative importance degree. A rational judgment by an expert about the relative importance ratio of evaluation index x_k and x_{k-1} is:

$$\omega_{k-1} / \omega_k = r_k \quad k = m, m-1, m-2, \dots, 3, 2 \quad (1)$$

Refer to table 1, we can get the value of r_k .

Step 3, to calculate the weight coefficient: if the rational assignment by the expert (or decision makers) meets the requirements of Table 1, then ω_k will be,

$$\omega_k = \left[1 + \sum_{k=2}^m \prod_{i=k}^m r_i \right]^{-1} \quad (2)$$

$$\omega_{k-1} = r_k \cdot \omega_k$$

Table 1. Reference table about the value of r_k .

r_k	importance degree
1.0	x_{k-1} and x_k are equally important.
1.2	x_{k-1} is more important than x_k .
1.4	x_{k-1} The index is more important than x_k .
1.6	x_{k-1} is Strongly more important than x_k .
1.8	x_{k-1} is extremely more important than x_k .

According to the cabin risk evaluation index system, based on the G1 method we can use expert questionnaire, to give the importance evaluation results about the two adjacent indexes. In order to ensure the reliability of the weight of the index system, some experts are from the airline cabin safety distribution and the other experts are cabin safety inspector. These experts have been engaged in flight and cabin management work over the years. 12 experts have been consulted, and 11 experts gave the effective results.

According to results from the experts, the index weights of two layers can be determined by using G1 method. The index weight calculation results are in Table 2.

To Determine the Possibility of Risk Factors. The possibility of risk can be divided into 5 levels [5], respectively, for the 5 levels value interval has been given in Table 3.

Table 3. possibility table for the cabin risk.

Possibility level	possibility description	value interval
5	Frequent	0.85-1.00
4	Accidental	0.65-0.85
3	infrequent	0.35-0.65
2	impossible	0.15-0.35
1	extremely unlikely	0.00-0.15

By expert questionnaire, based on Table 3, experts give the possibility value of the standard risk factors. Refer to table 2.

Table 2. Factors and risk value.

item	risk factors	Severity	possibility	risk value
1	The familiarity level of cabin crew manuals, standards and duty.	0.05257	0.73636	0.03871
2	The purser leadership, knowledge and experience.	0.05330	0.70455	0.03755
3	Safety awareness and attitude of cabin crew.	0.05066	0.67727	0.03431
4	Route weather, such as turbulence probability and frequency and intensity.	0.04435	0.76364	0.03387
5	Timeliness of the issue and emendment for the manual and program.	0.05148	0.63182	0.03253
6	The implementation of training program. (education and training implementation)	0.05516	0.57273	0.03159
7	The implementation of CRM.	0.04563	0.67727	0.03090
8	The experience of the cabin duty personnel and management personnel, to support safe operation.	0.04681	0.64091	0.03000
9	Operation fatigue or extra work burden.	0.03765	0.79455	0.02992
10	The dispatch, personnel arrangement and certificates management.	0.04380	0.62273	0.02727
11	Use of all kinds of service equipment, especially oven and other electrical equipment.	0.03936	0.66818	0.02630
12	Standard use of all kinds of the emergency equipment.	0.04408	0.53182	0.02344
13	The rationality of service program and impact on safety procedures.	0.03105	0.73182	0.02272
14	The characteristics of landing airport, such as high altitude airport or busy airport.	0.04094	0.54545	0.02233
15	Check for the number, location and function of all kinds of safety and emergency equipment, to give failure reports.	0.03844	0.57273	0.02202
16	Regularly check for effectiveness of toilet fire protection equipment during the flight.	0.03781	0.56364	0.02131
17	The rationality of company regulations on complaints, and the effects for cabin crew safety duties.	0.02790	0.72273	0.02016
18	All kinds of flight delays.	0.02192	0.69545	0.01525
19	Timeliness, integrity and unity for information transfer between The flight crew and cabin crew.	0.02584	0.55455	0.01433
20	The quality of preparation coordination meeting among crew members.	0.02212	0.63182	0.01398
21	The timeliness and effectiveness of the emergency disposal by cabin crew.	0.02747	0.47273	0.01298
22	The supervision for passenger luggage in the process of boarding and flight. (Attention to lithium battery)	0.02173	0.58636	0.01274
23	The effective monitoring of the cabin, cabin door, kitchen and toilet by cabin crew.	0.01519	0.66818	0.01015
24	Standard operation of slide separator by cabin crew.	0.01628	0.58182	0.00947

25	Effective control of large and heavy luggage.	0.01233	0.67636	0.00834
26	Implementation of safety briefing and special passenger briefing before takeoff.	0.01543	0.51364	0.00792
27	Confirm for passengers qualification about safety emergency exit.	0.01473	0.53182	0.00783
28	The whole monitoring for equipment and other dangerous equipment such as axe.	0.01771	0.37727	0.00668
29	The full supervision of goods in the cabin.	0.01846	0.33636	0.00621
30	The timeliness and effectiveness for Treatment of sick passengers by cabin crew.	0.01045	0.57273	0.00598
31	During flight cabin crew injure passengers accidentally.	0.00967	0.58182	0.00562
32	Effective monitoring of cabin safety during refueling.	0.00966	0.51818	0.00501

To Determine the Risk Value of Risk Factors. The weight of risk factors and possibility value risk are included in Table 2, then the risk value of each risk factor is calculated. The risk value is equal to the possibility value multiplied by the severity value of the risk. The weight is based on the relative importance determined by experts about the relative compared severity of factors.

The Control of Risk Factors. Cabin risk control is an active method. Before the accident, to analyze various dangers, to take preventive control measures in advance, and to ensure the safe operation of the cabin. According to the risk values of risk factors, for each specific operation, to analyze the risk control measures in advance, and then the accidents can be prevented.

Summary

After identification of the cabin risk factors, to evaluate the risk factors identified by qualitative analysis and quantitative analysis. Based on the qualitative method of expert analysis, the probability and severity of risk factors can be evaluated. According to the specific flight operation, about the risk factors with larger risk values, effective risk control measures can be proposed.

References

- [1] Lu Wang, To eliminate the cabin safety dangers. Air Transport & Business. 245 (2009) 8-10.
- [2] Wenjun Chen, The problems in cabin safety management and the suggestions, Safety & Security, 145 (2013) 31-33.
- [3] Xiaoning Sun, Tianmei Feng, Airline cabin safety audit, Safety & Security, 128 (2011) 54-56.
- [4] Yajun Guo, The principle and application of comprehensive evaluation method, (2007).
- [5] Yonggang Wang, Study on the risk management of airport apron, Journal of Safety and Environment, 7 (2007) 131-133.

Using Different Methods Estimate the Tensile Strain at the Bottom of the HMA Layer

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Keywords: FWD, TSD, Pavement, Nondestructive testing.

Abstract. In this paper, we compare the calculated tensile strain at the bottom of the asphalt layer obtained from FWD deflection measurements and TSD deflection slope measurements. The results here suggest that further investigation of the causes that lead to the TSD measuring higher strain levels is needed. Although the applied load was of the same magnitude (40 kN), the dynamics of how the load is applied in each case are much different. In the case of the FWD, the load is dropped onto the pavement whereas for the TSD the load is applied by a moving wheel.

Introduction

Measurement of the existing structural capacity of a pavement is a critical input structural analysis of in service pavements. State Departments of Transportation routinely use deflection measurements at the project level, using mainly the Falling Weight Deflectometer (FWD), and some are starting to use them also at the network level^[1]. Although the FWD provides a very useful tool to assess the pavement structural or bearing capacity of a pavement, and can be used to determine the moduli of the component layers, this technology has the limitation of only allowing stationary measurements at discrete points along the pavement sections. The need to stand on the road for a short period of time disturbs traffic and requires traffic control.

As part of the SHRP2 R06 (F) project, the Traffic Speed Deflectometer (TSD) was selected for further evaluation. Part of this evaluation included comparison of the TSD with the FWD. The Surface Curvature Index (SCI) and Base Damage Index (BDI) were two of the chosen parameters used in the comparison. The SCI and BDI were chosen because they can be calculated from measurements obtained from either the FWD or the TSD and because the two parameters have been shown to be good predictors of the tensile strain at the bottom of the HMA layer^[2]. In this paper, we compare the calculated tensile strain at the bottom of the asphalt layer obtained from FWD deflection measurements and TSD deflection slope measurements.

Falling Weight Deflectometer (FWD)

The FWD applies a load pulse that simulates the load of a rolling vehicle wheel to the pavement. The load is produced by dropping a weight onto a circular plate (generally of 300 mm diameter) placed on the pavement surface. The applied load and vertical pavement deflection at various radial distances (e.g. 0 mm, 200 mm, 300 mm, 450 mm, 600 mm, 900 mm, 1200 mm, and 1500 mm) from the center of the load are measured and recorded.

The collected data obtained from FWD testing is used to determine the structural condition of the pavement.

Traffic Speed Deflectometer (TSD)

The TSD (Fig. 1) is mounted on an articulated truck with a rear axle load of 100 kN (22 kips) which, in the model evaluated, utilizes four Doppler lasers mounted on a servo-hydraulic beam to record the vertical deflection velocity of a loaded pavement under one of the dual wheel assemblies. Three Doppler lasers are positioned such that they measure deflection velocity at a range of distances

in front of the rear axle. The fourth sensor is positioned 3.6 m in front of the rear axle largely outside the deflection bowl, acting as a reference laser. The beam on which the lasers are mounted moves up and down in opposition to the movement of the trailer in order to keep the lasers at a constant height from the pavement surface. To prevent thermal distortion of the steel measurement beam, a climate control system maintains the trailer temperature at a constant 20 °C. Two prototypes had been developed at the time of the evaluation by the manufacturer Greenwood Engineering A/S of Denmark.



Figure 1. Computer render of the TSD, with the TSD in operation (inset).

One is owned and operated by the Danish Road Institute (DRI) and the other is owned by the UK Highways Agency (HA) and operated on their behalf by the UK Transport Research Laboratory. Newer production devices have incorporated more Doppler laser sensors ^[3].

The lasers are mounted at a small angle to measure the horizontal vehicle velocity, the vertical and horizontal vehicle suspension velocity, and the vertical pavement deflection velocity. Due to its location, mid-way between the loaded trailer axle and the rear axle of the tractor unit, the reference laser is expected to measure very little vertical pavement deflection velocity, and its response can therefore be used to remove the unwanted signals from the three measurement lasers. When accurately calibrated, the TSD produces measurements of deflection velocity that depend on driving speed. To remove this dependence, the deflection velocity is divided by the instantaneous survey speed to give a measurement of deflection slope. Deflection velocity is measured in mm/s whilst survey speed is measured in m/s; therefore, deflection slope measurements are output in units of mm/m ^[4].

Calculating the SCI and BDI

The calculated SCI is SCI300 which is defined as $d_0 - d_{300}$, while BDI is defined as $d_{300} - d_{600}$, where d_0 , d_{300} and d_{600} are the deflections at 0 mm, 300 mm, and 600 mm from the applied load, respectively. For the FWD, d_0 , d_{300} , and d_{600} , are directly measured and calculation of the SCI and BDI is straightforward. In the following section, we present the methodology used to calculate the SCI and BDI from TSD slope measurements.

The TSD measures the slope of the deflection bowl. Therefore, the deflection can in principle be obtained from the TSD slope by integration if a sufficiently detailed representation of the full deflection slope bowl is available. Integration is however only specified up to a constant value and therefore we cannot recover the deflection without a reference deflection measurement. We can however obtain the difference between two deflection readings (the constant cancels out) which gives the SCI or BDI. The relationship between slope, deflections and SCI and BDI is presented in Eq. (1).

$$\int_a^b s(x)dx = d(b) - d(a) = SCI \quad (1)$$

where, $s(x)$ is the slope at location x and $d(x)$ = deflection at location x . To calculate the SCI from TSD measurements, the TSD slope was integrated numerically. To calculate SCI, the integration interval is [0mm; 300 mm] while for the BDI the integration interval is [300 mm; 600 mm]. TSD slope measurements were obtained at 100, 300 and 756 mm from the applied wheel load and the deflection slope at 0 mm was assumed to be equal to zero. This assumption is valid if the load is

uniformly (or approximately uniformly) applied over a specific area rather than being a point load, which is the case for a wheel load, and if viscoelastic effects are neglected. Note that viscoelastic effects will cause the response to the applied load to be slightly delayed so that the slope will actually be zero at a location slightly behind the applied wheel load. However, the calculation of SCI is not significantly affected by the location where the deflection slope is set to zero.

Results

To demonstrate the applicability of the methods described earlier, the strain at the bottom of the asphalt layer was estimated for a tested pavement section approximately 2.2 km long. Fig.2 shows the TSD deflection slope measured 100 mm from the applied wheel load and the FWD d0 both at 20 m intervals. Although the measurements from each device have different units, they both follow the same trends. The figure shows that like the FWD, the TSD was able to capture the structural variation in the pavement sections.

The comparison of the SCI calculated from data gathered by the TSD and FWD are presented in Fig.3.

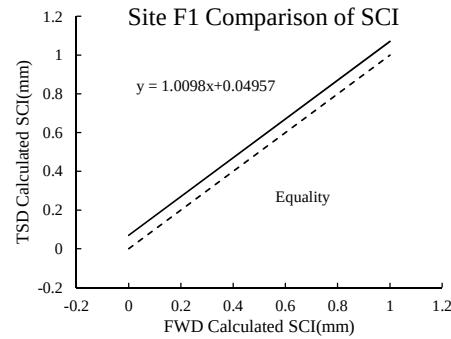


Figure 2. TSD calculated SCI versus FWD calculated SCI.

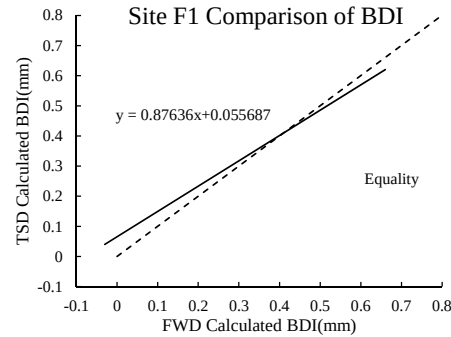


Figure 3. TSD calculated BDI versus FWD calculated BDI.

The relationship shown in each case was determined using orthogonal regression, since both estimates contain measurement errors. For the SCI, the slope of the regression is close to 1 and the TSD calculated SCI is, on average, about 0.05 mm higher than the FWD calculated SCI. For the BDI, the two devices follow the same trends however the slope of the regression in this case is about 0.88 which suggests that the relationship between the TSD BDI and the FWD BDI while still good, is not quite as good as the relationship in the case of the SCI. Similar to the case of the SCI, in general, the TSD calculated BDI values were larger than the FWD calculated ones.

Conclusion

This paper compared the TSD and FWD estimated strain at the bottom of the asphalt layer. The results show that, in general, the calculated strain from either device follow the same trends with TSD

calculated tensile strain on average larger by about 65 microstrain than FWD calculated tensile strain. Furthermore, there is a significant variability in the relationship between the tensile strain obtained from each device. This variability could result in differences of as much as 200 microstrain.

The results here suggest that further investigation of the causes that lead to the TSD measuring higher strain levels is needed. Although the applied load was of the same magnitude (40 kN), the dynamics of how the load is applied in each case are much different. In the case of the FWD, the load is dropped onto the pavement whereas for the TSD the load is applied by a moving wheel.

References

- [1]. Xu B., Ranjithan, S.R. and Kim Y.R. (2002), “New Relationship Between Falling Weight Deflectometer Deflections and Asphalt Pavement Layer Condition Indicators”, Transportation Research Record, Vol. 1806, pp. 48-56.
- [2]. Flintsch G.W. and McGhee (2009), “Managing the Quality of Pavement Data Collection,” NCHRP Synthesis of Highway Practice 39-01, Transportation Research Board, Washington, D.C.
- [3]. Flintsch G.W., Ferne B., Diefenderfer B., Katicha S.W., Bryce J., Nell S. and Clark T. (2013), “Assessment of Continuous Pavement Deflection Measuring Technologies,” Second Strategic Highway Research Program (SHRP2), Report S2-R06F-RW-1, The National Academies, Washington, D.C.
- [4]. Ferne B.W, Langdale P. and Round N (2009), Development of a calibration procedure for the UK Highways Agency Traffic Speed Deflectometer.

LS-SVM Based Fast Multi-class Classification Method and Its Application

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Keywords: Multi-class classification, Support vector machine, Face and fingerprint recognition.

Abstract. Support vector machine (SVM) is widely used in classification problems for its good performance. The processes of most of existing methods for multi-class problem are not simple, and more than one SVM classifiers should be trained in each of these methods. This paper presents a novel multi-class SVM method named Multi-class Least Square Support Vector Regression (MCLS-SVR), using only one classifier to solve multi-class problem. The main idea of this method is to transform classification problem to regression problem, which is much simpler, by treating the class label of each sample as the regression output value, then LS-SVR is used to solve the regression problem, and a rounding operation is added when testing samples because their class labels are positive integers. The result of experiments on face and fingerprint recognition show that the proposed MCLS-SVR method yields high recognition rate and fast calculation speed.

Introduction

Support vector machine (SVM), which is based on statistical learning theory, is one of the most popular classification methods due to its better data generalization performance and being able to solve nonlinear, small sample size and high-dimension problems. Since SVM was proposed by Vapnik [1], a lot of research has been done to apply and improve this method.

The SVM was originally developed for binary classification problems, but in practice, there are usually more than two classes of samples in most of the classification problems, so we should extend the binary SVM to multi-class SVM. Currently there are two major types of approaches:

One approach is to decompose classification problems into several binary SVM classifiers and then reconstruct them. For the first approach there are many methods: One-Against-Rest (OAR) [2], One-Against-One (OAO) [3], Directed Acyclic Graph (DAG) SVM [4], etc. These methods all have some deficiencies due to they didn't consider the classification problem as an entirety. However, lots of researches have been done to improve these methods and overcome their deficiencies, such as: Shiladitya Chowdhury et al. [5], Henry Joutsijoki [6], etc.

The other approach is to consider all data in one optimization formulation directly [7, 8]. This formulation is very promising because it deals with all the samples and classes at the same time, without losing any relevant information for arriving to the best solution for each problem. However, the SVM classifier may be not easy to design and the training speed may be slow when sample size is large.

This paper presents a novel multi-class classification method belonging to the second approach. This method is called Multi-class Least Square Support Vector Regression which introduces the idea of regression into classification problem. This method considers all the data and contains only one classifier which is easy to train, experimental results shows that this method performs very well and has a high classification rate.

Least Square Support Vector Regression

Least Square Support Vector Regression (LS-SVR), proposed by Suykens J. A. K. and Vandewalle J. [9], is an improved methods for SVM.

Consider a binary classification problem, the training data set of which is as follows:

$$(x_1, y_1), (x_2, y_2), \dots, (x_l, y_l), \quad x_i \in R^m, y_i \in \{-1, 1\}, \quad (1)$$

the goal of LS-SVR is to solve the following problem:

$$\begin{aligned} \min Q(\omega, e) &= \frac{1}{2} \|\omega\|^2 + \frac{C}{2} \sum_{i=1}^l e_i^2 \\ \text{s.t. } y_i &= \omega \cdot \phi(x_i) + b + e_i, \quad i = 1, 2, \dots, l, \end{aligned} \quad (2)$$

where $e = [e_1 \ e_2 \ \dots \ e_l]^T$. This problem can be solved by Lagrange multiplier method, and the regression function is as follows:

$$y(x) = \sum_{i=1}^l a_i K(x_i, x) + b \quad (3)$$

where a_i is Lagrange multiplier.

Multi-class Least Square Support Vector Regression

In this section, we proposed a LS-SVR based multi-class classification method. Compare a N classes classification problem with a regression problem, we can see that: in the classification problem, category labels $y_i \in \{1, 2, \dots, N\}$ are all positive integers; and in the regression problem, outputs $y_i \in R$ are real numbers. So if we can convert between regression outputs and category labels, the multi-class classification problem can be treated as a regression problem, which is more easily to be solved.

Given a N -class classification training sample set:

$$(\hat{x}_1, \hat{y}_1), (\hat{x}_2, \hat{y}_2), \dots, (\hat{x}_l, \hat{y}_l), \quad \hat{x}_i \in R^m, \hat{y}_i \in \{1, 2, \dots, N\} \quad (4)$$

We use the LS-SVR method to train these samples and get Eq. (3) as the classifier. To classify an unknown sample, we apply it into Eq. (3) and round $y(x)$ to get the final classification result, that is:

$$y(x) = \text{round} \left[\sum_{i=1}^l a_i K(x_i, x) + b \right], \quad (5)$$

where $\text{round}(x)$ is defined as follows:

$$\text{round}(x) = \begin{cases} p & p - 0.5 \leq x < p + 0.5 \\ p - 1 & p - 1.5 \leq x < p - 0.5 \end{cases}, \quad (6)$$

where p is positive integer and $p \geq 2$.

With this function, the multi-class classification problem can be directly solved by regression method, which is simple and fast; the robustness of the method is greatly enhanced due to the introduction of $\text{round}(x)$, and we can still classify correctly when there are noises in the input data.

Essentially, this method has omitted the intermediate process of the classification, and sought the mapping relationship between the input samples and their class labels. This method greatly simplifies the training process and accelerates the speed of operation by solving the classification problem with the idea of regression. Next, we should verify the validity of this method.

Experimental Results

To test the proposed method, we have selected three datasets from the UCI database: Iris dataset, Wine dataset and Image segmentation dataset, and the basic information of the datasets are shown in Table 1. To train the LS-SVR classifiers, we randomly selected 80% of samples from each dataset as the training samples, the rest as testing samples; and we chose the RBF kernel

$K(x, y) = \exp(-\|x - y\|^2 / 2\sigma^2)$. Let $\sigma = 30$, $C = 2000$, we then calculated the decision functions (Eq. (5)) respectively. The testing samples are used for verification of the method.

As a comparison, we solve these multi-class classification problems with the OAO method at the same time, and simulation results of classification accuracy rates are shown in Table 2. It could be seen that Multi-class LS-SVR method performed better than the OAO method and the classification accuracy rates were very high.

Table 1. Information of the three datasets.

	Number of instances	Number of attributes	Number of classes
Iris	150	4	3
Wine	178	12	3
Image segmentation	2310	19	7

Table 2. Classification accuracy rates of the two methods.

	Iris	Wine	Image segmentation
MCLS-SVR	100%	93%	100%
OAO	97%	86%	98%

We then applied the method to face recognition and fingerprint recognition to verify its effectiveness. Some of the face samples were selected from the Olivetti Research Laboratory (ORL) database and others were created by us, each image contains face region and non-face region, also contains the change of expression and illumination, including 80 faces images from 20 persons (4 images each). The fingerprint samples are collected by us too, including 60 fingerprints images from 30 persons (2 images each).

For higher recognition rate, the samples need to be pretreated before feature extraction. The pretreatment process in this paper is as follows:

- Gray processing for the original color images;
- Select feature regions from the images, shear on the gray images appropriately and normalize them;
- Filter the normalized images;
- Edge detection on the images;
- Binary processing to the images.

then the invariant moment [10] is used for image feature extraction after these pretreatment.

The step of face recognition using Multi-class LS-SVR method is as follows:

First take three samples from each category to form the training sample set, the rest as the testing sample set; then train the LS-SVR with training samples to get the decision function Eq. (5); finally randomly select a sample from the testing sample set and introduce it into the decision function, compare the output of decision function with its class label, and classification is successful if they are equal.

Following the step above, we picked up 60 face images from 20 persons to form the training set. We chose the RBF kernel $K(x, y) = \exp(-\|x - y\|^2 / 2\sigma^2)$, let $\sigma = 64$, changed parameter C to see its influence on results, and the result is shown in Table 3. The process of fingerprint recognition was basically the same with face recognition, 60 images were divided into training set and testing set, each contained 30 images. We let the parameter $\sigma = 40$ and testing result is shown in Table 4.

Experimental results of face recognition and fingerprint recognition showed that the Multi-class LS-SVR method also performed very well in practical application. We could notice that the parameter C should be large enough to get high classification rate.

Table 3. Recognition percentage of faces images.

C	10	100	1000	100000
Recognition rate	35%	65%	95%	100%

Table 4. Recognition percentage of fingerprint images.

C	10	100	1000
Recognition rate	56.67%	96.67%	100%

Conclusion

This paper presents a novel scheme for multi-class classification problem using Multi-class Least Square Support Vector Regression. The basic idea of this method is to convert the classification problem into a regression problem which is more easily to solve. The difference between LS-SVM and LS-SVR is their output values: integer for LS-SVM and real number for LS-SVR, so we add a rounding operation to SVR so that its output values can correspond to the class labels we need. Thus only one LS-SVR needs to be train as the classifier to solve the multi-class classification problem. Simulation and application results show that the proposed MCLS-SVR performs well and has high classification rate, at the same time, has good generalization ability for different classification problems.

References

- [1] Vapnik V. The nature of statistical learning theory [M]. New York: Springer-Verlag, 1995.
- [2] Kressel U. Pairwise Classification and Support Vector Machines [J]. Advances in Kernel Methods: Support Vector Learning, pp: 255-268 (1999).
- [3] Bennett K.P. Combining Support Vector and Mathematical Programming Methods for Classification [J]. Advances in Kernel Methods: Support Vector Machines, pp: 307-326 (1999).
- [4] Platt J.C., Cristianini N., Shawe-Taylor J. Large Margin DAGs for Multiclass Classification [C]. Neural Information Processing Systems, pp: 547-553 (2000).
- [5] Chowdhury S., Sing J.K., Basu D.K., et al. Weighted multi-class Support Vector Machine for robust face recognition [C]. 2012 International Conference on Communications, Devices and Intelligent Systems, pp: 326-329 (2012).
- [6] Henry J. Half-Against-Half Multi-Class Support Vector Machines in classification of benthic macroinvertebrate images [C]. Computer & Information Science (ICCIS), 2012 International Conference on, pp: 414-419 (2012).
- [7] Arenas-Garcia J., Perez-Cruz F. Multi-class support vector machines: a new approach [C]. Proc. ICASSP, pp: 6-10 (2003).
- [8] Weston J., Watkins C.. Support vector machines for multi-class pattern recognition [C]. ESANN, pp: 219-224 (1999).
- [9] Suykens J.A., Vandewalle J. Least squares support vector machine classifiers [J]. Neural processing letters, 9(3), pp: 293-300 (1999).
- [10] Gonzalez R.C. Digital image processing [M]. Pearson Education India (2009).

Diagnostics of Hydropower Objects' State Based on Results of Identification Measurement

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Keywords: Vibration signals, Identification measurement, State evaluation.

Abstract. The method of state evaluation of hydropower objects based on their results of identification measurement of vibration signals is described. The results of evaluation are presented in the form of two-dimensional hierarchical classification of objects, including qualitative (linguistic) interpretation of state.

Introduction

Nowadays to provide secure functioning of complex technical systems such an ideology is accepted, in compliance with which, firstly, machine maintenance and repair is performed only in case when it is necessary. Secondly, repairs are made, depending on the size of the object's defect. Such an ideology, unlike the concept based on scheduled maintenance, requires using high-efficiency facilities for monitoring the current state of the studied objects.

The most effective and commonly used technology for dynamic mechanical systems is the estimation of the current state according to the results of vibration level and spectrum measurement (vibration-based diagnostics [1]). In this case, vibration signal analysis is made on the basis of dedicated software developed according to the experts' experience particularly in this domain.

Currently more than 20 methods for technical system vibration-based diagnostics are known, however, the most effective methods are as follows: spectral analysis [2], analysis of specific diagnostic parameters [3,4,5], wavelet-based analysis [6], and neural network based analysis [7].

All the above-mentioned methods have their advantages and disadvantages. Among disadvantages that are common for all the methods are: incomplete defect and state identification of the studied objects, problem of result interpretation, low degree of unification and interference immunity.

In this article the possibility of using signal identification measurement theory (SIMT) is considered [8] in order to solve the problem of technical system vibration-based diagnostics that allows eliminating the above-mentioned disadvantages.

The possibility of using SIMT methods to solve the problems of object and process detection, identification and classification was proved in a number of monographies [8-12]. Some issues regarding the solutions for medical and technical diagnostics problems are presented in publications [13,14].

The actual hydro machine operated in one of the largest hydropower plants of Siberian region is used as an object under analysis.

The aim of the work was to perform analysis using the SIMT tools for vibration signals (Fig. 1) received from the shaft beat indicators located in the zone of lower generator bearing (LGB) – the

level of hydro machine's average height, and to develop methods for detection and diagnostics of their state.

Shaft beat is controlled by two induction sensors in cross-section of bearing located with an angle of 90° in the plane perpendicular to the shaft. Sensor No. 1 is placed at the side of "Head race" (HR) – upstream side, sensor No. 2 is installed at the side of "Right bank of the river" (RB) – low stream side. Vibration signals shown in Fig. 1 correspond to two states: of minor and large shaft beats [15].

In this work the problem of using two methods simultaneously, that are based on absolute and relative identification measurement, are considered; that allows us to increase the reliability of state estimate of such a complex object as hydro machine of a large hydropower plant. In this work, absolute measurement is deemed to be the measurement of an input signal when this signal forms the standard of reference. Thus, in such a case standard is "bound" to the signal. When using relative identification measurement, the standards are independent of an input signal.

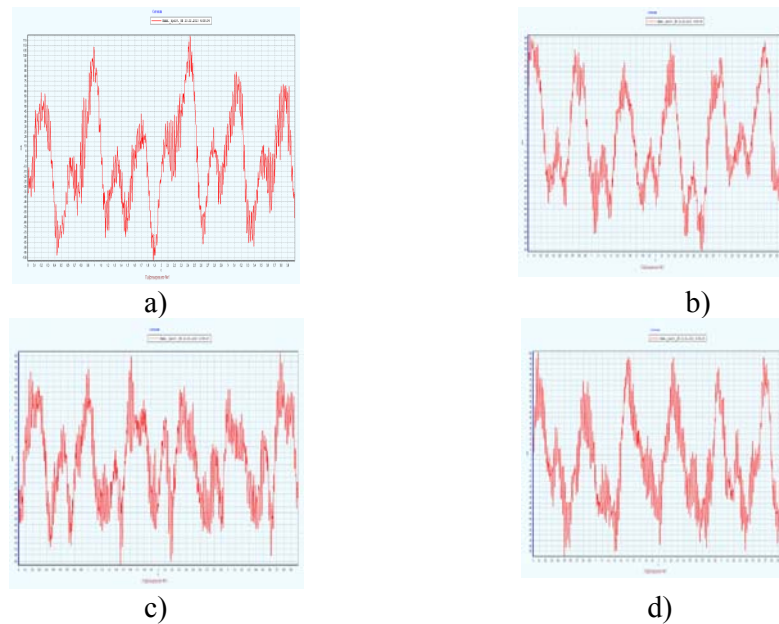


Figure 1. LGB vibration signal: a) sensor No. 1 – head race, minor beats; b) sensor No. 2 – right bank, minor beats; c) sensor No. 1 – head race, large beats; d) sensor No. 2 – right bank, large beats.

Identification Measurement Theory Elements

Identification Measurement Theory is a number of intellectual technologies (methods and computer aids) for signal detection and digital processing. The use of specific solutions allows solving some applied problems related to diagnostics, control, and management field.

SIMT is based on measurement of characteristics of signal waveform (momentary value distribution) and variability (time slot distribution) using *identification scale (IS)* the numbers of which include qualitative indices, represented, for example, with the help of proper names indicating the current state of the object under analysis. As a rule, proper names are names of probability distribution accepted in statistic measurement [16].

Technically, identification measurement consists of measurement transformation and reading data.

In SIMT measurement transformation is deemed to be transformation of a set, for example, time characteristic $x(t)$, into number IdP called *identifying parameter*. Data reading is the process of match making between identifying number (IdP) and standard ordered set (multivalued measure) - IS.

According to the theory identification measurement objects can be considered to be any sampled implementation of signals regardless of their nature (periodic, random, composite, fractal signals), and their characteristics:

- a) time characteristic $x(t)$ is a sample time function (in the form of electric signal or record);
- b) spectral characteristic $X(j\omega)$ is the Fourier transform of the function $x(t)$:

$$X(j\omega) = \int_0^T x(t) e^{-j\omega t} dt ; \quad (1)$$

c) correlative characteristic $K_{xx}(\tau)$ of stationary random process $x(t)$ for a finite observation interval $(0, T)$ in the form of

$$K_{xx}(\tau) = \frac{1}{T - \tau_{\max}} \int_0^{T - \tau_{\max}} x(t) x(t + \tau) dt ; \quad (2)$$

d) probabilistic characteristic (in the form of histogram) $P(a \leq e \leq b)$ possible numerical results of its numerical implementation for some interval $[a, b]$

$$P(a \leq x \leq b) = \int_a^b f_X(x) dx , \quad (3)$$

where a, b are interval boundaries; $f_X(x)$ is the distribution function of the random variable X .

Identifying parameters can be defined both for the signal $x(t)$ (IdP1), and its derivative (of increments) $\Delta x(t)$ (IdP2). Identification measurements related to 2 or more identifying parameters are called vectorial.

The logic of identification measurements is explained in Fig. 2.

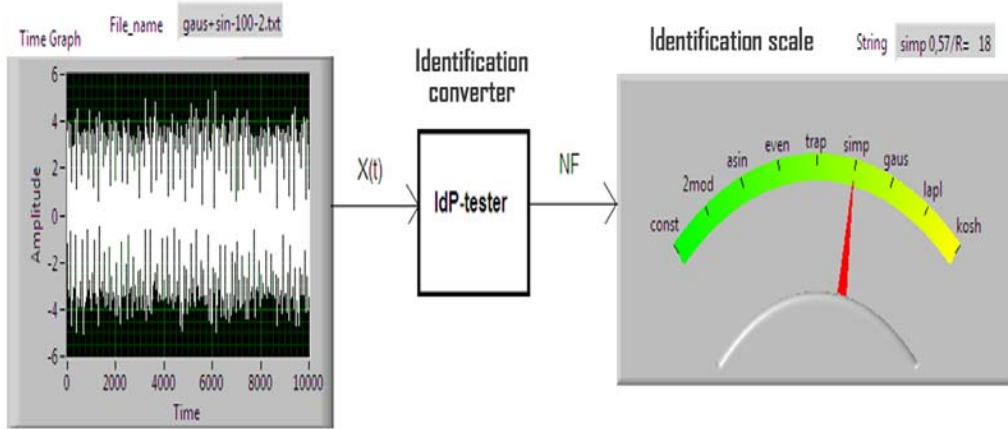


Figure 2. Identification measurement logic.

