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Improving Rotor Hole Shape to Reduce Force in Cold Pressing Rotor into Crankshaft

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ABSTRACT

Assembling rotor with crankshaft of a hermetic reciprocating compressor is, normally, heating of the rotor and then putting it into crankshaft. This assembly method allows larger magnitude of interference between rotor and crankshaft.

Tolerance band of crankshaft could be relaxed and facilitate the process, however, the process requires heating equipment and higher energy consumption, obviously, this will increase the manufacturing cost of the compressor. In this situation, using cold pressing rotor into crankshaft can avoid the disadvantage mentioned above effectively, moreover, easily mount the rotor at the accurate location. Using of the original design of the rotor and crankshaft cold press will lead large force which may damage the crankshaft. This article presents an analysis about a geometric proposal for the rotor hole that guarantees to withstand sufficient torque, while cold press force is under the crankshaft yield strength.

1. INTRODUCTION

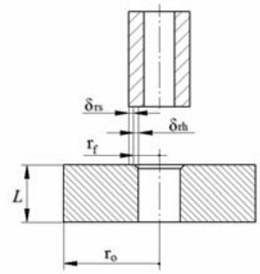
Nowadays, assembling rotor with crankshaft of a hermetic reciprocating compressor is, normally, heating of the rotor and then putting it into crankshaft. This heating process is simple and easily to operate. This assembly method allows larger magnitude of interference between rotor and crankshaft. Tolerance band of crankshaft could be relaxed and facilitate the process, however, the process requires heating equipment and higher energy consumption, obviously, this will increase the manufacturing cost of the compressor(Zhang Siwen, 2002). And after this process, we also require cooling equipment to cool the compressor pump, then worker could measure the gap between rotor and stator, this will cost some time and reduce production efficiency. Another disadvantage is that when the heating temperature is too high will result in some aluminum melt, when the heating temperature is too low it will cause the rotor can't reach the relative position with crankshaft. Those will cost a lot of repair time. Producing inverter compressor is another important reason to cold press rotor into crankshaft. When assembling rotor with crankshaft in inverter compressor by heating process, the magnetic steel will be demagnetized partially, and even worse, some magnetic steel will burst. So whether produce the inverter compressor or reduce energy consumption, we must implement that rotor was safely and quickly cold pressed into crankshaft.

When we don't change the original design of rotor and crankshaft, cold press rotor will cause excessive force. The stress of crankshaft will exceeds the elastic limit of material and deformation of crankshaft will not recover. Finally, the compressor will suffer quality problems. We did some experiments that under the limit load condition the compressors which apply cold pressing rotor into crankshaft process were broken after running for several hours, then we opened shell and found that the crankshafts were broken.

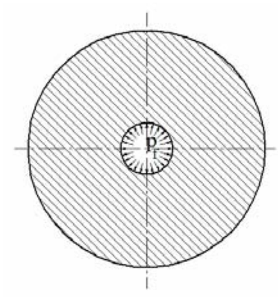
The dimensions and tolerances of the crankshaft is relatively fixed and it's processing technology becomes more and more mature. So changing the shape of the rotor hole is a realistic solution to reduce cold pressing force, and make sure the stress of crankshaft is under the elastic limit of material eliminate compressor quality problems.

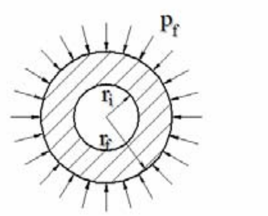
2. INTERFERENCE ASSEMBLE THEORY

The interference fit between rotor and crankshaft is contact problem，it's boundary conditions is highly nonlinear. There are complex contact state and stress state. Calculation interference fit usually based on Lame equation(Zhang Song et al., 2004).



