Simulation of the Deformation and Contact of Scrolls in the Scroll Compressor

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Abstract. The control of the clearances in a scroll compressor is crucial to ensure the reliability and efficiency of compressor, especially for the compressors without radial compliance mechanism. In this study, the fluid-solid-thermal simulation is conducted to get the temperature field, the pressure load distribution on scroll wraps. Then, the influences of the temperature, pressure and the inertial force on the deformation of scroll wraps are investigated. On these bases, the variation of gap size and potential contact between scroll wraps is analyzed under various orbiting radii. It is shown that the deformation caused by temperature is more significant than the pressure and inertial force. The contact occurs even though orbiting follows the theoretical orbiting radius due to the positive radial deformation of scroll wrap and negative radial deformation of inner wall of fixed scroll.

1. Introduction
The scroll compressor is widely used in the refrigeration system for its high energy efficiency, quite operation and low vibration. The control of the clearances in a scroll compressor is crucial to ensure the reliability and efficiency of compressor. The axial and radial clearances between orbiting scroll and fixed scroll greatly affect the internal leakage between working chambers [1]. Generally, intermediate chamber for orbiting scroll is used to balance the axial gas force and to control the axial clearance. For the radial gaps, some scroll compressors use radial compliance mechanism to ensure the scroll wraps radially engaged while some scroll compressors use journal bearings without radial compliance mechanism [2], which possess higher request on the control on manufacturing tolerances and on scroll wraps deformation to avoid radial contact or oversized gaps between scroll wraps [3]. Additionally, in this case, the real orbiting radius of orbiting scroll is not only determined by the geometry but also influenced by the operating conditions because the radial clearance of journal bearings allows the orbiting scroll slightly deviate from theoretical orbiting radius.

The radial gaps are simultaneously influenced by the thermal deformation, pressure deformation, inertial force deformation as well as the motion of orbiting scroll (OS). Computational Fluid Dynamics (CFD) simulation is generally used to obtain the temperature and pressure distribution [4]. Recently, many useful technologies such as structural dynamic mesh and fluid-structural-interaction (FSI) are introduced into the CFD simulation of scroll compressor to get a faster and more accurate result. Wang et al. [5] developed a novel structured dynamic mesh generation method for the scroll compressors and indicated that the structured mesh is much more accurate than the unstructured mesh. Hesse et al. [6] developed a software to generate dynamic mesh for scroll machines. Their simulation results show good
agreements with experiments. Sun et al. [7] presented the sharp decrease of pressure and temperature when leaked fluid flowed through the radial gaps with dynamic grids. Hsieh et al. [8] simulated the internal flow fields, thermal deformation of an oil-free scroll vacuum pump and compressor with fluid-solid-thermal coupling analysis. In conclusion, plenty of research has been done on the temperature-distribution on scroll wraps and its influence on the axial thermal deformation. Furthermore, few devotions have been made in the radial gaps and the possible contact between OS and fixed scroll (FS) under multi-field load.

Based on previous studies, this paper mainly focuses on the transient radial deformation of scroll wraps and possible contact between scroll wraps. The temperature and pressure fields are firstly obtained by the fluid-solid-thermal coupling simulation of compression chamber and scrolls. Then, temperature, pressure and inertial force loads are applied to scroll walls with one-way coupling method, where both the axial and radial deformation distribution of scroll wraps are determined. Moreover, the variation of orbiting radius of OS is taken into consideration to investigate the variation of radial gap and possible contact behavior between OS and FS wraps with finite element analysis (FEA).

2. Model and methodology

2.1. Geometry

The schematic of the scroll compressor is presented in Figure 1. The schematic of the scroll compressor is show in figure 1. In particular, models for fluid and solid domains are simplified in this paper. For the fluid domain: the working chambers, inlet, outlets (in order to avoid over-compression, eight relief valves are installed near the main discharge valve), upper-axial gap and lower-axial gap are considered while the intermediate chamber and high-pressure discharge chamber are ignored in the CFD simulation. For the solid domain, only the models of FS and OS is established in the fluid-solid-thermal coupling simulation and in the FEA simulation, while the shell, crank shaft and the main frame are not taken into consideration. The scroll wraps of OS and FS enclosed several cylindrical working chambers. Specific parameters of the scroll compressor are listed in Table 1. The materials properties are the same as those of the real scroll parts, both the OS or FS is made by HT300.