Fig. 2 shows: selective implementation (Time Graph) of the analyzed signal, the identification scale, the identification transmitter (IdP-tester) of NF-type [16] with the conversion function

$$NF = \left[\frac{\max\{x_i\} - \min\{x_i\}}{\sqrt{\frac{1}{N} \sum_{i=1}^N (x_i - \bar{X})^2}} \right]^2. \quad (4)$$

IS tabular presentation (Table 1) includes fixed point numbers (Rank), their names, compactly denoting symmetric distribution of random signals (bimodal distribution (2mod), arc sine distribution (asin), even distribution (even), trapezoidal distribution (trap), triangular distribution (simp), normal distribution (gaus), bilateral exponential (lapl), Cauchy distribution (kosh), and also numerical evaluations of their NF identifying parameter.

Table 1. Table presentation of identification scale.

IdP, N=10000	Type of distribution of random signals								
	const	2mod	asin	even	trap	simp	gaus	lapl	kosh
Rank	1	2	3	4	5	6	7	8	9
NF	0	4	8	12	18	24	48	96	N..2N

Note: Table 1 presents not real but conventional quantitative (numerical) characteristics of the FC "quality".

Physical meaning of such presentation is the following: you can work with signal *waveform* similarly as you work with any physical quantity, and perform various functional transformations. As it is shown in the work [17], waveform is the direct analog of *phase*, and therefore radians are used as natural units of measurement.

It is important to mention that in SIMT, as well as in classical measurement, the problem of interpolation occurs (reading data when the pointer is positioned between the IS numbers).

Methods and Tools of the Analysis

Structure diagram of studies (Fig. 3) consists of signal reader (SR), identification signal analyzer (ISA), comparator (C), data base of standard signals (DBSS), table and hierarchical classifiers (TC and HC), and also interpreter.

Signals under study are analyzed by two channels: absolute and relative measurement (AMC and RMC). ISA block measures several integral identification parameters, two of which are used for further analysis: degree of regularity (%Reg) and degree of asymmetry (%Asym). These parameters are used for two-dimensional table classification (TC block) shown in the top right corner of Fig. 3. C and DBSS blocks are used for measurement of input signal deviation from the system comprising 17 standard signals with particular distributions of instantaneous values. In order to define correctly accumulative value of deviations signals under study as well as standard signals must have an equal number of counts (equal sample sizes, N). The table of deviations are shown in the bottom of the Fig. 3, where column called Mean gives average values of deviations of each signal indicated in column called File_name, from all the standards. HC block allows standard signal sorting according to deviations for each input signal in order to make classification tree consisting of standard names (Fig. 4). Classification tree not only visualizes hidden regularity of communication structure between signals under analysis, but also allows giving quantitative estimation of these communications by calculating the relative number of standard names placed at equal positions in compared branches of the tree. Such estimations are presented in signal communication matrix placed in the bottom of Fig. 4. In the top right corner of Fig. 3 you can see the classification tree consisting of root and two trunks. Connection between the root and trunks is shown by a double line. Connection between standard names (vertically) inside trunks and branches is shown by a bold line. Connection between standard names (horizontally) in neighboring branches is shown by a dotted line.

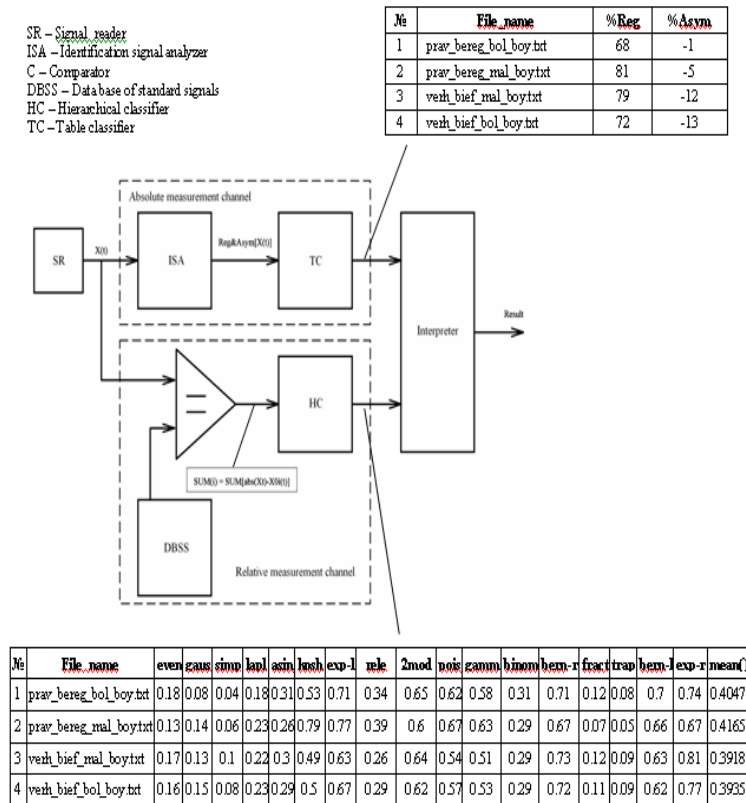


Figure 3. Structure diagram of studies.

The first (left) trunk (the name of the initial standard called kosh) has one branch with signal number #2 and name “prav_bereg_mal_boy.txt” at the end. The second trunk consists of 3 branches #3, #4, and #1 (with names verh_bief_mal_boy.txt, verh_bief_bol_boy.txt, prav_bereg_bol_boy.txt correspondingly). In the left corner of Fig. 3 signal diagrams and their histograms are shown.

Table of two-dimensional classification (the bottom table in Fig. 3) is made by dividing the identification parameter measurement ranges ($65 < \%Reg < 85$, and $0 < \%Asym < 15$) into subranges. The names of the signals under analysis are placed in the corresponding cells of the table, positional relationship of which allows giving them qualitative characteristic.

Program code structure of the system under study, made in graphical programming environment LabVIEW, is shown in Fig. 5. This structure forms relational database of the results of input signal identification parameter measurement by sorting and filtration of entries which form linear and tree-structured classification (in EXCEL environment). The system is an intellectual analyzer since it comprises the built-in data base of standards representing sorted distributions of instantaneous values (IVD) of random signals, and automatically adapted to a particular sample size of an input signal. System intellectual capacity and, correspondingly, resolving capacity of hierarchical classification are dependent on the number of the used standards. In our case, 17 standards are stored in the data base, the number of which can be increased according to the increasing requirements for the quality of identification. For these purposes it is necessary to take a number of technical measures regarding software updating. However, the existing system allows solving a lot of problems related to quantitative and qualitative estimation of complex object state.

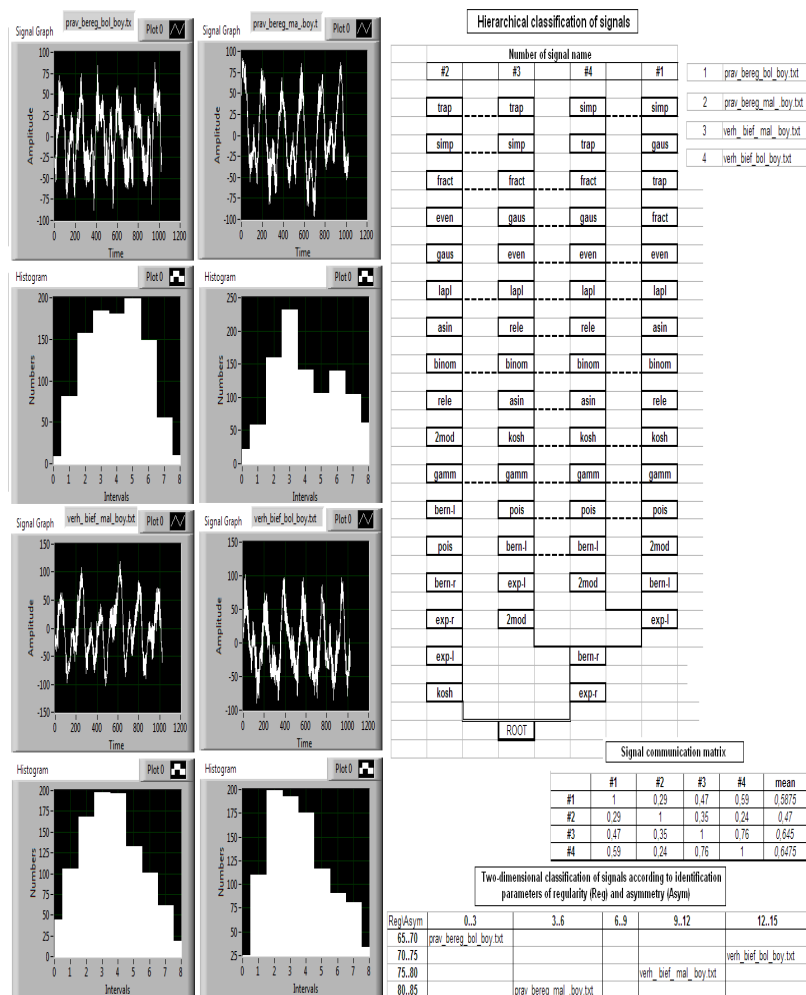


Figure 4. Signals under analysis and their classification.

Conclusions

1. In the work the application of signal identification measurement theory (SIMT) is considered in order to estimate the state of hydroelectric objects according to the vibrations signals comprising steady and nonsteady components. It is established that the suggested two-dimensional table classification made on the basis of such characteristics as regularity and randomness, symmetry and asymmetry allows dividing signals according to the object type into groups: “prav_bereg_” and “verh_bief”, and also according to the intensity of influencing factor into groups: “mal_boyt” and “bol_boyt”.

2. According to SIMT, if the combination of identification parameters of the signal under analysis does not allow you to estimate the state, you should increase the number of database standards or measurement dimension of identification parameters by using additional characteristics. It is established that to diagnose the state of hydro machine it is required and sufficient to measure time and probabilistic characteristics in the structure of 17 global signal standards.

3. Classification tree visualizes hidden regularity of communication structure between the signals under analysis. Moreover, it is established that hierarchical classification provides additional qualitative data on communication structure (type of the tree and sequence of standard names in the branches) and quantitative data on bonding force between signals (coupling coefficient matrix).

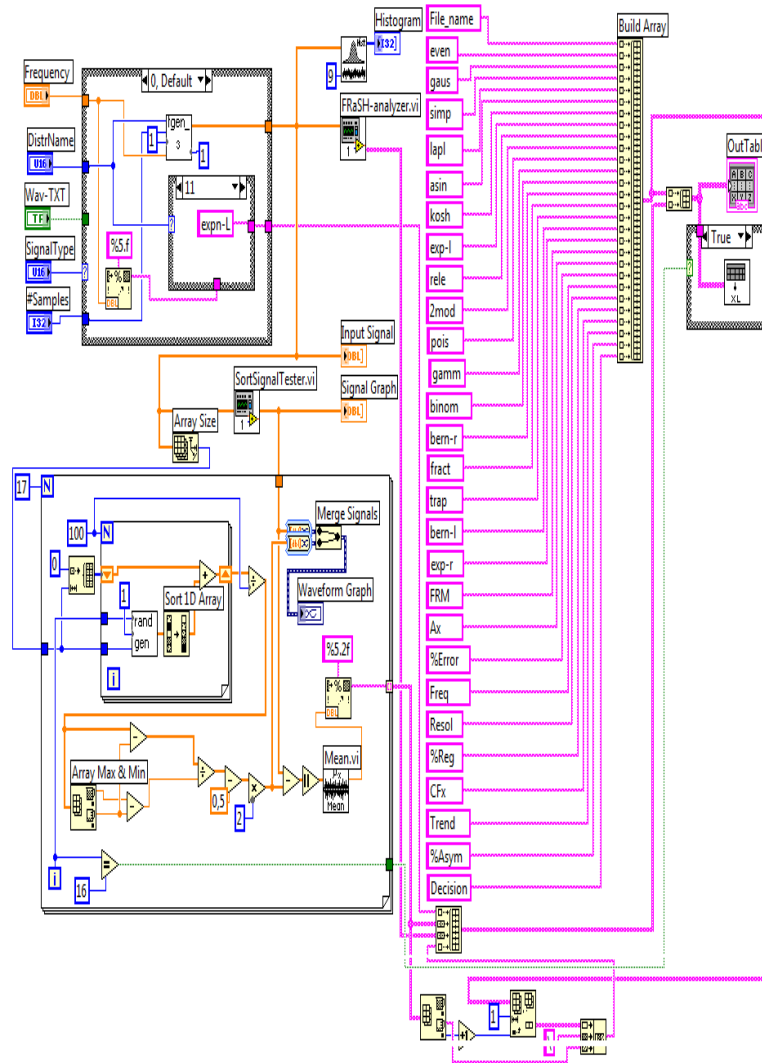


Figure 5. Program code structure of the system under analysis.

4. Suggested identification tools and methods of their use allow solving the problem of identification and state estimation of complex hydropower objects and provide the user automatically with linguistic descriptions in terms accepted by experts of this domain. It is supposed that further study will be related to development of software for automated intellectual system of technical diagnostics.

The work is a result of creative collaboration between North-Kazakhstan State University, Omsk State Technical University and DIAMECH 2000 Ltd, the leading Russian manufacturer of balancing machines and vibration diagnostic equipment.

References

- [1] R. Antunovic, Role and Importance of Vibrodiagnostics in Maintenance of Technical System // 15th International research-expert conference Trends in the Development of Machinery and Associated Technology. Prague, September 12-18, 2011, pp. 309-312.
- [2] V.A. Rusov, Spectral Vibration Diagnostics, Perm, 1996.
- [3] V.I. Smirnov, Methods and Facilities of Functional Diagnostics and Technological Process Control Based on Electromagnetic Sensors, Ulyanovsk State Technical University, Ulyanovsk, 2001.

- [4] J. Antoni, Blind Separation of Vibration Components: Principles and Demonstrations. // Mechanical Systems and Signal Processing, 2005, v. 19, pp. 1166–1180.
- [5] A. Jablonski, T. Barszcz, J. Antoni, Comparison of Vibration Signal Separation Techniques for Rotating Machinery // 6th International Conference Acoustical and Vibratory Surveillance Methods and Diagnostic Techniques. Compiegne, October 25–26, 2011, pp. 3-10.
- [6] I. Dobeshi, Ten Lectures on Wavelets, Research Center Regular and Random Dynamics, Izhevsk, 2001.
- [7] Application of Artificial Neural Networks for Expert System of Manufacturing Equipment Diagnostics. (Collection of articles Energy Saving, Industrial Automation, Smart Buildings and Automatic Control System of Technological Process, published on 06.04.2007) [Electronic resource] /A.V. Semenchenko. <http://d.17-71.com/2007/04/06/primenienieiskusstve>.
- [8] Yu.N. Klikushin, K.T. Koshekov, Methods and Facilities for Signal Identification Measurement, Publishing house of NKSU named after M. Kosybaev, Petropavlovsk, 2007.
- [9] Yu.N. Klikushin, Identification Tools for Signal Form Analysis and Synthesis, Publishing house of OmSTU, Omsk, 2010.
- [10] Yu.N. Klikushin, K.T. Koshekov, Theoretical Foundations of Identification Measurement and Modifications, LAP Lambert Academic Publishing, Saarbrucken, 2011.
- [11] Yu.N. Klikushin, Signal Identification Measurement (Anthology of electronic publications), Palmarium Academic Publishing, Saarbrucken, 2012.
- [12] Yu.N. Klikushin, A.A. Gorshenkov, Introduction into Signal Identification Theory, Publishing house SB RAS, Novosibirsk, 2012.
- [13] Yu.N. Klikushin, A.A. Gorshenkov, V.Yu. Kobenko, Methods of Computer Diagnosis of Parkinson's Diseases // Online article, M.: Journal of Radio Electronics, Publishing house: IRE RAN, No. 10 (October), 2012.-<http://jre.cplire.ru>
- [14] Yu.N. Klikushin, N.K. Nabiev, K.T. Koshekov, Diagnostic Method for Axle Box Bearings Based on Identification Measurement // VKGTU Herald. – Ust-Kamenogorsk: Publishing house: VKGTU, 2010, No. 1, pp. 81-86.
- [15] E.S. Trunin, O.B. Skvortsov, In-Service Inspection of Hydro Machine Technical State / Monthly Technical Journal Electric Power Plants. - Publishing house: Energoprogress, 2010, No. 6, pp. 38-45.

Research on the Method of Flowmeter Pulse Acquisition

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Keywords: Pulse acquisition, Double time method, Flowmeter counter.

Abstract. In order to solve the shortcomings of inaccurate pulse counting in the process of pulse flowmeter calibration, a new method of flowmeter pulse counting is proposed. The method, which is based on the principle of double time method of pulse counting, implements pulse counting by hardware-based foundation of PLC and relevant external circuits. The results show that the method can guarantee the accuracy of flowmeter pulse counting effectively.

Introduction

The flowmeter, which is used to measure fluid in a pipe or channel of an instrument, is widely used in the various fields of industrial, agricultural production, national defense, scientific research, foreign trade and people's lives [1]. During the flowmeter calibration process, the number of pulses which requires reliable and accurate measurement is the main premise to calculate the coefficient K meter [2]. Currently, most of the methods are based on the acquisition card, however, it only guarantees the pulse count in integer units at a lower cost conditions. In the large flow verification process, the pulse acquisition in integer units has little effect on the results, which could be neglected. However, in a small flow verification process, the number of pulses in integer units would directly affect the accuracy of the flowmeter and the factor K. In order to solve those problems, the paper based on the principle of double time method and the circuit of flowmeter pulse acquisition is designed. By this method, it could guarantee the accuracy of meter pulses effectively, therefore improve the accuracy of flowmeter indirectly.

Measuring Principle

The double time method could be used when the flow is constant [3, 4]. The pulse counter of PLC is recording the flowmeter pulse within the stipulated time. Through three parameters, the number of pulses could be calculated accurately. Under the conditions of stable flow pulse signals, the number of accurate pulses is based on the following formula.

$$N_a = N \frac{T_a}{T_b} \quad (1)$$

N is the number of pulses which counter recorded in a particular time.

N_a is the number of pulses based on the dual-time method.

T_a is the interval time from the detection signal beginning to end.

T_b is from the start of the first rising edge when the detection time started to the first rising edge when the detection time stops.

The diagram of double time method is shown in Fig. 1.

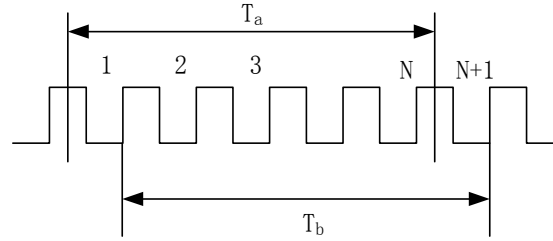


Figure 1. The diagram of dual-time method.

The number of pulses, which is obtained by the principle of double time method, should satisfy the following formula.

$$N-1 < N_a < N+1 \quad (2)$$

The double time method could calculate the number of pulses in theory accurately, however, the timer would not achieve the desired precision accuracy. For that reason, adopt the double time method to count had been limited in actually. The PLC highest precision of timer is 1ms. In the low flow, the number of pulses which calculated by the double time method could meet the formula (2). However, when a large flow rate is achieved, the number of pulses which calculated by the double time method could hardly satisfy the formula (2). For example, when the output frequency of flowmeter is approximately 5KHZ, T_a is 60.000s, T_b is 60.001s, the pulse difference between N_a and N has exceeded 16 which calculated through formula (1). So the double time method could not be used in any frequency of output pulses of flow meter.

The pulse of N_a and the pulse of N which is recorded by PLC counter, are compared on the foundation of double time method. When the N_a satisfies the formula (2), it's taken to the final pulse number; when the N_a dissatisfies the formula (2), the N is taken to the final pulse number.

Hardware Design

The hardware design includes: Power supply part, pulse process part and the PLC control unit. The power supply part includes 5V and 24V, the 5V power supply provides working power for the chips in pulse processing circuit and the 24V power supplies voltage is for driving a pulse voltage signal input [5, 6].

The hardware principle diagram is shown in Fig. 2.

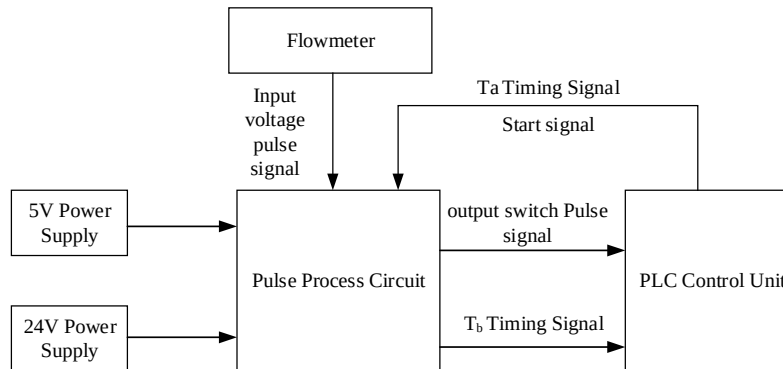


Figure 2. Hardware principle diagram.

In the verification process of flowmeter, the output pulse usually is voltage 0-5V or 0-24V pulse and the flowmeter signal that is susceptible to interference from other signal. Therefore, the PLC input interface get the voltage pulse signal of stable through the pulse processing circuit. The circuit input includes the pulse signal from the flowmeter, the negative pulse signal, a beginning test

signal, power signal. The circuit output contains the stable pulse signal which is distinguished by the PLC and time control signal T_b . The pulse processing circuit which was realize the filtering and shaping and ensure the reliability of the input signal. The process has carried on a signal drive, photoelectric isolation and gate processing in turn. The input pulse signal and control test signal got together through the D trigger ensures that only when the effective control of test signal come from PLC output interface, could the output pulse signal and double time T_b signal start to output valid signal [7]. Pulse processing circuit schematic diagram is shown in Fig. 3.

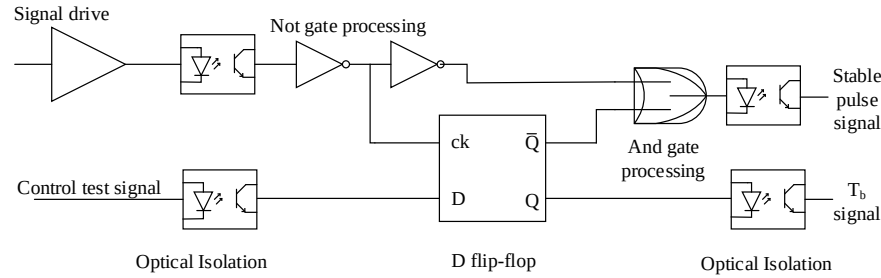


Figure 3. Principle diagram of pulse processing circuit.

The signal of diagram is shown in Fig. 4.

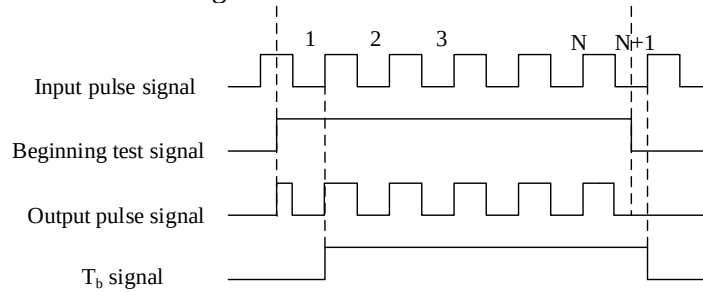


Figure 4. Signal schematic diagram.

The wiring diagram is shown in Fig. 5.

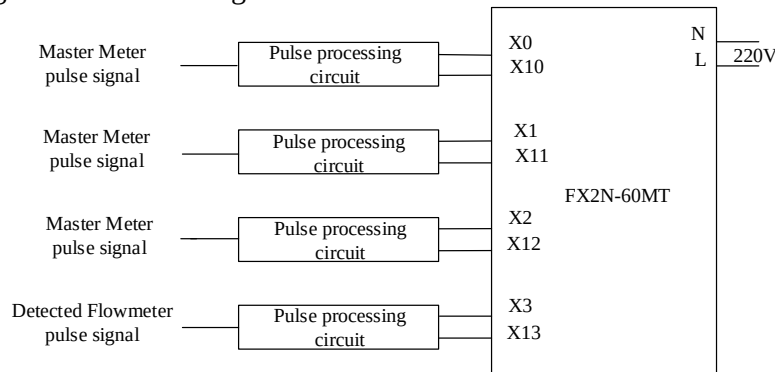


Figure 5. The wiring diagram.

The PLC could achieve pulse acquisition, time T_a and T_b in double time method [8]. The input ports X0-X3 are used as the pulse acquisition, the X10-X13 are used as T_b signal time ports. The output Y0 is used as the beginning test signal. Because the maximum pulse frequency of flowmeter is about 5KHZ, and the PLC could collect up to 60KHZ pulse signal, which effectively guarantees that pulses could be counted. The PLC only has four 1ms timer, while the longest test need measure 300s, adopts the iterative method is used to achieve T_a . The timing of T_b adopts 10ms timer.

Experimental and Processing

A pulse counting system based on double time method is developed, and experimental research is conducted. When the output frequency signal of flowmeter is 20Hz, the data are obtained through the system and shown in Table 1.

Table 1. The data of pulse of same frequency.

F(HZ)	T _a (S)	T _b (S)	N	N _a	N _a -N
20	19.995	20.000	399	398.90	-0.10
20	19.994	19.950	399	399.88	0.88
20	19.993	19.990	399	399.06	0.06
20	19.994	20.000	399	398.88	-0.12
20	20.000	20.000	400	400.00	0.00
20	19.994	20.000	400	399.88	-0.12
20	19.998	20.000	399	398.96	-0.04
20	19.993	19.950	399	399.86	0.86
20	20.001	20.000	400	400.02	0.02
20	20.000	20.000	400	400.00	0.00

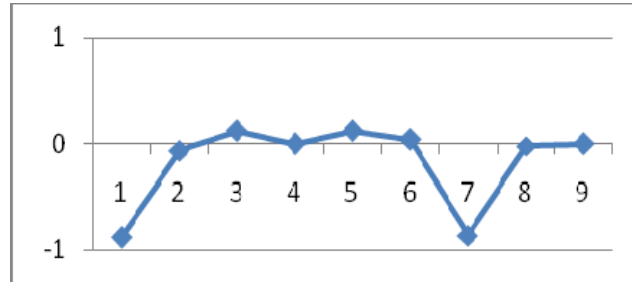


Figure 6. Error analysis figure of Na-N.

From Table 1 and Fig. 6 it could be seen clearly when the flowmeter is in low output frequency, the number of pulses N_a is calculated by the formula (1) and the value satisfies for formula (2). In order to ensure that the N_a is more reliability, the time of T_a and T_b should must be accurately. Therefore, using N_a could effectively guarantee the accuracy of flowmeter.

Under the premise of the stable output signal of flowmeter, the results are shown Table 2 and Fig. 7 at different output frequency signal.

Table 2. The data of pulse of difference frequency.

F(HZ)	T _a (s)	T _b (s)	N	N _a	N _a -N
10	19.997	19.990	200	200.07	0.07
20	19.997	19.990	399	399.14	0.14
50	19.998	19.990	1000	1000.4	0.4
100	19.992	19.990	1998	1998.2	0.2
200	19.999	20.000	4000	3999.8	-0.2
500	19.993	19.990	9997	9998.5	1.5
1000	19.992	19.990	19992	19994.0	2.0
2000	19.992	20.000	39986	39970.0	-16.0
5000	19.993	19.990	99969	99984.2	15.2

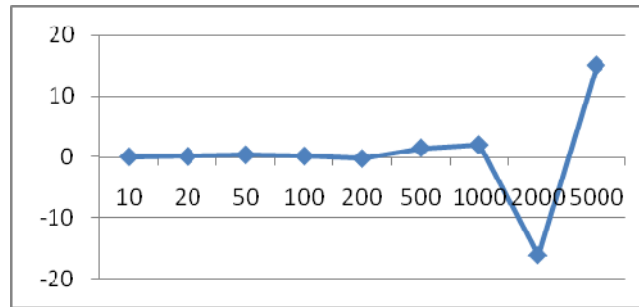


Figure 7. Analysis of the input frequency and the output error.

It could be seen from Table 2 and Fig 7, when the input pulse frequencies are below 200HZ, double time method could significantly improve the accuracy of pulse count. However, with the output frequency increasing, limit by precision of the T_a and T_b , the number of pulses using the double time method obviously does not satisfy the formula (2). Therefore, using the number of pulses which counter is recorded and calculated by double time method as the most final meter pulse value could guarantee the accuracy of pulse count effectively.

Conclusions

The article proposes a method of flowmeter pulse acquisition. The multi-channel acquisition simultaneous and accurate pulse count is achieved by PLC and associated external circuit. When the input signal is in low-frequency pulse, the number of pulses was modified adopt by the double time method make that improve the accuracy of pulses measurement. When the input signal is high-frequency pulse, PLC and associated external circuits also achieves a high-efficiency counting pulses. Experimental results shows that the method can guarantee the accuracy of meter pulse number effectively, improve the accuracy of flow meter indirectly.

References

- [1] Su Yansun, Liang Guowei. Flow Measurement and Test [M]. Beijing: China metrology publishing company, 2007.
- [2] Liang Guowei. Flow Measurement Technology and Instrument [M]. Beijing: Machinery industry publishing company, 2007.
- [3] Liang Guowei. Design and Application of Calibrator of Pulsed Flow Transducer [J]. Process Automation Instrumentation, 24(8), pp. 14-16, 2003.
- [4] Qi Lixiao, Research of Gas Flow Standard Facilities by Master Meter Method [J]. Control and Instruments in Chemical Industry, 37(2), pp. 34-38, 2010.
- [5] Li Hongwei. Study on PLC Based Multi-functional Digital Deep Indicator of Mine Hoist [J]. Mechanical Management and Development, (4), pp. 55-58, 2012.
- [6] Chen Peizhe, Zeng Gang. A research on equivalent sampling of impulse [J]. Electronic Measurement Technology, (10), pp. 55-58, 2014.
- [7] Yu Guowei. A Camera Calibration Method with Weight Factors [J]. Microprocessors, (6), pp. 58-61, 2011.
- [8] Guan Zhijian, Shen Feng. Implementation and Verification of Standard Device Diverter System for high Viscosity small Flow [J]. Measurement & Control Technology, (7), pp. 1-5, 2013.

Two-dimensional Imaging Algorithm Based on Linear Prognosis for Space Target in Bistatic ISAR System

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Keywords: Bistatic ISAR; Levions prognosis; Image resolution; Two-dimensional imaging.

Abstract. In bistatic inverse synthetic aperture radar (Bi-ISAR) system, its image resolution is lower than monostatic ISAR system. In order to solve this problem, the linear prognosis algorithm is adopted in the imaging process and the imaging algorithm based on linear prognosis is put forward. Space target Bi-ISAR imaging is taken as example in the research. The one-dimensional range profile is created through pulse compression. Before the azimuth compression, burg entropy maximum algorithm in Levions recursive method is used to estimate the prognosis coefficients and the azimuth echo data. Then Fourier transformation is used to compress the azimuth data in order to get the high resolution azimuth image. This imaging method can obtain the two-dimensional image with the resolution equal to the monostatic ISAR or even higher than it. Simulation experiments verify the effectiveness and availability of the algorithm.

Introduction

Due to the separated transmitting and receiving station, the bistatic Inverse Synthetic Aperture Radar (ISAR) could still get the 2D object image even when the relative radar line of sight has no rotational movement, which is the better characteristic than monostatic ISAR. So, bistatic ISAR possesses the higher imaging probability than monostatic ISAR [1]. Otherwise, when the receiving station is located in the front, the imaging range augments at the same time, which could improve the detection and imaging ability for stealth targets [2, 3]. In recent years, the main research of bistatic ISAR is focused on imaging principle and various compensation algorithms [4-6].

During the bistatic ISAR imaging process, the time-variant bistatic angle could lead to a time-variant bistatic imaging resolution, which makes the 2D image become blurred [7]. The movement of space targets is complicated, so it needs to image the object in a short integration time to avoid blurring of 2D image.

Because of the short integration time, the bistatic angle and image resolution could be looked as the constant during imaging time, which could effectively avoid image blurring. At the same time, the target character has relatively small changes when imaging during small rotation angel, which decreases the difficulty of movement compensation. While the coherent integration time becomes shorter, the resolution is definitely decreased. So it is the prerequisite to research the ISAR imaging algorithm during small rotation angle.

The ISAR imaging becomes spectrum estimation problem when the movement compensation is accomplished [8]. So, the imaging problem in small integration time could turn into a high-resolution spectrum estimation problem with short data sequence. For the problem of high-resolution spectrum estimation with short data sequence, there are two kinds of processing methods [9,10]. The first kind is spectrum estimation method based on spectrum functions such as MUSIC, ESPRIT and so on. This method could get the higher frequency resolution but has the fake amplitude information and is sensitive to noise. The second kind is spectrum estimation method based on DFT data extrapolation. This method complies with some principles which extrapolates new data based on the existent data. Then all the data are used to accomplish spectrum estimation based on DFT method which will equivalently increase data length and improves spectrum resolution. The second method attracts more attention and research because of its high resolution,

small count amount and the higher possibility of realization. So the second method is adopted in this paper to research the 2D imaging algorithm for bistatic ISAR according to space target.

Echo Modeling for Space Target Bistatic ISAR

The imaging geometry relationship of moving target bistatic ISAR is showed in Fig. 1.

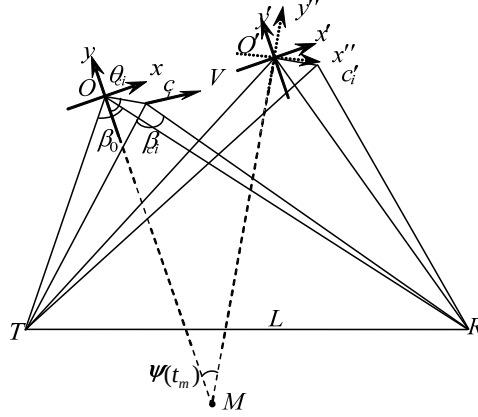


Figure 1. Geometry relationship of bistatic ISAR.

In Fig. 1, L is the baseline of bistatic radar; T is transmitting station; R is receiving station; y axis is the bisector of target when starting the observation and is the range direction of bistatic radar also; β_0 is bistatic angle; β_{ci} is the bistatic angle of scatter c_i in target, and $\beta_0 \approx \beta_{ci}$ is thought to be correct; dot O is the center of phase; θ_{ci} is the angle between position vector Oc_i of scatter and the negative direction of y axis; V is the target velocity; $\psi(t_m)$ is the rotation angle of bisector during imaging process. If we make the assumption that the bistatic radar is synchro ideally and LFM signal is transmitted with the echo sampled in intermediate frequency to desample it to baseband, the echo of scatter c_i could be denoted through (1) as follows.

$$s_{ci}(\hat{t}, t_m) = \sigma_{ci} \text{rect}\left(\frac{\hat{t} - R_{ci}(t_m)/c}{T_p}\right) \times \exp\left[-j2\pi f_0 \frac{R_{ci}(t_m)}{c}\right] \times \exp\left[j\pi\mu \left(\hat{t} - \frac{R_{ci}(t_m)}{c}\right)^2\right] \quad (1)$$

where $\hat{t}$ is the time-in-pulse; t_m is the pulse transmitting time; f_0 is carrier frequency; μ is frequency slope; T_p is pulse width; σ_{ci} is the non-back scattering intensity of scatter c_i in the i_{th} range bin.

