Multi-objective optimization design and reliability analysis of idler with hollow step-shaft

Jinhu SU*, Wenjun MENG* and Xiaoxia ZHAO*
*School of Mechatronics Engineering, Taiyuan University of Science and Technology
Waliu 66#, Wanbailin District, Taiyuan 030024, China
E-mail: 490179466@qq.com

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Abstract
Idlers, one of the important parts of belt conveyors, are substantial and constitute approximately one-third of the equipment cost. Therefore, idlers in conveyor systems must be optimized. This study designs a new kind of idler on the basis of theoretical calculation and coupling simulation, with the shaft, shell, and labyrinth gland of the idler as optimization variables and overall cost and life span of the idler as objective functions. According to each independent variable of the idler, this idler adopts the equal strength beam concept and maximizes the overall carrying capacity of the shaft and the bearing by using a hollow step-shaft. The idler effectively adopts the one-piece casting labyrinth gland, thereby automatically enhancing the sealing effect with a screw structure in different rotation directions. The simulation analysis shows that under certain working conditions, the stress of the new idler shaft is one-third of the stress of the traditional idler shaft, and the strain is one-half of the traditional idler shaft, and the effect is obvious. A reliability analysis of the idler system is described in this work through the establishment of a fault tree, determination of the logical relationship of each component and provides methods for further research. Finally, the idler is verified to be practical in engineering applications, with a 10% cost reduction and considerable economic benefits.

Keywords: Hollow step-shaft, Idler, Labyrinth gland, Multiple objective optimization, Reliability analysis

1. Introduction

It is necessary to optimize idler in the conveyor system. The idler, as an important part of the belt conveyor, is large in number and constitutes about one third of the cost of a conveyor.

The book by Andrzej et al. (2016) conducted experiments on roller sealing, bearing load, and bearing temperature in detail with an infrared camera and pointed out that improvement in sealing can considerably diminish bearing temperature and thus improve bearing life.

The influence of dust particles on vibration characteristics, surface temperature, and bearing life is obtained (Radivoje et al., 2016). The book by Furlan et al. (2009) presented a way to diminish dynamic load and increase bearing life by adding a cushion between the bearing and the shaft. The authors concluded that the greater the belt stiffness, the smaller the viscosity and the damping; the smaller the dynamic load and bearing vibration, the longer the bearing life span will be based on a simplified oscillatory differential equation.

The book CEMA (2005) presents that idler life is determined by various factors, such as labyrinth gland, bearing, idler shell thickness, belt speed, material density, and environmental temperature. Although idler life is often symbolized by bearing life, other variables (such as sealing effect) are also highly important to idler life. The book by Liu et al. (2018) made experimental research on the condition monitoring of belt conveyor idlers and pointed out that temperature measurement at roll shafts is a straightforward and effective approach to the condition monitoring of belt conveyor idlers. A developed methodology for testing idlers in detail is elaborated (Andrzej, 2013).

These studies mentioned above described idlers from the perspectives of the reasons for idler damage; distribution of external load on idlers; relationship between seal, vibration, and life; and life detection method, but they did not conduct in-depth research on the idler structure. In this study, the concept of beam of constant strength, combined with the existing bearing inner ring size, market price, and comprehensive consideration of shafts and bearings, is designed. This study aims to design a self-sealing device on the basis of the property of roller rolling resistance. This method theoretically realizes a light sealing device that is not loose or deformed. Then, a finite element analysis of idlers is conducted to compare the theoretical calculation results.
In view of previous research, this study designs a new kind of idler and performs a reliability analysis with the idle shaft, shell, and labyrinth gland as optimization variables and overall cost of idler and life as objective functions.

2. Proposed idler design

Despite the great variety in idlers, the structure of traditional idler is the same, comprising a solid flux shaft, deep groove ball bearings, bearing block, spring, and labyrinth gland. While the structure of new idler is based on different principles. The structure is compared as shown as shown in Fig. 1.

