Testing Research of Liner Stress in Process of Side Pressing

Xianzhang FENG
Zhengzhou Institute of Aeronautical Industry Management / School of Mechatronics Engineering, Zhengzhou, China
E-mail: phdfxz@163.com

***Abstract*—Liner is key parts of equipment of slab side pressing, its main function is supporting the weight of lateral pressure, and are under some impact pressure, which is impact force up to 2000 tons, it make liner frequent breakage in the process of working, an led to entire production line stop, seriously affect the yield and efficiency of hot rolled strip. In order to obtain the reasons of liner fracture, the means of non-linear finite element and test combination were used to test and verify the correctness of finite element model, it is a new solution on how to settle liner breakage by calculation and analysis. The result of study indicate that it can effective increase the output and reduce the cost of cut end for hot-rolled strip, and has obtained the anticipated effect.**

***Index Terms*—testing, contact mode, strain, stress distribution, waveform signals, liner**

I. INTRODUCTION

In recent years, with developing rapidly of the continuous casting and rolling technology, it can be built the connect between continuous casting and hot rolling for the width, to realize the slab join between casting and rolling width, It is one of effective measures to carry out inline changing in slab width during rolling Such as the widely used vertical roll rolling and width reduction method. Slab side pressing is an online regulation width technology for continuous casting blank. It can reduce the width types of slab casting, and improve the lever of unification in casting and rolling. [1-9]

When the continuous lateral pressure reached 250mm, the mill is under the high load condition, as weight of the lateral pressure the framework is nearest 100 tons. Maximum lateral pressure of more than 2200 tons, a great shock and vibration in the process of working, irrational allocation of clearance and the amount of interference, exacerbated the impact of shock and vibration. Some of the key device to bear a larger load. In the course of rolling mill failure frequently affecting the productivity and product quality for the hot-rolled strip.[10-15]

The FE model of liner force was established by the theory of nonlinear finite element. Based on the model, analysis the mechanical of the mill, it is a new solution on how to settle liner breakage, Analysis of key components of life and the corresponding countermeasures maintenance and Repair. The influence of different side pressures on rolling force was also studied systematically in stable and no-stable area. The change regulations of rolling force were attained in different side pressures and fixed width area respectively, which would offer reference to formulate the slab regulation width. The result of study indicates increased the output and lessened the cost for hot-rolled strip, and has obtained the anticipated effect. [16-20]

II. MODEL ANALYSIS

A Impact Theory

Assume that the size of relative velocity ratio of is the same before and after the collision for two objects, the ratio called the linear reply coefficient, where is used to indicate e, $0 < e < 1$, When the e is 1, indicating perfect elastic collision, When the e is 0, indicating perfect plastic collision, If the two objects collide, in which their mass is m_1 and m_2, their speed is v_1 and v_2 before collision, and their speed is v_1' and v_2' after collision, under the law of momentum, the formula is expressed as follows:

$$\left.\begin{aligned} v_1' &= v_1 - (1+e)\frac{m_2}{m_1+m_2}(v_1 - v_2) \\ v_2' &= v_2 - (1+e)\frac{m_2}{m_1+m_2}(v_2 - v_1) \end{aligned}\right\} \tag{1}$$

The expressions of kinetic energy losing is as follows:

$$\Delta E = \frac{m_1 m_2}{2(m_1+m_2)}(1-e^2)(v_1 - v_2)^2 \tag{2}$$

For the impact in terms of conventional, the Initial velocity is 0 for the impactors, that is $v_2 = 0$, the formula (2) can express as:

$$\Delta E = \frac{m_2}{m_1+m_2}(1-e^2)E_0 \tag{3}$$

Where E_0 is the kinetic energy of the impactor before the collision, the formula is expressed as follows: $E_0 = \frac{1}{2} m_1 v_1^2$.

Based on the mathematical description of principle of virtual work, the interaction between the two objects can be described in Principle of virtual in space.