The paper [6] makes the research result that whether the target moving track is in the same plane with baseline of bistatic radar, the range transformation of scatter could be expressed as follows (2).

$$R_{ci}(t_m) = R_{ref}(t_m) + R_{rot}(t_m) \\ = R_{ref}(t_m) - 2r_i \cos[\theta_{ci} + \psi(t_m)] \cos\left[\frac{\beta(t_m)}{2}\right] \quad (2)$$

where $t_m = mT$ ($m = 0, 1, 2, \dots$) is slow-time; $\beta(t_m)$ is target time-varying bistatic angle during the imaging process; $R_{ref}(t_m)$ is the range course between target phase center and transmitting-receiving station, which is the translational movement item of target; $R_{rot}(t_m)$ is the rotational range item of scatter c_i . It could be found from (2) that, in bistatic ISAR imaging, the range transformation could still be divided into two parts, i.e, translational movement item $R_{ref}(t_m)$ and rotational movement item $R_{rot}(t_m)$. So bistatic ISAR could also get the target image through

range compression, range alignment, initial phase correction, azimuth compression just as monostatic ISAR[6].

When the rotational angle is small which is equal to short integration time, the bistatic angle $\beta(t_m)$ could be thought as the constant approximately. It is assumed that the constant bistatic angel is thought to be β_0 . If the translational range item $R_{ref}(t_m)$ has been totally compensated, the range transformation item of scatter c_i could be simplified as (3)

$$R_{ci}(t_m) = -2r_i \cos[\theta_{ci} + \psi(t_m)] \cos\left[\frac{\beta_0}{2}\right] \quad (3)$$

At this time, target range resolution of and azimuth resolution could be expressed as (4), (5) respectively[11].

$$\rho_r \approx \frac{c}{2B_w \cos(\beta_0/2)} \quad (4)$$

$$\rho_a = \frac{\lambda}{2\Delta\theta_{rot} \cos(\beta_0/2)} \quad (5)$$

Bistatic ISAR Resolution Lower than Monostatic ISAR Cause Analysis

Transmitting station and receiving station separates from each other in bistatic ISAR system, which leads to the imaging resolution related with bistatic angle. If the imaging bistatic angle is β , then the target range resolution could be denotes as follows (6) [11, 12].

$$\rho_{rc_0} = \frac{c}{2B_w \cos(\beta/2)} \quad (6)$$

Where ρ_{rc_0} is the target range resolution; B_w is the transmitting signal bandwidth; β is the imaging bistatic angle; c is the speed of electromagnetic wave in vacuum.

It can be observed from(6) that, as the bistatic angel exists, the bistatic ISRA range resolution is lower than the monostatic ISAR range resolution $c/2B_w$ with the same signal bandwidth. For the convenience of research, bistatic ISRA is thought to be an equivalent monostatic ISAR in the bisector. If the transmitting carrier frequency is f_0 , signal bandwidth is B_w , bistatic angle is β , then the equivalent carrier frequency of monostatic radar is $f'_0 = f_0 \cos(\beta/2)$ and the equivalent signal bandwidth is $B = B_w \cos(\beta/2)$. So the resolution also could be expressed as (6) [12]. This equivalent model can explain why the range resolution decreases as bistatic angel, so this model has been widely used during the existent bistatic ISAR research.

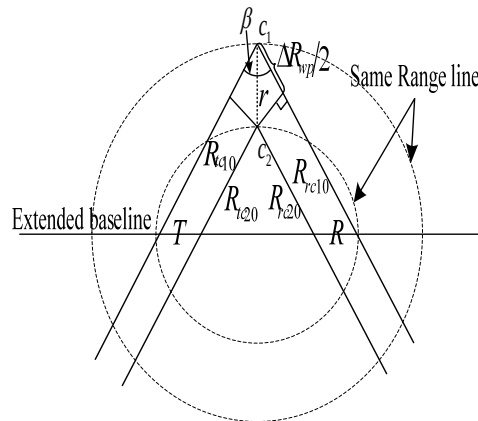


Figure 2. Wave distance difference.

In the Fig .2, the broken line is the range-same line for bistatic radar. R_{tc10}, R_{rc10} denote the distance between scatter c_1 and transmitting, receiving stations, respectively. R_{tc20}, R_{rc20} represent the distance between scatter c_2 and transmitting, receiving station, respectively. β is bistatic angle; T, R are transmitting and receiving station respectively; r is the radial distance between scatter c_1 and c_2 . When the target is far from radar, the assumption of far field is satisfied. So the wave-frontier is thought to be plane-wave. The wave distances between transmitting, receiving stations and scatter are parallel. With this assumption, the round wave distance difference ΔR_{wp} between scatter c_1 and scatter c_2 could be expressed as follows.

$$\Delta R_{wp} = 2r \cos\left(\frac{\beta}{2}\right) \quad (7)$$

Imaging Algorithm Based on Levions Prognosis

Imaging Flow of 2D Image

The ISAR imaging problem could be regarded as spectrum estimation problem [8]. The basic step of linear prognosis is firstly to estimate the filter coefficients and then extrapolate the data to the needed extent. The prognosis coefficients are estimated based on the least mean square error rule, which ensures the difference value between actual echo an the extrapolated data is the least.

If some range aligned range-bin echo could be expressed as $[x(t_{n-m}), \dots, x(t_{n-1})]$, so the extrapolated echo at time t_n of this range-bin could be expressed as

$$\hat{x}(t_n) = -\sum_{i=1}^m a_m(i) x(t_{n-i}) \quad (8)$$

In the equation (8), $a_m(i)$ is the i th coefficient of the m -order filter.

With the same analysis, the echo at time t_{n-m-1} after extrapolated of this range-bin echo could be showed by (9)

$$\hat{x}(t_{n-m-1}) = -\sum_{i=1}^m a_m^*(i) x(t_{n-m+i}) \quad (9)$$

In the equation (9), $a_m^*(i)$ is the conjugate complex of $a_m(i)$.

Combined with rang pulse compression, 2D imaging algorithm flow based on linear prognosis when the rotational angel is small which is demonstrated in the Fig.3.

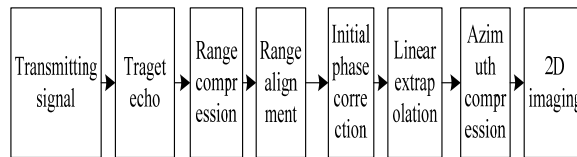


Figure 3. 2D imaging algorithm flow.

Construction of Prognosis Filter Coefficients and Its Order

The larger the linear prognosis filter is, more accurate the prognosis result is. But the count amount ascends rapidly also so the balance between algorithm count amount and prognosis error has to be satisfied. In the paper [13] shows that, when the Auto Regressive (AR) model is used to extrapolate data, the most accurate prognosis results could be caught when the filter order is chosen to be 1/3 of the data length. The Levinson prognosis of Burg entropy is used to estimate the filter order and its coefficients. The filter coefficients of different filter order are estimated firstly, then the prognosis error power with different filter is compared to choose the filter with the smallest prognosis error power. The basic step to get the prognosis order and its coefficients are summarized as follows.

Step1: Initial the prognosis power error and its back-and-forth prognosis error;

Step2: Count the reflection coefficient K_m that is used to extrapolate data;

Step3: Count the back-and-forth prognosis filter coefficients via (10);

$$\begin{aligned} a_m(i) &= a_{m-1}(i) + K_m a_{m-1}^*(m-i), \quad i = 1, \dots, m-1 \\ a_m(m) &= K_m \end{aligned} \quad (10)$$

Step4: Count the prognosis error power of the filter with m-order via (11)

$$P_m = (1 - |K_m|^2) P_{m-1} \quad (11)$$

Step5: Count the output of the filter with m -order, repeat Step 2 to Step 5 until the prognosis error power P_m hardly become smaller. And the order and coefficients of prognosis filter are got at the same time.

Simulation Experiment

Simulation scene and scatter model are expressed in Fig. 4 and Fig. 5, respectively.

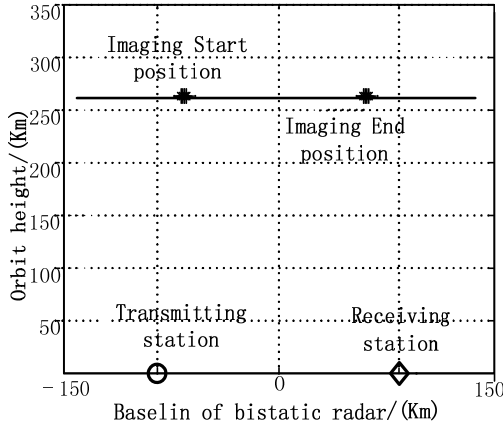


Figure 4. Target moving simulation scene.

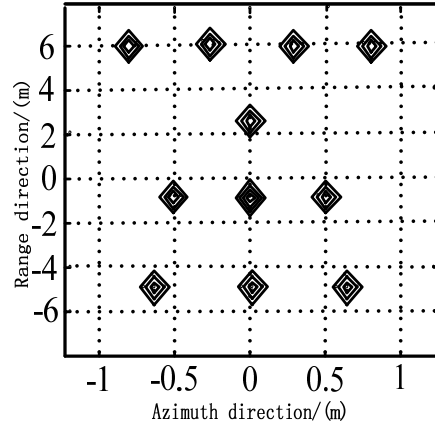


Figure 5. Target scatter model.

In the Fig. 4, circle stands for the transmitting station and rhomboid stands for the receiving station. Target orbit height is about 260Km; target round radial velocity is 3.5Km/s; In the simulation, it's assumed that translational compensation is totally completed. The target orbit data is simulated via orbit software. When the orbit data is generated, the orbit with constant bistatic angle is chosen to complete simulation to avoid the image blurring caused by the change of bistatic angle. The target is expressed by ideal scatters and the target model in range-azimuth is as shown in Fig.5. The echo SNR is counted by radar equation. In the simulation, the smallest receiving SNR is set to be 14dB and the target RCS is set as 4m². Because the scatters are separate, respectively so the RCS of every concrete scatter is 0.5m². Simulation radar parameters are shown in the Table 1.

Table 1. Simulation radar parameters.

parameter	index	parameter	index
Bistatic angel	120°	Imaging angel	2°
Signal width	950MHz	Pulse width	60μs
Carrier frequency	8GHz	Baseline length	200Km

According to the parameters in Table 1, the range resolution and azimuth resolution can be expressed as $\rho_r = 0.33$ and $\rho_a = 0.8596$ respectively through (4), (5). Simulation results are shown in Fig.6 and Fig.7.

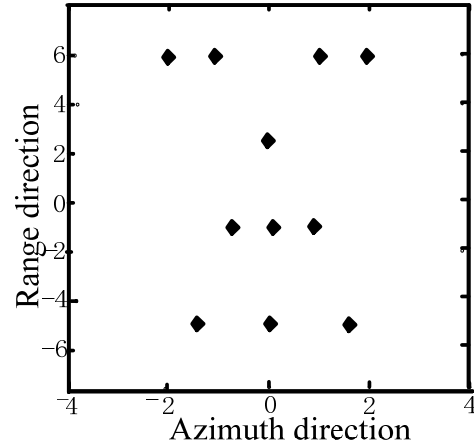
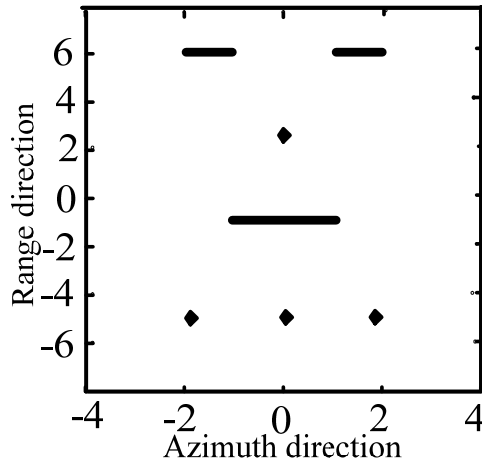


Figure 6. 2D image before azimuth data extrapolation. Figure 7. 2D image after azimuth data extrapolation.

From the Fig. 6 we can find that, before the azimuth data extrapolation range resolution is smaller than the interval of scatters so the scatters could be totally differentiated from range profile. But the azimuth resolution is not small enough, so the scatters could not be differentiated clearly form azimuth direction. Fig. 6 is the 2D image after data extrapolation and because the azimuth doppler resolution is improved, so the scatters could be totally differentiated from azimuth direction, and the correct 2D image is obtained.

Conclusion

ISAR imaging could be seen as high-resolution spectrum estimation problem after the movement compensation is completed. The space target bistatic 2D imaging problem in small rotation angle is researched in the paper. Simulation results prove that this algorithm could improve resolution in azimuth direction and could be used in practical environment. The research on how the algorithm could be used in lower SNR is also conducted.

Reference

- [1] Marco Martorella, James Palmer, John Homer, Brad Littleton, I. Dennis Longstaff, On Bistatic Inverse Synthetic Aperture Radar [J] IEEE Transactions on AES, 2007 (3): p. 1125-1134.
- [2] Delisle G.Y., Haiqing Wu, Moving target imaging and trajectory computation using ISAR [J]. IEEE Transactions on AES, 1994. 30 (3): p. 887-899.
- [3] Chen V.C., Shie Qian, Joint time-frequency transform for radar range-Doppler imaging [J]. IEEE Transactions on AES, 1998. 34 (2): p. 486-499.
- [4] Zhang Ya-biao, Zhu Zhen-bo, Tang Zi-yue, et al, Bistatic inverse Synthetic Aperture Radar Image Formation [J]. Journal of Electronics & Information Technology 2006. 28 (6): p. 969-972.
- [5] Zhu Yu-peng, Zhang Yue-hui, Wang Hong-qiang, et al, Modeling and Simulation of Bistatic ISAR Imaging for Moving Target [J]. Journal of System Simulation, 2009 (9): p. 2696-2699,2704.
- [6] Dong Jian, Shang Chao-xuan, Gao Mei-guo, et al, Research on Bistatic ISAR Speed Compensation of Space Target [J]. Journal of China Academy of Electronics and Information Technology, 2010 (1): p. 78-85.
- [7] Dong Jian, Study on key technologies of space target bistatic ISAR imaging [D]. Shijiazhuang: Ordnance Engineering College Dissertation, 2009.

- [8] Liang Hua-qiang, He Ming-yi, Zhang Lin-xi, Super-resolution ISAR imaging based on beam-space MUSIC [J]. Application Research of Computers, 2007. 24 (12): p. 361-363.
- [9] He Song-hua, Cheng Fan-yong, Chen Wei-bing, et al, Band-limited Signal Spectrum Estimation Based on Data Extrapolation and Maximum Likelihood Criterion [J]. Journal of Hunan University(Natural Science), 2009. 36 (3): p. 85-88.
- [10]Luo Jing-yang, Application of spectrum estimation in high-resolution radar imaging [D]. changsha: HUNAN UNIVERSITY DISSERATION, 2007.
- [11]Victor C. Chen, Andre des Rosiers, Lipps Ron, Bi-static ISAR range-doppler imaging and resolution analysis [C]. IEEE Radar Conference, 2009: p. 1-5.
- [12]Yang Zhen-qi, Zhang Yong-shun, Luo Yong-jun, Bistatic radar imaging system [M]. Beijing :Defense industry press, 1998.
- [13]Li Bing, Ci Lin-lin, Wan Jian-wei, et al, Comparison of ISAR Resolution Improvement Based on Two Cross Spectra Extrapolations [J]. Radar Science and Technology, 2006. 4 (6): p. 358-362, 381.

The Simulation Study of R410A Heat Pump System for Electric Automobile

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Keywords: Electric Automobile, R410A, Simulink, Heat Pump.

Abstract. R410A has a lot of advantages that are suitable for electric automobile air conditioning system. This paper discusses the research of R410A heat pump system for automobile by using Simulink simulation model, it offer help to the research of the electric automobile air conditioning in the future.

Introduction

At present, the quantity of automobile in our country is high, oil resources are very scarce, and a large number of automobile tail gas has seriously polluted the environment. Hybrid vehicles can only be a transitional product. In the future, the electric vehicle is bound to replace the traditional fuel vehicle. Without engine waste heat heating, electric vehicle can adopt heat pump air conditioning system. As a result of R12 destroy the atmosphere. R134a's efficiency is very low when it works in low temperature environment. In order to work properly in low temperature environment, the system can replace the refrigerant.

R410A includes two near azeotropic refrigerant R32 and R125, Two kinds of refrigerant each accounted for 50%. R410A has a stable, non-toxic, superior performance etc. R410A does not destroy the ozone layer, its air conditioning system working pressure is about 1.6 times of R22. R410A has good heat transfer performance and small pressure loss, it obtains good results when it works in the field of household air conditioning as a substitute for R22. Special compressor of R410A has also begun to be largely produced in our country.

Study on Properties of R410A

According to Fig. 1, R410A compared with R134a and R22, it has high saturation pressure, and has a good work pressure in low temperature environment. According to Fig. 2, with the decrease of the evaporation temperature, three kinds of refrigerant saturated steam specific volume increase, reduce refrigerant suction quantity of the compressor, lead to low system efficiency under the low temperature environment. R410A compared with R134a and R22, the specific volume is smaller under the low temperature environment, avoid low efficiency and insufficient heat when the system works under the low temperature environment. According to Fig. 3, R410A compared with R134a and R22, the Latent heat of vaporization of refrigerant is relatively high, it can take more heat away when it evaporate in the evaporator, and improve the efficiency of the system.

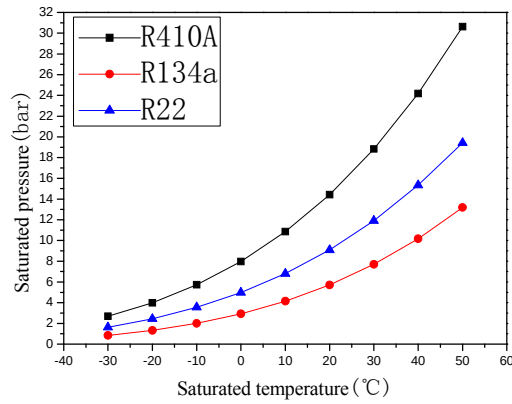


Figure 1. The relationship between evaporation pressure and evaporation temperature.

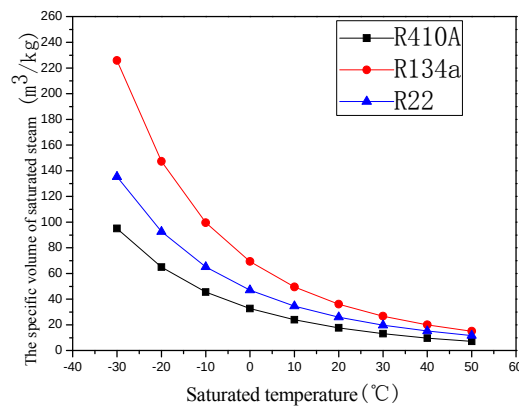


Figure 2. The relationship between suction volume and evaporating temperature.

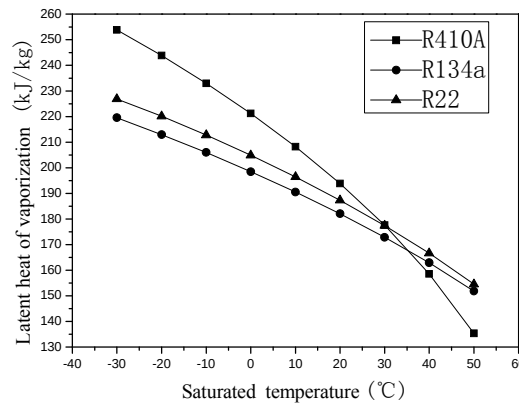


Figure 3. The relationship between latent heat of vaporization and evaporation temperature.

The advantages of R410A in various aspects it can replace the R134a used for automobile air conditioning.

The Simulation Study of Refrigerant for Automotive Air Conditioning

At the Zhongnan University, Su Jing et al fit R410A thermodynamic properties when it in When it is in the saturation temperature of $-40\sim 60\text{ }^{\circ}\text{C}$ and the superheat of $0\sim 60\text{ }^{\circ}\text{C}$, and analytical error. Compared with thermal models and reference data source, the error can be accepted in Engineering, it avoids the complex calculation program in the simulation and optimization calculation device, and improve the calculation speed and stability. [1]

At the Shanghai Jiao Tong University, Hong Yingchun et al simulate calculation for thermodynamic performance parameters and transmission parameters of R410a refrigerant based on Martin-Hou equation. Not only can guarantee the calculation speed and stability of the model, but also can reach a certain precision. [2]

At the Zhongnan University, Tang Weiwei et al research on the parallel flow condenser variable conditions, and compared R22 with the same environment. Through established the steady state distribution parameter model of the parallel flow condenser, simulated research on variable condition features of R410A and R22 under different condensing temperatures. Introduce the mass energy ratio, contrast of the two refrigerant of the required amount of refrigerant under the unit heat transfer. Results show that R410A has higher refrigerant heat transfer and low pressure drop. R410A has lower mass energy ratio. [3]

At the Wuhan Science and Technique University, Li Hongliang et al researched on R134a automobile air conditioning system by using the simulink simulation model. The results agree well with the experimental results. [4]

At the Shanghai Jiao Tong University, Zhao Yu et al researched on R1234yf as the automobile air conditioning system cycle fluid by using R1234yf as refrigerant and established a simulation model refrigeration system and components by using R1234yf as refrigerant, optimal performance experimental study and LCCP analysis. The analysis results show the superiority of R1234yf refrigerant, it can be used in the automotive air-conditioning system in the future. But R1234yf is expensive, is not suitable for automobile air conditioning system in China. [5]

Simulate Research on R410A Heat Pump Type Automobile Air Conditioning System Based on Simulink

Using Simulink to simulate the performance of R410A air conditioning system, can simulate the cooling capacity of the evaporator Q_0 , the heating capacity of the condenser Q_k , compressor power W_s , the refrigeration coefficient EER, the heating coefficient COP. The thermodynamic properties of R410A in the air conditioning according to Fig. 4, the condenser heat transfer process 2→3, the evaporator heat transfer process 4→1, the compressor working process 1→2.

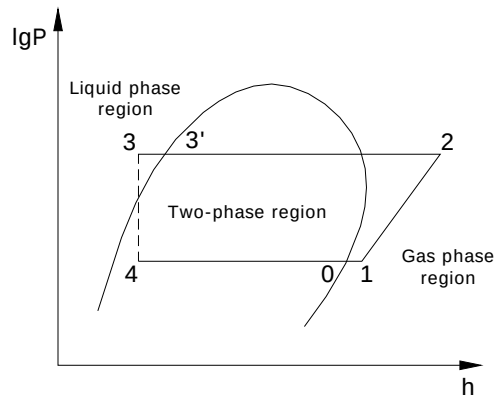


Figure 4. Pressure enthalpy diagram.

Calculate the compressor suction pressure and discharge pressure of R410A air conditioning by using Martin-Hou equation.

$$\lg P_{sat} = A + \frac{B}{T} + C \lg T + DT + E\left(\frac{F-T}{T}\right) \lg(F-T) \quad (1)$$

In the formula: P_{sat} —the refrigerant saturation pressure, kPa. T —the temperature of the refrigerant, K. A, B, C, D, E, F —calculated constants.

3.2 Calculate R410A enthalpy of saturated steam and superheated steam by using the Cleland formula.

$$h_{v1} = a_1 + a_2 t_{sa} + a_3 t_{sa}^2 + a_4 t_{sa}^3 \quad (2)$$

$$\frac{h_{v2}}{h_{v1}} = 1 + a_5 t_{sh} + a_6 t_{sh}^2 + a_7 t_{sh} t_{sa} + a_8 t_{sh}^2 t_{sa} + a_9 t_{sh} t_{sa}^2 + a_{10} t_{sh}^2 t_{sa}^2 \quad (3)$$

$$h = h_{v2} + a_{11} \quad (4)$$

In the formula: h_{v1} , h_{v2} —refrigerant enthalpy of saturation, superheated steam state refrigerant enthalpy, kJ/kg. t_{sa} , t_{sh} —refrigerant temperature of saturation, superheated steam state refrigerant enthalpy, °C. h —refrigerant enthalpy, kJ/kg. $a_1 \sim a_{11}$ —calculated constants.

The flow of refrigerant in the compressor can be calculated by the following formula:

$$G_r = 1.66 \times 10^{-8} V_h \lambda n / v_1 \quad (5)$$

In the formula: G_r —the refrigerant flow, kg/s. V_h —the displacement of the compressor, cm³/r. λ —the gas transmission coefficient compressor. n —compressor speed, r/min. v_1 — inspiratory volume compressor, m³/kg.

3.4 The flow of refrigerant into the evaporator can be calculated by the following formula:

$$G_r = k_a \sqrt{\rho(p_k - p_e)} \quad (6)$$

$$k_a = A_v + B_v t_e \quad (7)$$

In the formula: k_a —calculation of refrigerant flow coefficient. ρ — refrigerant density, kg/m³. t_e —the evaporation temperature of refrigerant, °C. A_v , B_v —calculated constants.

Through the above formula, we can calculate heat exchanger heat transfer and compressor power by using enthalpy difference method.

Summary

From comprehensive analysis we can know that R410A has advantages in various aspects. It is of great help for the study of R410A heat pump system for automobile by using Simulink simulation model, and it is bound to provide great help for researching and developing the electric automobile air conditioning system.

Reference

- [1] Su Jing, Hu Yixiong. Curve-fitting Method for Thermal Properties of R410A [J]. Refrigeration and Air Conditioning, 2012, 26 (1): 69-71.
- [2] Hong Ying-chun, Gu Bo, Han Hua. Fast Calculation Model of Thermodynamic and Transport Properties of R410a Refrigerant: II. Equation for Superheat Region [J]. Journal of Shanghai Jiaotong University, 2011, 45 (9): 1379-1383.
- [3] Tang Wei-Wei, Hu Yi-xiong, Zhu De-bin. Characteristic Comparison of Parallel-flow Type Condenser Using R410a and R22 [J]. Fluid Machinery, 2013, 41 (2): 66-70.
- [4] Li Hongliang, Xiong Wei. Simulation of R134a automobile air conditioning system with Simulink [J]. Journal of Wuhan University of Science and Technology, 2012, 35 (1): 69-72.
- [5] Zhao Yu, The performance research of R1234yf automotive air conditioning system [D]. Shanghai: Shanghai Jiao Tong University, 2013.

Target Parameter Estimation of Compressive Sensing Radar Using Sensing Matrix Optimization

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Keywords: Compressive Sensing Radar, Sensing matrix, Parameter estimation.

Abstract. To solve the problem that the parameter estimation accuracy of Compressive Sensing Radar (CSR) degrades when the sensing matrix and target information vector are mismatching, a CSR target parameter estimation method using sensing matrix optimization is proposed in this paper. Firstly, a distance-Doppler 2-dimensions sparse sensing model of CSR in the case of sensing matrix mismatching is established, and then the sensing matrix is calibrated through performing iterative optimization on sensing matrix mismatching error and target information vector. Compared with conventional CSR target parameter estimation algorithms, the proposed method dramatically reduces the estimation error of distance-Doppler parameter, consequently improving the accuracy and robustness of CSR target parameter estimation. Finally, the effectiveness and superiority of the proposed method are proven through computer simulations.

Introduction

In recent decades, utilizing sparsity and compressibility of signal to realize signal detecting and processing has been widely studied in information theory and signal processing field, which is used in pattern recognition, image processing, radar imaging and etc. [1]. Compressive sensing (CS) theory, proposed by Candès et al [2] in 2006, points out that through solving a norm optimization problem, a signal could be recovered from finite measured values with comparatively high possibility when it is sparse.

In reality, targets in interested area only occupy finite radar resolution cells. Since the radar echo signal is sparse, the applications of CS in radar signal processing has also achieved great attention [4]. Nevertheless, conventional Compressive Sensing Radar (CSR) fails to take the effects of sensing matrix mismatching error on target parameter estimation accuracy into consideration [5]. Furthermore, the corresponding Coherence of Sensing Matrix (CSM) of CSR is comparatively large and the matching relation between sensing matrix and target information vector is undesirable, giving rise to the low estimation accuracy of target parameter [6,7]. The recovery errors' upper bond through norm optimization algorithm is discussed in [8], while the sensing matrix and target information vector are mismatching. The effects of sensing matrix mismatching errors in CS on target information estimation accuracy have also been analyzed. It indicates that the recovering performance of sparse signals through conventional CS algorithms deteriorate drastically while the sensing matrix fails to represent targets precisely. The authors in [9] reduce the error of sparse signals recovery through increasing the times of measurements and the size of sensing matrix. However, with the improvement of sparse matrix resolution, the CSM of CSR increases correspondingly, resulting in the number of detectable targets decreasing as well as the ability of suppressing noise weakening.

To solve the problem that the parameter estimation of CSR degrades when the sensing matrix and target information vector are mismatching, a robust CSR target parameter estimation method is proposed in this paper. Firstly, a distance-Doppler 2-dimensions sparse sensing matrix of CSR in the case of sensing matrix mismatching is established. Secondly, the corresponding sensing model is optimized through performing iterative optimization on sensing matrix mismatching error and target

information vector. And then the distance-Doppler parameter is recovered with high-accuracy by using sensing matrix optimization. Eventually, a target parameter estimation algorithm using sensing matrix optimization is proposed. The proposed algorithm improves robustness and adaptability of CSR by increasing target parameter estimation accuracy in the case of sensing matrix mismatching.

CSR Sparse Sensing Model

Distance-Doppler 2-Dimensions Parameter Sparse Sensing Model

Assume that the plural envelop of radar transmitted pulse signal is $s_t(t)$, and there are K targets in the interested area, the distance of the k th target is r_k , the radial velocity is v_k , the scattering coefficient is α_k , and the wavelength of radar transmitted signal is λ . Taking no account of radar channel attenuation, the echo signal could be expressed as

$$s_r(t) = \sum_{k=1}^K \alpha_k s_t(t - \tau_k) e^{-j2\pi f_{dk}t} + n(t) \quad (1)$$

where $t = [1, 2, \dots, L]^T T_s$, T_s being Nyquist sampling period, $f_s = 1/T_s$, $L = T_r/T_s$, T_r being repetition period of transmitted signal, $\tau_k = 2r_k/c$ being the time delay of the k th target, c being velocity of light, $f_{dk} = 2v_k/\lambda$ being the corresponding Doppler phase-shift of k th target, and $n(t)$ being the noise vector of receiver.

Let $\mathbf{T} = \begin{pmatrix} \mathbf{0}_{L \times (L-1)} & \mathbf{1} \\ \mathbf{I}_{(L-1) \times (L-1)} & \mathbf{0}_{(L-1) \times 1} \end{pmatrix}$ be the unit time-delay matrix, $\mathbf{F} = \text{diag}(\omega_M^0, \omega_M^1, \dots, \omega_M^{L-1})$ be the unit frequency-shift matrix, where $\omega_M = \exp(j2\pi/M)$, $\Delta f_d = 2\pi/M$ is the size of Doppler resolution cell, $\text{diag}(v_1, v_2, \dots, v_M)$ denotes M -order diagonal matrix. Thus, Eq. 1 can be rewritten as

$$s_r = \sum_{k=1}^K \alpha_k \mathbf{F}^{f_k} \mathbf{T}^{t_k} s_t + n \quad (2)$$

Assume that the radar interested distance is $r \in [r_{\min}, r_{\max}]$ and Doppler frequency is $f \in [2v_{\min}/\lambda, 2v_{\max}/\lambda]$, let $t_i = K_{r_{\min}} + i, i = 0, 1, \dots, K_r - 1$, $K_{r_{\min}} = \text{round}(2r_{\min} f_s/c)$, $f_j = K_{f_{\min}} + j, j = 0, \dots, K_f - 1$, $K_f = \text{round}(2(v)/(\lambda \Delta f_d))$, $K_r = \text{round}(2(r) f_s/c)$ and $K_{f_{\min}} = \text{round}(2v_{\min}/(\lambda \Delta f_d))$.

Then the echo signal vector of single target can be denoted as $\mathbf{h}^{ij} = \mathbf{F}^{f_j} \mathbf{T}^{t_i} s_t$, and the Overcomplete Time delay-Frequency shift Basis (OTFB) matrix is $\mathbf{H} = [\mathbf{h}^{0,0}, \dots, \mathbf{h}^{0,(K_f-1)}, \mathbf{h}^{1,0}, \dots, \mathbf{h}^{(K_r-1),(K_f-1)}]$. Therefore, the equivalent sparse sensing model of Eq. 2 can be denoted as

$$s_r = \mathbf{H}\boldsymbol{\theta} + n \quad (3)$$

where $\boldsymbol{\theta} = [\theta_1, \theta_2, \dots, \theta_{K_r K_f}]^T$ denotes the target information vector, and when the condition $(t_i, f_j) = (\tau_k, f_k)$ is satisfied, $\theta_{i,j} = \alpha_k$.

CS Sparse Model under Sensing Model Mismatching

In general, the sensing matrix of CSR is established through uniform quantizing distance-Doppler 2-dimension parameter. When the targets do not locate in the CSR resolution grid exactly, which is called mismatching between sensing matrix and target information vector; it will be result in OTFB matrix incapable of representing targets, consequently impairing the accuracy and effectiveness of CSR target parameter estimation.

According to Eq. 3, when target distance-Doppler parameter mismatches with CSR sensing matrix, assume that K targets in interested area are randomly distributed in distance resolution unit grids, and the k th target's velocity mismatches with velocity resolution unit grid, pre-set in CSR system, that is to say $\tilde{f}_k = 2v_k/\lambda_\Delta f_d \notin \mathbf{K}_f$.