(a) Traditional idler structure. The outside of the bearing is a spring with labyrinth gland and generally made of nylon on the outside. A card slot directly cut on the shaft is present outside of the labyrinth gland in which the idler is positioned. When the system is run, the shaft, spring, outer ring of the labyrinth gland, and the bearing ring are kept still.

(b) Structure of new idler. This idler adopts the concept of equal strength beam and maximizes the overall carrying capacity of the shaft and bearing by using a hollow step-shaft. The bearing is fixed at the end of the idler by necking technology in place of traditional welding. The idler effectively adopts the one-piece casting labyrinth gland, thereby automatically enhancing the sealing effect with screw structure in different rotation directions.

The current major problems in traditional idlers are as follows.

1) Problem in load capacity difference between the shaft and the bearing. The model number is often increased to match with the others because the load capacity is determined by the small shaft or bearing, thereby leading to increased cost.

2) Sealing problem. The main cause of bearing damage is the wearing out of bearings due to dust, according to numerous surveys. The two main reasons for this occurrence are as follows. (1) As an idler runs at high speed, heat is generated with internal friction, and the grease becomes diluted and even evaporates, thereby leading to increased seal cavity pressure. In addition, when the idler stops running, bearing seal cavity pressure drops with a decrease in temperature, thereby resulting in a suction phenomenon with dust particles into the bearing seal cavity and greatly shortening the idler life span. (2) The sealing device deforms or loosens after running for a long period.

The new idler, compared with the traditional one, is mainly improved from the following aspects.

1) The main shaft of the new idler with a hollow step in place of the original solid shaft maximizes the comprehensive bearing capacity of the shaft and bearing while keeping weight as light as possible.

2) The position of the end of the idler shell and bearing seat is directly rolled and pressed to replace the all-welding method, which is time consuming and tends to produce stress concentration and belt scratch. The new technology simplifies the process, diminishes cost, and prevents the idler side from scratching the belt.

3) This one-piece labyrinth gland has the slot, labyrinth gland, and spring in one. The labyrinth gland is a regular hexagon in shape and affixed on the idler frame. Both ends of the idler shaft have screw threads that are positioned opposite each other and tighten in orientation when the conveyor belt is running. The method can effectively diminish the amount of dust entering the bearing and improve idler life compared with the previous structure, which has a card spring.
3. Force analysis of idler

As the conveyor runs, the conveyor belt bypasses the idler while the idler supports and hinders the conveyor belt. The conveyor belt exhibits resistance in the form of compressive stress to the rotating idler with constantly changing rotation resistance. Meanwhile, the belt and idler vibrate when the conveyor belt runs at a specific speed and tension.

The force of a single idler is shown in Fig. 2.

Fig. 2 The conveyor belt runs to the right, where G is the total weight of the material and conveyor belt, F is the tension of the conveyor belt on the idler, \( F_2 \) is the resistance of the bracket to the idler roller, and M is the rotation resistance of the bracket to the idler.

According to the fourth strength theory, resultant stress is

\[
\sigma_{\text{max}} = \sqrt{\sigma^2 + 3\tau^2}.
\]

(1)

The book by Paul and Craig (2016) studied resistance with different test parameters and applied the test data in the conveyor design to improve the accuracy of belt tension calculation. The wing conveyor idlers’ axial loads using the finite element method are examined (Miskovic et al., 2018). The rolling resistance calculation by comparing rubber stress relaxation models is studied (Munzenberger et al., 2018). The load on belt conveyor idlers operated under real-life conditions is measured (Lech et al., 2019). Resistance from the relaxation behavior of belt insulation is studied (Jayne and Craig, 2016). A technique of measuring the idler’s spin resistance is proposed (Robert et al., 2015). The vibration frequency of a conveyor belt is presented (Ronghe et al., 2006). During the operation of a conveyor belt, its vibration must be far from the resonance region (k. et al., 2006).

The book CEMA (2005) presents the idler misalignment load caused by idler height deviation and belt tension.

\[
P_{\text{add}} = \frac{HT}{a_0} \times 0.035,
\]

(2)

where H is height deviation (m), T is belt tension (N), and \( a_0 \) is idler spacing (m).