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$$\int_{t\Omega} \delta\tilde{\varepsilon} \cdot \sigma_t d\Omega - \int_{t\Omega} \delta u \cdot F_{Vt} d\Omega - \int_{Af_0} \delta u \cdot \tilde{q}_t dA -$$
$$\int_{\Omega_0} \delta u \cdot \rho \cdot a_t d\Omega - \int_{A_{ct}} F_{ct} \cdot \delta\tilde{u} \cdot dA = 0 \quad (4)$$

Where, Ω is the space of occupy for the impact of the collision of two objects. σ_t is Cauchy stress, $\delta\tilde{\varepsilon}$ is virtual strain, δu is virtual displacement vector, F_{Vt} is volume force, Af_0 is external load area, $\tilde{q}_t$ is external load vector act on the Af_0 , A_{ct} is contact area between space objects collision, F_{ct} is contact force act on A_{ct} , $\delta\tilde{u}$ is the relative virtual displacement for two points of contact basing on the F_{ct} , ρ is the mass density, a_t is acceleration vector, t is subscript, moment.

Assumed at a certain point in process of deformation, above the entity which is the volume V, surface is s_σ ,the component of the surface force vector is p_i , the component of the surface force vector is p_i , the force per unit volume is F_i ,basing on the Principle of virtual work, the deformation of the formula is as following:

$$\int_V \sigma_{ij} \delta v_{ij} dV = \int_{S_\sigma} p_i \delta v_i ds + \int_V F_i \delta v_i dV \quad (5)$$

σ_{ij} is component of Euler stress tensor, δv_i is component of virtue, δv_{ij} and is the virtual strain rate.

The more rigid system dynamics equation is as following:

$$\begin{cases} \dfrac{d}{dt}\left(\dfrac{\partial T}{\partial \dot{q}}\right)^T - \left(\dfrac{\partial T}{\partial q}\right)^T + \phi_q^T + \theta_q^T \mu - Q = 0 \\ \phi(q,t) = 0 \\ \theta(q,\dot{q},t) = 0 \end{cases} \quad (6)$$

Where T is system energy, $T = \frac{1}{2}\left[M \cdot v \cdot v + \omega \cdot I \cdot \omega\right]$, q is generalized coordinates array, Q is generalized force array, p is lagrange multiplier array for corresponds to the holonomic constraint, μ is lagrange multiplier array for corresponds to the non holonomic constraint, M is quality array, v s generalized velocity array, I is moment of inertia array, ω is generalized angular velocity array.

Here, which $\phi(q,t) = 0$ is holonomic constraint equations, $\theta(q,\dot{q},t) = 0$ is nonholonomic constraint equations, then the general form of formula 6 is as following:

$$\begin{cases} F(q, v, \dot{v}, \lambda, t) = 0 \\ G(v, \dot{q}) = v - \dot{q} = 0 \\ \Phi(q,t) = 0 \end{cases} \quad (7)$$

Where q is generalized coordinates array, $\dot{q}, v$ is generalized velocity array, λ is the array of constraint forces and forces, F is System dynamics equations and user-defined differential equations, Φ is algebraic equations describing the full array of constraints, G is the description array of non-holonomic constraint equations.[21-25]

B Model

Liner under side framing was established during side pressing, using the superposition theory of micro elastic displacement. The source program of dynamic characteristics lining board was edited with FORTRAN language. The model of contact described in Figure1:

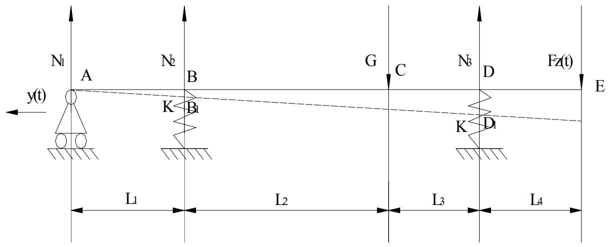

Figure 1. Force model of liner

The contact force law is:

$$F = Kx + c\dot{x} \quad (8)$$

Where F the normal force in stress direction of liner board, K was was the contact rigidity, c was the damping factor, x was the deformation displacement, $\dot{x}$ were the speed of contact point separately.