If $f_k \in \mathbf{K}_f$ is the closet grid to $\tilde{f}_k$, then the k th target's Doppler frequency-shift element can be approximately expressed as

$$\mathbf{F}^{\tilde{f}_k} \approx \mathbf{F}^{f_k} + \frac{d\mathbf{F}^{f_k}}{df_k}(\tilde{f}_k - f_k) \quad (4)$$

where $\mathbf{K}_f = [K_{f_{\min}}, K_{f_{\min}} + 1, \dots, K_{f_{\max}} + K_f]$. Thus, the k th target echo signal vector can be rewritten as

$$\mathbf{h}^{\tau_k \tilde{f}_k} \approx \left(\mathbf{F}^{f_k} + \frac{d\mathbf{F}^{f_k}}{df_k}(\tilde{f}_k - f_k) \right) \mathbf{T}^{\tau_k} \mathbf{s}_t \quad (5)$$

Therefore, the sparse sensing model in the case of sensing matrix mismatching can be denoted as

$$\mathbf{s}_r = \mathbf{H}_{\text{mis}} \boldsymbol{\theta} + \mathbf{n} \quad (6)$$

where $\mathbf{H}_{\text{mis}} = \mathbf{H} + \mathbf{Z} \cdot \text{diag}(\boldsymbol{\beta})$, $\mathbf{Z} = [\mathbf{z}_{0,0}, \dots, \mathbf{z}_{(K_r-1), (K_f-1)}]$, $\mathbf{z}_{i,j} = \frac{d\mathbf{F}^{f_j}}{df_j} \mathbf{T}^{t_i} \mathbf{s}_t$ and $\boldsymbol{\beta} = [\beta_1, \beta_2, \dots, \beta_{K_r K_f}]^T$

denotes sensing matrix mismatching error and satisfies

$$\beta_i = \begin{cases} \tilde{f}_k - f_k & , i \in L(\boldsymbol{\theta}) \\ 0 & , \text{else} \end{cases} \quad , \quad i = 1, 2, \dots, K_r K_f \quad (7)$$

Performing project measurement using $N \times L$ matrix $\boldsymbol{\Phi}$ on echo signal $\mathbf{s}_r$, then the $N \times 1$ measured value can be get

$$\mathbf{y} = \boldsymbol{\Phi} \mathbf{s}_r = \boldsymbol{\Phi} (\mathbf{H}_{\text{mis}} \boldsymbol{\theta} + \mathbf{n}) = (\mathbf{A} + \mathbf{B} \mathbf{v}) \boldsymbol{\theta} + \mathbf{e} \quad (8)$$

where $\boldsymbol{\Phi}$ is Gaussian random matrix, $\mathbf{A} = \boldsymbol{\Phi} \mathbf{H}$ denotes $N \times K_r K_f$ sensing matrix, $\mathbf{v} = \text{diag}(\boldsymbol{\beta})$, $\mathbf{B} \cdot \text{diag}(\boldsymbol{\beta})$ is the $N \times K_r K_f$ sensing matrix mismatching error corresponding to $\mathbf{A}$, $\mathbf{B} = \boldsymbol{\Phi} \mathbf{Z}$, and $\mathbf{e} = \boldsymbol{\Phi} \mathbf{n}$ represents $N \times 1$ noise vector.

Target Parameter Estimation Method while System Model Mismatching

Derivation of the Proposed Algorithm

When sensing matrix mismatches with target information vectors, conventional CSR algorithms take no account of the influences of sensing matrix mismatching errors, failing to ensure the effectiveness and robustness of target parameter estimation.

To solve these problems, a CSR target parameter estimation method using sensing matrix optimization is proposed in this paper. The proposed algorithm combines dantzig selector (DS) algorithm and least-absolute shrinkage and selection operator (LASSO) algorithm to achieve a high-resolution CSR target parameter estimation result by performing optimization on sensing matrix.

First assume the vector of sensing matrix mismatching error $\boldsymbol{\beta} = \boldsymbol{\theta}$ in lack of prior information. According to CS theory, the optimization problem expressed in Eq. 8 can be solved by working out L1 norm optimization under the circumstance of noise, thus obtaining target information vector

$$\begin{aligned} \hat{\boldsymbol{\theta}} &= \min \|\boldsymbol{\theta}\|_1 \\ \text{subject to } &\|\boldsymbol{\Omega}^H (\mathbf{y} - \boldsymbol{\Omega} \boldsymbol{\theta})\|_\infty < \mu \end{aligned} \quad (9)$$

where constant μ is relevant to noise variance and $\mathbf{\Omega} = \mathbf{\Phi} \mathbf{H}_{\text{mis}}$. We can solve this optimization problem by DS algorithm.

Take the estimation value of target information vector obtained by solving Eq. 9 to Eq. 8, then we can get

$$\mathbf{y} = (\mathbf{A} + \mathbf{B}\mathbf{v})\hat{\boldsymbol{\theta}} + \mathbf{e} \quad (10)$$

Then it can be achieved by compiling Eq. 10 that,

$$\begin{aligned} \mathbf{y} &= (\mathbf{A} + \mathbf{B}\mathbf{v})\hat{\boldsymbol{\theta}} + \mathbf{e} \\ &= \mathbf{A}\hat{\boldsymbol{\theta}} + \mathbf{B}\mathbf{v}\hat{\boldsymbol{\theta}} + \mathbf{e} \\ &= \mathbf{A}\hat{\boldsymbol{\theta}} + \mathbf{B}\boldsymbol{\varphi}\boldsymbol{\beta} + \mathbf{e} \end{aligned} \quad (11)$$

where $\boldsymbol{\varphi} = \text{diag}(\hat{\boldsymbol{\theta}})$.

From Eq. 5 we know that $\boldsymbol{\beta}$ shares the same sparsity with target information vector. So Eq. 11 can be transformed to

$$\mathbf{y}' = \boldsymbol{\gamma}\boldsymbol{\beta} + \mathbf{e} \quad (12)$$

where $\mathbf{y}' = \mathbf{y} - \mathbf{A}\hat{\boldsymbol{\theta}}$, and $\boldsymbol{\gamma} = \mathbf{B}\boldsymbol{\varphi}$.

Hence, Eq. 12 can be retreated as another CS optimization problem using $\boldsymbol{\beta}$ as the target signal and the optimization problem can be denoted as

$$\boldsymbol{\beta} = \underset{\boldsymbol{\beta}}{\text{argmin}} \|\mathbf{y}' - \boldsymbol{\gamma}\boldsymbol{\beta}\|_2^2 \quad \text{subject to} \quad \|\boldsymbol{\beta}\|_1 \leq \frac{1}{2} K\varphi \quad (13)$$

From Eq. 13 we know that the CS optimization problem can be perfectly solved by LASSO algorithm.

Take the vector of sensing matrix mismatching error $\boldsymbol{\beta}$ from solving Eq. 13 to Eq. 9, estimation value of target information vector can be re-solved. Repeat the process mentioned before until the minus between two measurement values of target information vector less than the certain predefined threshold. That is,

$$\left\| \hat{\boldsymbol{\theta}}^{(p+1)} - \hat{\boldsymbol{\theta}}^{(p)} \right\|_2^2 / \left\| \hat{\boldsymbol{\theta}}^{(p)} \right\|_2^2 \leq \Delta \quad (14)$$

Stop iteration at this moment, then $\hat{\boldsymbol{\theta}}^{(p+1)}$ we get is the target information vector, where Δ is the certain predefined threshold.

Implementation of the Proposed Algorithm

Assuming that the number of signals is known or correctly estimated, the proposed algorithm can be summarized as follows.

- 1) Initialize with $p=0$ and $\boldsymbol{\beta}^0=0$.
- 2) Solve Eq. 9 to get the estimate $\hat{\boldsymbol{\theta}}^{p+1}$,
- 3) Solve Eq. 13 to get the estimate $\boldsymbol{\beta}^{p+1}$, and then set $p=p+1$;
- 4) Stop the iteration if expression Eq. 14 is satisfied. Otherwise, go back to step 2).

Simulations

To verify the effectiveness of the proposed method, assume CSR interested distance range $[0, cT_r/2]$, interested Doppler frequency range $[\pi/3, 5\pi/6]$, the size of Doppler resolution cell $\Delta f_d = 2\pi/M$, $\omega_M = \exp(j 2\pi/M)$, $M = 128$.

Assume that there are two targets in interested area, the corresponding distance-Doppler parameter are $\{(5, 10.4); (10, 4.5)\}$, every target's scattering coefficient parameters are complex Gaussian distributed with mean value being 0, variance being 1, and Signal-to-Noise Ratio (SNR) range within

$[-10\text{dB}, 30\text{dB}]$, 500 Monte Carlo simulations are carried out to evaluate the performance of the proposed method.

Fig. 1 shows the Root-Mean-Square Errors (RMSE) of the proposed method and its three competitors versus different SNR. Fig. 2 shows the RMSE of sensing matrix mismatching error versus different SNR.

From Fig. 1, it can be seen that the target information vector estimation errors of CoSaMP, OMP, ROMP algorithms are comparatively large, for the reason that distance-Doppler resolution cells, pre-set in system, fail to represent targets precisely. These algorithms have no suppressive ability to sensing matrix mismatching errors, leading to estimate of targets deteriorating with SNR declining, severely impairing estimation accuracy of target parameters. In contrast, the proposed method performs estimations on system sensing matrix mismatching errors with high precision during iteration, and step by step uses the above results to calibrate quantized results of corresponding parameters in CSR interested area, improving matching relation between sensing matrix and target information vectors. Therefore, the proposed method possesses higher estimation accuracy of target scattering coefficients and better robustness.

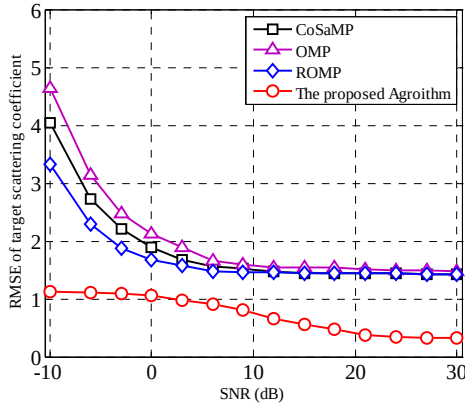


Figure 1. RMES of target scattering coefficient versus SNR.

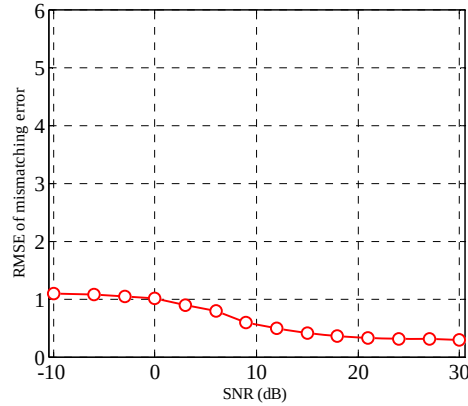


Figure 2. RMSE of sensing matrix mismatching error versus SNR.

Conclusions

Improving estimation accuracy of target information through calibrating and optimizing sensing model is a significant aspect of CSR, which has crucial theoretical and practical meaning of improving the effectiveness and robustness of CSR. To solve the problem that distance-Doppler parameter estimation performance degrades when sensing matrix mismatches with target information vectors, a novel CSR target parameter estimation algorithm using sensing optimization is proposed in this paper, which remarkably reduces the effects of sensing matrix mismatching errors on CSR. Compared with conventional CSR algorithms, the proposed method improve the estimation performance of target information vector, enhancing effectiveness and robustness of CSR target parameter estimation.

Acknowledgement

This research was financially supported by National Natural Science Foundation of China under Grant 61401204.

References

- [1] Li G., Zhu Z., Yang D., et al. On projection matrix optimization for compressive sensing systems [J]. IEEE Transactions on Signal Processing, 2013, 61 (11): 2887-2898.

- [2] Julio M., Cuo S., Lawrence C., et al. Task-driven adaptive statistical compressive sensing of Gaussian mixture models [J]. IEEE Transactions on Signal Processing, 2013, 61 (3): 585-600.
- [3] Julian W., Simon H. and Martin K. Analysis based blind compressive sensing [J]. IEEE Signal Processing Letters, 2013, 20 (5): 491-494.
- [4] Miosso C., Borries R. and Pierluissi J. Compressive sensing with prior information requirements and probabilities of reconstruction in l1-minimization [J]. IEEE Transactions on Signal Processing, 2013, 61 (9): 2150-2164.
- [5] Tello A., Lopez D. and Mallorqui J. A novel strategy for radar imaging based on compressive sensing [J]. IEEE Transactions on Geoscience and Remote Sensing, 2010, 48 (12): 4285-4295.
- [6] Liu Z., Wei X. and Li X. Aliasing-free moving target detection in random pulse repetition interval radar based on compressed sensing [J]. IEEE Sensor Journal, 2013, 13 (7): 2523-2534.
- [7] Zhu H., Geert L. and Georgios G. Sparsity-cognizant total least-squares for perturbed compressive sampling [J]. IEEE Transactions on Signal Processing, 2011, 59 (5): 2002-2016.
- [8] Chi Y., Louis S., Ali P., et al. Sensitivity to basis mismatch in compressed sensing [J]. IEEE Transactions on Signal Processing, 2011, 59 (5): 2182-2195.
- [9] Daniel C., Parastoo S. and Rodney K. Effects of basis-mismatch in compressive sampling of continuous sinusoidal signals [C]. Proceedings of the 2010 2nd International Conference on Future Computer and Communication, Canberra, 2010: 2739-2743.
- [10] H. Mohimani, B.Z. Massoud and C. Jutten. A Fast Approach for Overcomplete Sparse Decomposition Based on Smoothed L0 Norm [J]. IEEE Trans.Signal Process., 57 (1): 289-301, 2009.
- [11] R. Tibshirani. Regression shrinkage and selection via the Lasso [J]. Journal of the Royal Statistical Society, 58 (1): 267-288, 1996.

Analysis of Hollow Shaft Cross Wedge Rolling Forming Laws Based on Finite Element Theory

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Keywords: Cross wedge rolling, Hollow shaft, Forming law.

Abstract. Compared with the solid shaft parts, it is difficult to form the hollow shaft parts. Because of the existence of a free surface, there is an irregular flow in the forming process, so the inner circle is irregular. In this paper, the analysis of the rolling process of the typical hollow shaft is modeled by using DEFORM-3D software, the mandrel rolling process of the hollow shaft is simulated, and the useful data are obtained through the comparison and analysis of the whole rolling process force field and strain field variation, and without a mandrel the rolling process of hollow shaft is analyzed and the choice of parameters in the actual production has certain directive significance.

Introduction

Cross wedge rolling technology is with two rotating rolls of the wedge shaped die installed in the same direction, in the wedge shaped mould, wedge salient drives the workpiece rotation, and makes the blank generate continuous small deformation, eventually rolled into the wedge groove all stepped shaft. The deformation of cross wedge rolling is mainly radial compression and axial extension. Cross wedge rolling technology is mainly used for production of shaft parts, such as automobile gearbox gear shaft. They are mainly some different diameter of the solid shaft, of the traditional solid shaft in cross wedge rolling processing regularity in the process of analyzing people already have mature experience. With the continuous application of the hollow shaft in engineering, people begin to study the process of the cross wedge rolling.

Compared with the solid shaft parts, the difficulty of forming hollow parts greatly increases because of the free surface in the heart place. During the forming process it is very easy to produce irregular flow, so the round shape is irregular. In addition, because of the change of wall thickness during machining process, it can be converted, and it can't estimate the change of wall thickness before and after rolling. Because it is hollow, so the stiffness of some parts is greatly reduced, which results in that if inappropriate process parameters are chosen, it is easy to make some parts flattening and ultimately being unable to do the work. The cross wedge rolling process of hollow ladder shaft is simulated by the finite element technology, and we get the three-dimensional state of stress and strain distribution of cross wedge rolling, which provides reasonable theoretical basis for further solving practical problems.

Simulation Model of Hollow Shaft Cross Wedge Rolling

Double roll cross wedge rolling is a complicated process of plastic forming. It is necessary to analyze and calculate the whole process. Here we choose DEFORM-3D software as an analytical tool.

The interaction between the die and the workpiece in the process of cross wedge rolling. While the mold is installed on the upper and lower roll, to design the mold first according to the upper and lower two roll of the center distance to determine the base circle diameter. The model of the device is selected by the stable rotating condition of the workpiece. The maximum diameter of the blank is 53mm, so we choose the two roller center distance for the D46-630 630mm equipment to rolling.

After selecting a good device, it should determine the size of the base circle radius R_0 , which is calculated as follows:

$$R_0 = \frac{S - d_{r0} - 2\delta}{2} \quad (1)$$

Δ refers to the gap between the workpiece and the maximum diameter of the mold surface of the base circle, d_{r0} is the largest diameter hot blank; S is the center distance of the two roll. Then we must determine the combination of the shape and angle of the die in the experiment, and then can be designed by the mould design after the forming angle and the angle of the stretch.

Through the choice of element type, mesh and the selection of material model, contact and other boundary conditions, etc.. An analytical model is set up as shown in Fig. 1.

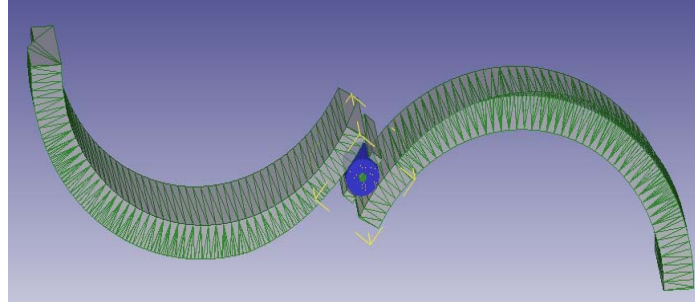


Figure 1. Analysis model.

The Process Simulation Analysis of Wear a Hollow Shaft Cross Wedge Rolling

For the hollow shaft of the cross wedge rolling process usually has two kinds of methods, one is the mandrel rolling, a is wearing a mandrel rolling. In the actual production to wear with mandrel rolling so we wear, analysis of rolling with mandrel.

The working process of cross wedge rolling is generally divided into: wedge section, wedge finishing, finishing, stretching stage, shear section. In the actual production, wedging fine whole usually cases are elided, and shearing segment is only use the cutting knife will cut off the head and the workpiece separately, there is no mechanical defects. Therefore, this paper mainly studies properties of wedged parts and broadening the two stages of the period. Wedge, wedge die tip has just entered into the maximum deformation of rolled piece until, at this stage of the workpiece stress from zero to a maximum value. The stretching stage is the main deformation stage of the workpiece, and most of the work in this section is finished. The following section mainly analyzes the wedge and widening section.

Simulation in the process of selecting the parameters of die for forming angle $\alpha=35^\circ$, spread angle $\beta=7^\circ$, section shrinkage rate $\psi=64\%$, the outer diameter of the workpiece blank 53 mm diameter 13mm. The stretching length of the workpiece is 50mm. Such a set of simulated data is of sufficient length of rolling, deformation can be more comprehensive analysis of the workpiece with different when should stress, strain and velocity distribution.

Stress Analysis of Rolling Process. Fig. 2 is a wedge section in the middle of the stress distribution, it is easy to see in the mould wedge hole began deformation because of the existence of the mandrel so in Z direction diameter unchanged, mandrel subjected to radial flow in the metal barrier, so here the equivalent should force and radial stress are significantly larger. Within the pore radius in the y to start larger. This is due to the friction in the forming mold surface and positive pressure in the cutting force to work together to push the metal in the tangential flow. Die and workpiece contact surface of workpiece applied axial pressure pushes the axial metal flow, and in places not in contact with the mould is subjected to tensile force is perpendicular to the maximum pressure reaches the maximum value. The distribution of tangential stress is similar to the axial stress distribution, which is formed by the pressure stress and the flow of metal tangential flow.

Fig. 3 is the stress distribution in the middle time of the two roll cross wedge rolling. In this stage the workpiece has entered a stable stage of rolling, experience the wedging zone after the die to form a V-shaped groove is more and more wide, the contact area of the workpiece and its dies reached maximum value. Fig. 3 (a) shows that the workpiece's equivalent stress is on the contact surface and mold, and for has been separated from the forming surface part has also been gradually by elliptic into several nearly circular, so this part of the amount of deformation with respect to the slope is much smaller, equivalent stress becomes small. This part of the equivalent stress to the equivalent stress of the wedging zone is also much smaller. In the farther distance contact the local equivalent stress is gradually decreasing, but here the strip has been rolled to flat, so and on both sides of the baffle plate occurred in the contact, resulting in additional stress

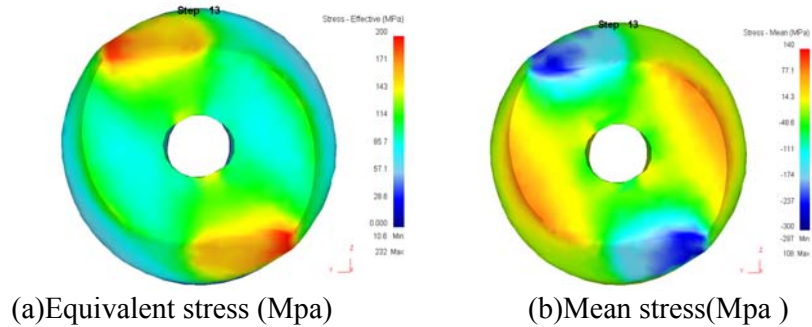


Figure 2. Wedge middle section stress distribution.

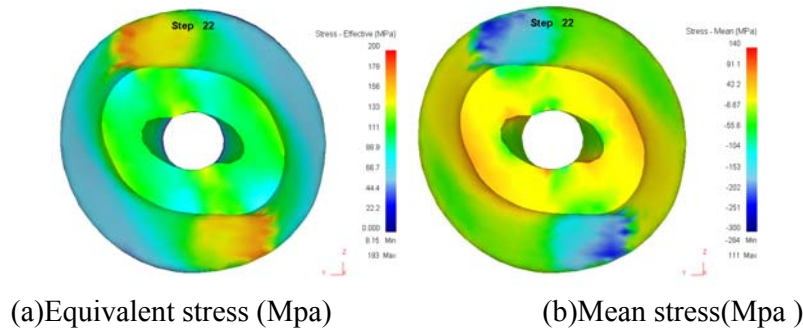


Figure 3. Stretched intermediate section stress distribution.

From the middle of the workpiece to take six points on the surface, the point of the location shown in Fig. 4, Fig. 6 is the equivalent stress and time of the corresponding diagram. In the initial stage of the die and the workpiece no contact equivalent stress is zero; in the die and workpiece contact equivalent stress sharply increased, probably in the 0.742s reached the maximum value at mould wedge to maximum depth, in after the completion of the wedge, symmetrical surface shape by elliptical to round rolling process in equivalent stress fluctuation; after deformation of the basic end, equivalent stress is relatively stable period; finishing section, equivalent stress rapidly decreased to zero. The point near the inner diameter of the effect is smaller, but inside the six point, due to the mandrel in the 1.48s appear larger values of.

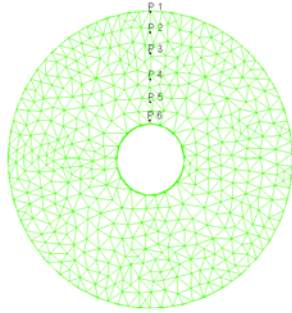


Figure 4. Position of a point on the plane of symmetry.

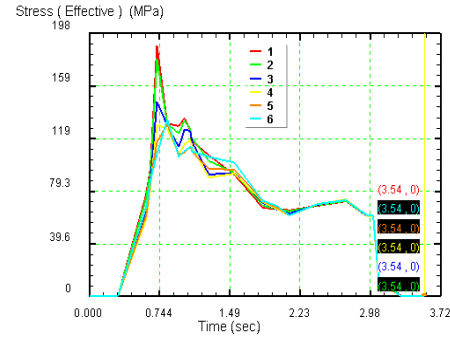
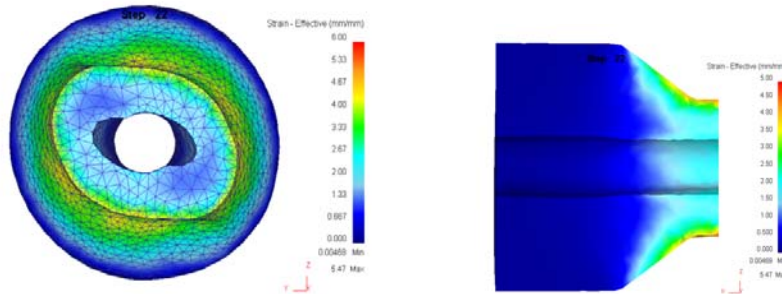


Figure 5. Relationship between equivalent stress and time.

Strain Analysis of Rolling Process. Fig. 6 is a strain distribution diagram of the stretching section of the double roll cross wedge rolling simulation. Strain is mainly focused on the outer surface of the workpiece in the spiral surface and the minimum diameter of the cylindrical surface intersection, strain depth is also the largest. Through the distribution of strain on the cross section can be seen strain increases as the diameter of the spiral surface, the inner hole strain in the diameter of the place of the minimum value is the biggest, because by plug flow in the local metal workpiece and the barrier. It can be seen from the longitudinal section that the larger the surface strain is close to the surface



(a)Equivalent strain of cross section

(b)Equivalent strain of longitudinal section

Figure 6. Strain distribution in the middle section of the stretched section.

Same as shown in Fig. 4, in the rolling plane of symmetry uniform obtain 6 points, these points are almost in a straight line and center distance and distance closer, using the software point tracking function can observation effect change with time change. As shown in Fig. 7. In rolling at the beginning of the mold and the workpiece is not exposed, strain have a zero; with the mold of the wedge.

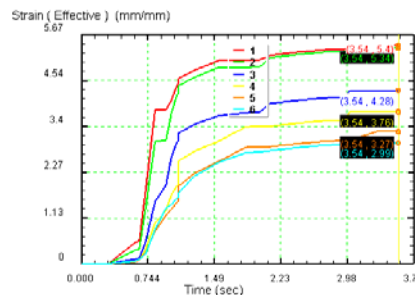


Figure 7. The relationship between the effect of each point and time.

With the contact part of the mould of metal by forming surface pressure to deformation, in wedge segment of the work-piece is rolling out a V-shaped groove, in this stage the other effect variable are increasing rapidly. With mold to wedge workpiece within a week have reached the minimum

diameter and metal deformation is mainly concentrated to flow on both sides and in the phase plane of symmetry does not have too big radial deformation. Therefore, at this stage the strain will a slow increase, the workpiece most of deformation are done on stretching segment, the metal side of the axial moving side radial movement and subsequent deformation of the effect variable will change big, and the choice of the six points in the process of stretching segment, and effect variable basic have reached a stable value. Then entered into finishing section, in the mold of workpiece has completed the broadening of shape finishes, so at this stage in the deformation quantity is not great, but some of the details of the adjustment. In addition to the 6 points, the more recent the distance center, the smaller the value of the 1 points, the value of the minimum 5 points. This trend, the more close to the center part of the deformation of the smaller. 6 point strain compared 4, 5 points is big, is mainly because of hole workpiece metal flow by the mandrel of the barrier, the radial motion changed to cut to the movement, thus inner hole of the effect variable is the corresponding increase.

Comparison of Rolling Process of not Wearing the Hollow Shaft Mandrel.

A rolling with mandrel rolling process and the influence of parameters of the rolling process discussed earlier, discussed below and mandrel of equivalent should force, the equivalent strain effects.

Fig. 8 is the distribution image of broadening stage under two kinds of equivalent stress, equivalent strain. It can be seen from the Fig. 8, equivalent stress in places far from the center of the not too big different, in near center place of mandrel equivalent stress significantly greater than those without mandrel to, and in the radial direction is especially obvious. This is mainly because the effect of the mold metal to flow inward, without mandrel diameter for free surface, and a mandrel will barrier metal radial flow, and in the radial produced larger stress. Two cases of effective strain distribution is basically the same, but also in near a mandrel of the effect variable is obviously larger than that without mandrel. Greater damage coefficients are concentrated in after forming the outer diameter of the edge portion, and a mandrel thickness is obviously larger than that without mandrel.

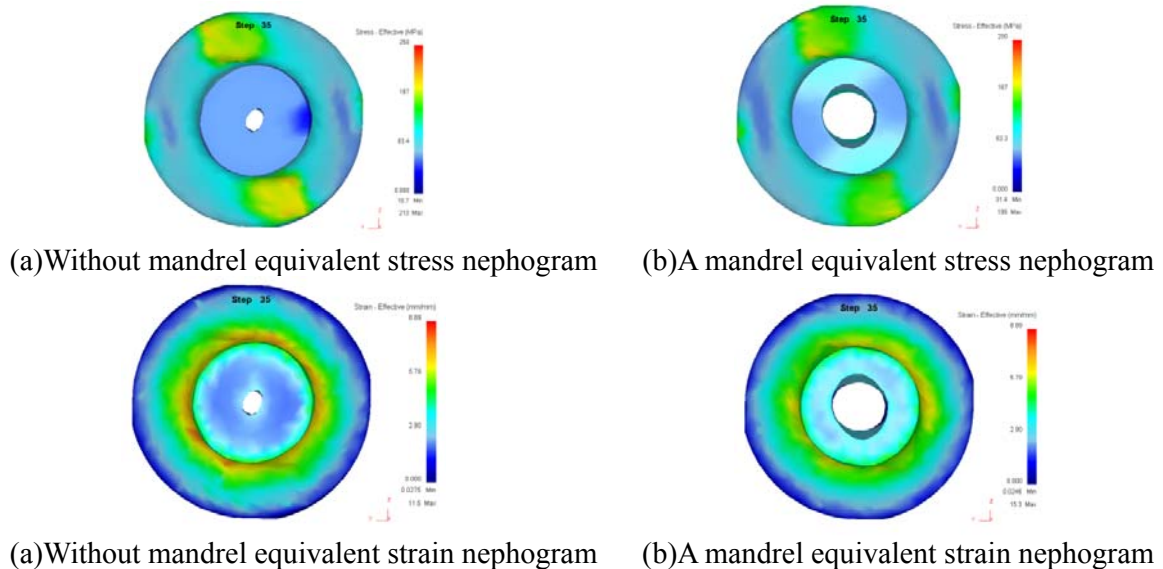


Figure 8. Equivalent stress, equivalent strain.

Equivalent with mandrel maximum stress than non mandrel at. In the wedging zone, with mandrel workpiece near the center point of the stress is significantly higher than that of without mandrel, mandrel, the radial stress is relatively large. By (c), (d), and with the mandrel of effect value is significantly greater than that in the non mandrel duration, without mandrel metal flow for radial flow edge to the axial flow and mandrel, metal radial flow by mandrel obstacles, mainly to the tangential and axial flow. The same speed wedge mold, the effect of wall thickness on the small

number etc.. In the process of rolling workpiece, workpiece by mandrel barrier, deformation is mainly distributed in the axial direction, thus causing large axial pull stress and strain, resulting in damage of large value. Using the core rod to ensure good internal hole diameter, but easy to because of too much should stress and strain caused by defects. Therefore, the use of mandrel should consider the influence of various factors in order to avoid the defects of.

Conclusions

(1) Finite element software is used in the simulation of rolling process of cross wedge rolling of hollow shaft, and we obtain the ideal hollow shaft rolling model, prove the feasibility of using mandrel rolling of hollow shaft.

(2) Based on certain parameters with a hollow mandrel shaft of the rolling process simulation analysis, it is concluded that the hollow shaft strength and strain at each stage of stress and the wedging zone and broadening the stress field, strain field distribution are analyzed, the mandrel rolling mechanism has a certain role in revealing.

(3) By means of a series of contrasting simulation experiments, the rules how different parameters influence the workpiece roundness can be concluded: the bigger the forming angle is, the better the roundness is; broadening angle is smaller, roundness is better; the greater the section shrinkage rate is, the better the roundness is; temperature is lower, roundness is better.

(4) By comparing without mandrel rolling, we use the core rod to ensure good internal hole diameter, but it is easy to cause defects because of too much should stress and strain. Therefore, the influence of various factors should be considered when using mandrel in order to avoid the defects.

Reference

- [1] YuanWen Sheng Wang Zhong Lei, Based on Delphi cross wedge rolling die CAD system. Forging and stamping equipment and manufacturing technology 2005, 06: 93-95
- [2] Hu Zhenghuan, Zhang Kangsheng, Wang Baoyu, et al. Parts forming technology and simulation. Beijing: Metallurgical Industry Press, 65-108124180-184, 2004: 4-5
- [3] WangZhong Lei, Yuan Wen Sheng, The cross wedge rolling forming process demonstration system research and development. Forging and stamping equipment and manufacturing technology. 2006.4:73-75 8.
- [4] Li Chuanmin, Wang Xiangli, Hua Jun door.DEFORM5.03 metal forming finite element analysis tutorial. Beijing; Mechanical Industry Press..2007;
- [5] Du Fengshan, Tu min, computer simulation and process optimization of microstructure evolution during controlled rolling process [J]. Journal of mechanical engineering, 2002,38 (increasing): 65-69
- [6] Chen Jun, Wang Yu Guo, Wang Yue Xian, et al. Metal plastic forming finite element simulation of hexahedral element subdivision technology [J]. I path of plastic, 2002, 9 (3): 1-4
- [7] Li Yao, the principle of metal plastic forming. Mechanical Industry Press, 2004.7
- [8] Zhang Kangsheng, Hu Zhenghuan, et al. Analysis of the factors affecting the mechanical parameters of the wedge rolling force of the [J]. heavy machinery, 2002, (4): 29-33
- [9] Yuan Wensheng. The motion principle and workpiece rotation [J].plate type of cross wedge rolling forging technology, 1994 (1): 53-56
- [10] Yuan Wensheng, Ma Hongfang, DuanHui, Development and application of cross wedge rolling equipment [J]. Journal of Shandong College of Construction Engineering, 2005, 20 (3): 80-85.
- [11]Ying Fu Qiang, Zhang Geng Chao, Xiao Yong Pan. 3D FEM simulation in metal plastic forming closed. Forging equipment and manufacturing technology, 2003 (5): 10-13

[12]Chen Jun, Wang Yu Guo, Wang Yue Xian, et. Metal plastic forming finite element simulation of hexahedral elements and then dividing the research field of technology progress. Journal of engineering plastic, 2002, 9 (3): 1-4.

Analysis on the Influence of Cross Wedge Rolling Process Parameters on the Process of the Hollow Shaft Parts

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Keywords: Cross wedge rolling, Hollow shaft, Process parameters, Process influence.

Abstract. For the hollow shaft parts by cross wedge rolling, due to the phase compared with the solid shaft, its stiffness is greatly reduced, which results in that if the process parameters are not suitably selected, it will be easy to make some parts flattening, and ultimately the hollow shaft cannot be formed. Through several groups of cross wedge rolling process parameters, using the finite element simulation technology to analyze the cross wedge rolling process of hollow shaft, we obtained the three-dimensional strain stress distribution in the hollow shaft in the rolling process. And the rules how parameters such as broadening angle, forming angle, section shrinkage rate influence the stress and strain of workpiece are also analyzed.