(a)





(b) (c)

Figure 1. (a) Assembling hollow shaft and hub (b) Hub showing interference pressure

(c) Hollow shaft showing interference pressure

The hub displacement is δ_{rh} .For internally pressurized, thick-walled cylinders, the radial stress of hub is σ_{rh} , the circumferential stress of hub is $\sigma_{\theta h}$ (Bernard Hamrock, 1999).

$$\delta_{rh} = \frac{r_f}{E_h}(\sigma_{\theta h} - \nu_h \sigma_{rh}) \quad (1)$$

Where E_h is modulus of elasticity of hub material , ν_h is Poisson's ratio of hub material.

$$\sigma_{rh} = -p_f \quad (2)$$

$$\sigma_{\theta h} = \frac{p_f(r_o^2 + r_f^2)}{r_o^2 - r_f^2} \quad (3)$$

Substituting Eqs.(2) and (3) into Eq.(1) gives

$$\delta_{rh} = \frac{r_f p_f}{E_h}(\frac{r_o^2 + r_f^2}{r_o^2 - r_f^2} + \nu_h) \quad (4)$$

The shaft displacement is δ_{rs} .For externally pressurized, thick-walled cylinders, the radial stress of shaft is σ_{rs} , the circumferential stress of shaft is $\sigma_{\theta s}$.

$$\delta_{rs} = \frac{r_f}{E_s}(\sigma_{\theta s} - \nu_s \sigma_{rs}) \quad (5)$$

Where E_s is modulus of elasticity of shaft material , ν_s is Poisson's ratio of shaft material.

$$\sigma_{rs} = -p_f \quad (6)$$

$$\sigma_{\theta s} = -\frac{p_f(r_f^2 + r_i^2)}{r_f^2 - r_i^2} \quad (7)$$

Substituting Eqs.(6) and (7) into Eq.(5) gives

$$\delta_{rs} = -\frac{r_f p_f}{E_s}(\frac{r_f^2 + r_i^2}{r_f^2 - r_i^2} - \nu_s) \quad (8)$$

The total radial displacement is shown in Figure 1 (a). Recall that outward deflection (expanding the inside diameter of the hub) is positive in sign and inward deflection (reducing the outside diameter of shaft) is negative. Thus the total radial interference δ_r is

$$\delta_r = \delta_{rh} - \delta_{rs} = r_f p_f [\frac{r_o^2 + r_f^2}{E_h(r_o^2 - r_f^2)} + \frac{\nu_h}{E_h} + \frac{r_f^2 + r_i^2}{E_s(r_f^2 - r_i^2)} - \frac{\nu_s}{E_s}] \quad (9)$$

The interference pressure p_f is

$$p_f = \frac{\delta_r}{r_f[\frac{r_o^2 + r_f^2}{E_h(r_o^2 - r_f^2)} + \frac{\nu_h}{E_h} + \frac{r_f^2 + r_i^2}{E_s(r_f^2 - r_i^2)} - \frac{\nu_s}{E_s}]} \quad (10)$$

The press force F_f is shown in Eq.11. μ is coefficient of friction at interference fit.

$$F_f = 2p_f \pi r_f L\mu \quad (11)$$

The torque T which can transmit shaft is

$$T = 2p_f \pi r_f^2 L\mu \quad (12)$$

3. THEORETICAL CALCULATION, SIMULATION, EXPERIMENT

We selected three couple of crankshafts and rotors, their diameters were measured by equipment. The dimensions of the assembly and material properties are listed in Table 1. The rotor is made of steel, and the crankshaft is made of cast iron. When the rotor was cold pressed into crankshaft without lubrication, coefficient of friction at interference fit is usually 0.3. Substituting the measured diameter of the rotor and the crankshaft into equation from(10)-(11), the press force can be calculated. Respectively, three rotors were cold pressed into crankshafts while the biggest press force value was recorded.

Table 1. Dimensions and material properties

Part	Outside diameter of rotor(mm)	Diameter of crankshaft (mm)	Interference length(mm)	Modulus of elasticity (GPa)	Poisson's ratio
rotor	55		15.75	110	0.3
crankshaft		12		200	0.3

In the three-dimensional software, we create assembly model about crankshaft and rotor, calculate the press force by the finite element software contact nonlinear iterative algorithm. The amount of interference is obtained by offsetting the contact surface.

