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Life Prolongation Technology of Mechanical Elements

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The body can be viewed from the next page.

Life Prolongation Technology of Mechanical Elements*



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1 Introduction

Life prolongation of mechanical elements is a very important problem from the standpoint of minimizing of maintenance costs and improving productivity by lengthening the shutdown cycle. As shown in **Fig. 1**, the ratio of trouble caused by mechanical elements to all equipment trouble is 80% in terms of the downtime rate and 75% in terms of the trouble occurrence rate. These troubles classified by type of elements are shown in **Fig. 2**. Bolts and nuts that are very commonly used as mechanical elements are actually the greatest cause of trouble at steel works. Furthermore, there are significant number of troubles in universal-purpose mechanical elements such as cylinders and universal joints.

A detailed analysis of trouble in these mechanical elements has revealed that major problems relate to loose of bolt and nut, damage of packing in cylinders and the stiffness design of cross bearings in universal joints.

At Kawasaki Steel, therefore, in-house development of mechanical elements that surpass general-purpose technology has been positively carried out by expanding the scope of development to the level of mechanical elements of equipment.

This report describes the development for looseness-free nut, a life prolongation technology of a hydraulic cylinders and the development of a universal-joint cross

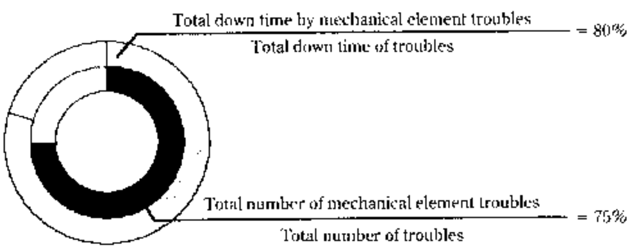

Fig. 1 Analysis of down time caused by mechanical elements

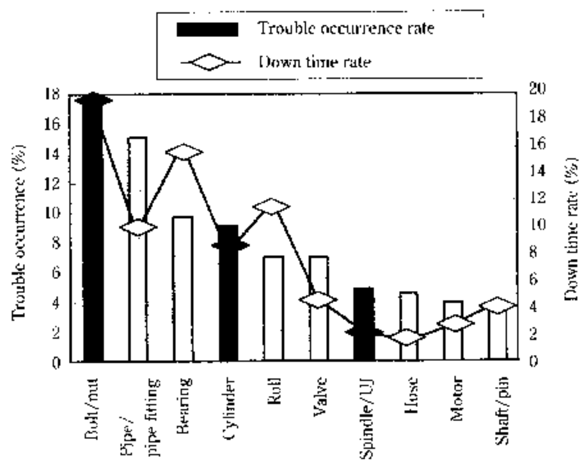

Fig. 2 Trouble occurrence rate and down time rate of several mecanical elements

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bearing with appropriate stiffness. In each case, long life has been achieved by essential improvement of the above problems.

2 Development of Looseness-free Nut

2.1 Analysis of Bolt and Nut Trouble

Bolt/nut trouble is shown by phenomenon and cause in **Fig. 3**. The number of cases of trouble in which bolt/nut looseness caused by vibration results in trouble is the largest, and cases where the bolt screw itself is broken due to looseness are the second. The total of these two classes of bolt/nut trouble caused by looseness accounts for approximately 85% of all bolt/nut trouble.

2.2 Mechanism of Looseness and Required Functions of Looseness-free Nut[1–3]

As shown in **Table 1**, nut looseness trouble can be roughly divided into looseness that occurs without rotation of the nut and looseness accompanied by rotation of the nut. In a typical example of the former, when the surface stresses under the bolt head and on the washer face are large, plastic deformation proceeds on the contact surfaces, producing caves, with the result that the bolt loses axial tension and the constrictive force decreases. In the latter case, when an impact force is applied by a vibration external force to a tightened portion, a slip occurs on the flank of a thread ridge, with the result that the coefficient of friction ultimately becomes zero, impairing the self-sustaining condition of the screw and causing looseness. In the type of looseness that occurs in the field, the above two phenomena proceed simultaneously, causing looseness. Therefore, the functions required of a looseness-free nut, from a standpoint of these mechanisms of looseness, were set as follows:

(1) A washer is provided in order to relive the surface stresses during tightening of a nut.
(2) In order to secure the resistance against vibration, a shearing force is applied in the direction vertical to the bolt shaft, thereby improving the looseness prevention effect.
(3) One-action tightening is made possible to ensure good workability.