Introduction

Cross wedge rolling technology as a kind of advanced metal plastic forming method, with the advantage of high material utilizing rate, high production efficiency, long service life of the die, small molding equipment tonnage, has attracted more and more attention. Presently, this technology is largely applied to producing a variety of shaft parts, such as automobile gearbox gear shaft. They are mainly some solid shafts of different diameters. With the continuous application of the hollow shaft in engineering, people begin to study the process of the cross wedge rolling. Compared with the solid shaft parts, the difficulty of forming hollow parts greatly increases because of the free surface in the heart place. It is very easy to produce irregular flow during the forming process, which results in the irregular round shape. In order to make the process of cross wedge rolling smooth, workpiece must be in the upper and lower mold driving to rotate, and the parameters such as main section shrinkage rate ψ , forming angle α , stretching angle β and so on selected in cross wedge rolling process have to meet certain conditions. Combinations of these parameters for different after rolling force on the workpiece, the influence of stress and strain to achieve the process of cross wedge rolling normal. Therefore, this paper studies the influence of different parameters on the process of the hollow shaft forming based on the finite element principle.

The Influence of Process Parameters on the Rolling Force of the Hollow Shaft

The size of rolling force has important influence on the quality of the workpiece. Rolling force too large may be causing workpiece necking or other defects, which is not conducive to the workpiece forming; on the contrary, if the rolling force is too small, light cannot penetrate, serious work may appear unable to achieve the desired size requirements. In addition, the size of the rolling force is also an important parameter for selecting and designing the cross wedge rolling mill.

Parameters selected in the simulation of forming die are: forming angle $\alpha=35^\circ$, spread angle $\beta=7^\circ$, section shrinkage rate $\psi=64\%$, the outer diameter of the workpiece is 53 mm and the inner diameter is 13mm. The stretching length of the workpiece is 50mm. Such a set of simulated data is of sufficient length of rolling and allows for a more comprehensive analysis of workpiece deformation under different internal variation.

Influence of Forming Angle α on Rolling Force. Two sets of experiments were chosen to analyze the influence of forming angle on rolling force. These two groups have common

broadening angle, section shrinkage, inner diameter, outer diameter, temperature. Respectively for the broaden angle, section shrinkage rate, inner diameter of $d=13\text{mm}$, outer diameter $D=53\text{mm}$, temperature $T=1150$. Forming angle: $\alpha_1 = 35^\circ$, $\alpha_2 = 30^\circ$

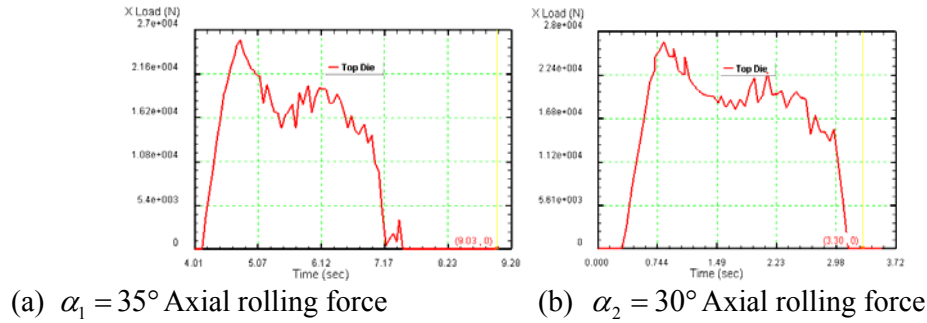


Figure 1. the rolling force of different forming angles.

From Fig. 1 It is visible that the bigger the forming angle is, the bigger the maximum value of rolling force is, this is mainly because of the forming angle is the rolling force in the axial component is bigger also. In the stage of forming die, the bigger the wedge angle is, the greater the axial pressure is. In the stable stage, the size of the axial rolling force is not obvious with the change of the forming angle. The value of axial rolling force reaches the peak at the end of the wedge and radial rolling force of maximum value appear in the middle section of the broadening.

Effect of Stretching Angle β on Rolling Force. Two sets of experiments were chosen to analyze the influence of stretching angle on rolling force. They have common widened angle, section shrinkage, inner diameter, outer diameter, temperature, namely forming angle $\alpha=35^\circ$, section shrinkage rate $\psi = 64\%$, inner diameter $d=13\text{mm}$, outside diameter $D=53\text{mm}$, temperature $T=1150$. The two groups of the widening angle is: $\beta_1=5^\circ$, $\beta_3=7^\circ$.

Fig. 2 is the change curve of the rolling force at different angles. In the wedging zone, the axial rolling force is bigger and increases faster when the stretching angle is bigger mainly because of the widening of the angle, equivalent to rolling speed increase, at the same time, the volume of the same volume of deformation, so this part of the deformation rate to be faster. The larger the stretching angle is, the flow rate of the metal in the axial direction will increase, so that the radial and tangential rolling force will reduce, and the axial rolling force will become more and more. Likewise, the contact area of the workpiece and the die increase with the increasing of the stretching angle, which can cause the metal flow to be blocked, but decrease the axial rolling force.

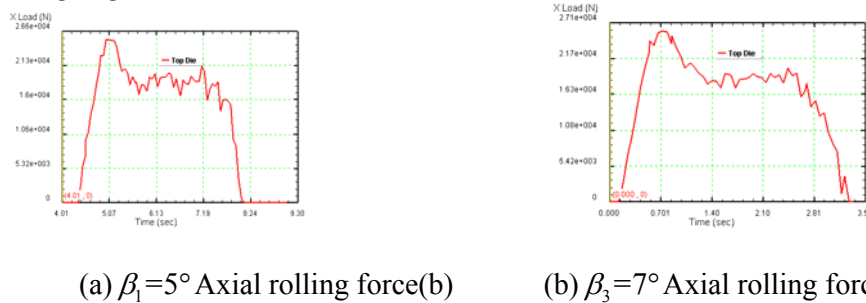


Figure 2. the rolling force of the different angle of stretching.

Section Shrinkage Rate ψ on Rolling Force. Two sets of experiments were chosen to analyze the influence of forming angle on rolling force. These two groups have the common broadening angle, the section shrinkage rate, the inner diameter, outer diameter, the temperature, the forming angle $\alpha=35^\circ$, the stretching angle $\beta=7^\circ$, the inner diameter $d=13\text{mm}$, the outer diameter $D=53\text{mm}$, the temperature $T=1150^\circ\text{C}$. The section shrinkage rate of the three groups was: $\psi_1=54\%$, $\psi_3=74\%$

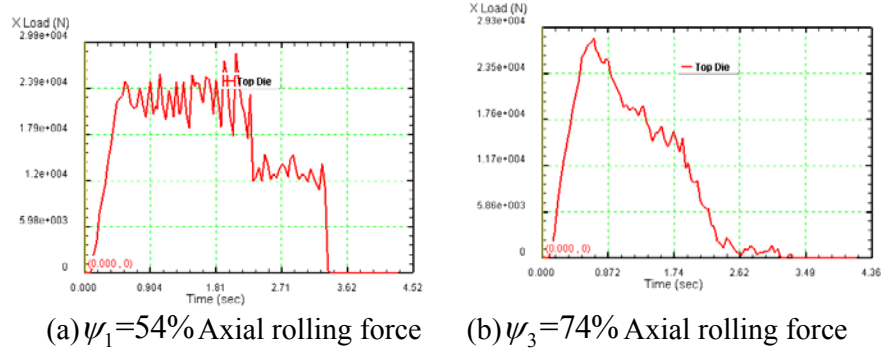


Figure 3. Rolling force of different section shrinkage.

Fig. 3 is the variation of the rolling force of the different section of the rolling force. In $\psi_1=54\%$ At the time of the rolling process, there is a slight slippage in the rolling process. In $\psi_3=74\%$, the phenomenon of shrinkage occurred during the middle part of the rolling, and the fracture occurred in the later stage of the rolling. The greater the deformation of the metal is, the axial rolling force, the rolling force and the shear force will increase. In the contraction section rate of wedge rolling force is obviously some stretching stage. This is mainly because the wedging zone if section shrinkage rate is high, the rolling depth, metal flow will be violent. However, in broadening stage, deformation of the workpiece has become stable and gentle, so the impact degree will be smaller. Maximum axial rolling force values appeared at the end of the wedge segments and radial and tangent to the rolling force is completed in the stretching segment.

Effect of Process Parameters on the Equivalent Stress of Hollow Shaft

In the rolling process of the workpiece, the change law and distribution of the equivalent stress have a direct influence on the forming of the workpiece. It can directly reflect the size of the force in the workpiece, and can determine the risk. Therefore, it is of great significance to study the magnitude and change law of equivalent stress. Forming angle, stretching angle, shrinkage rate, temperature and other factors have a significant effect on the equivalent stress of rolling. Because the equivalent stress of each point in the workpiece has a different value at different time points, the contrast analysis is not possible. Select a point at the center of 10mm in the distance on the plane of symmetry, this analysis at different time stress.

Effect of Forming Angle α on the Equivalent Stress. In the same experiment, the effect of the forming angle on the equivalent stress was analyzed in two groups. These two groups have a common broadening angle, section shrinkage, inner diameter, outer diameter, and temperature, respectively, the broadened angle $\beta=7^\circ$, the section shrinkage $\psi=64\%$, the inner diameter of $d=13\text{mm}$, outer diameter $D=53\text{mm}$, temperature $T=1150^\circ\text{C}$. Forming angle: $\alpha_1=35^\circ$, $\alpha_2=30^\circ$.

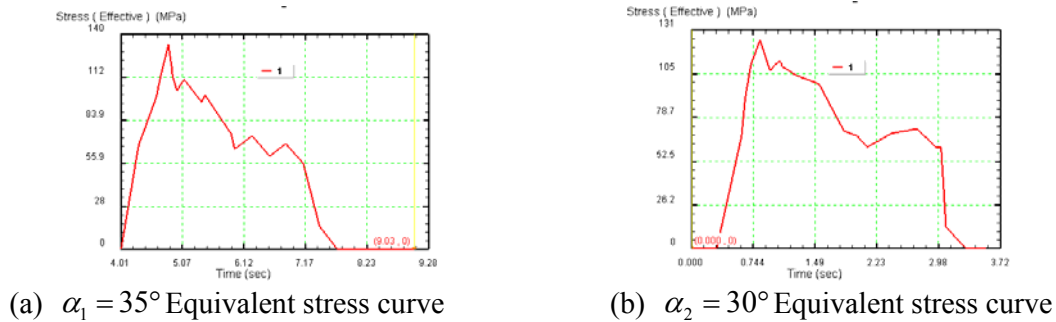


Figure 4. Equivalent stress curve of different forming angles.

From the image above, the maximum equivalent stress increases with the increase of the forming angle. The wedging zone, forming angle is bigger, equivalent should be faster growth, the same value, due to the wedging zone of deformation for equal radial and axial deformation, forming angle when the axial stress and radial stress force is small, the size difference between the two. In the stretching section, the larger the forming angle is, the larger the equivalent stress is because the deformation of this stage is mainly axial extension, therefore, the axial stress is greatly increased. Wedging zone at the end of the equivalent stress reaches the maximum, then the stable rolling, so the equivalent stress begins to decrease, in fine the whole rolling force rapidly decreases to zero.

Effect of Broadening Angle β on the Equivalent Stress. In the experimental group, the two groups had the same size, the section shrinkage, the inner diameter, outer diameter, the temperature, the forming angle $\alpha=35^\circ$, the section shrinkage $\psi=64\%$, the inner diameter of $d=13\text{mm}$, the outer diameter of the $D=53\text{mm}$, the outer diameter of the $T=1150^\circ\text{C}$, the. The three groups of the widening angle is: $\beta_1=5^\circ, \beta_3=7^\circ$.

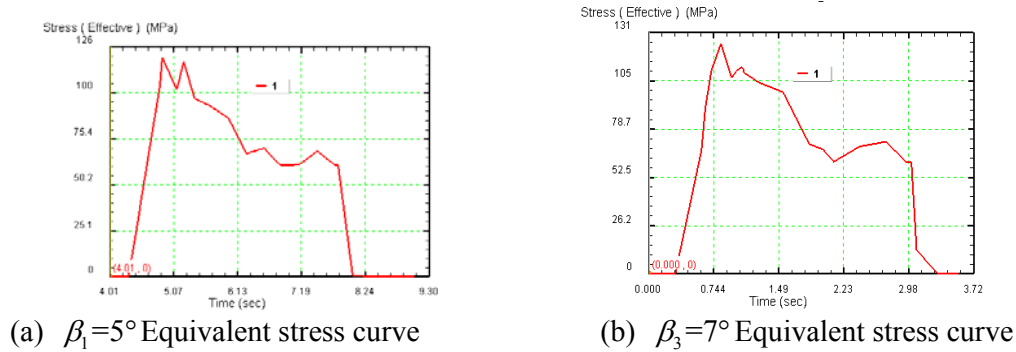


Figure 5. The equivalent stress curve of different stretching angles.

By the graph, the maximum equivalent stress of the workpiece with the increasing of the stretching angle is also increased. This is because the larger the angle of the larger the distortion of the workpiece, the faster the flow of the metal, so it caused a larger equivalent stress. The smaller the stretching angle, the longer the maximum equivalent stress. Also due to the widening of the angle, the greater the speed of the die in the axial direction of the mold, the greater the angle, the more quickly the equivalent stress is reached.

Effect of Section Shrinkage Rate ψ on Equivalent Stress. The selected two groups have the common broadening angle, the section shrinkage rate, the inner diameter, outer diameter, the temperature, the forming angle $\alpha=35^\circ$, the stretching angle $\beta=7^\circ$, the inner diameter $d=13\text{mm}$, the outer diameter $D=53\text{mm}$, the temperature $T=1150^\circ\text{C}$. The section shrinkage rate of the three groups was: $\psi_1=54\%, \psi_3=74\%$

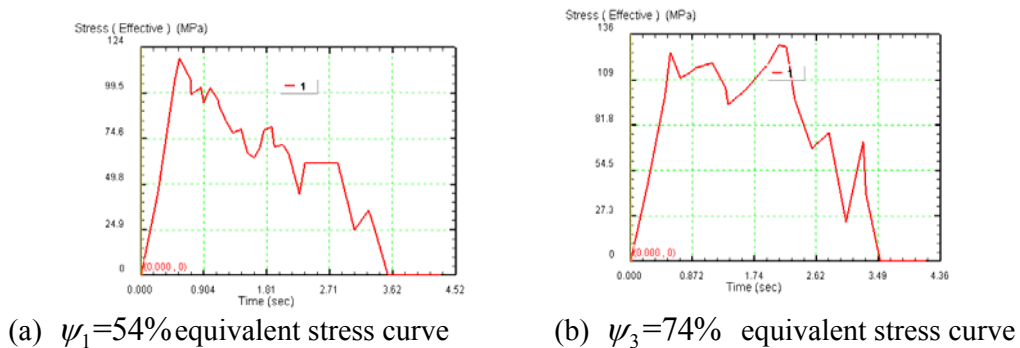


Figure 6. Equivalent strain curve of different section shrinkage.

From the graph, we can know that the maximum equivalent stress value of the workpiece increases with the increase of the section shrinkage rate. Section shrinkage size decision diameter

variation of the rolling stock and by the preceding section shrinkage rate influence on rolling force shows its value is bigger, anisotropic rolling force are larger. Thus the equivalent stress with variable. In the stretching section, the equivalent stress increases with the increase of the section shrinkage rate, but in this stage, the stress trend is decreased. But in (b) we found on stretching segment stress still increases. This is mainly because at this stage due to the section shrinkage rate is too large to produce the necking defect, so stress increases first and then decreases.

Effect of Process Parameters on Equivalent Strain of Hollow Shaft

In the rolling process of the workpiece, the change law of the equivalent strain and the distribution of the workpiece have a direct influence on the forming of the workpiece. It can directly reflect the deformation of the workpiece around the size, and can determine the risk. Therefore, it is of great significance to study the magnitude and change law of equivalent strain. Forming angle, stretching angle, shrinkage rate, temperature and other factors have a significant effect on the equivalent strain of rolling. Because the effect of each point of the workpiece changes at different time points with different values, the comparison analysis is not possible. Select a point at the center of 10mm in the distance on the plane of symmetry, this analysis of strain variation in different time condition.

Effect of Forming Angle α on the Equivalent Strain. In the experimental group, the two groups have the common broadening angle, the section shrinkage rate, the inner diameter, outer diameter, the temperature, the stretching angle $\beta=7^\circ$, the section shrinkage rate $\psi=64\%$, the inner diameter $d=13\text{mm}$, the outer diameter $D=53\text{mm}$, the temperature $T=1150^\circ\text{C}$. Forming angle: $\alpha_1=35^\circ$, $\alpha_2=30^\circ$.

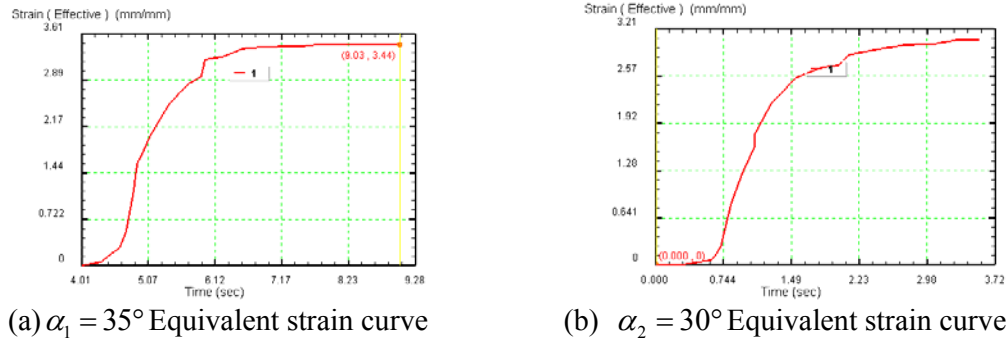


Figure 7. Equivalent strain curves of different forming angles.

The size of the forming angle has little effect on the equivalent strain of the workpiece. With the increase of the forming angle, the effect of such effects is increased, but the increase is limited. Mainly because of the forming angle is bigger, the axial stress is bigger, which has caused a large axial strain, and so the effect becomes large. In addition, the change of the equivalent strain of the forming angle is faster.

Effect of Stretching Angle β on the Equivalent Strain. In the experimental group, the two groups had the same stretching angle, the shrinkage rate of the cross section, the inner diameter, outer diameter, the temperature, the forming angle $\alpha=35^\circ$, the section shrinkage rate $\psi=64\%$, the inner diameter of $d=13\text{mm}$, outer diameter $D=53\text{mm}$, the temperature $T=1150^\circ\text{C}$. The two groups of the widening angle is: $\beta_1=5^\circ$, $\beta_3=7^\circ$

We can see from Fig. 8, the wedge section and stretching segment of the first half, the strain changes with the stretching angle is not great. Into broadening in the second half and the fine whole and effect of slow increase and stable at a value, (a), (b) value is always increasing, and the former value is bigger than that of the latter. This is mainly because in stretching angle was less than 7, the workpiece produced a certain degree of necking phenomenon, axial strain change ambassador to

effect change continues to increase. Which (a) than (b) of the shrinkage of the neck to be serious. In addition, stretching angle is bigger in the wedging zone strain rate is greater.

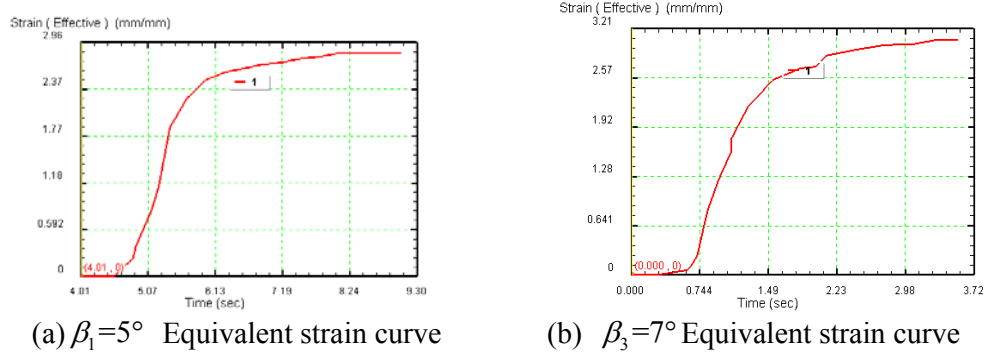


Figure 8. Equivalent strain curves of different broadening angles.

Effect of Section Shrinkage Rate ψ on Equivalent Strain. Two groups were selected to analyze the influence of the forming angle on the equivalent strain. The two groups have the same size, the section shrinkage, the inner diameter, outer diameter, the temperature, the forming angle $\alpha=35^\circ$, the stretching angle $\beta=7^\circ$, the inner diameter of $d=13\text{mm}$, outer diameter $D=53\text{mm}$, the temperature $T=1150^\circ\text{C}$. The section shrinkage rate of the three groups was: $\psi_1=54\%$, $\psi_3=74\%$.

As shown in Fig. 9, the section shrinkage rate is significantly affected by the equivalent strain. With the increase of the section shrinkage rate, the effect of the change is also increased. This is because the cross section shrinkage rate is more easy to penetrate the rolling force into the workpiece, it is more prone to deformation. But the selected section shrinkage rate cannot be too large, otherwise there will be shrinkage neck or even pull off etc..

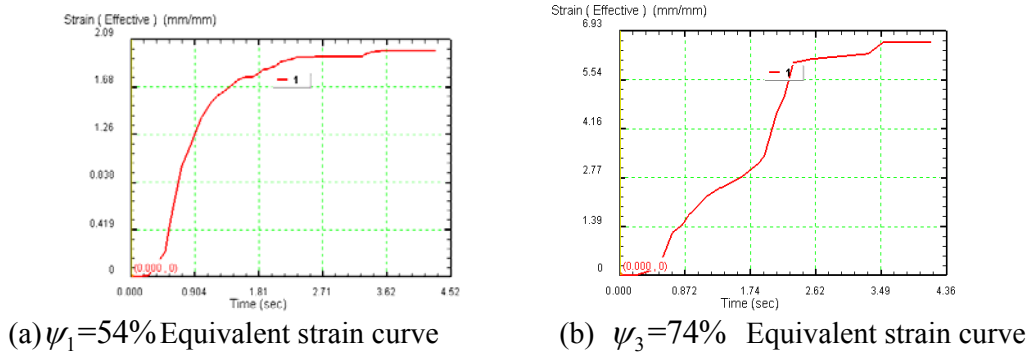


Figure 9. Equivalent strain curve of different section shrinkage.

Conclusion

(1) Analysis of the influence of different process parameters (forming angle, spread angle, section shrinkage rate) on the rolling force is: the larger the forming angle is, the greater the axial rolling force is, and radial rolling force and rolling force is small: broadening angle is bigger, axial rolling force is large, and radial rolling force and cut to the rolling force is smaller; section shrinkage rate is high, axial rolling force, radial rolling force, cut to the rolling force are increased.

(2) A detailed analysis of the effects of forming angle, stretching angle, section shrinkage rate and other parameters on the workpiece equivalent stress is made, and we get the conclusion that the larger the forming angle is, the bigger the equivalent stress is, the less influence on the effect; the bigger the stretching angle is, the bigger the equivalent stress is, and the smaller the effect is; the larger the section shrinkage is, the bigger the equivalent stress and the equivalent stress is, etc..

(3) By means of a series of contrasting simulation experiments, the rules how different parameters influence the workpiece roundness can be concluded: the bigger the forming angle is, the better the

roundness is; broadening angle is smaller, roundness is better; the greater the section shrinkage rate is, the better the roundness is; temperature is lower, roundness is better.

Reference

- [1] Song Yuquan. Development prospect of continuous local plastic forming [J]. China Mechanical Engineering, 2000, 11 (1-2): 65-67.
- [2] Hu Zhenghuan, Zhang Kangsheng, Wang Baoyu, et al. Parts forming technology and simulation. Beijing: Metallurgical Industry Press, 65-108124180-184, 2004: 4-5.
- [3] Zhang Kangsheng, Wang Baoyu, Jin Ping Liu,. Wedge cross rolling of hollow parts wall thickness variation rules of the experimental study [J]. Forging technology, 2000, (2): 34-3.
- [4] Li Chuanmin, Wang Xiangli, Hua Jun door.DEFORM5.03 metal forming finite element analysis tutorial. Beijing; Mechanical Industry Press. 2007.
- [5] Du Fengshan, Tu min, computer simulation and process optimization of microstructure evolution during controlled rolling process [J]. Journal of mechanical engineering, 2002, 38 (increasing): 65-69.
- [6] Chen Jun, Wang Yu Guo, Wang Yue Xian, et al. Metal plastic forming finite element simulation of hexahedral element subdivision technology [J]. I path of plastic, 2002, 9 (3): 1-4.
- [7] Li Yao, the principle of metal plastic forming. Mechanical Industry Press, 2004.7.
- [8] Zhang Kangsheng, Hu Zhenghuan, et al. Analysis of the factors affecting the mechanical parameters of the wedge rolling force of the [J]. heavy machinery, 2002, (4): 29-33
- [9] Yuan Wensheng. The motion principle and workpiece rotation [J].plate type of cross wedge rolling forging technology, 1994 (1): 53-56
- [10] Yu Hanqing, Chen Jinde. Principle of metal plastic forming. Beijing: Mechanical Industry Press, 1998: 141-142.
- [11] Ying Fu Qiang, Zhang Geng Chao, Xiao Yong Pan. 3D FEM simulation in metal plastic forming closed. Forging equipment and manufacturing technology, 2003 (5): 10-13.
- [12] Chen Jun, Wang Yu Guo, Wang Yue Xian, et. Metal plastic forming finite element simulation of hexahedral elements and then dividing the research field of technology progress. Journal of engineering plastic, 2002, 9 (3): 1-4.
- [13] Shangwu Xiong, Xianghua Liu, Guodong Wang. Analysis of Non-Steady State Edging in Roughing Trains by a Three-dimensional Rigid-plastic Finite Element Method.International Journal of Machine Tool&Manufacture, 2000, (40): 157.

Design of Orientation and Turning Device for Oral Thermometer

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Keywords: Oral thermometer, Automatic orientation, Automatic feed, Swing action, Classification.

Abstract. The slot on the conveyor tray is designed based on the analysis on the shape of the oral thermometer. The thermometers arrived at the tray through the bucket bin, and show the preferred orientation after turned over up to twice by the claws. And then, the thermometers enter the rotary table in order to take the photo of the height of the mercury column clearly. The vertex angle of the thermometer will swing smoothly due to the rotation of the rotary table. It is indicated by actual use that the productivity of calibration and classification is increased from 1/sec to 4/sec.

Introduction

The oral thermometer is one of the conventional instruments used in health system and household. There is a big market for the oral thermometer because of its low price and ease of use [1].

The oral thermometer is a triangular prismatic glass tube with a capillary inside connected to a mercury reservoir. Through the convex side of the triangular prism which plays a magnifier role, the height of the mercury column in the capillary can be observed conveniently. According to the oral temperature in measurement, the height of the mercury column in the capillary will rise to a certain graduation due to the thermal expansion [2].

As a result of the inaccuracies that existed in the diameter of capillary and the volume of the mercury reservoir, the absolute height and lift magnitude shown in different thermometer are different even if under the same temperature. In other words, the position of graduation mark and the scale space in different thermometer should also be different. Therefore it's necessary to calibrate the thermometer for the benchmark temperature and then to classify and print the thermometer according to a certain accuracy standard [3]. It relies on manual operation before now with low productivity. The production quality is also unsatisfactory.

The automatic calibration device and classify device are designed [4][5][6][7]. The image measurement technology is introduced in the device instead of manual measurement. Through the PLC control system, the calibration and classify operation are done automatically [8][9][10]. In order to take the photo of the mercury column [11][12][13][14], the aligner feeder is requested. Through which, the function of automatic orientation and swing action could be achieved.

Theory of Orientation and Swing Action

Analysis on the Shape of the Oral Thermometer

It's necessary to know the geometric dimension of the triangular prism oral thermometer. According to the national standard of glass thermometer (GB1588-2001), the dimension of this kind of thermometer is the same as Fig. 1.

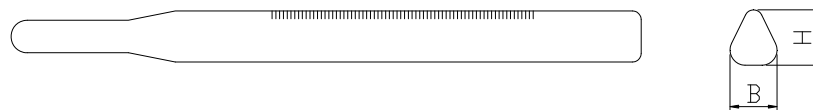


Figure 1. structure and dimensions of the oral thermometer.

As shown in Fig. 1, dimension H of cross section is 5.2mm, dimension B is 4.5mm which show that cross section of the thermometer is close to the isosceles triangle. According to this feature, the orientation scheme is analyzed below.

Method of Orientation

According to the section shape of the oral thermometer, bucket bin is used to feeding. The thermometer is let out to the tray on the conveyor belt. The tray is designed to be rigid, with some slots on it whose section shape is similar to that of thermometer. Several possible cases are shown in Fig. 2. It's obviously that when the vertex angle of the thermometer faced to the left side, the slot fits the thermometer well. The thermometers in other position are above the lever drawn in Fig. 1. In order to get desirable position, two claws are installed above the tray to turn the thermometers over (only twice at the most) which exposed above the level.

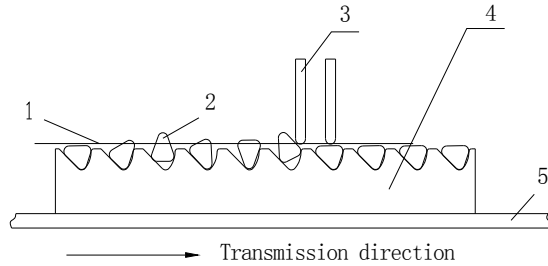


Figure 2. principle of orientation (1.level; 2.thermometer; 3.claw; 4.tray; 5.conveyor belt).

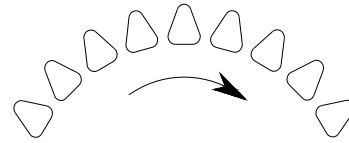


Figure 3. principle of swing action.

Principle of Swing Action

A device is designed to get the swing action in convey the thermometers as illustrated in Fig. 3. Through this device, the vertex angle of the thermometer can face to the camera above in different angle.

General Structure Designs

Based on the theories described above, the orientation device and swing device of the thermometer have been designed together in transport line as illustrated in Fig. 4. Each of the trays, with the thermometers on it, is connected with the conveyor belt by a bolt. Through the belt wheel (which is not shown in Fig. 4), the conveyor belt transports the thermometers from left to right. After passing through the claw, the thermometers will show a preferred orientation. And then, the thermometers will be sent into the slots located in the edge of the rotary table. These slots have the same slot pitch as those on the trays. As a result, the feed movement of thermometers plays the role of driving source, which led to the clockwise rotation of the rotary table. Through this, the thermometers on the rotary table will swing smoothly.

Design of the Orientation Device

Structure Design of Tray

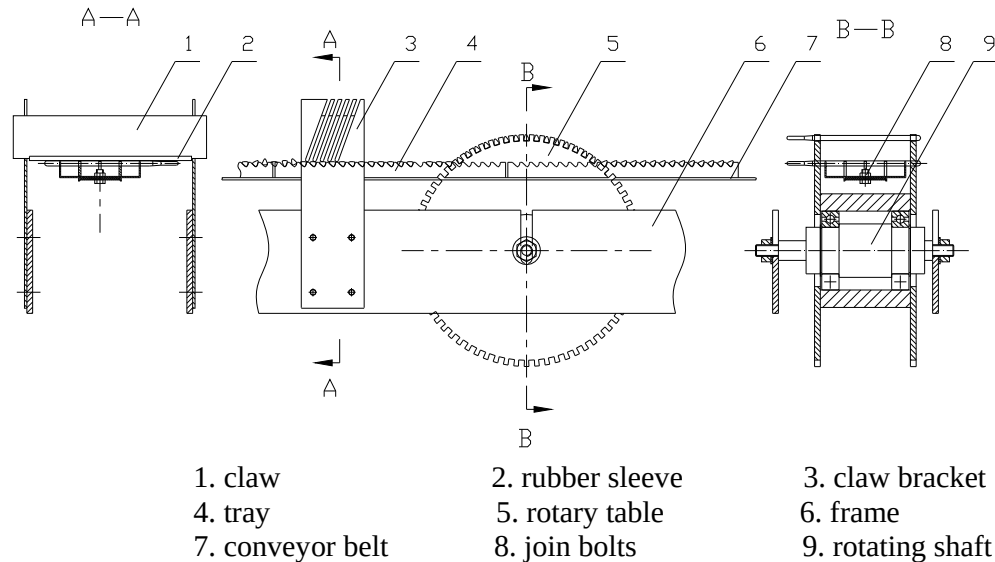


Figure 4. structure of orientation and swing device of the thermometers.

According to the orientation theory above, using the stainless steel plate with the thickness of 1-2mm, the tray frame of "U" shape is got through the bending processing. On both edges of the tray, uniformly spaced positioning slots are produced. These slots are used as sliding guides. Through these guides, the thermometer is pushed into collection box in follow - up classification process. As the guide slot, two fulcrum is not stable. Therefore, three supporting ribs made by stainless steel plate with the thickness of 1-2mm are welded in the tray. And slightly larger size slots are cut on the surface of ribs. Larger size of slots can avoid over-positioning and guide the thermometer.

The tray unit is assembled with several independent trays, because the unit is fixed to flexible conveyor belt during the transmission.

Each tray has the same structure and size at both ends so that the pitch between the adjacent trays installed on the conveyor belt is equal.

There are two methods. In the first, the adjacent trays are connected where the two slots are connected. With this method, burrs at the end of the tray will be a problem, or enlarging the pitch between the adjacent trays will decrease the efficiency of the transmission. In the second, the connecting part is designed to occupy the width of just one pitch, and this vacant position will be recognized using the special software. In this way, the transmit rhythm and tray structure will be not affected. The second method is adopted in this case, as shown in Fig. 5.

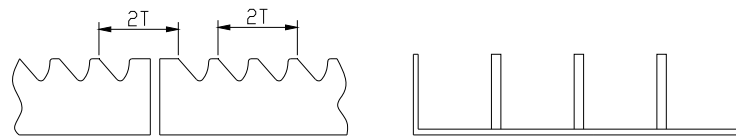


Figure 5. tray structure and relative position.

Connection Type of Tray and Conveyor Belt

As mentioned above, several independent trays work droved by the metal conveyor belt. Through this, the feeding of the thermometers is carried out.

During the transmission, it's necessary that the connection between the tray and belt is reliable and the belt with the trays connected must rotates around the belt wheel smoothly. The connection

type of tray and conveyor belt is shown as in Fig. 6. The tray is connected to the belt with a bolt in the center of the tray. At the bottom of the tray, some bayonets downward are produced to make the tray parallel with the belt, and to avoid rotation of the tray around the bolt. The slots are cut in the middle of the belt wheel to contain the head of the bolt.

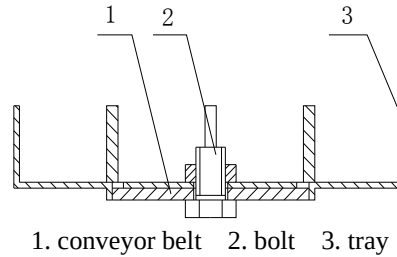


Figure 6. connection of tray and conveyor belt.