![Figure 1 Schematic of the scroll compressor](image)

| Parameter                          | Value        |
|------------------------------------|--------------|
| Type of scroll profile             | algebraic spiral |
| Height of scroll wrap [mm]         | 41.4         |
| Stroke volume [mL/rev]             | 96           |
| Theoretical orbiting radius [mm]   | 4.52         |
| Numbers of relief valves           | 8            |

Table 1 the parameters of the scroll compressor
|                            |       |
|---------------------------|-------|
| Refrigerant               | R410a |
| Suction pressure – Absolute [MPa] | 0.998 |
| Discharge pressure – Absolute [MPa] | 3.35  |
| Designed intermediate pressure – Absolute [MPa] | 1.4   |
| Suction temperature [℃]   | 18    |
| Rotational speed [rpm]    | 1800 to 7200 |
| Mass of orbiting scroll [kg] | 1.50  |

2.2. CFD model

The commercial software ANSYS is used to solve the complex fluid-solid-thermal coupling simulation. As Figure 2 shows, the three-dimensional grids for inlet, outlets, upper axial gaps are stationary region, while the dynamic mesh strategy is used for working chambers and lower axial gap that fixed with OS. The radial clearance of scroll wraps is set as low as 20 μm to avoid negative volume and to reduce the leakage between chambers in present work. Additionally, axial gaps are also included in the fluid domain and its height is 10 μm, where the refrigerant could radially leak to low-pressure chamber from high-pressure working chamber. Moreover, a pressure inlet boundary at is applied to the outside of lower-axial gap to realize the effect of the intermediate pressure chamber.

![Figure 2. Illustration of the grids for fluid domain](image)

Groups of grids are tested as a grid-independence check, where the total number of fluid domains varies from 300,000 to 1,000,000 with various layers in radial and axial directions for working chambers. The results of volume efficiency of the scroll compressor stand at about 84.8% under given operating condition, showing no significant changes with different grid densities. Thus, about 500,000 of total number of fluid domain elements is used in the fluid-solid-thermal simulation, taking the computational cost and performance into consideration.

The solid domain is composed of OS and FS with their discs respectively. The motion of OS is constrained properly as that in the reality. Fluid-structure interfaces (FSI) are established to handle the heat transfer between fluid and solid. Thus, the FSI boundary is employed for scroll wraps and discs of FS and OS as well as the inner wall of FS. An assumption is made that the temperature is fixed at 88 ℃ for outer side of the FS because the average discharge temperature is 88 ℃. The adiabatic boundary is used for outer wall of OS. The grid-independence check for solid domain is not done and the total number elements is 100,000.

2.3. FEA model

Since the temperature and pressure distribution of FS and OS are obtained, FEA simulation is firstly done to investigate the temperature, pressure and inertial force deformation respectively. The temperature and pressure distributions are exported as external date at each timestep and then import into structural analysis. The results of CFD simulation are imported into FEA module respectively. Additional discharge pressure boundaries are manually applied at the outer walls of FS and OS, because
they are not given in the results of CFD simulation. The inertial force of scroll wrap of OS is calculated by the rotational speed and it is manually added too.

As the main frame, the crankshaft and the shell are ignored to reduce grid size, additional boundary conditions are needed for the OS and FS. In the deformation analyses, all the degree-of-freedom of FS is constrained at the red solid lines in Figure 3, while the axial and radial displacement of OS is fixed at zero at the red solid and dot lines. The influences of temperature, pressure and inertial force are calculated respectively. Then, in contact analysis, the multi-field results are imported simultaneously. The radial displacement at dot lines of OS is set according to the eccentricity, which is decided by the shaft angle (θs) and the orbiting radius of OS (Ror), at the red dot line to simulate the drive of crankshaft. Specifically, only the contact between side walls of scroll wraps is considered in the contact analysis.