In Figure 1, between the line segment AE is the framework of lateral pressure, point A is connection point of acetabular seat, point B and D are points of contact between wheels and liners, point C is center of gravity position of lateral framework, point E is point of force of module along Z direction. As the lateral deformation of the framework it is smaller than the wheel contact with the liner deformation. So consider the rigid framework of lateral pressure, spring force represents the contact force of liners and the wheels, K is the contact stiffness of the liner and the wheels.

The required parameters for calculation are as follows:

Slab material is 45 steel, its specifications is 1300mm ×230mm×9900mm, Friction factor is 0.3, Elasticity coefficient is E=12×104Mpa, The angle of die bevel is 12°, The movement cycle of die is 1.2 s, and the temperate of slab is 1050 ℃. The force curve was shown in Figure 2.

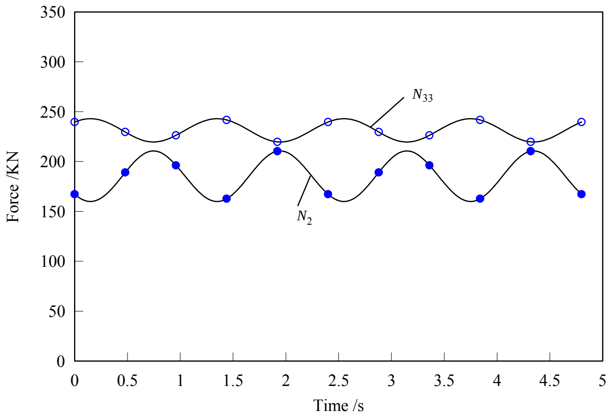

Figure 2. Force curve of liners

It is significantly different force for the three liners, in which is support the weight of lateral pressure framework. The largest force lie in D points under the module The role of rolling contribute further exacerbated this phenomenon of uneven force, Theoretical results in good agreement with the production site.

III. THE MESHING

A Solid Model

According to geometric symmetry of the structure of slab size pressing mill, it can establish solid contact model for three-dimensional as shown in Figure 3.

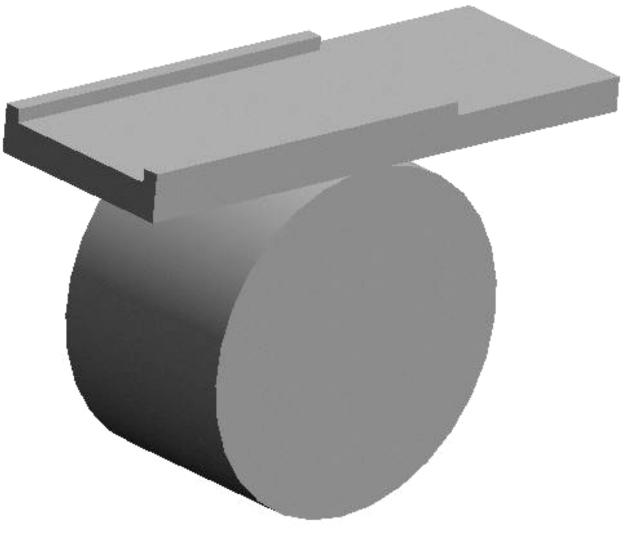

Figure 3. Contact mode of liner

In the course of division the grid in the contact model, according to the extent of deformation, the liner board as elastic mass, and the liner and bottom of side frame was connected with screws. Because they were connected with two different material parts, it was simplified to elastic body, and the wheel was regarded as rigid body. Since the wheel did the periodicity roll on the contact surface of liner board, we could get the displacement curve of wheel in a period. The force boundary condition that dynamics analysis got was exerted on the axis of wheel. The liner board and its linking part were regarded as motionless, which could analyze the internal stress change in different contact district of liner board. The related parameter of the component mechanics shows Table 1 in coures of analyzing basing on the contact model.

B Boundary condition

TABLE I.
MECHANICAL PROPERTY TABLE

Component	E/MPa	Poisson's ratio / υ	Friction
Liner	205	0.3	0.29
Screw	200	0.29	0.3
Body 1	190	0.29	0.3
Body 2	180	0.29	0.3

The calculation results of post processing finite element, stress cloud which lie in bolt hole center of liner was shown in Figure 4.