Structure Design of the Claw

The design goals of the claw are, the thermometers showing a preferred orientation after passing through the claw. Many factors must be considered to achieve the goals. First, fixed type of the claw is not feasible because the thermometer might be crushed for deadlocking. Second, It's necessary that there is a friction when the thermometer passing through the claw. With this friction, the thermometers turn over directionally. As is shown in Fig. 4, the claw is made by some pieces of rectangular metal plates with rubber sleeve installed at the lower end. The claws are obliquely plugged in to the slots on the claw bracket and its lower end contact the thermometer because of its deadweight. The thermometers turn over due to the effect of the friction from the rubber sleeve.

Design of Swing Device

As is shown in Fig. 4, the rotating shaft 9 is fixed on the frame 6, a pair of rotary tables rotate around the shaft. The driving force of rotation is from the thermometers transmitted. When the trays move horizontally, they guide the thermometers inset into the slots on the rotary tables. So the rotary tables rotate under the action of pushing force produced both when the thermometers enter into the slots on the rotary tables and when they get off those slots, as shown in Fig. 7. There are some key points when designing the swing device. The first is the shape and number of the slots in the edge of the rotary table. It should be easy for the thermometers to enter into or get off the slots on the rotary tables, which have the same slot pitch as those on the trays. The second is that, it's necessary to adjust the installation position of swing device carefully. The number of thermometers entering into the rotary tables should be the same as those of the slots on the trays obstructed by the rotary tables. For this reason, from which slot that the thermometer entered into the rotary tables, after swing action, the same thermometer will come back to that slot.

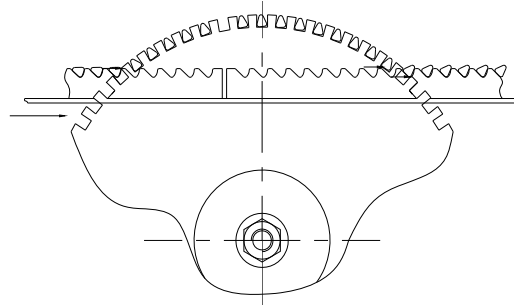


Figure 7. driving schematic diagram of swing device.

Practice and Improvement

According to the using condition, the devices were improved further. For example, the number of claws in the orientation device is added to about 5-6 because of the deadlock of the thermometer. When the thermometer failed to turn over, it will be deadlocked. The deadlocked thermometer will arise the claw. The deadlock will not disappear until the next claw could work normally. As a result, more than two claws are needed to increase reliability. The slots used to fit the claws are tilted 30 degrees backward to make the claws move up flexibly. It is necessary to keep the bearings on the axis of swing device clean and lubricated so that the rotary table can rotate flexibly. The shape of the two sides on the slots of the rotary table is different. The right sides are made to be round-cornered to make the thermometers enter the slots smoothly. And the left sides are made to be keen-edged, so that the thermometers will drive the rotary table to rotate.

It is indicated by actual use that the action of orientation and swing is satisfactory and can meet the need of the graphics processing. With this device taken into the use, the productivity of calibration and classification is increased from 1p/sec to 4p /sec.

References

- [1] Zhou Hong-lei, Li Guang-ling, Pan Yong-hui, Production line design for thermometer calibration and classification [J], Machine Buiding & Automation. 05 (2011) 22-25.
- [2] Zhou Hong-lei, Li Guang-ling, Pan Yong-hui, Automated calibration method for mercury thermometer benchmarks scale [J], Journal of Taiyuan University of Technology. 03 (2011) 305-308.
- [3] Li Guang-ling, Sha Yin, Structure design for rotary of thermometer classification machine [J], Manufacturing Automation. 16 (2013) 147-150.
- [4] Li Guang-ling, Zhou Hong-lei, Sun Yi et al, CN Patent 201110150156.4. (2012).
- [5] Li Guang-ling, Sha Yin, Zhou Hong-lei et al, CN Patent 201110150155.X. (2013).
- [6] Zhou Hong-lei, Li Guang-ling, Yin Guo-hong et al, CN Patent 200710135105.8 (2009).
- [7] Zhou Hong-lei, Li Guang-ling, Yin Guo-hong et al, CN Patent 200710135103.9 (2009).
- [8] Lli Guang-ling, Zhou Hong-lei, Yang Li-zhi, Study of thermometer classification method based on image recognition and PLC technology [J], Journal of North University of China (Natural Science Edition). 06 (2014) 763-768.
- [9] Sun Yi, Zhou Hong-lei, Application of PLC control technology to classification machine for mercury thermometer [J], Journal of Chongqing Technology and Business University (Natural Science Edition). 09 (2013) 60-64.
- [10] Chen Guo, Design and implement of PLC control stepping motor indexing [J], Industrial Control Computer. 05 (2008) 83-84.
- [11] Liu Wen-jun, Pan Yong-hui, Automatic temperature demarcation equipment of thermometer based on WDM [J], Industrial Control Computer. 11 (2010) 28-29.
- [12] Yang Li-zhi, Pan Yong-hui, Research on techniques for browsing visualized information based on Zooming-Panningmode [J], Computer Applications. 08 (2008) 2067-2070.
- [13] Gu Yao-lin, Yang Li-zhi, Method of smooth-transition images navigation [J], Computer Engineering. 03 (2007) 120-123.
- [14] Zhu Hong-tao, Wang Wei, Design of thermometer inspection system based on machine vision [J], China Instrumentation. 04 (2008) 64-66.

An Assessment of Random Splitting and Cross Validation Methods Based on Decision Trees Used to Identify Accident Event Aviso Category

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Keywords: Patient Safety Reporting System (PSRS), Decision trees C4.5 algorithm, Random splitting, Cross validation, Classification models.

Abstract. In the healthcare industry, health, safety, and environment (HSE) attracts much attention from patients and their family. Practically, how to explore the problem creating a safe medical environment for hospitalized patients is an interesting and important issue to academicians and practitioners. Applications of patient safety reporting system (PSRS) for classifying accident event category become useful works to identify potential hazards of medical errors. Unfortunately, few studies have examined addressing the accident category problems from the PSRS, and the related analysis against these problems is still lacking. This knowledge gap must be closed accordingly. By the way, to accurately classify the event reporting categories, analyze their determinants, and provide meaningful decision rules for interested parties, this study proposes a classification model based on expert recommendations, data splitting methods, and decision trees C4.5 algorithm to classify the accident event categories. Furthermore, random splitting and cross validation methods are assessed for assessing performance of the proposed classification model. A practical data set was collected from the PSRS of existing hospital databases with patient-focused used to analyze and identify hazards and risk practices that may harm patients. Analytical results show four key points: (1) The proposed classification model effectively determines 18 core condition-attributes and performs better 92.15% accuracy than other listed models, (2) the 70%/30% random splitting on given data is more suitable to be used in the healthcare industry than cross validation, (3) for the random splitting method, the splitting ratio of 70%/30% on training/testing sub-sets has a better classification result than other ratios, and (4) the analytical results offer useful information to both academicians and practitioners.

Introduction

To increase detection in recording medical error or accident reporting (or aviso) events for decreasing risk of harm to patients is important and one of main activities on error management for patient safety. Highly accident reporting rate implicitly indicates the low risk level of the reported events that can be avoided in the future. Thus, the constructing accident event aviso system is the key to ensure and create a prospect of health, safety, and environment (HSE) for patients. Particularly, Kaplan et al. [1] showed that the event severity level should be taken as an indicator of successful error management when the detection rate remains high. Through an integrated set of activities to record the events of medical or human error practices will build on the systematic progression made with a primary focus on minimizing the risk of preventable injuries. Achieving a goal of recording patient safety reporting system (PSRS) reliably identify potential hazards to patients.

Given the above reasons, identifying determinants to influence the medical errors to patients becomes an important issue, and the determinants are to be reliable and valid for interested parties. Thus, developing a classification model to represent the adverse medical errors is an interesting challenge for benefiting patient-centered efforts and is urgently needed for interested parties. Furthermore, prior studies had constructed methods for processing the related HSM problems for detecting adverse events by using the statistical techniques, such as six sigma [2], patient interviews

[3], and combined modalities [4]. However, the conventional methods are subjected to the characteristics, such as the restrictive assumptions of linear separability and multivariate normality [5]. Recently, soft computing techniques in data mining fields have emerged as a promising approach to the scientific study as well as alternatives to the conventional statistical methods for addressing real-world problems [6]. This study has thus the three objectives: (1) Develop a classification model to intelligently classify event reporting category in the healthcare industry; (2) use automatic feature selection approach to identify determinants when determining event reporting category; (3) assess the performance of the proposed classification model. Generally, this study proposes an application of classification model based on expert recommendations, data splitting methods, and decision trees C4.5 algorithm to classify the accident event categories.

Related Works

This section introduces patient safety reporting system, decision trees and its C4.5 algorithm, random splitting and cross validation, and other soft computing algorithms of classification.

Patient Safety Reporting System

Based on a medical report, it is a red flag that about 98,000 Americans will be dead in avoidable medial errors each year [7]. Specially, a more significant number of patients are being harmed than the death on the process of health care in hospitals. With the above reasons, making a principle of HSE for eliminating medial errors to patients is major concerns in hospitals. The medial errors contain several key events, such as incident, medical adverse event, no-harm event, preventable adverse event, high-alert drugs, adverse drug reaction, and adverse drug event, which are classified into three categories, human error, drug error, and systemic error. However, the errors with hazard events caused by a human lead to negative consequences to patients. To achieve an excellent medical-patient relationship and to promote medical quality satisfaction for hospitals, a way to providing the high information quality for patient safety is to record best practices of integrating set of activities to design a PSRS for reducing medical errors. The PSRS builds a safety culture of hospitals, records a tangible evidence of medical professional efforts, and integrates the risk management activity, involving both medical professional and patients. Developing such an accident reporting system from the PSRS can improve the quality of health care and address preventable errors by reducing the risk of patients being harmed in the process of health care of physicians.

Decision Trees and Its C4.5 Algorithm

In data mining fields, classification is an important function; numerous classification models, such as distance base, statistical base, neural networks base, and decision tree base [8], are studied from earlier literatures, particularly for decision trees techniques. Decision trees are a flow-chart-like tree structure and have particularly useful for working through complex multistage decisions for classification and prediction. Decision trees attract wide use for decision-making in supervised classification learning algorithm. ID3 (iterative dichotomiser 3) is one of well-known algorithm in decision trees learning, which is based on information theory invented by Quinlan [9] to produce a decision tree. The ID3 used the highest information gain to choose splitting attributes. C4.5 is designed by Quinlan [9] on a software extension of the basic ID3 algorithm to address the issues not dealt with by ID3. Decision trees have five distinct advantages. First, decision trees are a white box model; thus, the output of decision trees for analytics is easily explained and interpreted for interested parties. Second, decision trees can be implicitly regarded as a feature selection tool. Third, data preparation for decision trees needs little or unnecessary effort from users. Fourth, declaration of nonlinear relationships between variables is unnecessary. Finally, large amounts of data can be well experienced in decision trees work in a short time.

Random Splitting and Cross Validation

This study uses two types of dividing data, including random splitting and cross validation. (1) In random splitting method, all the instances in experimental data set are split into two different parts, training subset and testing subset, at random or by chance. (2) K-fold cross validation is a common technique for estimating the performance of a classifier. Given a set of n training instances, a single run of K-fold cross validation proceeds is presented in four steps as follows:

Step 1. Use the training instances in a random order.

Step 2. Divide the training instances into K folds equally and randomly.

Step 3. For $i = 1, \dots, K$, implement the following three sub-steps.

(1) Training all the instances that do not belong to Fold i by using the classifier.

(2) Testing all the instances in Fold i by using the classifier.

(3) Computing n_i that is the number of instances in Fold i but it was wrongly classified.

Step 4. Return the calculated numbers to count the classifier error, and repeat run Steps 1-3 several times accordingly to obtain an accuracy of a classifier in a K-fold cross validation.

Other Soft Computing Algorithms of Classification

To verify the proposed classification model, three comparable classification methods are used in this study. First, multilayer perceptron (MLP) is one of the most widely used learning machine based on mathematical computation of neural network [10] and is a feed-forward neural networks trained by the backpropagation algorithm. Second, Bayes net (BN) has well performance in accuracy in data mining fields and is a probabilistic graphical model applying Bayes' theorem to complex problems [11] to represent a set of random variables and their conditional dependencies. Finally, support vector machine (SVM) applies kernel-based methods to do regression estimations and binary classifications through classifying, clustering, and ranking the experimental examples [12].

The Proposed Classification Model

To reduce medical errors is curial to improve hospital-patient relationships. Instead, identifying accident event reporting category will have assistance to increase patient safety in the healthcare industry. Assistance is needed to support hospitals and patients by designing suitable classification models for mining hidden information and lowering medical errors from the PSRS. This study builds a classification model for classifying the accident event reporting category and extracting its knowledge-based decision tree rules based on soft computing techniques, such as the DTs [9]. Although the DTs-based rule classifiers have the flexibility to adequately perform in a variety of application areas, including finance, manufacturing, or health care, it is rarely used in the context of the accident event reporting category based on our research. This study focuses on filling the above knowledge gap based on the DTs-based rule classifiers. The above factors comprise the rationale for the proposed classification model. The proposed classification model as well as its computational processes by experiencing a real data set is implemented step by step as follows:

Step 1: Attributes Extraction and Data Collection. Initially, from the PSRS of a Taiwan's specific medical center database, select all data and attributes covered the period 2004–2007 under the target patient-centered information and is thus called the Patient-focused data set in this study. The observations in the Patient-focused data set are addressed with Taiwan's medical instances in practice from the PSRS to record all the accident event reporting occurred in the past.

Step 2: Expert Suggestion and Data Preprocessing. Based on an expert suggestion, 33 extracted condition-attributes are collected and characterized in an un-unified format, such as 'Event no.,' 'Notifier grade,' 'Notifier position,' 'Notifier capacity,' 'Seniority,' 'Date of birth,' 'Notifier age,' 'Education,' 'Level,' 'Patient age,' 'Patient gender,' 'Sickroom (or Ward),' 'Division,' 'Event location,' 'Related personnel,' 'Reason,' 'Cause,' 'Notification phase,' 'Notification type,' 'Nationality,' 'Medical care type,' and 'Category,' from Patient-focused data set; the attribute of accident event reporting category (renamed Class) is the decision-attribute. Accordingly, handle all the instances to process three types of missing attribute values, such as attribute-concept values, lost

values, and irrelevant values based on the past studies for their satisfied performance, and transform the format of the collected Patient-focused data set into an EXCEL file in CSV format to speed experimental operations. Specific attributes are normalized to improve the operational efficiency of classifying accident event reporting category. For decisional attributes, there are three classes: drug event, human event, and other. A total of 927 instances are remained in the target data set.

Step 3: Feature Selection and Key Attributes. To facilitate performance of experiment execution and DTs-based classifier, the feature-selection function of DT is first worked to remove redundant/irrelevant attributes that will be perhaps harmed to classifiers. The work of feature-selection gives better performance from the determined key attributes set for constructing a classification model for reducing computational time and increasing classification accuracy of results. After the implementation of the DTs-C4.5, 18 key condition-attributes are identified, such as Age, Gender, Phase, Type, Location, and Division... which are used in next step as input attributes.

Step 4: Various Data Splitting Methods. This step is divided into two sub-steps.

Step 4-1: Random Splitting and Classifier Execution. For evaluating performance of different data splitting methods, the 18 key condition-attributes and one decision-attribute with data of 927 instances in the Patient-focused data set are randomly split into five various ratios of training/testing sub-datasets with a 10% threshold; they are 50%/50%, 60%/40%, 70%/30%, 80%/20%, and 90%/10%. For example, the 50% training set has 464 records, and the 50% testing set is 463 records for the 50%/50% ratio. The training and testing sub-sets are used as to training and testing the given data, respectively. The DTs-C4.5 algorithm is repeat run under the above five various ratios.

Step 4-2: Cross Validation and Classifier Execution. Similar to the previous sub-step, the same setting on attributes and data in the Patient-focused data set are experienced by 5-fold and 10-fold cross validation methods with a five-fold threshold. By the way, the training and testing sub-sets are used in the following experiments to verify the proposed classification model.

Step 5: Performance Evaluation of Internal Model. This step uses DTs-C4.5 algorithm for classifying the event reporting category under the five various ratios of training/testing sub-datasets and 5-fold and 10-fold in Steps 4-1 and 4-2, respectively. Table 1 lists the applicable classification performance in various data splitting methods for the experimental medical data. Clearly, the accuracy of 70%/30% random splitting has better classification performance than other ratios and cross validation, and the best accuracy will be used in the following comparison of external model.

Table 1. The comparison results in various data splitting method by using DTs-C4.5 algorithm.

Method	50%/50%	60%/40%	70%/30%	80%/20%	90%/10%	5-fold	10-fold
Accuracy (%)	91.37	90.98	92.15	91.37	88.19	91.92	91.99

Step 6: Performance Evaluation of External Model. This step applies the three classification models, MLP, BN, and SVM for classifying the event reporting category with 70%/30% random splitting. Table 2 shows the applicable accuracy in the four classifiers for the medical data. Definitely, the DTs-C4.5 algorithm of the proposed model performs well compared to other listed classifiers.

Table 2. Comparison results of the four classifiers in the Patient-focused data set.

Method	The proposed model	MLP	BN	SVM
Accuracy (%)	92.15	90.28	91.36	91.67

Experimental Analysis

The proposed classification model creates empirical results from the above experimental analysis:

(1) A Suitable Data Splitting Method for Medical Data: Based on Table 1, the 70%/30% of random splitting is more suitable than other ratios, and the ratio of 70%/30% method has the best accuracy in the medical work in data mining fields. The study offers successful application examples.

(2) Key Condition-attributes are Identified: According to the DTs-C4.5 algorithm, the 18 determinants of influencing the category of accident event reporting are determined, and the remaining 14 redundant condition-attributes can be removed in classifying the accident event reporting category. Importantly, this hidden information implies that the DTs-C4.5 algorithms can be used as attributes-evaluating from a given data set for interested parties.

(3) A High Classification Accuracy: Clearly, the proposed classification model has higher accuracy than the listed classifiers in this study. Particularly, the 70%/30% random splitting is suitable ratio for the training/testing sub-data sets in the health care industry.

Conclusions

This study has proposed an effective classification model for addressing accident event reporting category issues in the healthcare industry. The proposed classification model experiences practical Patient-focused data sets. Two key points are concluded. (1) The analytical results show that the proposed classification model has a satisfactory accuracy of 92.15% and outperforms other listed methods. (2) The interpretability of a DTs-based classifier can construct a tree rule-based system to imitate the reporting process to interested parties. The interested parties have varied purposes on accident event reporting category during their evaluations. (1) For hospitals, low medical errors for improving specific facilitates of hospitals to benefit their reputations. (2) For patients, the proposed classification model thereby simplifies the evaluation of HSM. (3) For researchers, this study provides alternative potentials to further identify and analyze at the accident event errors in the subsequent researches. Future research can apply the proposed classification model for in-depth analysis of other data sets in various industries to handle different classification problems.

Acknowledgement

The author would wish to sincerely thank the National Science Council (Ministry of Science and Technology) of the Republic of China, Taiwan, for financially supporting this research under Contract No. MOST 103-2221-E-146-003-MY2.

References

- [1] H. Kaplan, J.B. Battles, T.W. Van der Schaaf, C.E. Shea, S.Q. Mercer, Identification and classification of the causes of events in transfusion medicine, *Transfusion*. 38 (11-12) (1998) 1071–1081.
- [2] M.R. Chassin, Is health care ready for six sigma quality? *The Milbank Quarterly*. 76 (4) (1998) 565–591.
- [3] A.J. Forster, H.J. Murff, J.F. Peterson, T.K. Gandhi, D.W. Bates, The incidence and severity of adverse events affecting patients after discharge from the hospital, *Ann Intern Med*. 138 (3) (2003) 161–167.
- [4] R.L. Evans, R.T. Connis, Risk screening for adverse outcomes in subacute care, *Psychological reports*. 78 (3) (1996) 1043–1048.
- [5] A. Ravi, H. Kurniawan, P.N.K. Thai, P. Ravi Kumar, Soft computing system for bank performance prediction, *Applied Soft Computing*. 8 (2008) 305–315.

- [6] E. Cesario, F. Folino, A. Locane, G. Manco, R. Ortale, Boosting text segmentation via progressive classification, *Knowledge and Information Systems*. 15 (3) (2008) 285–320.
- [7] Kohn L., Corrigan J., Donaldson M., eds. *To err is human: building a safer health system*, Washington, DC: Institute of Medicine, Committee on Quality of Health Care in America, 2000.
- [8] M.H. Dunham, *Data mining: introductory and advanced topics*, New Jersey: Prentice Hall, Upper Saddle River, 2003.
- [9] J.R. Quinlan, Induction of decision trees, *Machine Learning*. 1 (1) (1986) 81–106.
- [10] D.E. Rumelhart, G.E. Hinton, R.J. Williams, Learning internal representations by error propagation, in: Rumelhart, D.E. et al., (Eds.), *Parallel Distributed Processing*, MIT Press, Cambridge, MA, 1986, pp. 318–362.
- [11] K.P. Murphy, *Bayes Net ToolBox*, Technical report, MIT Artificial Intelligence Laboratory. <<http://www.ai.mit.edu/~murphyk/>> (2002).
- [12] V. Vapnik, *The nature of statistical learning theory*, Springer, NY, 1995.

The Design of Corn Straw Bundling Machine Based on SolidWorks

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Keywords: Straw, Bundling, Wrap net, Crushing.

Abstract. We designed the pull-type corn straw crushing machine, the collecting machine and the bundling compound operation machine based on the characteristics of corn straw. The solid modeling was designed using Solidworks software. We carried out the stress analysis of suspension pull-type device. Through the solid modeling and simulation analysis, we designed the machine and provided the basis for the optimization design.

Introduction

Crop straw is the most renewable resource on earth and is abundant in our country, but using ratio is low. Comprehensive utilization of straws needs reliable supporting machine and equipment especially a complete set of collecting, crushing and bundling pull-type operation machine. Bundling machine can be divided into two types by shape, namely the circle and the square ones ^[1]. The square bundling machine has a larger compression ratio and consumed more energy, which needs the complete set of horsepower tractor; The circle bundling machine has the characteristics of small volume, strong adaptability, easy operation and low cost ^[2]. In this paper, the circle bundling machine was designed to fit the present situation of agricultural machinery in China.

The Overall Structure Design of Corn Straw Bundling Machine

In this paper, the bundling machine was powered by the power output shaft of the tractor. The speed changing box and drive system would respectively pass power to straw collecting and crushing device, straw feeding device, straw wrap net device with control devices ^[3]. That could achieve molding and bundling of the straw. The overall structure design of corn straw bundling machine was shown in Fig.1. There were many devices in this system such as the traction device, the power input shaft, net bundling place box, steel roller gearing, wrap net device, molding chamber open device, crushing toss lifting device, screw feeding device, crushing toss device and so on. The width of the bundling room was 1m. The forming roller diameter was 0.18m. The interval of the forming roller was 0.008 m. The bundling working width was 1.58 m.

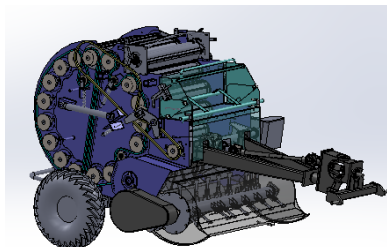


Figure 1. The overall structure diagram of the bundling machine.

The Design of the Key Components

The Design of the Collecting, Crushing Device. Collecting, crushing device is one of the key components of the bundling machine. The field straw is collected and broken in order to achieve the bundling requirements. The collecting device of corn straw bundling machine is the spring-tooth

type in general. It doesn't have the function of crushing ^[4]. The crushing toss device was shown in Fig. 2. This device was designed with reference to the part structure of the straw counters-field straw crushing. It was composed of straw crushing blade axis and flail cutter.



Figure 2. The crushing toss device.

The Design of the Spiral Feeding Device. The spiral feeding device was placed between the crushing device and the bundling chamber. The working width of crushing device was 1.58m. The entrance of the bundling chamber was 1m. In the spiral feeding device, the smashing straws shredded forcibly were pushed into the bundling chamber to avoid the straw blockage. The spiral feeding device was shown in Fig.3. The spiral blade would smash straw from both ends to the middle at working and push the straw into the bundling chamber by the middle of the spiral dial tooth.

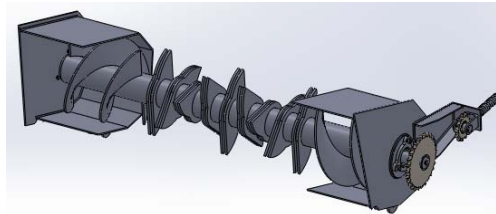


Figure 3. The spiral feeding device.

The Design of the Forming Roll. The forming roller played a key role in the formation of bale and determined the efficiency and quality of bundling. In this paper, we designed the cylinder type of the forming roller. It was shown in Fig. 4. For the convenience of unloading the bale, the forming roller was designed with two groups. The crushed straw would rotate along the same direction like a snowball and gradually formed the bale.

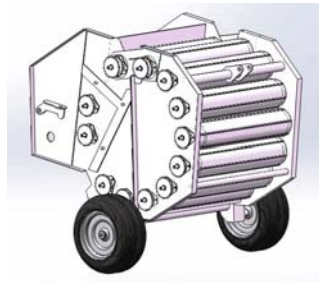


Figure 4. The forming roll.

The Design of the Straw Wrap Net Device. This design adopted the wrapped net fixed bale. It not only tied up good bale not easy to loose but also make the control of put net, wrap net, cut net very simple. It should accelerate the speed of wrapping net and improve the production efficiency. The structure diagram of wrap net device was shown in Fig.5.



Figure 5. The wrap net device.

The Design of the Suspension Traction Device. The suspension traction device was shown in Fig. 6. We adopted the tractor under suspension traction machines and the bundling machine that would realize traction of the machine. The power input shaft was placed above the suspend beam. We ensured the stably driving force by universal joint and the speed changing box of the bundling machine in the complex working environment.

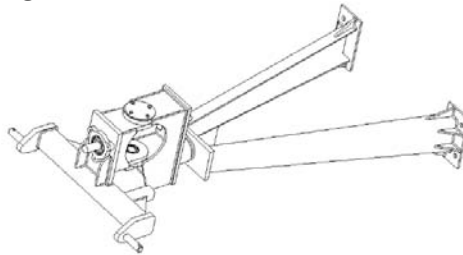


Figure 6. The suspension traction device.

The Simulation Analysis of the Key Components

The Simulation Analysis of the Forming Roller. We carried out the model analysis using ANSYS Workbench for the forming roll, obtained its inherent frequency and vibration model and analyzed the dynamic characteristics of the forming roller [5]. We added the parts material in the ANSYS Workbench (the material of the forming roller was Q235), constraints, mesh and calculation. Then, the order vibration mode was shown in Fig.7. The natural frequency was obtained by simulation. It was shown in Table 1.

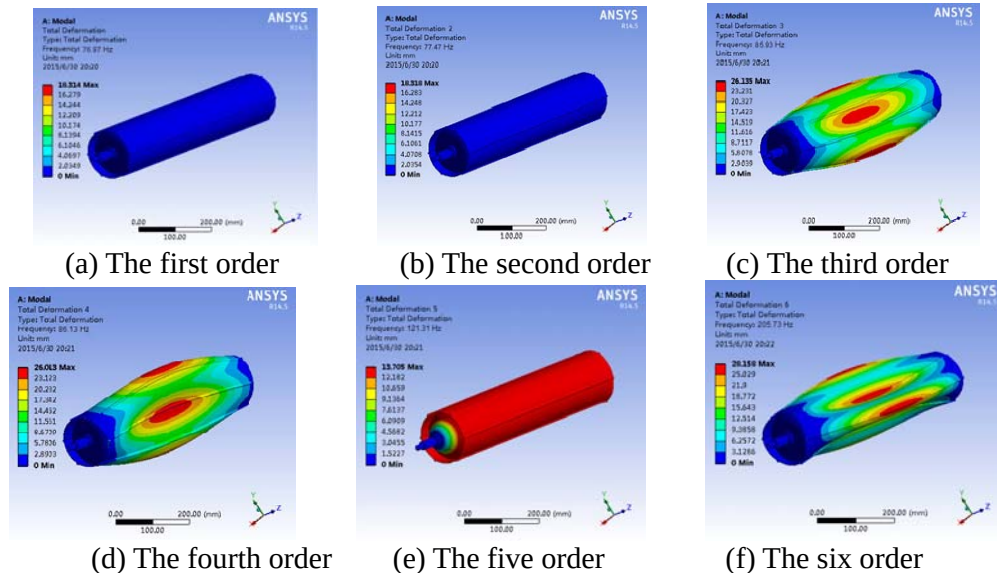


Figure 7. Each order vibration mode of the forming roller.

Table 1. The first six order natural frequency.

order	1	2	3	4	5	6
natural frequency (Hz)	76.97	77.47	85.93	86.13	121.31	205.73

It could be seen that the first six order natural frequency of the forming roller was gradually increasing with the increase of the order, the range is between 76.97-205.73 Hz. By the first order natural frequency $F_1=76.97$ Hz, we could get the lowest critical speed (4618.2r/min). At work, the speed of the forming roller was about 360r/min and far below the minimum critical speed. So, the design of the forming roller would meet the needs of the practical work.

The Simulation Analysis of Traction Beam. The traction beam model material was Q235. According to the working resistance of bundling machine, we added level load of 30000N to the traction beam. The model was divided mesh and calculated. Then, the stress analysis results were obtained as shown in Fig.8 and Fig.9. We would obtain the minimum value 2.08 of the safety coefficient. That would meet the operation requirement.

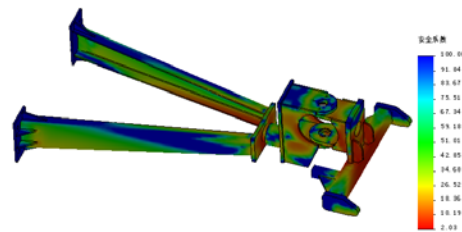


Figure 8. The distribution diagram of the safety coefficient.

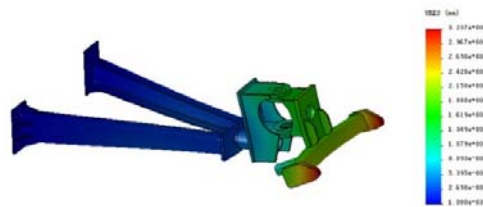


Figure 9. The distribution diagram of displacement.

Conclusions

The crushing toss device, instead of the traditional spring tooth pick-up device, could not only better adapt the ridge culture of corns but also pick up the standing straws, which played a key role in collecting straws. Both ends of the spiral feeding device were adopted spiral blade and the middle broken blade was also the spiral arrangement, which made the inhale of the straw to the molding chamber more efficient. We used the wrapped net device which tied up good bale not easy to loose, this design makes storage and transportation easily.

Acknowledgements

This paper was supported by Jilin province industrial technology innovation strategy alliance project (NO. 20130305043N Y).

References

- [1] Wang Chunguang, Ao Encha, Xing Jihui, Zhai Gaixia, He Gang. Design and Experiment on Round Baler with Revolute Steel Roller [J]. Transactions of the Chinese Society for Agricultural Machinery, 2010, S1: 103-106+102.

- [2] Yan Jingfeng, LI Yuanyuan, Song Wei. Design of Bundling Machine by Round Bundle for Corn Stalks [J]. Agricultural Science & Technology and Equipment, 2013, 09: 21-22+25.
- [3] Hua Rong-jiang, Tang Zun-feng, Ye Hong-yan, Wang Ying-yao, Jia Jing-xia. Research and development tendency about balerin domestic and overseas [J]. Chinese Agricultural Mechanization, 2012, (3): 23-26
- [4] Wang Chunguang, Ma Weimin, Ma Sai, Zhang Qiang, Liu Baojun. Design on Round Baler with Prepressing Device [J]. Agricultural Engineering, 2011, 02: 17-20+7.
- [5] Gao Dongming, Wang Decheng, Wang Guanghui, Sun Jie, Ma Xiaogang, Fu Zuoli, Self-wrapping Net Mechanism for Round Baler [J]. Transactions of the Chinese Society for Agricultural Machinery, 2012, 08: 99-104.

The Simulated Analysis on the Smash Device of the Straw Returning Machine Based on ADAMS

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Keywords: Cutter roller, ADAMS, Assembly scheme, Simulation.

Abstract. The smashing cutter roller was one of the most important parts in the straw returning machine. The blade arrangement on the cutter roller had a great influence on its performance. Therefore, we would simulate to analyze the arrangement form of the blade using the dynamics simulation software ADAMS for the single helix, double helix line respectively. The results showed that the double helix arrangement could effectively reduce the vibration of the grinding device. The scheme would provide the effective data for optimizing the straw returning machine of the corns and establish the theoretical foundation for designing the smash parts of the straw returning machine.

Introduction

The smashing cutter roller was the most important part in the straw returning machine. When crushing roller was working at high speed, it would generate large vibration and noise. That would affect the performance of the whole machine and reduce the service life of the machine [1-2]. In this paper, the arrangement of the blades on the crush cutter roller was the research object. The vibration performance of the cutter rollers with various arrangements of blades would be simulated using virtual prototype technology and the dynamics analytical software ADAMS of mechanical system. Ultimately, we would determine the best arrangement of the smash blade.

The Establishment of the Cutter Shaft Assembly Model

The Choice of the Blade Form. The blade of the straw returning machine could generally be divided into hammer claw, flail cutter and straight cutter [3]. Compared with other type of cutter, the collecting rate of the blade of flail cutter was high at work. The smashing effect was good. At the same time, it has a lot of advantages such as the small volume, light quality, less power consumption and so on. The design of the straw returning machine should match the straw tying machine. So, the smash blade of the flail cutter was chosen in this paper.

The Number of Blades. The number of flail cutter would determine the smashing effect. When the number was too little, the straw smashing effect would not be good, there would be the question of the picking up missing; when the number was too much, the energy consumption of machine would increase and there would appear even the blocking phenomenon. Setting reasonable blade number could refer to the following formula [4]:

The blade number (N) = The density of cutter (C) × Machine width (L)

The density of cutter was chosen between 0.02 piece/mm~0.04 piece/mm for the blade of the flail cutter. The width of the baler operation machine was 1.5 m. The blade number (N) was 30~60 pieces which was calculated by the above formula.

The Arrangement Plan. The reasonable flail cutter distribution could reduce the vibration of the machine, improve the effect of crushing and reduce the occurrence of machine fault. We would establish the three-dimensional model of cutter axis and flail cutter based on SolidWorks. According to the symmetrical arrangement and helix arrangement for assembly, the assembly model was established. We would simulate to make analysis on three kinds of blade arrangement[5-6].