2.3 Structure of Looseness-free Nut and Mechanism of Prevention Looseness

The structure of the newly developed nut and the looseness prevention mechanism are shown in **Fig. 4**. An inner nut with an eccentricity of δ in its threaded portion is housed in a mother nut, and the inner nut is secured by caulking the upper portion of the mother nut. Upon tightening of the mother nut, this eccentric inner nut gives the bolt a bending stress due to a shearing force (F) at right angles to the shaft, thereby improving the looseness prevention effect.[4,5]

2.4 FEM Analysis

An FEM analysis was conducted in order to evaluate the effect of the amount of eccentricity of the developed eccentric nut on the threaded portion of a bolt.

2.4.1 Analysis model

A two-dimensional analysis model, as shown in **Fig. 5**, was used. In order to evaluate the local stress on the threaded portion, the number of elements was set at

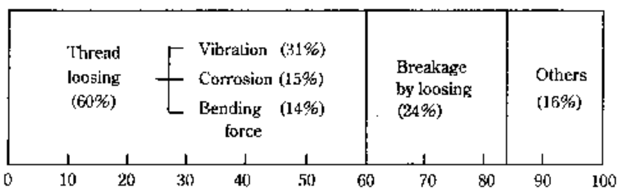


Fig. 3 Analysis of bolt and nut trouble

Table 1 Classification of thread loosing cause

Thread loosing without rotation	Early stage loosing Depression loosing (Thread face and base face) Micromotion wear (Thread face) Overladen external force Thermal factor
Thread loosing with rotation	Vibration at axial direction Vibration at horizontal and vertical directions Vibration at rotation direction

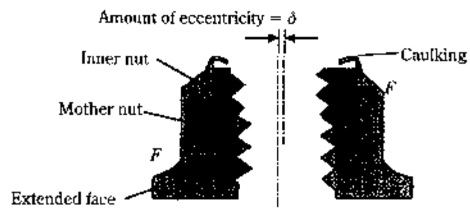


Fig. 4 Section of looseness-free nut

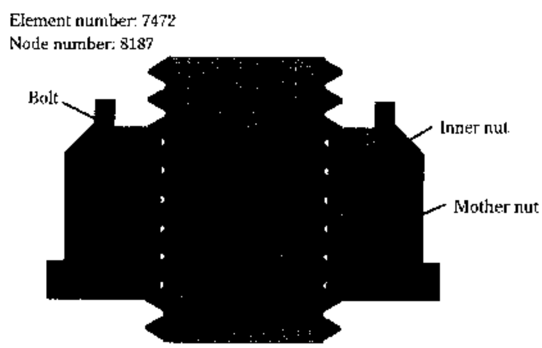


Fig. 5 Model of FEM analysis

7 472 and the number of nodes was set at 8 187. This analysis was treated as a contact problem because it is necessary to consider the contact among three parts, the inner nut, the mother nut, and the bolt.

2.4.2 Results of analysis

Figure 6 shows the Von Mises stress distribution in a case where the axial stress of the effective section of the threaded portion in ordinary bolt/nut tightening is 0.7 times as large as the yield point of a high tensile strength material. The highest stress is generated in the minor-diameter portion of the first thread of the engaged part of the screw between the bolt and the nut, and stresses decrease gradually in order, beginning with the second thread, third thread, etc. The local surface stress on the threaded portion of this first thread has reached a level almost equal to the allowable surface stress. When an external force is given by vibration or other factors, the thread face of the first thread first forms a plastic cave (i.e., undergoes permanent set in fatigue) and then the permanent set in fatigue proceeds to the second thread, the third thread, etc., releasing the axial tension of the bolt and causing looseness.

Figure 7 shows the Von Mises stress distribution for the bolt/nut when inner nut is in an eccentric condition (amount of eccentricity = 0.20 mm). The bending stress due to the shearing force at right angles to the bolt shaft given by the inner nut affects both the bolt portion on the eccentric side of the inner nut and the inner nut housed on the mutually opposed side. In the latter, the load is received by the whole surface of the threads of the bolt and it is possible to ensure uniform distribution of the surface stresses on the threaded portions of the bolt/nut by giving an appropriate amount of eccentricity. Therefore, it is possible to prevent the bolt looseness due to local permanent set in fatigue in thread ridges that occurs in general bolt/nut tightening.