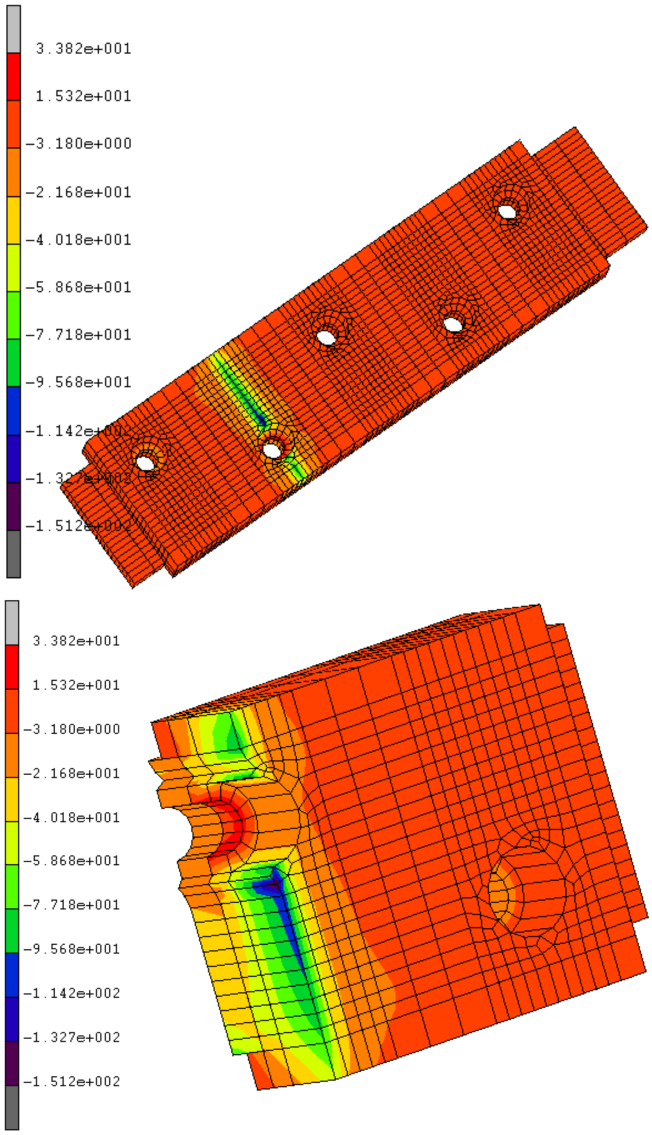

Figure 4. Contact stress cloud in center of adjacent bolt holes

The Fig4 is Shown the stress cloud overall view and local view in bolt hole center of liner.

Stress cloud which lies in bolt hole edging of liner was shown in Figure 5.