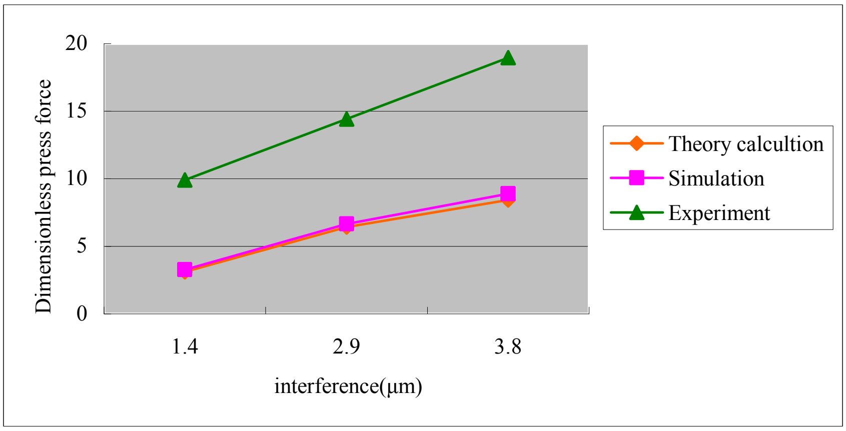


Figure 2. Press force comparison

The press force obtained by theoretical equation, simulation, experiment at the different amount of interference was shown in Figure 2. The press force divided by 500N is dimensionless press force. As we can see from Figure 2, the theoretical results are consistent with simulation results with less than 5% error, however, which is much smaller than the experimental values. When interference is 2.9μm and 3.8μm, the experimental measurements press force is about 2.1 times of the value of simulation.

The main reason of large difference between experimental measurement and simulation is rotor produce process. The rotor core is made of 0.5mm thick silicon steel laminations stacked together, and then form the integral cast. So the cylindricity and roughness become worse. While using simulation, we can't consider the real shape of rotor hole, just use the ideal cylinder instead.

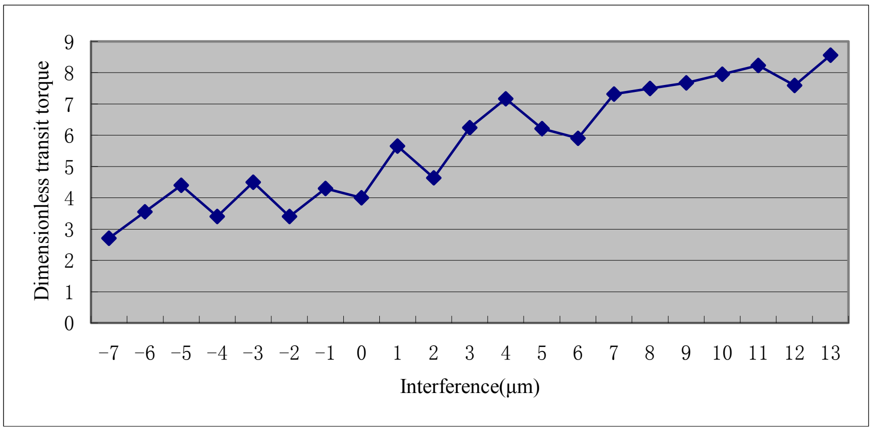


Figure 3. Transit torque with interference

The relationship between transit torque and interference is shown in Figure 3, which is measured a number of rotors and crankshaft. The transit torque divided by 5Nm is dimensionless transit torque. As we can see from Figure 3, some interference values were negative that means there is a gap at one position of the rotor hole. But, the cylindricity of the rotor hole is so big that the transit torque is still about 3(dimensionless) to 5(dimensionless).

In finite element analysis, the press force value from experiment is applied to the crankshaft, the result showed that the stress on the crankshaft has exceeded the elastic limit of the material, some deformation of crankshaft couldn't recover forever. Therefore, we need to improve the design of the shape of rotor hole, but also ensure the reliability of connect between rotor and crankshaft.

4. IMPROVE THE ROTOR HOLE

We redesign the rotor hole, Figure 4 is original shape of the rotor hole, Figure 5 is improved shape of the rotor hole. Before modification, we measured the range of press force and found that the max press force is about twice of limit force which crankshaft can bear. So we want to reduce the half contact area. The shape of the rotor hole is cylinder, in contrast with that the new shape of rotor hole is made of three equal slots. The radius of slot is 1mm lager than original rotor hole. At the sharp edges, we use 0.5mm radius chamfer to avoid stress concentration. The interference length between rotor and crankshaft is the same as original. It was marked as scheme 1.