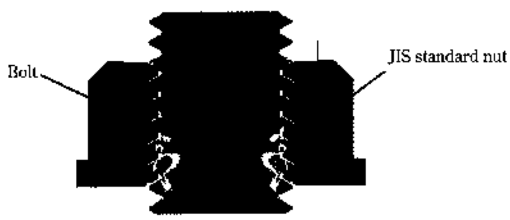


Fig. 6 Von Mises stress distribution of JIS standard nut (only axial force applied)

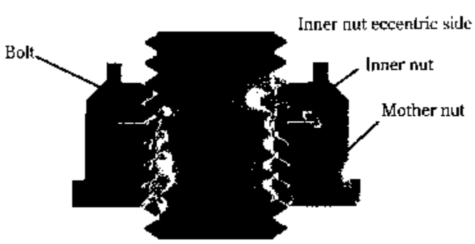


Fig. 7 Von Mises stress distribution of looseness-free nut (axial force and amount of eccentricity (δ) = 0.20 mm)

Bolt and nut material: SS400 Tt = Tf + Tp

Tightening torque (N · m) Tt	Prevailing torque (N · m) Tp	Axial tension torque (N · m) Tf
33	25	8
33	16	17
41	8	33
41	11	30

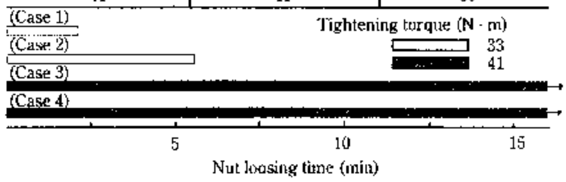


Fig. 8 Result of the nut loosing experiment

2.5 Evaluation of Looseness Prevention by High-Frequency Vibration Experiment

The looseness prevention performance of the nut was evaluated by conducting a high-frequency vibration test[6]. The results of the test are shown in **Fig. 8**. From the relationship that tightening torque is equal to prevailing torque plus contributory torque for axial tension, contributory torque for axial tension is approximately 8 N·m and 17 N·m in Case 1 and Case 2, respectively, where contributory torque for axial tension is low. In these cases, looseness occurred in an early stage because it was possible to obtain an axial tension equivalent only to 20–40% of the yield point of the material of the bolt/nut used in the experiment. In Case 3 and 4, contributory torque for axial tension is approximately 70% as in general bolt/nut tightening, and the looseness prevention effect worked, as a result of the shearing force acting on the bolt due to the eccentricity of the inner nut. Consequently, the tested nut could pass a vibration test of 17 min (30 000 vibrations). From these test results it became apparent that an effective looseness prevention function can be imparted to bolts by an appropriate amount of axial tension and an appropriate amount of eccentricity.

2.6 Example of Application at Steel Works

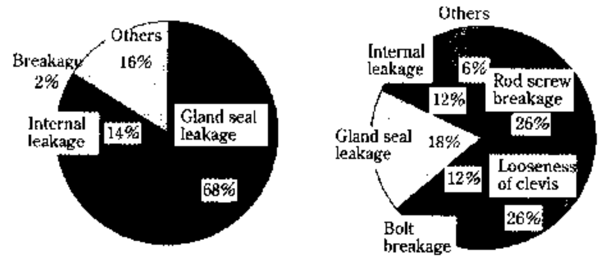
The newly developed looseness-free nut[7] is applied to rail connections and parts where vibration occurs in the steel works. An example of application to the rail of a torpedo car is shown in **Photo 1**.

2.7 Summary

The company developed a new looseness-free nut that has an excellent looseness prevention characteristic and has excellent workability, enabling one-action tightening, thereby contributing to the prevention of trouble in the steel works.