![Figure 3. Illustration of displacement boundary condition and imported temperature and pressure boundary conditions from CFD results for the FEA simulation](image)

3. Results and discussions

3.1. CFD results

The gas force on OS could be obtained by integrating the pressure on the scroll wrap walls and the discs. The forces with θs are presented in Figure 4. Lateral forces consist of radial force (Fr) and tangential force (Ft) in cylindrical coordinate (and Fx and Fy in cartesian coordinate). The value of Fr nears constant, standing at about -400 N, and the Ft fluctuates from 4650 N to 5200 N in a cycle. However, the Fz is much large than lateral force, almost reaching -19,500 N, which needs to be balanced by the force of intermediate pressure chamber. It should also be noticed that the Fr is larger than the eccentric force (240 N) of OS when rotational speed is 1800 rpm, which could lead to the separating tendency between OS and FS. Fortunately, the limited clearances of journal bearings could provide extra constrain for OS to avoid radial clearances out of control.

![Figure 4: Gas forces on orbiting scroll](image)

Since asymmetrical chamber design is used, the pressure variations are also asymmetrical for different working chambers as Figure 5 shows. As the compression angle (θc, θc = θs - 137°) for outer of OS with inner of FS is 180° earlier than that of inner of OS with outer of FS, the pressure of working chamber formed by them is higher at the same θs. It should be also pointed out that the pressure...
distribution on the lower axial gap is affected by the working chamber and the intermediate chamber. Similarly, the upper axial gap is also affected by the discharge ports, as the pressure remains unchanged when the scroll wrap hides the relief ports.

Figure 5. Pressure distribution on working chamber and lower and upper axial gaps

The temperature distribution on the middle cross-section of working chamber, OS and FS is shown in Figure 6. In a single chamber, the uneven temperature distribution is caused by the leakage high-temperature fluid and heat transfer of solid. The temperature at center of scroll wraps is the highest, reaching 110 °C, and it gradually decreases along the wrap. The temperature distribution of FS is greatly affected by the environmental temperature and the minimum temperature of FS is about 45 °C, while the minimum temperature of OS is 18 °C. Overall, the variation of temperature distribution on solid is insignificant in a cycle.

Figure 6. Temperature distribution on middle cross-section of working chamber, OS and FS

3.2. FEA results

Since the temperature and pressure fields are obtained by CFD simulation, the transient results are exported and imported into structural module as external loads. Other boundary conditions are also applied as the section 2.3 described. In this section, a cylindrical coordination origin in the center of FS and OS is established to illustrate the radial and the axial deformation of solid domain.

3.2.1. Temperature deformation. Figure 7 presents the radial (dR) and axial (dZ) deformation of FS and OS respectively. Generally, the scroll wraps show a tendency to expand in radial direction and positive deformation tendency in axial direction. The maximum radial deformation of FS wrap occurs near the inlet, reaching about 30 μm, which is as about twice larger as that of OS wrap. However, the negative dR of FS inner wall at bottom reaches -22 μm, which is different from the wraps. The maximum axial deformation of scroll wrap occurs at the center of OS and it gradually decreases at the tail of scroll wrap, while the dZ of FS is quite uniform along its wrap. The dZ of discs shows opposite trend. The dZ of OS disc remains near zero because the constraint is close. The maximum dZ of FS disc, being 85.8 μm, occurs at the center of FS and it decreases radially as the constraint is applied on the bottom surface in the FS. The transient deformation is not given as the variation of temperature loads on FS and OS nearly remains unchanged in a cycle as described in section 3.1.
3.2.2. Pressure deformation. Figure 8 shows the deformations in radial and axial directions of FS and OS over the time respectively. The pressure deformation is slightly small than that caused by temperature load, especially for the dZ. The largest dR of scroll wrap is about 10 μm at the top due to the cantilever effect of wrap. Generally, positive and negative dR occur around the engage line of OS and FS. Taking the pressure distribution into consideration, positive dR is caused by the pressure difference between inside and outside of the wrap chambers. The negative dR could not only be influenced by the nearby deformed scroll wrap region, but also the pressure difference on symmetric chambers of inner and outer OS, which is caused by the phase difference of θc for theses chambers. Thus, both positive and negative dR occur in scroll wrap under pressure load. As the pressure distribution changes with the orbiting of OS, mirror distribution dR is obtained when θs is 360°. However, similar deformation tendency of both FS and OS could be concluded as those of 180°.