Based on the above research, the force applied to the idler can be divided into two (i.e., vertical and horizontal) regardless of the way idler resistance is calculated. Only the vertical force is considered in this study because idler resistance is smaller than the vertical force.

Several researchers have contributed to the analysis of belt deflection. Grimme and Berumer considered that the real situation and position of the material are obscured. In theory, two possibilities are considered. (1) The material stays at the center of the conveyor belt while the conveyor belt runs. (2) The material completely follows the conveyor belt. Under extreme conditions, when the material is kept at the center of the conveyor, the carrying idler is in the ideal load. When materials completely follow the belt deflection, conveyor belt deflection is generally not more than 10% of the belt width.

Since the running deviation of the belt does not influence the distribution of pressure on the cross section of the conveyor. Thus, the impact of the conveyor belt operation can be ignored in idler analysis.

When the conveyor belt is not deviating, and adjacent idlers are kept at the same height, and force analysis of the idler can be divided into two according to the idler location.
1) Parallel idler for returning and middle parallel idler for carrying (Fig. 3)

![Diagram](image)

Fig. 3 The force of the beam element in a subsection is analyzed. Then, the superposition method in material mechanics is used to perform the overall analysis. A and B are the supporting points (i.e., the position of the idler card slot), O is the symmetrical center, C is the connecting position of the fine and thick diameter shafts, and $\frac{P}{2}$ is the bearing center.

The shaft is calculated by symmetry and superposition as follows.

The idler is divided into two parts by symmetry, and the left part is regarded as a cantilever beam. Then, the effect of the left small diameter shaft (AC), which is considered a cantilever beam affixed in section C and acting on the deflection angle $\theta$ of the force position, is analyzed.

In Figure 3, based on the superposition method, using the cantilever beam principle, the overlying of the rotation angle $\theta_1$ produced by the idler shaft can be described by the following formula (Fig. 1).

$$\theta_1 = \frac{P_m(L-2n)I_1}{4EI_1} + \frac{2P_m(n-m)}{4EI_1}$$  \hspace{1cm} (3)

In the same way, the rotation angle $\theta_2$ produced by the idler shell can be described.

$$\theta_2 = \frac{P_m(L-m)}{4EI_3}$$  \hspace{1cm} (4)

The deflection angle is generally less than 0.1' under normal load conditions with great rigidity. Therefore, the influence of the deformation of the bearing seat is ignored in the calculation.

$$\theta = \theta_1 - \theta_2 = \frac{P_m(L-2n)I_1}{4EI_1} + \frac{2P_m(n-m)}{4EI_1} - \frac{P_m(L-m)}{4EI_3}$$  \hspace{1cm} (5)

$$\rightarrow P = \frac{2ml_3(n-m)-ml_1(L-m)+m(L-2n)}{2ml_3(n-m)-ml_1(L-m)+m(L-2n)} \frac{I_1}{I_2}$$  \hspace{1cm} (6)

where

- $P$ = idler load, $P = P_{nomal} + P_{add}$ (kg);
- $E$ = elastic modulus of material (kg/cm$^2$);
- $I_1$ = moment of inertia of fine diameter shaft (cm$^4$), $I_1 = \frac{(1-d_1^4)\pi d_1^4}{64}$;
- $I_2$ = moment of inertia of thick diameter shaft (cm$^4$), $I_2 = \frac{(1-d_2^4)\pi d_2^4}{64}$. 