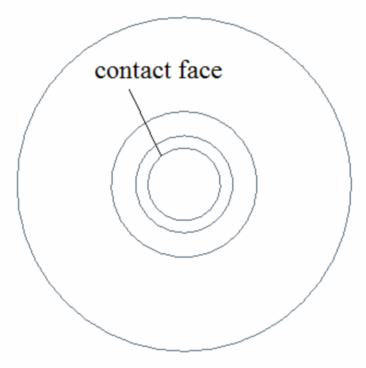


Figure 4. Original scheme

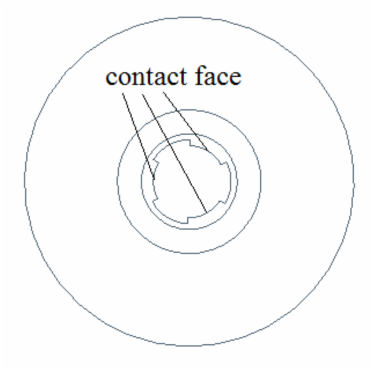


Figure 5. Scheme 1

In scheme 2, the shape of rotor hole is the same as the scheme 1, but the interference length between rotor and crankshaft is reduced from the original 15.75mm to 13mm.

Due to the hole shape of scheme 1 is no longer cylindrical, we can't apply Lame equation to solve the problem. We will calculate the press force of two scheme using the finite element method.

The finite element model of Original scheme and scheme 1 was shown in Figure 6,7. To facilitate comparison, the interference between rotor and crankshaft use the same value 3.8μm. The frictional contact was disposed at interference face. The simulation process include the entire assembling process, the maximum press force was recorded.

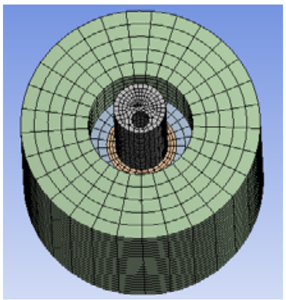


Figure 6. Finite element model of Original

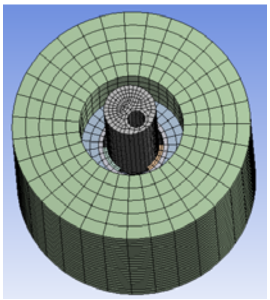


Figure 7. Finite element model of scheme 1

5. RESULT DISCUSSION

The press force result of two schemes were listed in Table 2, while the amount of interference is 3.8μm. It is important to notice that the press force of scheme 1 is reduced by 45.49% compared with the original scheme, the press force of scheme 2 is reduced by 54.52%. This illustrated that modifying the shape of the rotor hole and reducing the interference length is helpful for decreasing press force.

According to the ratio of the original scheme between simulation and experiment, we could estimate the real press force value of scheme 1,2 shown in Table 2. 10.466 is maximum pressing force which ensure that crankshaft is under the elastic range. It can be see from the result that the estimate of real press force at scheme 1 was close to 10.466(dimensionless). The estimate of real press force at scheme 2 was 17% less than limit of crankshaft.

Transit torque can also be estimated by the press force and radius of crankshaft as shown in Table 2. The transit torque of scheme 1 is 6.2(dimensionless), scheme 2 is 5.16(dimensionless). The maximum operating torque is 0.1845(dimensionless)which was measured by dynamometer at maximum capacity of the compressor. The transit torque of scheme 1 is about 31 times of compressor maximum operating torque, scheme 2 is about 27 times of maximum. It is easy to observe that scheme 1,2 could transmit torque reliably between rotor and crankshaft.

Table 2. Result of scheme1,2

Model	Dimensionless press force	Real dim. press force	Dimensionless transit torque	Real dim. transit torque
Original	8.908	18.952	5.34	11.38
scheme1	4.856	10.332	2.92	6.2
Scheme2	4.0506	8.618	2.43	5.16

6. CONCLUSIONS

The conclusions of this work are:

- It is convenient to calculate the press force and transit torque of cylindrical interference face based on Lame equations.
- The theoretical results are consistent with simulation results with less than 5% error, while the experimental press force is about 2.1 times of the value of simulation.
- Due to rotor produce process, the cylindricity and roughness of rotor hole become worse.
- Modifying the shape of the rotor hole and reducing the interference length is helpful for decreasing press force. The estimate of real press force at scheme 2 was 17% less than limit of crankshaft.
- Scheme 1,2 could transmit torque reliably between rotor and crankshaft.

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