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ABSTRACT

The analysis model of oil supply system was established for rotary compressor in this study. Oil supply system was simulated by using a CFD software —FLUENT. It gets the oil supply capacities through the shaft at different oil levels in oil sump. We have developed a CFD model for oil supply system of the rotary compressor including all components of the oil supply system and proposed a method to analyze it with CFD software including establishing the physical model, making some assumptions, setting boundary conditions. All these were validated by experiments. Then the oil supply system of S-80 compressor was studied and the oil supply rate of main bearing was obtained. In the original design, the oil supply rate is adequate at the low oil level. But in the actual extremely working condition, the oil supply of main bearings will be lack.

1. INTRODUCTION

The oil supply system has a great effect on the reliability of the components of the rotary compressor. Appropriate lubricant oil is helpful to generate oil film, reduce friction losses and take the heat away, which generated by friction and wear debris. Inadequate lubricant oil will make the average temperature of lubricant oil and bearings increase and the viscosity of the lubricant oil decrease. Excess lubricant oil also has problems on the separating of oil, at the same time the oil will be adhered to the tube of heat exchanger, which deteriorates the heat transfer performance, further more friction losses will increase and the performance and reliability of the compressor will decline. Consequently, it is essential to study the oil supply system of rotary compressor.

The rotary compressor's oil supply system which involves a lot of design parameters is complex. Studies on rotary compressor oil supply system are so lack that the design of oil supply system depends on experience. That makes design parameters optimization difficultly and can't guarantee the oil supply rate to the bearings. Some researchers employed equivalent circuit in oil supply system and regarded the elements as resistance. This method solves a few of problems. The lubricating elements also were analyzed by pressure difference and some simple formulas which include experience factor, and then the formulas constitute equations. Not so many factors were considered in the formulas of elements that are one-dimensional equations which include experience factors. In fact this method has great limitations. In recent years, some researchers have used CFD software to simulate oil system of scroll compressor.
The project has applied CFD software to oil supply system of the rotary compressor successfully. This study is the first time of using the advanced technology in the field of oil supply system on rotary compressor.

In recent years, CFD software was used to simulate start-up process oil system of scroll compressor in USA, South Korea and Denmark. H.J.Kim et al. (2000) employed equivalent circuit in oil supply system with the analogy method. Honghyun et al. (2002) simulated oil supply system of variable-speed scroll compressor with CFD software and proposed a method to improve it. Michael M. CUI (2004) simulated oil supply system of scroll compressor including all the lubricating elements using CFD software and got a good result. In this paper a CFD model for oil supply system of the rotary compressor has been developed. This model includes all components of the oil supply system. At the same time, a method to analyze oil system with CFD software by establishing the physical model, applying some assumptions and setting boundary conditions has been proposed. There was a good agreement between calculations with CFD software and experiments for the oil pumping rate. At the same time the oil supply system of S-80 compressor was studied and the oil supply rate of main bearing was obtained. In the original design, the oil supply rate was possible at the low oil level. But in the actual extremely working condition, the oil supply of main bearings will be lack.

2. NUMERIAL SIMULATION

A CFD model for oil supply system of the rotary compressor was established which includes all components of the oil supply system. We solved the fluid field and got the results through establishing the physical model, applying some assumptions, generating mesh and setting boundary conditions.

2.1 The Structure of Oil Supply System

Figure 1 shows oil supply system of twin-cylinder compressor. Oil in the sump at the bottom of the shell will be pumped as the shaft rotates. The pumped oil passing through the radial hole in the shaft is supplied to all the sliding surfaces, such as rotor and bearings. The oil to lower journal bearing flows back to sump through the groove. The oil is supplied to the clearance between rotor and shaft through the radial hole in the shaft. The oil to upper journal bearing is pumped to high pressure shell through the groove. As a result, the oil supply system is a whole. The arrows show the direction of oil flowing.