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\[ I_3 = \text{moment of inertia of idler shell (cm}^4), \quad I_3 = \left(1 - \frac{d_5}{d_6}\right)^4 \frac{\pi d_6^4}{64}; \]

\[ \theta = \text{rotation angle in middle of bearing (')}; \]

\[ m = \text{length from support point to bearing center (cm)}; \]

\[ L = \text{length of shaft (cm)}; \]

\[ n = \text{length from support point to end of thick diameter shaft (cm)}; \]

\[ c = \text{distance from center of bearing to end of idler shell (mm)}; \]

\[ a = \text{length from support point to end of idler shell (mm)}; \]

\[ b = \text{thread width (mm)}; \]

\[ d_1 = \text{inside diameter of small diameter shaft (mm)}; \]

\[ d_2 = \text{outside diameter of small diameter shaft (mm)}; \]

\[ d_3 = \text{inside diameter of major diameter shaft (mm)}; \]

\[ d_4 = \text{outside diameter of major diameter shaft (mm)}; \]

\[ d_5 = \text{outer diameter of idler shell (mm)}; \]

\[ d_6 = \text{inner diameter of idler shell (mm)}. \]

2) Two outer idlers for carrying trough idlers

The load on a single idler is generally designed to 63% of the total load of carrying trough idlers in current engineering. Specifically, the force is a uniformly distributed load and generally accepted, thereby neglecting axial force and load unevenness. This condition is due to the nonlinear load from one end of the idler to the other and the absence of a unified formula.

The book DIN2123 (2012) expresses the normal force distribution on a conveyor belt. The book by K. Grabner et al. (1991) studied the influence of bulk material properties, belt tension and characteristics, and idler configuration on the stress profile. The bulk solid interactions on a conveyor belt are obtained (Dusan and Craig, 2013, 2017). The relationship between tension and idler resistance by a mathematical method was analyzed (Vieroslav et al., 2014, 2015, 2016) and showed that the force in the contact position is closely related to the conveyor belt tension, material property, conveyor belt, and hatch areas of the material; they offered basic research for the latter design of belt conveyors. The book by Liu et al. (2015, 2016) proposed a pressure distribution model of bulk material on the cross section of the trough conveyor with a discontinuity method and verified its accuracy, as shown in Fig. 4. This model points out that the percentage of the normal force on the left wing belt section varies from 16.8% to 23%, and that at the center belt section changes from 74.2% to 66.2%.

![Fig. 4 (A) Areas on either side of a stress discontinuity with (B) Mohr circles.](image)

Subpanel A presents stress discontinuity that separates two bodies of bulk material under different stress states. Each body consists of numerous infinitesimal bulk elements. In subpanel B, Mohr circles are created to separately describe the stress states of bulk elements in bodies 1 and 2 (circles C1 and C2, respectively). With the two intersected Mohr circles, the relationship of the effective stresses across the stress discontinuity can be retrieved. Moreover, the rotation of direction of the major principal stress can also be obtained.
Where
\[ \theta = \text{rotations of the major principal stress} \],
\[ \delta = \text{strength mobilized on the stress discontinuity} \],
\[ \sigma_{oc1}, \sigma_{oc2} = \text{average effective stresses in active stress state (N/m}^2) \],
\[ R_1, R_2 = \text{radiuses of Mohr circles} \],
\[ \varphi_i = \text{internal friction angle of bulk material} \],
\[ \Delta = \text{angles in Mohr circle} \],
\[ \sigma_1, \sigma_2 = \text{normal stress and shear stress (MPa)} \],
\[ \sigma_2 = \text{major principal stress (MPa)} \].

The model has been repeatedly verified in engineering experiments. Therefore, in engineering, the two sides of an idler are designed to be equal to the middle idler in specification, which has been proven to be efficient.

4. Simulation analysis

According to the abovementioned algorithm, the new idler’s shaft is compared with the traditional one by simulation analysis. The maximum permissible angle in the bearing of the shaft is 12’. The working conditions are as follows: Quality = 6000 t/h, Belt widths = 1400 mm, and Velocity = 3.15 m/s; material: iron ore.

![FE model of traditional idler](image)
![FE model of new idler](image)

Fig. 5 The A represents gravity, B represents the load on idlers \( P = q/3.6v \), and C represents the constraint conditions, i.e. the supporting position of idlers.