(a) Appearance of a torpedo car (b) Fastening condition

Photo 1 Application example of looseness-free nut



(a) Causes of repair (b) Analysis of trouble

Fig. 9 Damage analysis of hydraulic cylinder

3 Life Prolongation Technology of Cylinders at Kawasaki Steel

3.1 Actual Situation of Cylinder Life

There are many cases where the cylinders used in the environment of steel making plants have a very short life due to accelerated breakage and wear of their components by forced degradation factors such as vibration, impact, heat, and corrosive dust particles. The analysis results of causes of cylinder repairs and trouble at steel works are shown in **Fig. 9**. These short-life cylinders cause equipment stoppages and an increase in the maintenance load and, therefore, life prolongation techniques for cylinders suited to the environment of steel making plants are required.

3.2 Improvement of Strength Balance of Cylinder Components

The most vulnerable part of all cylinder components is the threaded portion on the tip of rod. An example of a strength balance calculation (ϕ125 mm × ϕ71 mm × 14 MPa) is shown in **Fig. 10**. The safety factor for fatigue of the threaded portion obtained when a static full rated load is repeatedly applied is 1.4. However, when surge pressure is considered, this factor decreases to about 1.1, and when bending stress generated by misalignment caused by the wear of an actuated sliding surface (amount of ejection: 200 mm, misalignment: 0.1 mm) and decreases in fatigue strength are taken into consideration, the safety factor further decreases to 0.6. The strength balance of each cylinder component is in agreement with the fact that breakage of the threaded portion of the rods of cylinders used in rolling mills and other equipment occurs frequently.

Furthermore, it is suggested that the safety factor for fatigue of tie-bolts, which is 5.8, decreases to about 0.7, resulting in fatigue breakage, when surge pressure, looseness due to cave of the washer face, vibration, and corrosion are taken into consideration.

Measures to improve the reliability of these components were taken as follows. For the thread on the tip of the rod, the rod and the crevice were integrally formed to increase strength and the material of the rod was changed to stainless steel to prevent corrosion. The gap of the crevice pin was expanded to prevent bending load.

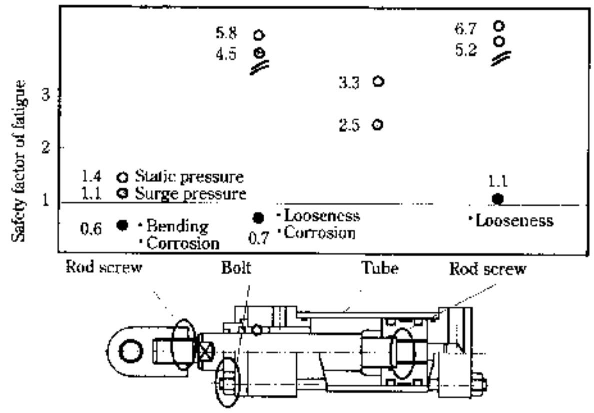


Fig. 10 Calculated result of fatigue strength

For tie-bolts, corrosion protection measures were taken and the looseness-free nut developed by Kawasaki Steel, which was described in Chapter 2, was applied.

3.3 Wear of Cylinder Components and Its Measures

As shown in Fig. 9(a), cylinder life is determined by leakage from the gland seal. In the gland real, sealing is performed by pumping action. Leakage occurs when the actual thickness (hm) of the oil film carried away along with the rod increases due to the wear of the seal and a decline in the contact pressure gradient resulting from a decrease in elasticity[8], as expressed by Eq. (1).

$$\mathrm{hm} = \sqrt{\frac{8\pi\mu U}{9\,|\mathrm{d}p/\mathrm{d}x|_{\max}}} \quad \cdots\cdots (1)$$

hm: Oil film thickness, U: Slide speed,
μ: Viscosity of hydraulic oil,
dp/dx: Contact pressure gradient inlet of hydraulic oil

Seal wear is governed by the wear resistance of the seal material itself, the surface roughness of the rod and debris contained in the oil, which accelerate wear. The relationship between the amount of wear V and the physical properties of rubber is given by Eq. (2)[9]

Table 2 Rod surface treatment to prolong cylinder life

Environment	Surface treatment	Surface hardness (Hv)	Thickness of treatment (mm)
General	Hard chrome plating	800	0.05
Corrosion	Hard chrome carbide plating	900	0.05
Wear corrosion	Tungsten carbide spray coating	1 200	0.05

$$V \propto \mu WL/(H\sigma\varepsilon) \cdots\cdots (2)$$

V: Amount of wear, W: Load, L: Slide distance,
H: Hardness of rubber, μ: Viscosity,
σ: Tensile strength, ε: Fracture elongation

The elasticity of the seal is reduced by oxidative deterioration of the seal material.