2.2 The Model of Oil Supply System

In the CFD model, the calculated domain is the domain of oil flowing. So, the model of oil supply system neglects the other parts of the compressor. Figure 2 shows the 3-D model of oil supply system. Without consideration of the radial hole in the eccentric shaft, the model will be a simplified one which including upper and lower oil rings, spiral oil groove and oil pump. When the shaft rotates at high speed, oil will be pumped to those radial holes along the stirring blade. To prevent oil entering high pressure shell through shaft, there is an oil-limiter put in the shaft top. Figure 3 shows the section of oil sump.

2.3 Fundamental Assumption

The CFD model is based on the fundamental assumptions: 1) Reynolds number is not so big correspondingly that oil flowing can be considered as incompressible laminar flow. 2) Oil has no phase change in the model and oil doesn't exchange heat with the refrigerant vapor. Then oil density and velocity will be constant because of neglecting heat exchange. 3) Oil in control volume is continuous.

2.4 Mesh Generation

Mesh generation is the base of CFD numerical simulation. The accuracy of the physical model is correlated to the correctness and the stability of the result directly. The quality of mesh has much more influence on the simulation results than the method of simulation format. It will save a lot of calculating time and computer memory if the mesh is reasonable.

In this study, tetrahedron cell was used to mesh the flow domain which was divided into several parts. Considering the complexness of flow domain, admixture mesh that means map mesh for the regular and submap mesh for the irregular domain parts was introduced. Where the gradient is large, mesh generation must be refined here. Figure 4 shows mesh of the flow domain.

2.5 Boundary Condition

The flow domain needs to set boundary conditions after generating mesh. In this study, we set pressure conditions at inlet and outlet of the model. Acceleration of gravity must be considered at the same time. Walls need to be set as stationary wall condition when the walls do not move and need to be set as rotary wall condition when the walls move. In this study the rotational speed is 2850 n/min. We also simulated oil supply rate at different oil level.

2.6 Result of Simulation

After calculation, we got the corresponding results. From pressure distribution chart, pressure contours can be obtained. In the inlet of shaft, throttle makes the pressure fall distinctly. On the wall, it will form high static pressure because of the rotating shaft, and oil will be pumped to upper part by the rotating stirring blade. Turbulent flow will occur and vortex will appear in the radial hole. Figure 5, 6, 7,8,9,10,11 show the results of simulation. From the velocity vector and pressure contour graph, we can see that there have extremum in the upper and lower bearings radial hole. The extremum is related to the direction of rotary shaft. The pressure distributions in the oil rings are asymmetrical and they are related to the position of radial holes sometime. Pressures are larger comparatively at the back of the holes. Pressures are less relatively at the front of the holes. In the spiral oil groove, oil flow is rotational flow which is caused by the rotating shaft.

3. EXPERIMENTAL VERIFICATION

3.1 Experimental Verification

The aim of this study was to find out the variety of oil supply rates at different oil levels. Thereby, getting the effect of design parameters offers a gist for the designing of oil pump and compressor.

Actually, in the steady working condition, it is hard to determine the oil supply rate of each part. Further more, the cost of measurement is high and it will take a long time. Without consideration of the pressure difference between inlet and outlet, compressor runs at an opening condition. The oil supply rate measurement becomes feasible and simple. According to experience, we know that oil flowing out spiral oil groove is a maximal part. Therefore, we can measure and compare it with the simulation results and this can validate the simulation model.

Experiments include measurements of oil supply rate at different oil level in unit time. In the opening condition, we adopt coal oil in place of lubricant oil. In the actual condition, there is a lot of refrigerant dissolving in lubricant oil. The oil viscidity will vary at a large extent as the temperature lifts, so that the experiment does not use special compressor lubricating oil. The test of oil supply system is divided into two parts. 1) Water: at room temperature, the viscidity of water is close to the real lubricating oil's. At working condition, the viscosity of lubricant oil is about 0.002 m^2/s. The viscosity of water is 0.001 m^2/s at room temperature. 2) Oil: in this study, we adopt coal oil in place of real compressor lubricant oil. At room temperature, the viscosity of coal oil is 0.002 m^2/s and it is close to the lubricant oil's at working condition.

Figure 12 is the schematic diagram of the experimental setup of the total oil supply rate measurement. When the crankshaft rotates, oil will rise along the inner wall of the shaft hole. A lot of oil flows out the spiral oil groove of the upper bearing. Through the organic glass spigot, oil flows out directly from the hole and it can be measured with a graduated flask.