![Deformation of traditional idler](image)
![Deformation of new idler](image)

(a) Deformation of traditional idler                         (b) Deformation of new idler

![Stress of traditional idler](image)
![Stress of new idler](image)

(c) Stress of traditional idler                         (d) Stress of new idler

Fig. 6 (a) shows the idler and shaft diameters are 133 and 30 mm, respectively. The bearing specification is 6306, the maximum deformation is 0.08 mm, and the deformation at the bearing position is 0.036~0.044 mm. Fig. 6 (b) shows the diameter of the idler is 133 mm. The internal and external diameters of the coarse shaft are 30 and 40 mm, respectively, and the internal and external diameters of the fine shaft are 20 and 30 mm, respectively. Fig. 6 (c) shows the maximum stress is 127 MPa, and the stress concentration occurs at the clamping spring. Fig. 6 (d) shows the maximum stress is 31 MPa, and the stress distribution is relatively uniform.

The maximum displacement allowed at the bearing is 0.16 mm at the maximum allowable corner of 12’. The new
and traditional idlers can meet the deformation and stress requirements. However, the new type of idler has less deformation and stress. In addition, the weight of the hollow step-shaft is 2.16 kg, whereas that of the traditional shaft is 3.14 kg.

The economic reasons for this technique are also presented. In addition, differences in the price of idlers mainly lie in the shaft and the bearing. The shaft can be sold at $1300/t on the market. Table 1 shows a comparison of the idlers in Yantai port.

Table 1 Comparison between the traditional and the new idlers under the same conditions.

| Working condition | Q = 10,000 t/h, B = 1600 mm, V = 3.75 m/s, material: iron ore |
|-------------------|---------------------------------------------------------------|
|                   | Bearing no. | D (shaft) | Lifetime | Cost variance |
| Traditional idler | 6308        | 40 mm (solid) | 31*10³ h |              |
| New idler         | 30,307      | 35 mm (hollow) | 100*10³ h | Save $8       |

According to Figs. 6 and Tables 1, the new idler can diminish the cost by approximately 25% compared with the conventional roller under the same working conditions. In addition, the overall conveyor system cost can be diminished by 10%.

5. Reliability analysis of the new idler

Few studies on the reliability of the conveyor are available due to difficulty of testing and statistics. Tewaril et al. (1991) studied the reliability of a conveyor system. Amid et al. (2015) examined the reliability of an armored face conveyor, beam stage loader, and conveyer belt is examined.

The fault tree model of the idler is established as shown in Fig. 7.

![Fault Tree Model](image)

Fig. 7 The relationship between \(X_1, X_2, X_3\) and A expresses several common causes of failure in the process of design and installation of labyrinth gland. The relationship between \(X_9, X_{10}\) and E expresses two common causes of failure due to lubricant problems. The relationship between \(X_4, X_5, X_6\) and C expresses several common causes of failure in bearings. The relationship between \(X_7, X_8\) and D expresses two common causes of failure of the shaft in the process of design and installation.

The idler is divided into several parts, and fault diagnosis of each part is conducted.

Where

\[T = \text{idler failure},\]
$A$ = labyrinth gland failure,
$B$ = shaft failure,
$C$ = bearing failure,
$D$ = idler shell failure,
$E$ = grease failure,
$X_1$ = nonstandard installation,
$X_2$ = unreasonable design of sealing arrangement,
$X_3$ = fixed installation loosening,
$X_4$ = nonstandard bearing installation,
$X_5$ = bearing installation loosening,
$X_6$ = dusty entering through sealing arrangement,
$X_7$ = inappropriate idler arrangement,
$X_8$ = excessive tension on idler,
$X_9$ = irrelevant selection of grease,
$X_{10}$ = abnormal evaporation of grease.

The dependability system of the idler is a typical tandem.

$$R = \prod_{i=1}^{n} R_i = R_1 \ast R_2 \ast R_3 \ast R_4$$  \hspace{2cm} (7)

where $R_1$, $R_2$, $R_3$, and $R_4$ are reliabilities of the bearing, shaft, idler shell, and labyrinth gland, respectively.

Subsequently, $R_1$, $R_2$, $R_3$, and $R_4$ are separately described.