The cutter seat was adopted the symmetrical arrangement that was called the symmetric arrangement scheme. The symmetrical arrangement scheme was shown in Fig. 1 for the cutter roller of the smashing device. The single helix arrangement was adopted on the cutter seat that was called the project of the single helix arrangement. That was shown in Fig. 2. The double helix arrangement was adopted on the cutter seat that was called the project of the double helix arrangement. That was shown in Fig. 3.

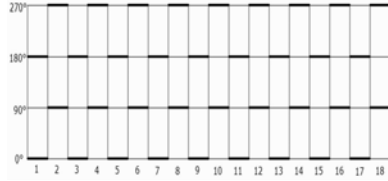


Figure 1. The symmetric arrangement.

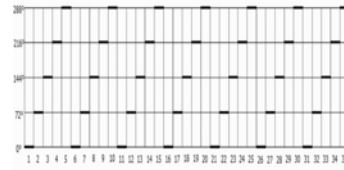


Figure 2. The single helix arrangement.

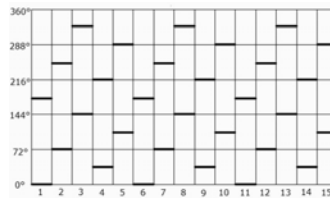
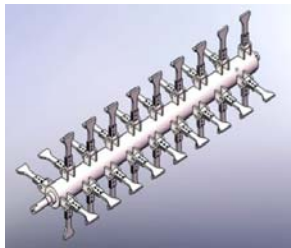
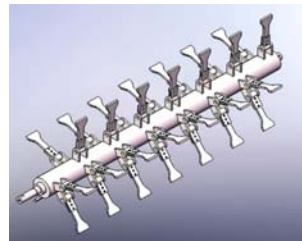


Figure 3. The double helix arrangement.

The assembly drawing of the symmetrical arrangement plan was shown in Fig. 4 (a). In this project, the blade was the staggered vertical distribution [7]. The width was 1.5m. We would design the flail cutter of nine groups. So, the total number of the blades was 36. The assembly drawing of the single helix arrangement plan was shown in Fig. 4 (b). The intersection angle of two cutters was 180° on the same cross section. The intersection angle of the adjacent two flail cutter was 90° . The scheme right adjacent blade angle was 72° that would meet the requirements of more than 60° angle between the adjacent blades. In order to reach the goal of not missing straw, we would set up 7 groups the single helix lines on the cutter roller. Every helix line was composed of five flail cutters. So, the total number of the blades was 35. The assembly drawing of the double helix arrangement plan was shown in Fig. 4(c). The scheme between two blades on the same circumference angle was 180° . The adjacent blade angle was 72° that would meet the requirements of more than 60° angle between the adjacent blades. We would set up 7 groups the double helix lines on the cutter roller. Every helix line was composed of ten flail cutters. So, the total number of the blades was 30.



(a)the symmetrical arrangement



(b)the single helix arrangement



(c)The double helix arrangement

Figure 4. The assembly drawing of arrangement plan.

ADAMS Simulated Analysis

Simulation time was 0.5. The step length was 200. The gravity was set to y axis direction. The constraining force cutter shaft, the bounce of the shaft and x, y direction of the acceleration curve could be obtained for the single helix arrangement scheme, the symmetry and double helix arrangement scheme. The simulation results were shown in Fig. 5(a),(b),(c), Fig. 6(a),(b),(c) and Fig. 7(a),(b),(c). Table 1 showed three kinds of permutation schemes simulation results

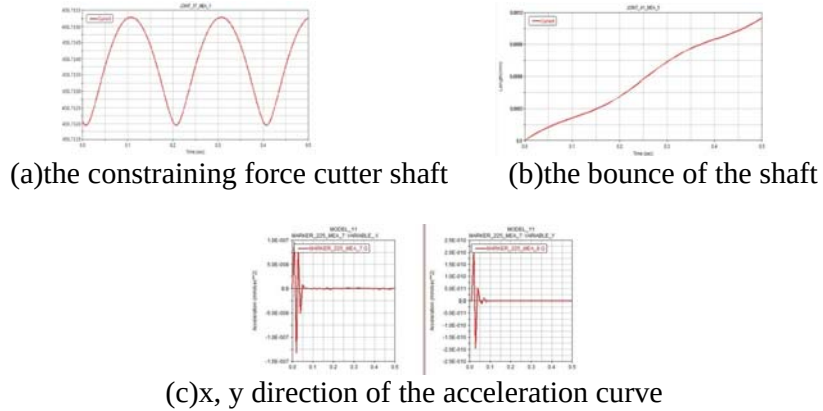


Figure 5. The simulation results of the single arrangement.

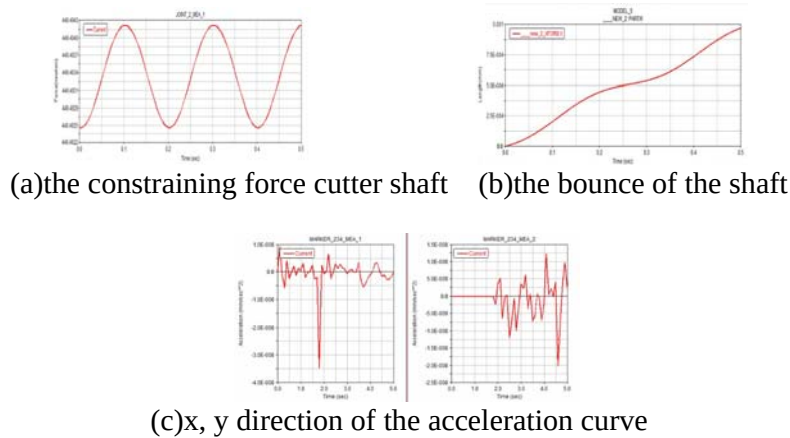


Figure 6. The simulation results of the symmetric arrangement.

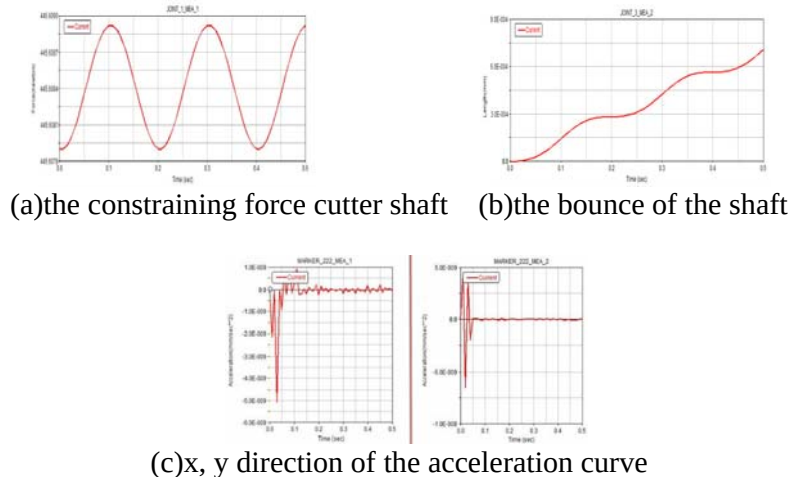


Figure 7. The simulation results of the double helix arrangement.

Table 1. Three kinds of permutation schemes simulation results.

index	value	the single	the symmetric	the double helix
the constraining force cutter shaft (N)	Min	450.7119	449.4924	445.6089
	Max	450.7156	449.4942	445.6079
the bounce of the shaft (mm)	Min	0	0	0
	Max	0.0011	9.48e-4	7.0e-4
x direction of the acceleration (m/s^2)	Min	-1.387e-7	-3.495e-8	-5.098e-9
	Max	9.132e-8	9.214e-9	9.894e-10
x, y direction of the acceleration (m/s^2)	Min	-1.942e-10	-2.014e-8	-6.873e-9
	Max	2.142e-10	1.248e-8	7.568e-9

We could see that the change rule between the force of gravity and centrifugal force of the cutter roller presented a sine curve. The value of the constraining force was close between the single helix and the double helix arrangement. On the whole, the single helix arrangement, symmetry arrangement and the double helix arrangement showed decreasing trend. In the shaft end bounce, the single helix was the most obvious. The maximum value was 0.0011mm. The double helix arrangement performed more smoothly on the job. It could be seen from the acceleration curve of x direction that the flexible impact force from small to large in the cutter shaft end was: the double helix arrangement, the symmetrical arrangement, the single helix arrangement. It could be seen from the acceleration curve of y direction that the flexible impact force from small to large in the cutter shaft end was: the single helix arrangement, the double helix arrangement, the symmetrical arrangement.

In conclusion, using the double helix arrangement scheme could effectively reduce the vibration problem of the smashing toss device. In addition, the blade number of double helix arrangement scheme was less. The effect of smashing toss would be better. Therefore, we would determine the smashing toss device on the cutter roller blade that adopted the double helix arrangement.

Conclusions

In this paper, the cutter shaft arrangement of the straw smashing toss device was researched similar to the design of the straw returning machine. The curve of the shaft end bounce and the constraining force of the cutter shaft were measured for three schemes. Through the simulation analysis, the double helix arrangement was determined the best scheme. In addition, ADAMS was applied in the agricultural machinery design using virtual prototype technology in this paper. It could greatly improve the efficiency of agricultural machinery and provide a new way for other agricultural machinery design.

Acknowledgements

This paper was supported by Jilin province industrial technology innovation strategy alliance project (NO. 20130305043N Y).

References

- [1] Fu Xue-gao, Li Ming, Lu Jing-ming, Wang Jin-li, Deng Yi-guo, Zhang Jin. Research on the cutter of straw crushing, machine to field [J]. Chinese Agricultural Mechanization, 2011, (1): 83-87.
- [2] Zhang Xirui, Gan Shengbao, Zheng Kan, et al. Design and experiment on cut roll feeding type horizontal shaft flail machine for banana pseudostem crushing and returning [J]. Transactions of the Chinese Society of Agricultural Engineering (Transactions of the CSAE), 2015, 31(4): 33-41. (in Chinese with English abstract).

- [3] Li Ming, Wang Jinli, Deng Yiguo, et al. Structural design and experiments on sugarcane leaf shattering and returning machine [J]. Transactions of the Chinese Society of Agricultural Engineering (Transactions of the CSAE), 2008, 24 (2): 121-126.
- [4] Liu Xu, Zhang Xiaohui, Wang Chuanjun. Research and improvement of 1JFH-34 type of field wheat straw chopper [J]. Journal of Agricultural Mechanization Research, 2010 (10): 98-100.
- [5] Liu Xiaoliang. Design and Experiment of the new Smashed Straw Machine Blade [D]. Changchun: Jilin University, 2012.
- [6] Li Yan. Design on Multi-function Smashed Straw Machine for Corn [D]. Jinan: Shandong Agricultural University, 2007.
- [7] Zhang Fengwei, Dai Fei, Han Zhengsheng, et al. Improve design on blades of straw returning machine [J]. Mechanical Research & Application, 2009 (5): 83-85.

Experimental Study on Crack Growth Rate of B780CF Steel Welded Joints

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Keywords: Crack growth rate, Welding process, Welding heat input.

Abstract. The fatigue crack growth rate under different welding process and welding heat input was tested with standard compact tension specimen, the crack growth rate curve under different welding process and welding heat input was obtained. The effect of different welding process and welding heat input on the crack growth rate was analyzed, and the fatigue fracture under welding process and welding heat input degree was analyzed, the mechanism of the crack growth rate under different welding process and welding heat input was discussed.

Introduction

Since the invention of the electrode, welding has become the most widely used technology. However, the fatigue fracture of welding structure often occurs, it will result in disastrous consequences. B780CF steel is used for domestic 800MPa production station pressure pipe, fork tube and spiral case in hydropower high-strength steel. In order to ensure the safety of welding production, a series of welding experiments of B780CF are needed to determine the reasonable welding way. But at present the researches on the steel B780CF welding joint crack growth rate are rarely reported. In this paper, fatigue crack growth behavior of the steel B780CF welded joint under different welding process and welding heat input were studied and compared to ensure the best welding modes.

Specimen and Test Method. Sample materials for B780CF steel, using W=50mm, B=12.5mm, a₀=9.5mm standard compact tension C (T) specimen. The basic mechanical properties of the samples are shown in Table 1.

Table 1. Tensile properties of B780CF steel.

Yield strength σ_s /MPa	Tensile strength σ_b /MPa	Elastic Modulus E/GPa	Poisson's ratio
770	820	210	0.3

Test using MTS810 material test system, the maximum load $F_{\max} = 15\text{kN}$, minimum load $F_{\min} = 1.5\text{kN}$, prefabricated fatigue crack length for 15mm, stress ratio $R = 0.1$, crack growth tests, the maximum load is still $F_{\max} = 15\text{kN}$, in different welding heat input and process under test, the frequency is 10Hz, load waveform is a sine wave.

Data Processing

Seven Point Incremental Polynomial Method. The da/dN according to (a_i, N_i) can be gotten by seven point incremental polynomial method.

$$a_i = b_0 + b_1 \left(\frac{N_i - C_1}{C_2} \right) + b_2 \left(\frac{N_i - C_1}{C_2} \right)^2 \quad (1)$$

In the formula, b_0, b_1, b_2 is a regression parameter.

The crack growth rate of crack length a_i :

$$C_1 = \frac{1}{2}(N_{i-n} + N_{i+n}) , C_2 = \frac{1}{2}(N_{i+n} - N_{i-n}) \quad (2)$$

$$\left(\frac{da}{dN} \right)_{a_i} = \frac{b_1}{C_2} + \frac{2b_2(N_i - C_1)}{C_2^2} \quad (3)$$

Fig.1~4 are a-N curve under various welding modes, welding process and heat input .

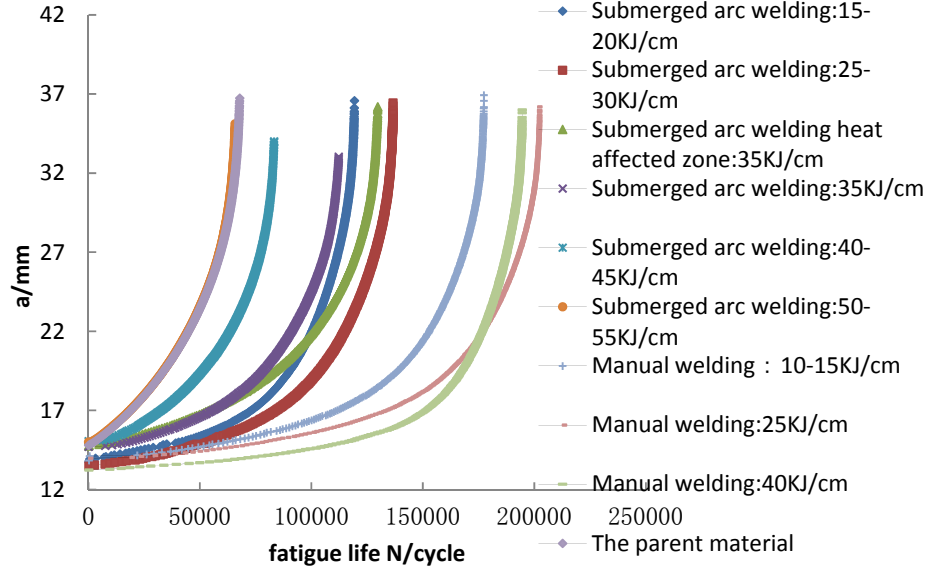


Figure 1. a-N curves under various welding modes.

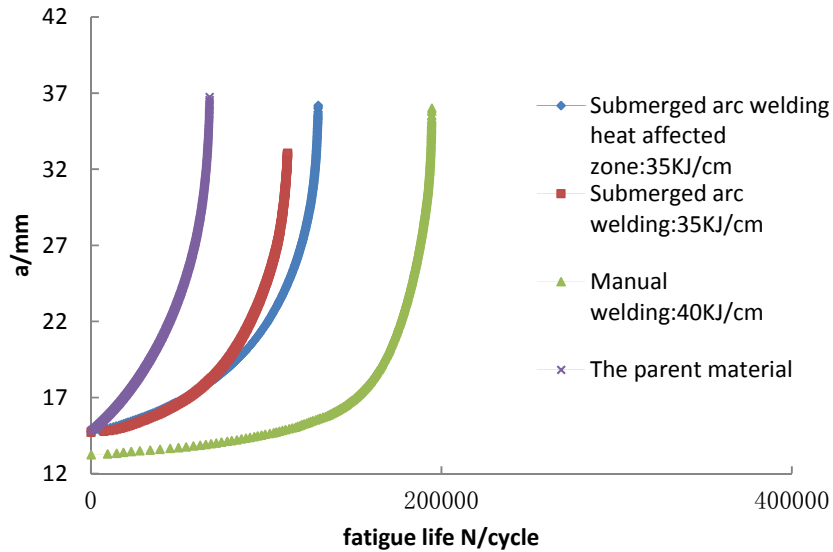


Figure 2. a-N curves under various welding processes.

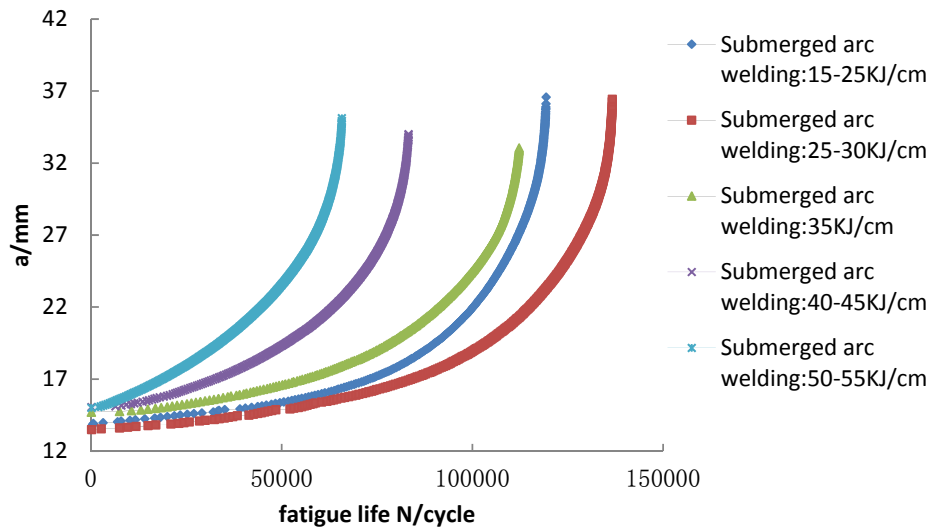


Figure 3. a-N curves of different heat input under submerged arc welding.

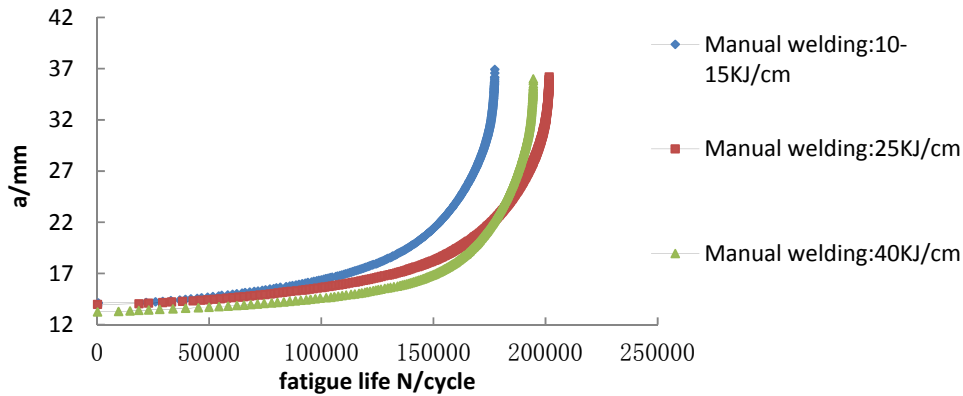


Figure 4. a-N curves of different heat input under manual welding.

Processing Results

The Influence of Welding Modes on Fatigue Life N. It can be seen from Fig. 1 that the difference of the welding modes has a great influence on the fatigue life of B780 (CF) steel. Under the condition of the same crack length a , the influence on fatigue life N , from Fig. 2 available, parent material life < submerged arc welding life < heat affected zone life < manual welding life. From Fig. 3, submerged arc welding 40-45 KJ/cm life < submerged arc welding 35 KJ/cm life < submerged arc welding 15-20 KJ/cm life < submerged arc welding 25-30 KJ/cm life < submerged arc welding 50-55 KJ/cm life. Can be obtained from Fig. 4, manual welding 40 kJ/cm life < manual welding 25 kJ/cm life < manual welding 10-15 kJ/cm life.

Crack Growth Rate. The crack growth rate da/dN and stress intensity factor range can be calculated by (2) and (3) type, Fig. 5 is the fatigue crack growth rate curve under different welding modes.

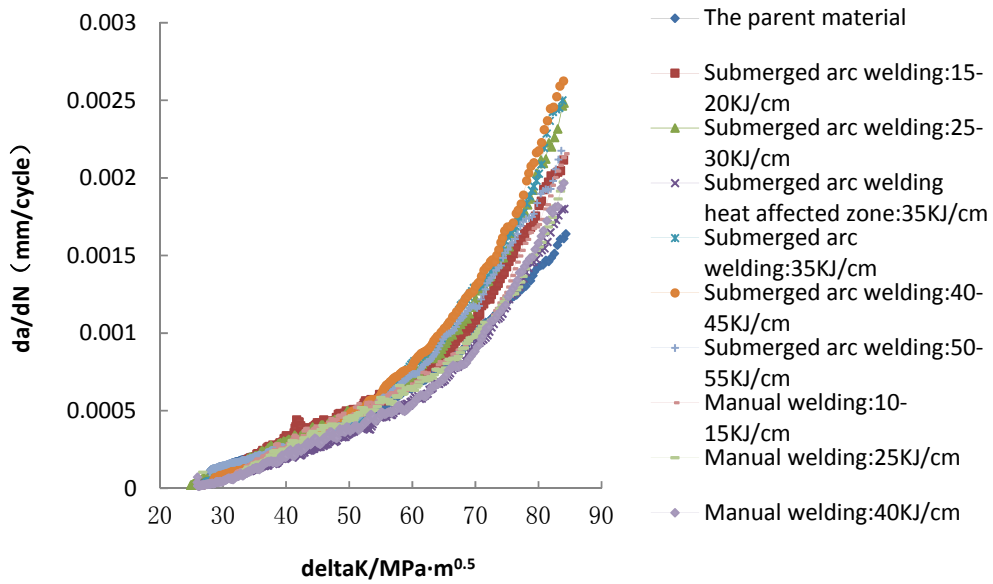


Figure 5. Fatigue crack growth rate curve under different welding modes.

Observe Fig. 5:

(1) da/dN with ΔK show an increasing trend, and increase more and more quickly.

(2) Welding mode has a influence on B780CF steel fatigue crack growth rate, which is mainly reflected in the middle and later stage.

Observe Fig. 6:

The parent material, submerged arc welding heat affected zone, submerged arc welding 40-45KJ/cm, submerged arc welding 50-55KJ/cm: fatigue crack test $\lg(da/dN) - \lg(\Delta K)$ curve basically doesn't show the fatigue crack growth in the first stage. And fatigue crack growth rate about double logarithm curve of the threshold values don't overlap under different welding modes, indicating that the heat input degree and welding process have a large influence on crack growth rate of the first stage.

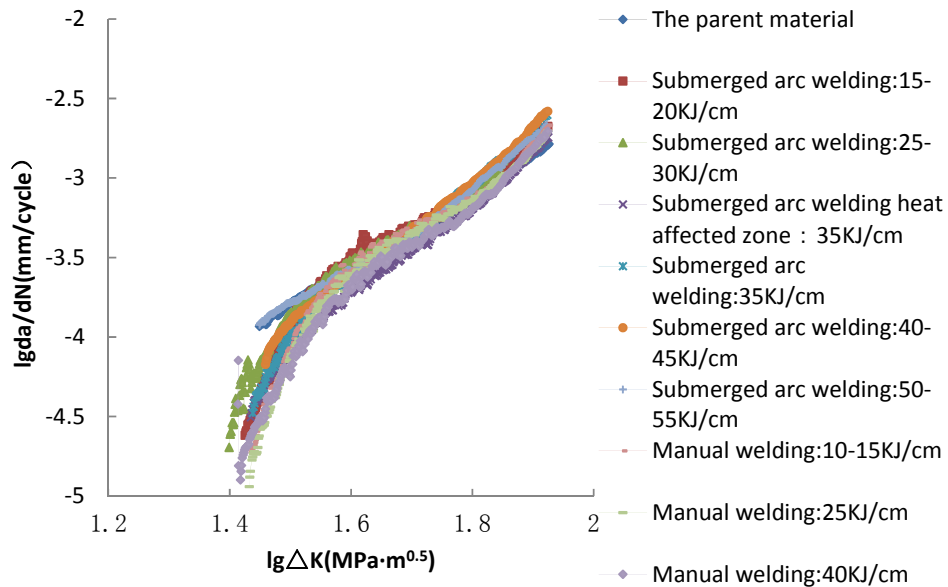


Figure 6. Fatigue crack growth rate curve under different welding modes.

Mechanism Analysis

The fracture images of the crack growth regions in different welding processes are obtained by scanning electron microscopy:

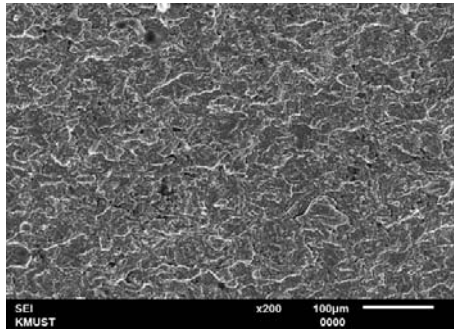


Figure 7. Fracture morphology under parent material (200×).

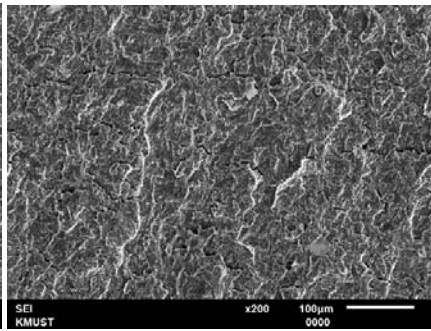


Figure 8. Fracture morphology under Submerged arc welding heat affected zone (200×).

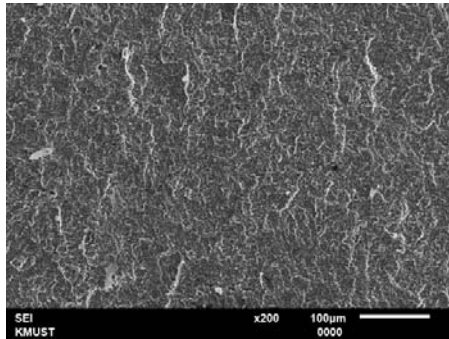


Figure 9. Fracture morphology under Submerged arc Welding 35KJ/cm (200×).

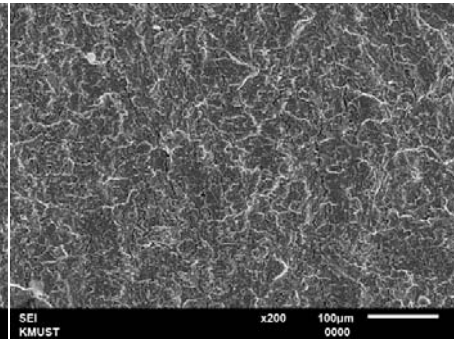


Figure 10. Fracture morphology under Manual welding 40KJ/cm (200×).

It is visible in Fig. 7 to 10 that fatigue fracture is flat under different welding processes and submerged arc welding heat affected zone, the reason is that under cyclic loading maximum load, crack mouth open, while under the smallest load, the load is not enough to provide the energy needed for crack mouth to open, therefore, friction is generated on the crack surface. According to the theory of Griffith, as the welding mode is different, the difference is reduced between strain energy in per unit area for crack extension and the energy for the formation of the free surface, so the section is more and more flat.

Conclusions

- [1] Different welding method has a great influence on the fatigue life of B780 (CF) steel. Under the same crack length, fatigue life: Parent material < Submerged arc welding < Submerged arc welding heat affected zone < Manual welding;
- [2] In submerged arc welding with welding way, under the same crack length, fatigue life: submerged arc welding 40-45 KJ/cm < submerged arc welding 35KJ/cm < submerged arc welding 15-20KJ/cm < submerged arc welding 25-30 KJ/cm < submerged arc welding 50-55KJ/cm;
- [3] In a manual welding with welding way, under the same crack length, fatigue life: manual welding 40kJ/cm < manual welding 25kJ/cm < manual welding 10-15kJ/cm.

References

- [1] Guoshou Liu. The Experiment of Studying on Crack Growth Rate of Welding-joint for ADB610 steel.(Kunming University of Science and Technology 2011) (In Chinese).
- [2] Tianhui Zhang. The Reliability Study on Crack Growth of New Low-carbon Bainitic Steel Weld Joints. (Kunming University of Science and Technology, Kunming 2012) (In Chinese).
- [3] Zhentong Gao, Junjiang Xiong. Fatigue Reliability. Beijing University of Aeronautics and Astronautics Press 2000.

Study on Fault Diagnosis Expert System of Reciprocating Compressors

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Keywords: Reciprocating Compressor, Fault Diagnosis, Expert System.

Abstract. Reciprocating compressor is the key mechanical equipment widely used in petroleum chemical industry. In order to prevent and reduce equipment failure, improve equipment operation safety and stability, it is very necessary to research on fault diagnosis technology of reciprocating compressor. This paper mainly researches the application of expert system in fault diagnosis of reciprocating compressor. The fault diagnosis expert system of reciprocating compressor is designed and systematization is realized. The expert system function will be gradually strengthened with the use and learning.

Introduction

As major key equipment which was widely used in the chemical industry, the reciprocating compressor has complex structure, poor operating conditions, so it is prone to all kinds of trouble; and each section are closely connected, and fault of a part often causes paralysis of the entire unit, resulting in huge economic losses and even casualties. Therefore, the reliability, availability, economy and safety of the reciprocating compressor are put forward. In the running process of the equipment, through condition monitoring and signal analysis, equipment fault forecasting and warning, accidents will be reduced and prevented effectively, reliability and safety of the equipment will be largely improved, which directly improve the economic efficiency of enterprises.

According to the above problem, a reciprocating compressor fault diagnosis expert system is proposed, which will be used to guide the safe and stable operation of the unit. The establishment of expert system will enrich the theory and technology of fault diagnosis for reciprocating compressors.

Fault Diagnosis Expert System

Basic Principles of Expert System. The basic principle of expert system is that using a language that both users and computers will be able to understand to express the experts' experience and knowledge, and to store in the computer, so even for non-professional staff, it is possible to apply expert knowledge and experience and learning within the system, and the system can easily be maintained. Expert system is to simulate the experts, experts through on-site known conditions (expert system by entering information), the use of professional knowledge and experience within the mind (expert system through the knowledge base), after a series of reasoning thought process (expert system by norms established mode of reasoning), to give the final conclusion (expert system output a conclusion), and being able to answer other non-professional staff questions about the reasoning process (expert system for the user to answer doubts by explaining the mechanism).

Fault Diagnosis Expert System. Expert system can be divided into various types, including diagnostic expert system, interpretation expert system and predictive expert system. Reciprocating compressor fault diagnosis expert system constructed in this paper belongs to the Diagnostic Expert System. Expert system has many advantages in fault diagnosis of auxiliary equipment.

(1) *Knowledge capacity.* Utilizing the huge storage capacity of the computer, the expert system can have all the knowledge of the fault diagnosis of reciprocating compressor.

(2) *Able to simulate expert reasoning.* This advantage of the expert system benefits from the advantages of computer technology and artificial intelligence, as well as in-depth human sociology, ethics and psychology. A combination of these research results makes it possible to simulate human thinking.

(3) *Able to output conclusions and interpretation.* Using the input information, and through the above reasoning process, the expert system can be used for the user to output expert-level conclusions and recommendations, and each step has reasonable basis. When it is needed, an explanation can be provided.

(4) *Good interaction.* An appropriate knowledge presentation makes it possible that the user can interact with the system similar to human communication, both input and output information is readable and easy to understand.

System Objectives and Function Design

System Objectives. A fault diagnosis expert system for reciprocating compressor is designed and developed in this paper, which is applied to fault diagnosis of reciprocating compressor. This system is designed to achieve the following objectives.

(1) To gather the knowledge and experience of expert all together, and to store in the knowledge base through the standard and unified way, which can guarantees the inheritance of knowledge and avoid the loss of knowledge.

(2) To make the average users have expert knowledge, so the time to determine the cause and the location of the fault will be greatly shortend.

(3) To provide users with expert maintenance recommendations, thereby reducing maintenance investment and avoiding outage or shortening outage downtime and improving machine utilization.

(4) To improve the working mechanism of the equipment fault diagnosis, and to realize the operation reliability, the use efficiency and the comprehensive optimization of the service life.

Function Design. On the basis of the previous analysis, combining with the actual needs, the system function is shown in Fig. 1.

(1) *Failure warning.* Real-time monitoring of the main operating parameters of the reciprocating compressor, such as the circulating hydrogen out and inlet pressure, the new hydrogen at all levels of the out and inlet pressure, the circulating hydrogen and hydrogen gas pressure, etc. When the parameters are abnormal, a warning will be sent to the user by some way.

(2) *Fault Diagnosis.* Upon receiving the warning message that will be transmitted to fault diagnosis module, the system will start to analyze the cause of the fault, locate the fault location, and analyze the influence of the fault, and formulate the strategy of fault response. According to the actual situation the experts can modify the results and maintenance strategies provided by the system, so as to complete the maintenance and improvement of the knowledge base.

(3) *Historical records.* This module is used to record the previous fault diagnosis and maintenance information, including fault time, location, equipment, repair scheme, the fault picture, etc., and manually entry the maintenance process evaluation, experience and other information, so as to sum up the lessons, accumulate experience, and constantly improve the maintenance work.

(4) *Equipment management.* The module is for the management of equipment information, including equipment nameplate parameters, drawings etc.

(5) *Parameter setting.* The module is used to manage the parameters which represent the operating state of the device.

(6) *Fault tree.* The module is used to construct, maintain and manage the fault tree.

(7) *Authority management.* To ensure the system security, only registered users can log on the system. The module features include user registration and user rights settings.