There are some physical qualities to be measured, such as oil level, oil supply rate, and the time of oil pumped. Oil level is measured by a scale ruler. At the beginning of test, it needs to measure the oil level. Oil supply rate is confirmed by measuring oil pumped. We can measure the volume flux of lubricating oil. We can read the scale of the organic glass spigot which can confirm oil supply rate when it is too little. The oil supply rate can be confirmed by reading the scale of the organic spigot when it is too little.
Time of oil pumped can also be confirmed by a machine stopwatch. In this proceeding, we must measure the time several times and take an average value.

In the following, the calculation results and the experimental results were compared. In above mentioned theoretical analysis, we have set up an oil supply system physical model and calculated the pumped oil rate at different oil level. In order to verify the accuracy of model, the results of theoretical calculations were compared with the experimental results. Table 1, 2 show the results of supply rate of coal oil and water flowing out spiral oil groove for the characteristics of test.

By comparing the experimental results and calculated results, it can be found that the calculation results are in good agreement with the experimental results. Calculated values of coal oil for the characteristics of test are smaller than the experimental values. The calculation error does not exceed 5%. To a certain extent, it proves oil supply system of numerical simulation accuracy. And to conduct experiments on water, due to the reasons of the viscous and some other unknown factors, it makes the results error large relatively.

Table 1: Coal oil for the characteristics of test

Oil level	**Calculation results (g/s)**	**Experimental results (g/s)**	**Error (%)**
At the bottom of upper bearing	1.015	1.011	0.4
At the top of lower bearing	0.908	0.883	2.8

Table 2: Water for the characteristics of test

Water level	**Calculation results (g/s)**	**Experimental results (g/s)**	**Error (%)**
At the bottom of upper bearing	1.521	1.310	13.9
At the top of lower bearing	1.203	0.997	20.7

3.2 Application Examples

Through the above research, we consider that it is feasible to apply CFD software to oil supply system numerical calculation. Therefore, we used this new method to study the S-80 type compressor's oil supply system and calculate the oil supply rate of upper bearing. In the original design, the oil supply rate is adequate at the low oil level. But in the actual extremely working condition, the oil supply of main bearings will be lack. Thus it will affect reliability of the upper bearing. By calculating, we identify the problem and improve it in the producing process. According to the company's feedback, abrasion of the shaft and bearing has been effectively controlled. Figure 12, 13 show model and mesh of the S-80 type compressor.

4. CONCLUSIONS

Through this study, we have reached the following conclusions:

1. Oil level has a great influence on oil supply rate of the compressor. When the oil level is too low, oil supply rate is not enough or almost no oil pumped into the compressor shaft. When the oil level is too high, oil supply rate is excess. Therefore, oil level (filling of oil to the sump) should be controlled in a suitable range.

2. Oil viscosity also has a great influence on oil supply of the compressor. In the same test condition, flow rate of water is larger than coal oil's although the result is not obvious in a low oil level since viscosity playing an important role.

3. In the experimental process, we find that a lot of oil jets from the cylinder end surface at the time of starting the machine. This phenomenon will be more serious if the oil level is higher. Switching on/off the air-conditioner frequently will impacts on the longevity seriously. This is likely to result in damaging the compressor.

REFERENCES

1. H.J. Kim H.J. Kim, E.S. Lee, S.H. Kwag, 2000, A Study on the Oil Supply System of a Rotary Compressor. Fifteenth International Compressor Engineering Conference at **153** Purdue University, West Lafayette, IN, USA **-** July ***25-28,2000***
2. Honghyun Cho and Byoung-Kil Yoo, Yongchan Kim, Jin Taek Chung, 2002, CFD Simulation on the Oil Pumping System of A Variable Speed Scroll Compressor. Compressor Engineering Conference at Purdue,C24-1, 2002.
3. Michael M. CUI, 2004, Investigation on the Oil Supply System of a Scroll Compressor. International Compressor Engineering Conference at Purdue, July 12-15, 2004.
4. Kim, D. H., Shin, D,K., and Cho, S.O., 1998, An experimental analysis on the flow rate in scroll compressors,

Proc. Int. Compressor Engineering Conference at Purdue, pp 637-642.