$R_1$ is designed according to GB/T6391. The fatigue life of the bearing obeys the Weibull distribution.

$$t = \eta [-\ln R(t)]^{\frac{1}{\beta}} = L_{10} \left[ \frac{1}{\ln(0.9)} \right]^{\frac{1}{\beta}}$$  \hspace{2cm} (8)

$$L_{10} = \frac{10^6}{60n} L_{105}$$

$$L_{10} = \left( \frac{C}{P} \right) \epsilon$$

$$\rightarrow R(t) = e^{\left( \frac{tP}{\epsilon C} \right)^{\beta}} + 0.9$$  \hspace{2cm} (9)

Where

$L_{10}$ = basic rating life (million transfer),
$C$ = basic static load rating,
$P$ = equivalent Dynamic Load,
$\epsilon$ = life exponent,
$L_{105}$ = basic rating life (h),
$n$ = Operating Speed ($\text{r} / \text{min}$),
$\eta$ = Size parameters,
$\beta$ = shape parameters,
$t$ = bearing life

$R_2$:

Through the analysis of idlers, the main failure mode of the shaft comes from excessive bending of the shaft, and the strength of the shaft fully meets the operational requirements, that is, the strength reliability of the shaft is equal to 1. Therefore, $R_2$ is calculated on mandrels’ rigidity as a simply supported beam to establish the following flexure line equation:

$$y = f(F, I, m, E, I, x)$$  \hspace{2cm} (10)

Force $P$, span $l$, and position $a$ are taken as random variables in the design of the axial diameter.
Since \( y' = \theta \) \( y'(0) = 0 \), the formula (5) is integrated

\[
y = \frac{P m^2 (I_2 - I_1)}{4 E I_1 I_2} + \frac{P m^3 (I_1 - 2I_2)}{12 E I_1 I_2}
\]

(11)

Based on formula (13), mean of deflection \( y \) is

\[
\bar{y} = f (\bar{F}, \bar{I}, m, \bar{E}, \bar{I}, \bar{x}) = y
\]

(12)

The standard deviation is

\[
S_y = \left[ \left( \frac{\partial y}{\partial F} \right)^2 S_F^2 + \left( \frac{\partial y}{\partial I} \right)^2 S_I^2 + \left( \frac{\partial y}{\partial m} \right)^2 S_m^2 + \left( \frac{\partial y}{\partial E} \right)^2 S_E^2 + \left( \frac{\partial y}{\partial I} \right)^2 S_I^2 + \left( \frac{\partial y}{\partial x} \right)^2 S_X^2 \right]^{1/2}
\]

(13)

Since each random variable in equation (12) is basically a normal distribution, the deflection \( y \) is also approximated to a normal random variable. Thus, if the allowable value of the deflection \([y]\), the mean \( \bar{y} \) and the standard deviation \( S_y \) are known, the coupling equation can be used to determine the reliability of the stiffness of the shaft under given conditions,

\[
Z_\text{R} = \frac{[y] - \bar{y}}{S_y} = \frac{[y] - \bar{y}}{S_y}
\]

(14)

where

\( P(\bar{F}, \sigma_\alpha) \) = average value and standard deviation of loading force,

\( l(\bar{l}, \sigma_l) \) = average value and standard deviation of beam span,

\( \bar{a}(\bar{a}, \sigma_a) \) = average value and standard deviation of position,

\( E \) = elasticity modulus,

\( I \) = moment of inertia of the shaft,

\( x \) = the distance from the maximum bending moment to the support point,

Reliability is highly prone to fatigue failure at the concentrated force section, which is a dangerous section. The concentrated cross-section is adopted for the calculation of the cross-section.