In consideration of these relationships, hydrogenated nitrile rubber, which has high strength and excellent oxidation stability, was adopted as the standard seal rubber material and, at the same time, wear due to debris in the oil was prevented by expanding the oil cleaning systems at all works. Furthermore, in order to prevent the deterioration of rod surface roughness, which is a factor responsible for accelerating wear, tests were conducted to compare various methods of rod surface treatment. As listed in **Table 2**, appropriate rod surface treatments were adopted according to environmental conditions.

3.4 Life Prolongation Technology for Large Cylinders

A large number of large high-pressure cylinders, such as the hydraulic screwdown cylinders of rolling mills, are used in steel works. Many such large cylinders are of special construction, and it is necessary to adopt a life prolongation technique suited to the individual cylinder type. The following is an example of life prolongation in which optimization of the packing gland seal is applied to a hydraulic screwdown cylinder of a rolling mill. In the past, U-packing was used in the gland seal of this cylinder. As shown in **Fig. 11**(a), shredding occurred in the heel and cracking was initiated from the bottom of the lip groove. The shredding was caused by an increase in the projection gap (the gap between the rod and the packing case) due to plastic deformation during the pressurization of the packing case, and the cracking was caused by a decrease in the strength of the packing material due to the sliding frictional heat resulting from insufficient lubrication between the packing and the rod. An FEM calculation was performed using the shape and material of the back-up ring and the relief allowance of the packing heel as parameters to obtain the optimum seal shape that produces only a small projection gap due to the deformation of the packing and has a short length of contact with the rod, as shown in Fig. 11(b). Substantial life prolongation has been achieved through the use of this packing.[10]

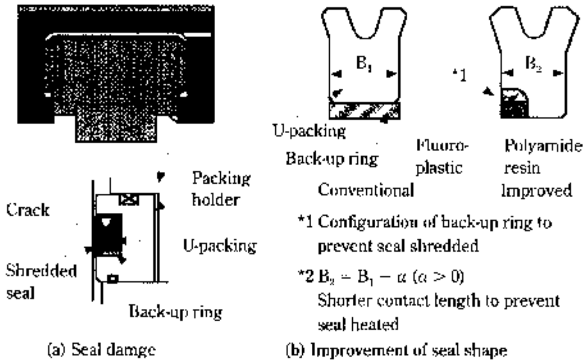


Fig. 11 Damage and improvement of high-pressure cylinder

3.5 Summary

Based on the results of an analysis of trouble in hydraulic cylinders, trouble was reduced to 35% of the former level and life was extended by 1.8 times by increasing the toughness of the rod and crevice and by adopting design techniques suited to each packing component.

4 Development of Long-Life Cross Bearing in Universal-Joint Spindle[11,12]

4.1 Background

Universal-joint spindles are used in many rolling mills as the spindles for main drive units because of their advantages such as small friction energy loss, high transmission efficiency, low noise level, and small vibration. On the other hand, because of the complex structure of the joint portion, there are many cases where the spindle requires high repair costs among the main drive units. In general, the fatigue life of a cross bearing in the joint portion determines the replacement cycle, and life prolongation of the cross bearing is therefore a major problem for reducing universal joint repair costs. The general structure of a typical universal-joint spindle is shown in **Fig. 12**.