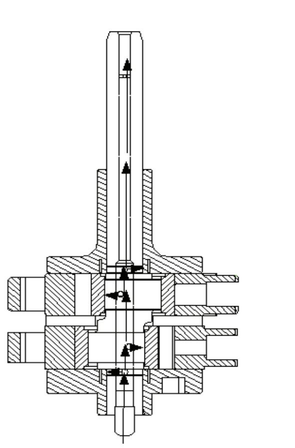

Figure 1: The structure of oil supply system

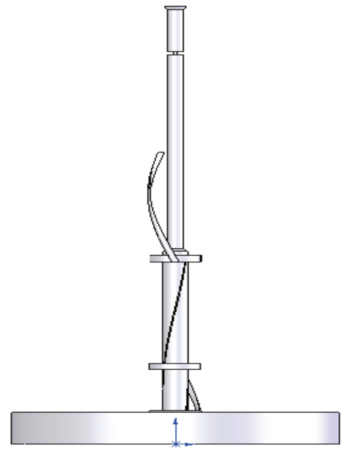

Figure 2: The 3-D model of oil supply system

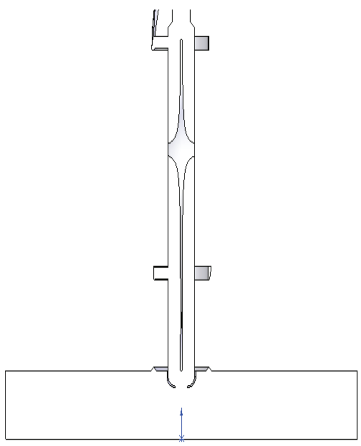

Figure 3: the section of oil sump.

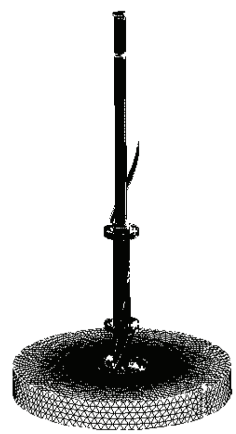

Figure 4: mesh of the flow domain.