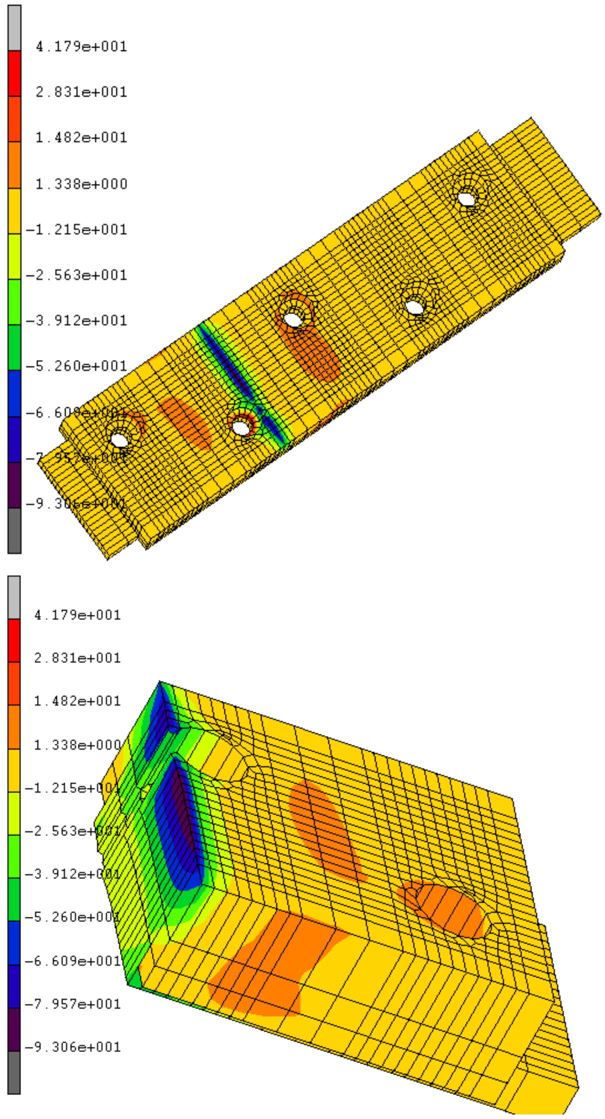

Figure 5. Contact stress cloud in bolt hole edging

The Fig5 is Shown the stress cloud overall view and local view in bolt hole edging of liner.

Stress cloud which lie in the meddle of adjacent the bolt holes of liner was shown in Figure 6.

The Fig6 is shown the stress cloud overall view and local view in the middle adjacent the bolt holes of liner.

Under the conditions, including bolt hole diameter 36mm, thickness 76mm, research the contact cloud stress in different positions for liner, in order to test the correctness of the calculated results basing on the finite element model, according to the results of really testing, in the same operating conditions, it compare with the variation of strain in the same location for liner.

As the liners undertake most of the load by the wheel contact in vertically, the maximum stress lie in z direction. It can gain the results of stress distribution of liners by analyzing the FE model. The results of calculation are as shown in Figure 7:

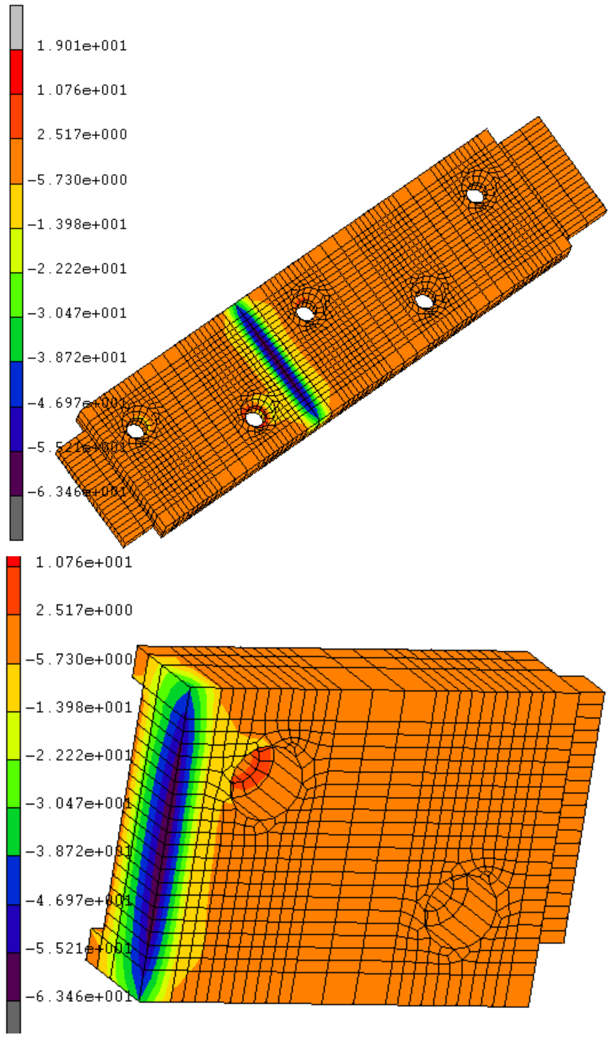

Figure 6. Contact stress cloud in the meddle of adjacent the bolt holes

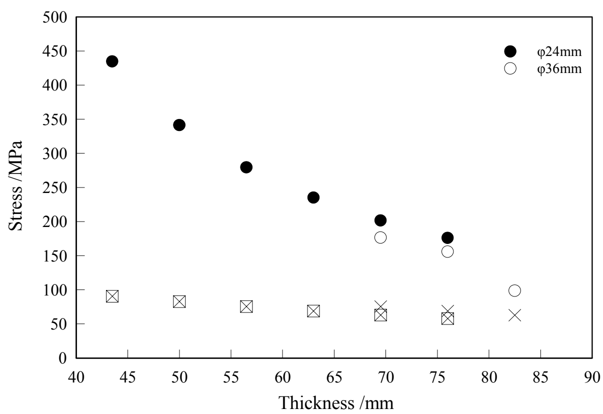

Figure 7. Stress distribution in liner

In order to study the relation between the liner thickness and bore diameter, taking the stress as the study object along the z orientation, giving two combinations of its thickness and the tap hole diameter: one was that the liner board was 43.5, 50.0, 56.5, 63.0, 69.5 and 76.0mm separately and the bore diameter was 24mm, another was that the liner board thickness was 69.5, 76.9 and 82.5mm, but the tap hole diameter was 36mm. Base on the finite element model, it could know that, a section which was parallel to zy plane and went through the center of tap hole was taken in the contact area of liner board and wheel. Figure 7 was the change regulation of the biggest stress of z direction on this section. From Figure 7 we could draw a conclusion that, when the liner board thickness was within the scope of 43.5-82.5 mm, and the tap hole diameter was 24mm and 36mm, the biggest stress value was in 434.6-98.69 MPa on z direction. Along with the increasing of liner board thickness, the biggest stress had the decreasing tendency on z direction. Under the condition of liner board with identical thickness, with the increasing of tap hole diameter, the biggest stress submitted to decrease tendency on z direction.

Similarly, a section which was parallel to zy plane and went through the center of two tap holes was taken in the contact area of liner board and wheel. And, Figure 7 was the change regulation of the biggest stress of z direction on this section. From Figure 4 we found that the biggest stress was in 90.47-57.79 MPa on z direction. The stress would decrease along with the liner board thickness increasing. Under the condition of liner board with identical thickness, along with the increase of tap hole diameter, the biggest stress would present the tendency of increase on z direction.

IV. TESTING ANALYZING

Since the liner board is located in the bottom of side frame, it will make reciprocating movement in the rolling process. The strain change on z direction will be attained through strain gauge directly pasting on liner board side during constant width course. Its measure result will offer reference for the finite element analysis of liner board. The methods were adopted by strain gauge on the edge of liner, its Layout shown in Figure 8.

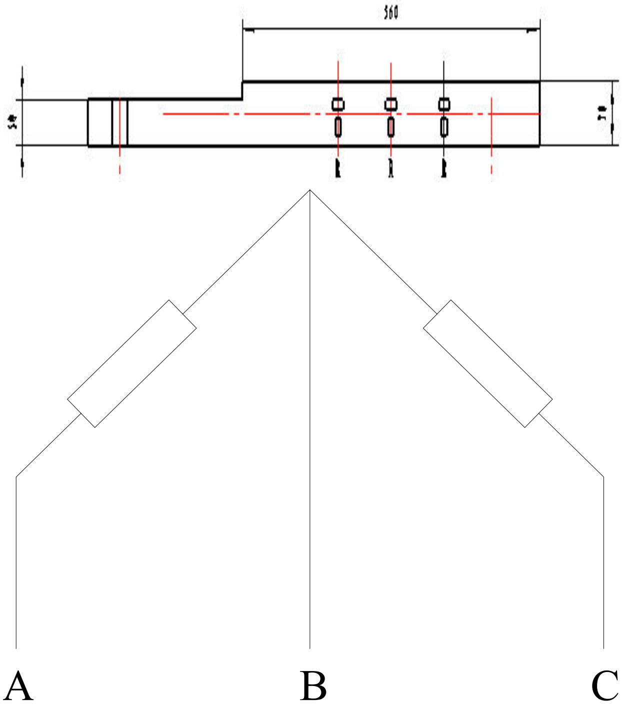

Figure 8. Layout of strain gauge

The waveform signals of collection for strain shown in Figure 9.