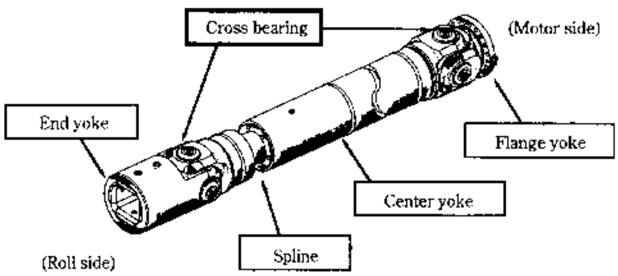


Fig. 12 Structure of universal joint

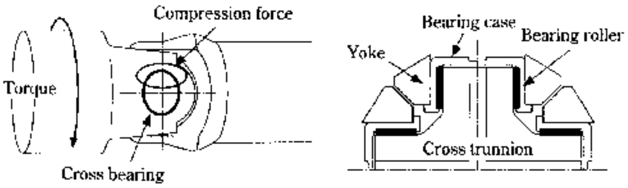


Fig. 13 Section structure of cross bearing

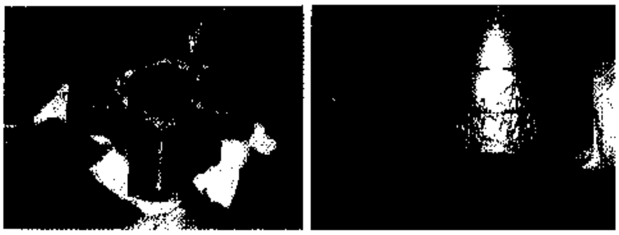


Photo 2 Damage condition of cross trunnion

4.2 Estimation of Causes from Damage Patterns of Cross Bearings

Increasing the bearing capacity based in a review of dimensions is the most effective means of prolonging cross bearing life. However, because the outer dimensions of mill spindles are limited by the diameters of the rolling mill rolls, a large increase in capacity cannot be expected.

Because cross bearings undergo local contact under high surface stresses due to an oscillating motion, they have had the problem that a fatigue separation phenomenon occurs in the surface layer in a short period. A representative example of the structure of a cross bearing is shown in **Fig. 13**, and damage is shown in **Photo 2**. In general, a cross bearing comprises multiple rows of cylindrical rollers (2 to 5 rows). In terms of the power transmission mechanism, the maximum load acts on the roller at the leading end of the cross pin (outermost diameter portion), and fatigue separation of the surface layer of the cross pin occurs. Therefore, it is considered that life prolongation can be achieved by smoothing this eccentric load distribution and reducing the maximum load value per roller.

Attention was given to the stiffness balance between the yoke and the cross pin. It was considered that in the original type, eccentric load tends to affect at the leading end because the stiffness of the cross pin was higher than that of the yoke.

4.3 Development of Low-Stiffness Cross Pin

4.3.1 Analysis model

The simplest method for lowering the stiffness of a cross pin is to change the shape of the lubrication hole through the center of the pin. A finite element method for the elastic body nonlinear contact problem based on a combination of a yoke and cross pin, which is shown in **Fig. 14**, was used in studying the optimization of the hole shape. The bearing rollers were modeled as a two-dimensional elastic body gap. In order to verify the significance of the FEM analysis model, FEM analytical values were compared with values obtained by an experiment in which bearing roller loads were measured. As shown in **Fig. 15**, the FEM analytical values were basically in agreement with the experimental values, confirming the effectiveness of the model.

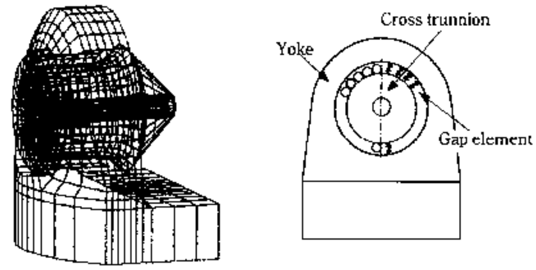


Fig. 14 Finite element model for 3-dimensional analysis

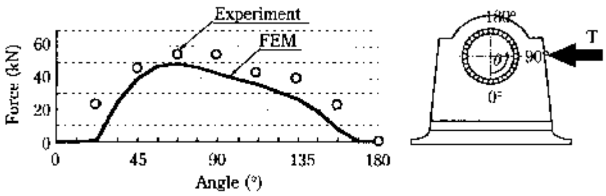


Fig. 15 Comparison of force distribution on the first line roller

4.3.2 Determination of shape

Because the amounts of deflection of the yoke and cross pin are influenced by loading torques, the average torque and maximum torque were used as evaluation torques in this examination. This is because it is considered that the average torque affects the fatigue life of the cross bearing and the maximum torque affects the fatigue strength of the cross fillet. The evaluation items used in the study of the shape were as follows:

(1) The maximum load value per bearing roller under average torque is minimized.
(2) The fatigue strength of the cross fillet under maximum torque is ensured.
(3) The maximum load value per bearing roller under maximum torque must be not more than the maximum load value of the original type.