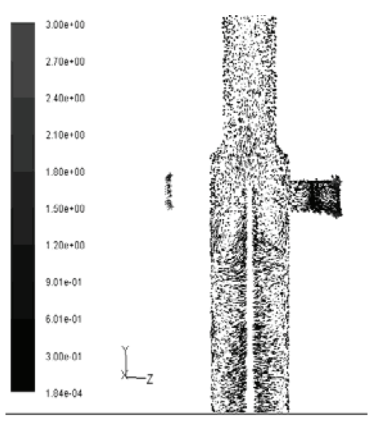

Figure 5: Velocity vector at shaft section

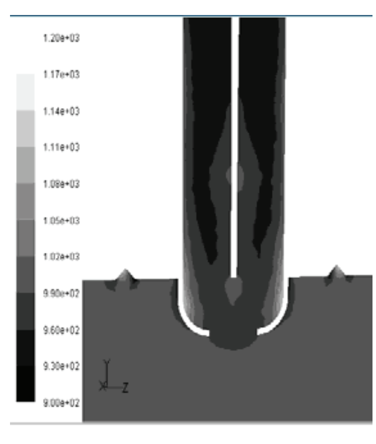

Figure 6: Pressure at inlet of shaft pump

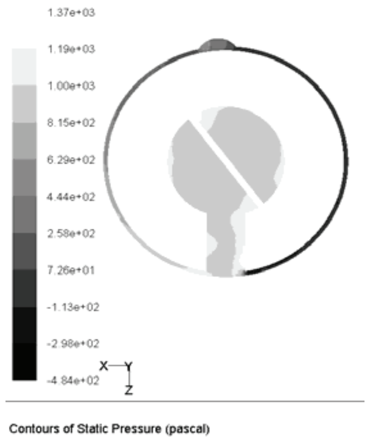

Figure 7: Pressure at lower bearing section

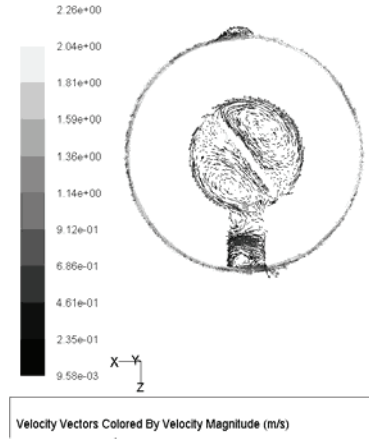

Figure: 8 Velocity vector at lower bearing section

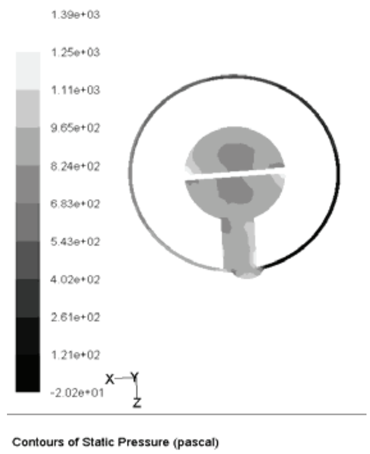

Figure 9: Pressure at upper bearing section

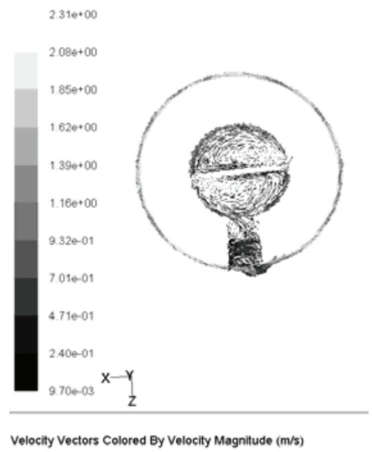

Figure: 10 Velocity vector at lower bearing section

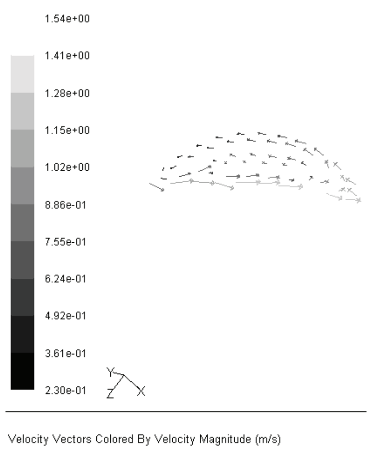

Figure 11: Velocity vector at spiral oil groove

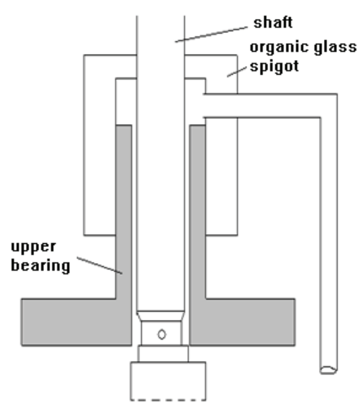

Figure 12: Diagrammatic installation

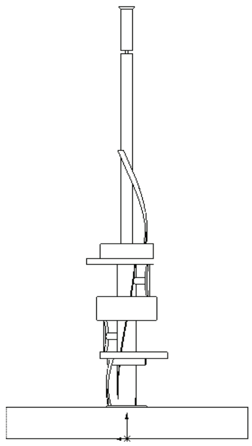

Figure 13: oil supply system model of S-80 compressor

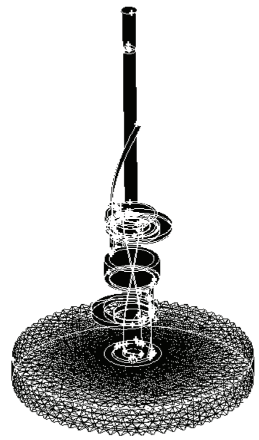

Figure 14: mesh of the flow domain for S-80 compressor