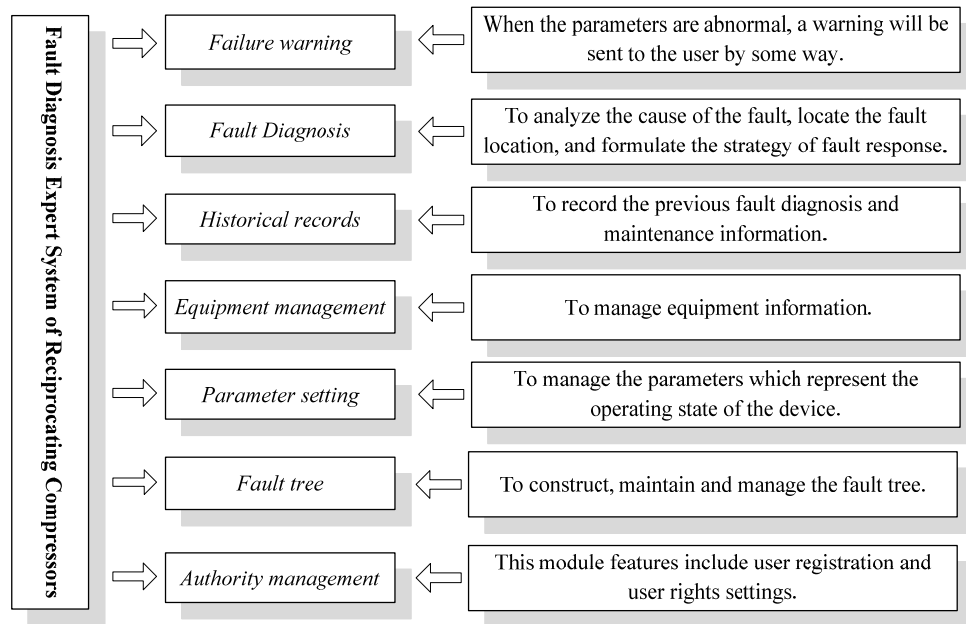


Figure 1. Function design of the system.

Typical Application

On the basis of the above functional design, the system has been developed and tested, and the typical application is as follows.

Failure Warning. According to the functional design, the failure warning module is realized as shown in Fig. 2, and the module functions are as follows.

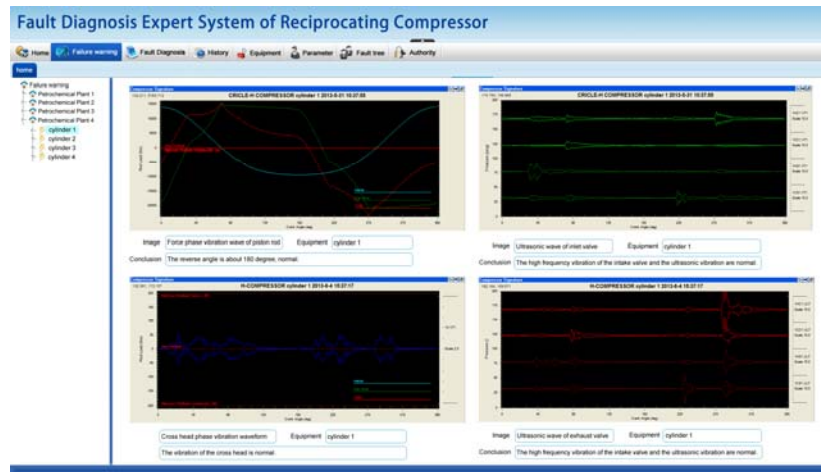


Figure 2. Failure warning.

(1) The home page shows the curve of the main parameters of the cylinder, such as the piston rod stress phase vibration waveform, crosshead phase vibration waveform, intake valve ultrasonic waveforms, and exhaust valve ultrasonic waveform. When a parameter is abnormal, the corresponding parameters become red, and the system puts forward a warning to the user via voice.

(2) The early warning module is associated with the fault diagnosis module, and the parameters can be transmitted to the corresponding diagnostic module when there is a warning.

(3) System administrators can adjust not only the parameters which are displayed on the home page, but also the number.

Fault Diagnosis. According to the functional design, the Fault Diagnosis module is realized as shown in Fig. 3, and the module functions are as follows.



Figure 3. Fault diagnosis.

(1) When receiving the warning message, the system will start to analyze the cause of the fault, locate the fault location, analyze the influence, and formulate the strategy of fault response.

(2) According to the actual situation the experts can modify the results and maintenance strategies provided by the system, and the results will be pushed to the historical record module

(3) The system can generate reports, output and print the final fault diagnosis information to assist in the maintenance of the site.

Summary

Fault diagnosis expert system is the collection and incarnation of human expert, and has a lot of knowledge and experience of domain experts. Because of the huge storage capacity of the computer, and because the program can be permanently saved and copied, the system has a stronger ability to work and more experience than human experts. The fault diagnosis expert system of reciprocating compressor is designed and systematization is realized. The expert system function will be gradually strengthened with the use and learning.

References

- [1] Cox I.J. Expert systems [J]. Electronics and Power, 1984, 30 (3): 237.
- [2] Gary L.W., Bhaskaran G., Magdy A., et al. Expert Systems [J]. Professional Safety, 2005, 50 (8): 45.
- [3] Kocyigit N, Bulgurcu H, Lin C-X. Fault diagnosis of a vapor compression refrigeration system with hermetic reciprocating compressor based on p-h diagram [J]. International Journal of Refrigeration, 2014, 45: 44-54.
- [4] Liang Z., Li S., Tian J., et al. Vibration cause analysis and elimination of reciprocating compressor inlet pipelines [J]. Engineering Failure Analysis, 2015, 48: 272-282.
- [5] Becerra J.A., Jimenez F.J., Torres M, et al. Failure analysis of reciprocating compressor crankshafts [J]. Engineering Failure Analysis, 2011, 18 (2): 735-746.
- [6] Kim H.C., Cho M.G., Shim J. Multi-stage diagnostic system for reciprocating compressor using DTW technique [J]. Journal of Mechanical Science and Technology, 2013, 27 (8): 2293-2298.

Research on Design and Test of Corn Straw Forage Device

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Keywords: Corn straw, Forage device, Test bench.

Abstract. To measure the work performance of test bench of corn straw forage device and optimize the structural parameters, single-factor tests were carried out respectively with the circular blade front angle, snapping roll and tearing roll speed variation and harvest period as the factor, and forage qualified rate, wrapped grass amount and torque size as the evaluation indicators. The parameters meeting the requirements of straw forage while minimizing the power consumption were: to adopt the circular blade with a front angle of 30°, control the rotating speed of tearing roll within the reasonable interval of 1200-1400r/min and harvest and process the corn straws as early as possible under the premise of ensuring the maturity of the corn grains. The data can provide basis for the further optimization of forage device.

Introduction

At present, as the yield of corn increases year by year, the yield of corn straw has increased accordingly, with a huge utilization space in the corn straw resources [1,2]. Having a higher nutrient value than other crop straws and with total energy equivalent to herbage, the corn straw can become quite an important source of feed for livestock [3]. However, due to huge volume, short collecting time and inconvenient transportation, corn straw has a low utilization rate and is limited in the use of livestock culture and is wasted a lot by burning or abandoning [4-6]. Using the straw corn as forage can solve this problem effectively [7,8]. A corn straw forage device was designed. It can be installed on the vertical roller harvesters to combine operation of corn harvest and straw forage. For obtaining the actual device work condition and evaluating forage device performance index, the corn straw forage device test bench is set up. The forage performance test was carried for comprehensively measuring the equipment in terms of qualified rate, grass wrapping condition and maximum torque under different parameter conditions, the parameters meeting the straw forage requirements while reducing the power consumption were obtained, thus providing basis for the further optimization of the device.

Test Equipment

Corn Straw Forage Device. A corn straw forage device was designed (as shown in Fig. 1), the device consists of a pair of snapping rolls, a pair of tearing rolls, support frame, and transmission mechanism etc. Distance of two rolls can be adjustable, tearing rolls consists of a pair of square shaft, saw blade and location spacer (as shown in Fig. 2). To test the influence of the front angle of blade on the tearing rolls (as shown in Fig. 3) on the device foraging performance, the tearing blades on the tearing rolls can be changed, and blades of three angles 30°, 20°, -15° are designed (as shown in Fig. 4). In addition, the tearing roll speed of this test bench can be regulated through the control system.

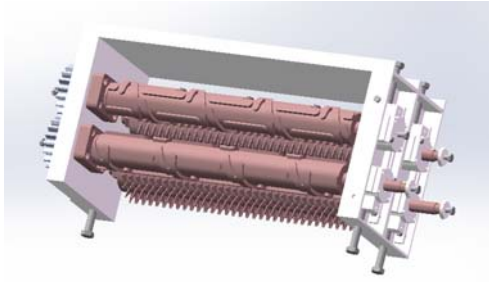


Figure 1. 3D model diagram of a corn straw forage device.



Figure 2. 3D model diagram of tearing rolls.

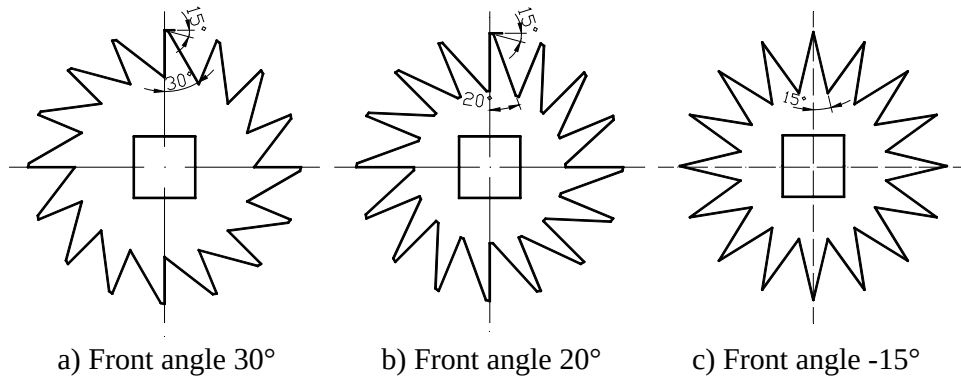


Figure 3. Three kinds of saw blade of different blade angle.

Test Bench. The test bench of corn straw forage device (as shown in Fig. 4) consists of chassis, a pair of snapping rolls, a pair of tearing rolls, feeding station, discharge hole, control system and torque test system etc. During work, the control system is adjusted, so that the two motors respectively output power to rotate the pair of snapping rolls and tearing rolls in the opposite direction. After the four rolls reach the test speed with stable rotation, the corn straws are fed from the feeding station, and before they reach the snapping rolls, the snapping rolls separate the clusters from straws. The clusters fall under the effect of gravity, and the straws are preliminarily flattened via the snapping rolls, and then directly enter the tearing rolls, and are torn into several sections via more zigzag-shape tearing blades on the tearing rolls, and the stems are destructured into fine filaments, and then output from the discharge hole. The torque test system is installed at the power output end of the motor which provides power for the tearing rolls, and it can measure the torque of the tearing rolls. The system consists of torque sensor, dynamic deformeter, adaptor and a computer installed with dynamic signal test and analysis system software.

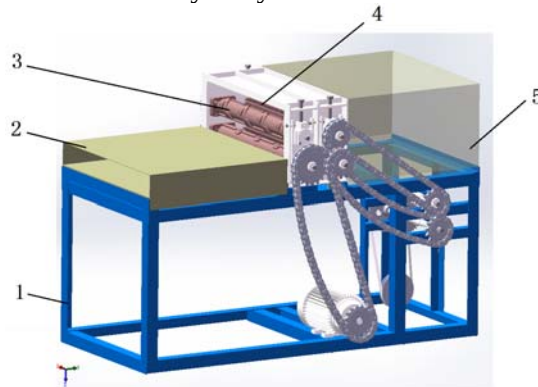


Figure 4. The device of tearing corn straw into forage test bench.

1. chassis 2. feeding station 3. snapping rolls 4. tearing rolls 5. discharge hole

Test Materials and Method

Test Materials. The corn species used in this test is “Xianyu 335” corn planted in the test field of Jilin Agricultural University. The corn straws with normal growth and development were selected as the test materials

Test Factor and Level. Single-factor tests were carried out respectively with the circular blade front angle, snapping roll and tearing roll speed variation and harvest period as the factor, and forage qualified rate, wrapped grass amount and torque size as the evaluation indicators. blade front angles are respectively 30°, 20° and -15°; The corn straws selected on September 10, September 20, September 30, October 10 and October 20, 2014 were respectively marked as 1 (early period), 2 (early-middle period), 3 (middle period), 4 (middle-late period) and 5 (late period for test); the rotation speed of snapping rolls was 800r/min, and the rotation speed of the tearing rolls was determined as 800r/min, 1000r/min, 1200r/min, 1400r/min and 1600r/min.

Calculation Method of Foraging Qualified Rate. The calculation method of foraging qualified rate in this experiment is: in 5 repeated tests under the same test condition, 15 corn straws were weighted in advance before processing, with total mass recorded as M1, and after processing via the test bench, the straws with average lobe width below 6mm were the qualified foraging products (as shown in Fig. 5), with total mass recorded as M2, so as to calculate the foraging qualified rate:

$$Q = \frac{M_1}{M_2} \cdot 100\% \quad (1)$$

where Q is the foraging qualified rate under the test condition.



Figure 5. Qualified product.

Evaluation Method of Wrapped Grass Amount. The wrapped grass situation of tearing rolls was observed during test. The wrapped grass is reflected through grading from 1 to 5, Without grass wrapping, foraging is smooth when 5 scores, the more serious the grass wrapping is, the lower the score of the amount of wrapped grass will be.

Test Result and Analysis

Result of Single-Factor Test of Blade Tooth Angle. Fig. 6 and Fig. 7 show the difference of change of tooth angle on the qualified rate, score of grass wrapping and torque. During test, speed of snapping rolls was 800r/min, and speed of tearing rolls was 1400r/min. Samples were taken on September 30. It can be seen from these figures that:

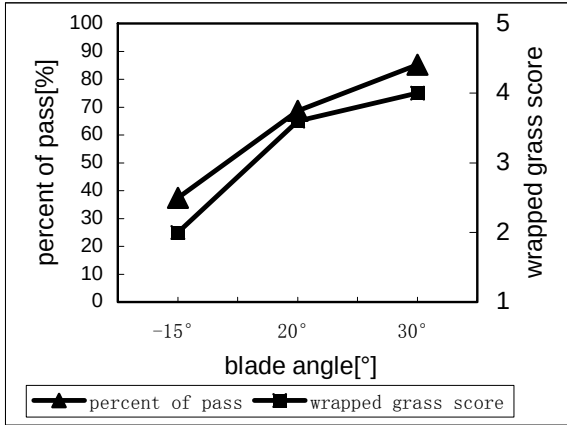


Figure 6. Percent of pass, wrapped grass score with blade angle.

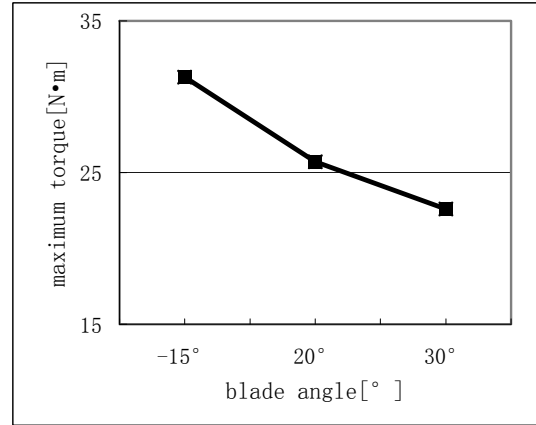


Figure 7. The maximum torque changes with blade angle.

With the increasing of blade angle, the qualified rate increased and the foraging effect was improved; the score of amount of wrapped grass increased gradually, while the amount of wrapped grass gradually decreased. The torque decreased successively. According to the calculation formula of power consumption:

$$P(W) = \frac{M \times n}{9549} \times 1000 \quad (2)$$

where n is the motor speed, the calculation of which is:

$$n = i' \times n_s \quad (3)$$

where n_s is the rotation speed of tearing rolls, and according to the design of the device, the value of transmission ratio of tearing rolls i' is taken as 0.59, and in combination with formulas (2) and (3), it is calculated that the minimum power is 1955W when the blade front angle is 30°.

Single-Factor Test Result of Speed Variation of Snapping Rolls and Tearing Rolls. Fig. 8 shows the influence of speed variation on qualified rate, score of wrapped grass and torque. Fig. 9 shows the influence of change of speed variation on torque and power consumption. It can be seen from two figures that:

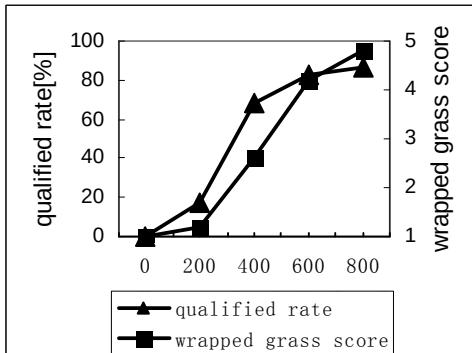


Figure 8. Qualified rate, wrapped grass score with rolls speed variation.

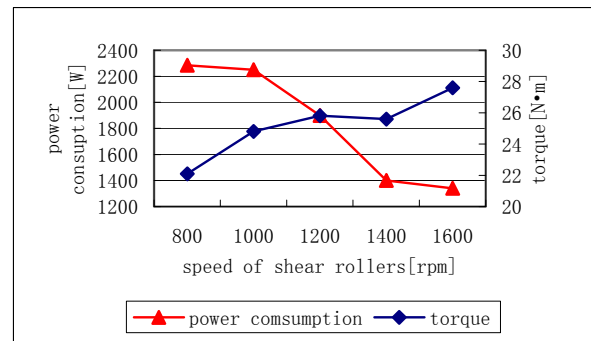


Figure 9. Torque and power consumption change with speed.

During the test of corn straws harvested in the middle period with a circular blade of 30° front angle, when the same speed was adopted for the snapping rolls and tearing rolls, the qualified rate was 0%, with serious grass wrapping to the extent that the operation could not continue, this is

because the same rotation speed was used for the snapping rolls and tearing rolls, without relative speed. With the increasing of speed variation and increasing of foraging qualified rate, the foraging effect was improved.

The change of tearing roll speed caused by the speed variation was an important factor to influence the torque size, and the torque decreased with the increasing of speed variation, when the speed variation reached 800r/min, i.e. when the speed of tearing rolls reached 1600r/min, the minimum value of maximum torque was 21.2N.m.

When the speed was between 800-1200r/min and 1400-1600r/min, with the increasing of speed of tearing tools, the power consumed by the tearing rolls maintained growth basically. When the speed of tearing rolls was 1200-1400r/min, the growth of power consumption tended to be stable and even decrease. Therefore, the reasonable interval of the speed of tearing rolls was 1200-1400r/min, and the power consumed at this time was maintained at about 1880W.

Result of Single-Factor Test of Corn Straws during Harvest Period. Fig.10 and Fig.11 respectively show the influence of change of different harvest periods on qualified rate, score of wrapped grass and torque. It can be seen that:

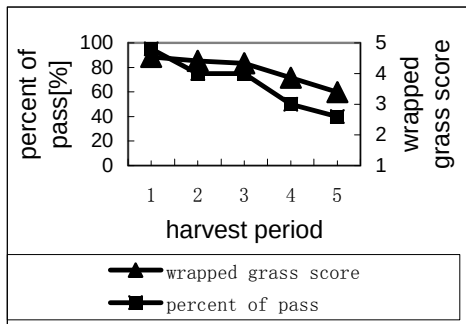


Figure 10. Percent of pass, wrapped grass score with the harvest period.

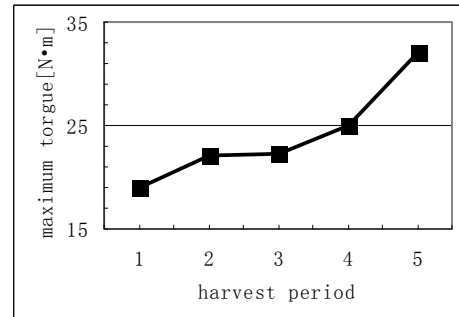


Figure 11. The maximum torque changes with the harvest period.

During the test of corn straws with the circular blade of 30° front angle at the tearing roll speed of 1400r/min, the harvest period had a great influence on the foraging qualified rate, with the harvest period ahead of time, the qualified rate increased, and foraging effect was improved; the amount of wrapped grass score increased gradually, and the amount of wrapped grass decreased gradually.

In the harvest of the middle period, the corn straws were foraged with a qualified rate of >80%, and if the harvested corn straws were foraged in the early period, the qualified rate could be up to 95.4%, indicating that the corn straw foraging device had a very good work performance and that this machine can be used to forage the straws.

The torque increased with the postponing of the harvest period, when the corn straws harvested in the late period were processed, the torque increased to 32.06 N·m.

Conclusion

1. The increasing of front angle of circular blades, increasing of speed variation of snapping rolls and tearing rolls and the advancing of harvest period could increase the foraging qualified rate, increase the score of wrapped grass, improve grass wrapping condition and reduce torque and power consumption.
2. During the speed variation of snapping rolls and tearing rolls, it was found that with the increasing of speed of tearing rolls, the torque decreased, and the power consumption basically maintained growth, but when the speed of tearing roll was 1200-1400r/min, the growth of power consumption tended to be stable and even decreased, and then increased with the increasing of speed and decrease of torque.
3. According to the test result above, it was determined that the optimal parameters were to adopt the circular blade with a front angle of 30°, control the rotating speed of tearing roll within the

reasonable interval of 1200-1400r/min and harvest and process the corn straws as early as possible under the premise of ensuring the maturity of the corn grains. This plan can effectively improve the qualified rate of the device, avoid grass wrapping and reduce torque, so as to maintain the power consumption at a lower level, thus foraging the corn straws with low energy consumption.

Acknowledgements

This paper was supported by Jilin Scientific and Technological Development Program (20130305043NY) and program of the Ministry of Science and Technology on serving enterprises with scientific and technological personnel (2009GJB0008)

References

- [1] Zhang Jie, Fang Hanpu, Problems Existing in Utilization of Corn Straws and Solutions [J], Animal Husbandry and Feed Science, 2014, 05: 101-102.
- [2] Wang Tielin, Design and Test of Corn Straw Processing Equipment [D] Shandong Agricultural University, 2013
- [3] E.D. Thomas, P. Mandebvu, C.S. Ballard, C.J. Sniffen, M.P. Carter, J. Beck et al. Comparison of corn silage hybrids for yield, nutrient composition, in vitro digestibility, and milk yield by dairy cows [J]. J Dairy Sci, 2001, 84 (10): 2217-2226.
- [4] Hu Junmei, Analysis of Resources Utilization of on Crop stalks [J] Guangdong Agricultural Sciences, 2010, 04: 207-210.
- [5] Liu Yi, Huang Yaoming, Brief Discussion on Technology and Development for Comprehensive Mechanical Utilization of Straws [J], Shaanxi Journal of Agricultural Sciences, 2012, 03: 149-151.
- [6] Yu Yinglin, Dong Yunzhe, Xie Tao, Li Mingsen, Shi Yuntian, Brief Discussion on Comprehensive Utilization of Straws [J], Agriculture and Technology, 2009, 06: 73-74.
- [7] Yang Yazhou, Xu Jie, Wang Jingli, Lu Chaohui, Liu Qingfu, Design of a new chopper for green corn stalk [J], Journal of Yangzhou University (Natural Science Edition), 2010, 02: 41-43+56.
- [8] Dong Dejun, Chen Haixia, Wei Yunfeng, Current Status and Development Trend of Coarse Feed Processing Machinery [J], Journal of Agricultural Mechanization Research, 2006 (5): 42-43.

Design and Comparative Trials of Three Kinds of Tearing Rolls for Maize Straw Forage

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Keywords: nail-like roll, saw-like roll, rotate-knives, maize straw, tearing roller, comparative trial.

Abstract. In this paper, three kinds of stem-tearing rollers for maize straw forage were designed: nail-like roller, rotate-knives roller and saw-like roller, which could be connected with com snapping roll to form the maize straw foraging part. They could be installed on vertical roll harvester to realize the combination operation of corn harvest and straw forage. The working performances of three stem-smash rollers were tested on the self-made corn straw forage test bench under the same experimental conditions, and comparative analyses of the results were made. The results showed that the saw-like tearing roller had the best foraged effect for straw, and then the nail-like roller. The rotate-knives stem-smash roller had the worst effect.

Introduction

At present, the methods of the comprehensive utilization of maize straw are breaking into the soil, rural and industrial fuel, fungi crop culture medium and livestock fodder [1, 2]. The related maize straw processing machines mostly are small separate operating machines, such as hay cutter, silk rubbing machine and grinder [3-5], the coarse straw fodders processed by some machines have different problems: in the processing, the internodes of straw are not appropriately processed. The ingestion rate is low and this will cause esophagus block of the livestock; while some coarse straw fodders processed are too smashed, which are hard for livestock to chew and the palatability is bad [6]; some of machines have the shortcomings of high energy consumption, low production efficiency and bad safety [7,8]. Aiming at the problems above, three kinds of stem-tearing rollers for maize straw forage (nail-like, saw-like and rotate-knives) were designed in this paper, which could be connected with com snapping roll to form the component of maize straw pulverization. It could be installed on the vertical roller harvesters to realize the combination operation of corn harvest and straw pulverization. Straw could be torn into pieces along the axis direction and could be quickly dried and compacted into bundles to solve the problem of the loss of nutrient substance caused by longtime drying and the transportation difficulties caused by the large volume. Or the torn straw can be used to feed the livestock directly.

Structure and Working Principle of Straw Forage Components

Structural Composition. Straw forage component is (as shown in Fig. 1) composed of support frame, a pair of snapping rolls, a pair of tearing rollers and the drive mechanism. The designed component of maize straw forage can be installed on side suspended combine harvester. The effect of the installation is shown in Fig. 2.

Working Principle. When working, the harvested maize straw is transported into com snapping rolls through conveyor, then the processed maize straw is flattened and pushed into tearing roller. Smashed by the tearing roller, the maize straw is ripped into pieces, and then casted onto the platform floor. Then it is passed through the chain drive on the platform floor to be compiled on the ground to dry.

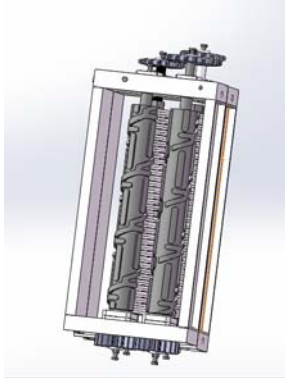


Figure 1. 3D assembly drawing of straw forage components.



Figure 2. The picture of the complete machine with components installed.

Structural Design of Tearing Roller

Design of Nail-like Tearing Roller. The nail-like tearing roller is composed of three parts: the short axis for installing the bearing and chain wheels, tearing nail teeth and cylinder (as shown in Fig. 3). The cylinder is the welded carrier of the nail teeth. The nail teeth are the main working parts which have a ripping effect on the maize straw. Total height of nail teeth is 36mm. The diameter of it is 4mm, with the cone height of 6mm. The outsider diameter of the cylinder is 60mm, and the length of 455mm. The circumference included angel of adjacent two rows of nail teeth is 30° , and the distance of the nail teeth of the same row is 20mm.

Design of Rotate-Knives Stem-Smash Roller. The components of rotate-knives stem-smash roller are cylinder, blunt blazed and short axis (as shown in Fig. 4). Dimension of cylinder and short axis is same as nail-like roller. The circumferential welding section of cylinder is six steel blunt blade of $5\text{mm} \times 25\text{mm}$, with the length of 455mm.



Figure 3. nail-like tearing roller.



Figure 4. Rotate-blade smashing roller.

Design of Saw-like Stem-Smash Roller. The saw-like stem-smash roller is composed of square shaft, saw blade and location spacer. The saw blade is made by 1.8mm steel board slicing with 16 teeth. The dip angel of saw tooth is 30° , and the tooth length is 25mm. The saw blade is circumference square shaft locating. And the adjacent blades are separated with spacers with the distance of 14mm (as shown in Fig. 5 and Fig.6).

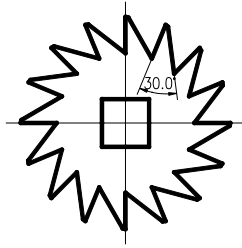


Figure 5. The 2D diagram of the blade.



Figure 6. Saw-like smashing roller.

Comparative Trials on the Working Performances of Three Stem-Smash Rollers

Experimental Equipment. The test bench of corn straw forage device is composed of the rack, snapping rolls, tearing roller, drive system, and the control cabinet (as shown in Fig. 7). When device was working, add experimental maize straw in a simulating homogeneous velocity of the harvester, after processed by snapping rolls, the straw were torn into several section by tearing roller. The rotate speeds of snapping rolls and tearing roller could be adjusted by the frequency changer in the control cabinet.

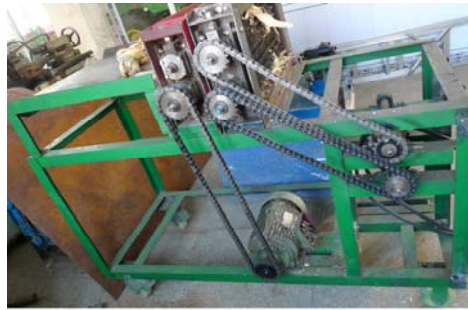


Figure 7. The test bench of corn straw forage device.

Trial Conditions. To compare the working performances of three smashing rollers, the same trial conditions were adopted, as shown in Table 1.

Table 1. Experimental conditions.

Experimental parameter	value
com snapping rolls rotate speed	1000r/min
Stem-tearing roller rotate speed	1300r/min
stem-tearing roller centre distance	125mm
Date of sampling	2014-9-28
Trial sample size	20
Straw sample length	2.2m—2.4m
Trial sample average knobs	13

Trial Evaluation Index

1. Straw smashing length

The diameter of straw decreases from the root to the top. The upper part is too thin to be torn into slip. We hope that the smashing length of the straw could be as long as possible. Smashing length was used as trial index. 20 straws were taken as samples and the average value of smashing length and the variance were recovered.

2. Tearing pieces of straw

Due to the decrease of diameter of the straw above 5 knobs, the tearing effect was worse than that of the root. So the tearing pieces number is calculated into two parts from 1 to 5 knobs and from 5 to 10 knobs. 20 straws were taken and the average value of tearing pieces and the variance were recorded.

Comparative Analysis of Trial Results

Comparative Analysis of Stem Tearing Effect. The trial results and variance comparisons of three stem-tearing rollers are shown in Table 2. From Table 2, among the three stem-smash rollers, the tearing lengths of saw-like roller, nail-like roller and rotate-knives roller accounted for the proportion of the overall length respectively as 69%, 57.7%, 47.7%; the smashing pieces of 1—5 knobs were 11.7, 7.05, 5.2; the smashing pieces of 5 -10 knobs were 6.5, 3.05, 2.55. It showed that tearing effect of saw-like tearing roller was the best, and then came the nail-like ones. The stem pulverization effect of rotate-knives stem-smash roller was the worst.

The overall length of straw could not be torn in pieces by all the three rollers. But it would not influence the straw pulverization effect, because the overall length of straw tested in this paper included staminate inflorescence. Its diameter of the upper part for almost 0.8 meter was very thin which could be quickly dried and feed directly without being smashed.

Table 2. Average and variance of test result about three tearing rollers.

stem-tearing roller	Name of parameter	Average value	variance
nail-like	Overall length of straw(mm)	2346.3	
	tearing length(mm)	1355	
	the average tearing pieces of 1—5 knobs (p)	7.05	2.15
	the average smashing pieces of 5 -10 knobs(p)	3.05	0.63
Saw-like	Overall length of straw(mm)	2278.5	
	Smashing length(mm)	1579.5	
	the average smashing pieces of 1—5 knobs (p)	11.7	1.39
	the average smashing pieces of 5 -10 knobs(p)	6.5	3.85
rotate-knives	Overall length of straw(mm)	2358.7	
	Smashing length(mm)	1125	
	the average smashing pieces of 1—5 knobs (p)	5.2	2.16
	the average smashing pieces of 5 -10 knobs(p)	2.55	0.35

Analysis of the Working Performances of the Three Stem-Smashing Rollers. Observing the working process of the three kinds of stem-tearing rollers, rotate-knives stem-smash roller had bigger working noise, when adding too much straw into rotate-knives roller, there would be choking phenomenon. Nail-like stem-tearing roller ran well with small noise, but after a period of time grass would wrap the roller which needed periodic cleaning. The stem pulverization effect of saw-like stem-smash roller was the best. The component of saw-like stem-smash roller ran well with small noise, and no grass wrapped in the current velocity.

Conclusion

1. Three different kinds of stem-tearing rollers were designed, they respectively were nail-like roller, rotate-knives roller and saw-like roller, which could be connected with com snapping roll to form the maize straw foraging part. They could be installed on vertical roll harvester to realize the combination operation of corn harvest and straw forage.
2. Comparative experiments on trial platform were made for three tearing roller. The results showed that: the tearing length of saw-like roller was the longest; the average smashing pieces was the most. During test, saw-like roller ran well with small noise, and no grass wrapped it in the current velocity.
3. To sum up, the tearing effect and working performance of saw-like roller was the best.

Acknowledgements

This paper was supported by Jilin Scientific and Technological Development Program (20130305043NY) and program of the Ministry of Science and Technology on serving enterprises with scientific and technological personnel (2009GJB0008)

References

- [1] Liu Yu, Shi Tongrui, Zhu Dandan, etc. Utilization Status of Corn Stalk and Research Progress of the biological Fermentation Technology. [J]. Chinese Journal of Animal Science, 2014, 50 (6): 61-64.
- [2] Hou Jie. Related Study on Mechanical Characteristic and Physicochemical Property of Corn Straw [D]. Heilongjiang: Northeast Agricultural University, 2013.
- [3] Duan Tianqing, Sheng Guocheng, Zhang Xiong. Research and Design of 9RS-40 Straw Cutting and Rubbing Machine [J]. Journal of Chinese Agricultural Mechanization, 2013, 34(2): 106-108, 130.
- [4] Chen Liqing, Wang Li. Zhang Jiaqi, et al. Design of 1JHSX-34 straw crusher for whole-feeding combine harvesters [J]. Transactions of the CSAE, 2011, 27 (9): 28-32. (Written in Chinese with English abstract)
- [5] Han Lujia, Liu Xiangyang, Yan Qiaojuan, et al. Design of 9LRZ-80 Vertical Straw Rubbing and Cutting Machine [J]. Transactions of the CSAE, 1999, 15 (3): 137-141. (Written in Chinese with English abstract)
- [6] Wang Fang, Wang Chunguang. Elementary Introduction on Development of Straw processing machineries in China [J]. Journal of Agricultural mechanization Research, 2007, (8): 211-216.
- [7] Yang Yazhou, Xu Jie, Wang Jingli, etc. Design of a new chopper for green corn stalk [J], Journal of Yangzhou University (Natural Science Edition), 2010, 02:41-43+56.
- [8] Meng Haibo, Han Lujia. Analysis of application and prospects and study status of the processing properties of straw materials [J]. Journal of China Agricultural University, 2003, 8 (6) :38-41.

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