A taper shape having a hole gradually expanding to the leading end, which is shown in **Fig. 16**(b), was adopted as the hole shape of the low-stiffness cross pin, and the optimum shape was determined using the taper

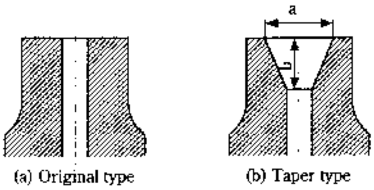


Fig. 16 Analysis case (Constant diameter of taper edge)

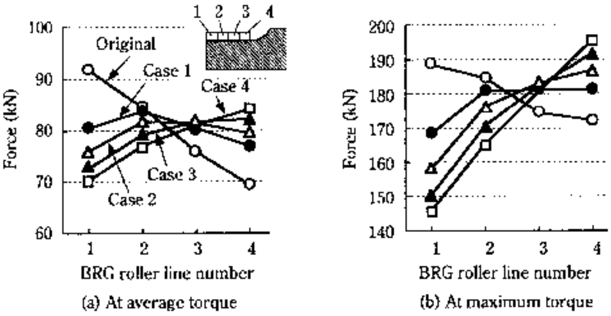


Fig. 17 Distribution of force in cross axial direction at maximum force angle

depth b and the end diameter a as parameters. An analysis was conducted by changing the depth b with the diameter a fixed at 135 mm. The results of the analysis are shown in **Fig. 17**. As is apparent from this figure, the reduction in the bearing roller load under average torque is greatest in Case 2. Also in Case 2, under maximum torque, the maximum load value per bearing roller is not more than the maximum load value of the original type, and this condition is also satisfied.

Similarly, various analysis were carried out using the diameter a as parameter. The results of these analyses showed that the shape of Case 2 is the optimum one in terms of the stiffness balance.

4.3.3 Examination of strength of cross fillet

The results of a stress analysis of the fillet in Case 2 are shown in **Fig. 18**. From the maximum value of generated stress, it could be ascertained that the safety factor for fatigue is Sf = 1.11 and that there is no problem in terms of fatigue strength, although the strength is slightly lower than that of the original type (Sf = 1.17).

4.4 Result of Application to Commercial Production Equipment

Low-stiffness cross pins were applied to the finisher mills (F1 and F2) of the hot strip mill at Mizushima Works, and as a result, the fatigue separation phenomenon was not observed even after use for approximately twice as long as the period in which fatigue separation occurs with conventional cross pins. Thus, the life prolongation effect was ascertained.

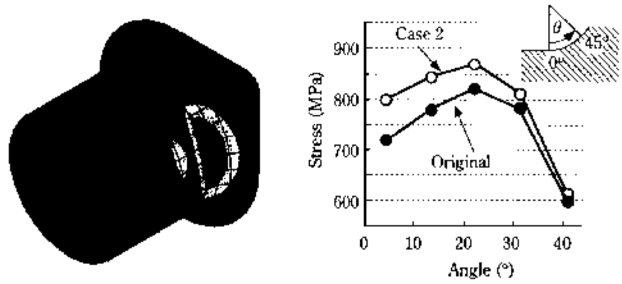


Fig. 18 Distribution of stress in cross fillet (At maximum torque)

5 Conclusion

Development of life prolongation techniques for general-purpose mechanical elements in steel making plants was carried out with the following results were obtained:

(1) A looseness-free nut has been developed to reduce screw looseness, which is the largest cause of bolt/nut trouble at steel works, thus contributing to the prevention of trouble.

(2) Based on the results of an analysis of hydraulic cylinder trouble, the strength and toughness of the rod and crevice were increased and packing design techniques suited to individual parts were adopted. As a result, trouble decreased to 35% of the former level and life was substantially extended by 1.8 times.

(3) For universal joints, a low-stiffness cross pin has been developed. In an example of application to commercial production equipment, life was extended by approximately two times.

The authors would like to thank those concerned at Taiyo Nut Mfg. Co., Ltd. and Nakamura Jico Co., Ltd. for their generous assistance in the development of the looseness-free nut and the universal joint, respectively.

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