Through the query, \( F \) can be got according to the \( Z_\text{R} \), and then \( R_2 \) can be calculated according

\[
R_2 = 1 - F
\]

(13)

\( R_3 \):

Like the shaft, the strength of the idler shell fully meets the requirement, the strength reliability is approximately equal to 1, and \( R_3 \) is calculated with stiffness. In designing the idler shell as a mandrel, the impact of the torque is neglected (with the torque much smaller than the bending moment), and the stress approximates the normal distribution. As the shell runs, the size and direction of load \( F \) are constant, the stress of the shell symmetrically changes, and the cycle characteristic \( r = -1 \). Therefore, \( R_2 \) is calculated on mandrels’ rigidity as a simply supported beam to establish the following flexure line equation (10):

\[
y = f (F, I, E, I, x)
\]

Deriving \( y \) using a method similar to equation (5) to derive equation (13):

\[
y = \frac{-5P_1^3}{384EI_1}
\]

(13)

Similar to formula (12), (13), mean of deflection \( y \) is

\[
\bar{y} = f (\bar{F}, \bar{I}, \bar{E}, \bar{I}, \bar{x}) = y
\]

(14)
The standard deviation is

$$S_y = \left[ \left( \frac{\partial y}{\partial F} \right)^2 S_F^2 + \left( \frac{\partial y}{\partial I} \right)^2 S_I^2 + \left( \frac{\partial y}{\partial \varepsilon} \right)^2 S_\varepsilon^2 + \left( \frac{\partial y}{\partial \varepsilon} \right)^2 S_\varepsilon^2 \right]^{\frac{1}{2}}$$

(15)

The rest of $R_3$’s calculations and lookup tables are consistent with $R_2$, no more details are made here.

Axial preload $P$ is an important index of the idler’s sealing performance. The second-order response surface model of the labyrinth gland is established on the basis of the radial size $D$, tolerance $\varphi$, inner and outer rings of the sealing device, and frictional coefficient $f$ as variables by taking $P$ as the response. Then, reliability optimization analysis is conducted on the basis of the non-probability method.

Some of the parameters require a large amount of experimentation, the detailed calculation of $R_4$ is not performed. Since the sealing device is closely matched with the bearing and guarantees the bearing life, the reliability of the bearing is taken as its reliability.

The operating conditions as shown in Table 1 were selected and substituted into formula (11)-(19), respectively. Then, the reliability $R_1$, $R_2$, $R_3$, and $R_4$ of the traditional and new idlers are separately calculated and substituted into formula (9) to obtain the total reliability ($R$). The two total reliability($R$) are compared as shown in Table 2.

### Table 2  Reliability comparison between the traditional and the new idlers under the same conditions.

| Working condition | $Q = 10,000$ t/h, $B = 1600$ mm, $V = 3.75$ m/s, material: iron ore | Idler diameter = 159mm, Length of the roller = 600mm |
|-------------------|---------------------------------------------------------------|---------------------------------------------------|
| $R_1$ (bearing)   | $R_2$ (shaft)       | $R_3$ (idler shell) | $R_4$ (labyrinth gland) | $R$       |
| Traditional idler| 0.9001             | 0.992               | 0.998                 | 0.9001    | 0.802    |
| New idler         | 0.9002             | 0.994               | 0.998                 | 0.9002    | 0.803    |

It can be seen from Table 2 that under a certain working condition, the reliability of the traditional and new idlers is almost the same, but in Table 1, it can be seen that the cost of the new idler is reduced by 10% compared with the traditional idler. That is, in the case of almost the same life and reliability, the new idler is 10% lower than the traditional roller, which is significant in engineering.

### 6. Conclusion

This study designs a new kind of idler on the basis of theoretical calculation and coupling simulation with the shaft, shell, and labyrinth gland of the idler as optimization variables and overall cost and life span of the roller as objective functions. Moreover, this study describes the reliability analysis of the roller system in detail through establishing a fault tree, determining the logical relationship of each component, and conducting a reliability analysis of each component and provides methods for further research. The calculation shows that the reliability of the new idler is approximately equal to that of traditional idlers. The idler is proven to be practical in engineering applications, with a 10% cost reduction and substantial economic benefits. At the same time, some problems remain in the calculation, such as more accurate calculation and simulation of the reliability of sealing device, and comparative calculation of the idler life in the case of shock load.

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