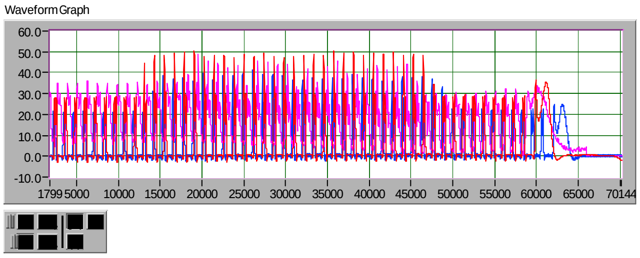

Figure 9. Waveform signals of strain

The liner thickness was 50mm before reformation in the equipment of slab size pressing rolling mill, and the diameter of tap hole was 24mm. After deformation, the thickness of liner board was 76mm and diameter of tap hole was 36mm. This test went on the SP rolling mill after deformation.

The calculation results of liner for strain were shown in Figure 10.

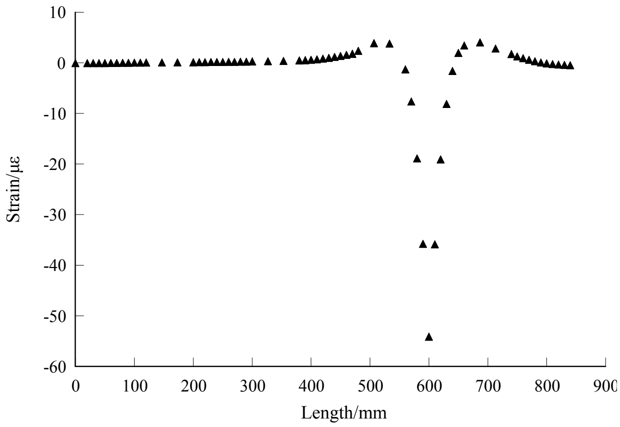

Figure 10. Research results

From Figure 9 and Figure 10, the peak value is 52.224 $\mu\varepsilon$ for testing, and 54.15 $\mu\varepsilon$ for the calculation for liner strain. Both are less only 3.66 %, from the engineering point of view, the error of theoretical value and testing value of is smaller. Therefore, it can be considered the model and boundary conditions imposed is reasonable, the model could well reflect the actual working state in line.

To improve the force state of liner board, through the simulated emulation and analysis, we could find the stress concentration would appear at the point of thread hole because of the thread hole existence on liner board. The size of thread hole and liner board thickness was much small, the stress that the liner board got would increase. The diameter of thread hole and the thickness of liner board increased, the stress would reduce. But the installation of liner board had a certain size restrictions, it was very important to seek the suitable size combination.

According to the calculation result, we could know that when the thickness of liner board was 50.0 mm, and the diameter of thread hole was 24mm, the biggest stress on its z direction was 341.4 MPa, and when the thickness of liner board was 76.0mm, the diameter of thread hole was 36mm, the biggest stress was 176 MPa on z direction. The biggest stress reduced 48%. At the same time, its stress distribution coefficient also fell from 4.12 to 2.27. So, it would improve apparently the force of liner board by selecting the latter combinations, and this also coincided with the use of liner board on the spot. The service life of liner board was three months probably, after reformation, its life had exceeded six months.

V. SUMMARY

The finite element analysis model was established of liner in course of working by non-linear finite element theory, it gained the law of force distribution online board with time, and proposes a new solution on how to settle liner breakage. The result of study indicates increased the output and lessened the cost for hot-rolled strip, and has obtained the anticipated effect, the stress change should meet the rules, they were as follows：

With the increase of thickness of the liner, the peak of stress decreases, their relationship is approximately linear between the ranges of 43.5mm-82.5mm for thickness of the liner.

The peak of stress is trending up with the increase of bolt hole diameter under the same conditions

The more bolt hole diameter, the more stress distribution between bolt holes two is, Due to the presence of liner bolt hole, certainly there is stress concentration in course of contacting between liner and wheel.

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Xianzhang Feng is an associate professor of school of mechatronics engineering in Zhengzhou Institute of Aeronautical Industry Management, Zhengzhou, China. He received the Ph.D. degree from the Yanshan University in 2004. He specializes in software development for finite element, optimization technology, virtual Prototyping and mechanical design and theory.