3.2.3. Inertial force deformation. Figure 9 illustrates the total deformation of OS driven by inertial force at various θs when rotational speed is 1800 rpm. It seems that the deformation mainly occurs at the tail region of scroll wrap. The value of deformation caused by inertial force is fairly smaller than those caused by temperature and pressure load as the rotational speed is small.
3.2.4. Coupling deformation. The coupling deformation under temperature, pressure and inertial loads when θs=360° is shown in Figure 10. It should be noticed that only the loads are applied while the radial displacement boundary of OS is set to zero for a clear view of the deformation distribution. As have been noticed in 3.2.1, the negative dR of inner wall of FS attracts our attention again, while the positive dR of scroll wrap pairs are relatively uniform in the cross-section. The negative dR of inner wall of FS as well as the positive dR of scroll wrap could provide extra constrain to the motion of OS, even though the OS have not reached theoretical orbiting radius. Though the deformation of OS wrap tends to reduce the axial gap, the maximum deformation of FS disc reaches 80 μm that is much larger than that of OS. Thus, the axial gap is increased under multi-field loads. It is also shown the uneven axial gap distributes along the wraps. The axial gap decreases from the center region to the outer region of scroll wraps.

3.2.5. Contact analysis. The results of contact analysis without or with multi-field loads on solid are shown in Figure 11. When loads and theoretical Ror are applied, the contact only occurs at the bottom outer wall of OS and inner wall of FS on the suction side of scroll wraps, which agrees well with the deformation results discussed in section 3.2.4. The maximum contact pressure is 485 MPa. Moreover, since the contact occurs on limited region of engaging lines, the sealing of chambers is not geometrically granted now. However, it is noted that, in reality, the lubricating oil in the working chamber could improve the sealing between chambers by filling in the gaps and reduce the contact pressure by hydrodynamic effect.

The variation of contact force with the change of Ror and θs is shown in Figure 12. The orientation of contact forces is normal to the rotation direction. The average contact force is 9100 N in a cycle when Ror keeps the theoretical value, being 4.52 mm. Since the eccentric force of OS is much smaller than 9100 N, the deformation and contact of wraps could also cause the deviated motion of OS as well as the tilting crankshaft. As the OS and FS have separating tendency by the contact force, the full sealing of
the engage line could not be secured. Additionally, potential reliability risks could be raised due to the misalignment of shaft in journal bearings on this condition. There is a drastic decrease of contact force to 400 N when \( \theta_s \) increase to 120°, where then the ending profiles for outer of OS and inner of FS start to engage. The drastic decrease might be caused by the positive dR of FS wrap near suction port as Figure 7 shows. Thus, a shock load could occur on the tail of OS wrap during its engagement according to the simulation results, especially at a higher rotational speed.

![Figure 7](image_url)

**Figure 7. The variation of contact force with different dR in a cycle**

4. Conclusion

In this paper, the fluid-solid-thermal CFD simulation is conducted to obtain the pressure and temperature distributions of an asymmetric algebraic scroll compressor. Then the multi-field loads are imported into fixed scroll and orbiting scroll in the FEA analysis to get the deformation results. At last, the contact behavior under various orbiting radius of OS is analyzed. Following conclusion can be made:

- The average radial gas force exerted on OS is -400 N while the eccentric force of OS at 1800 rpm is about 240 N. The larger radial gas force could lead to the separating tendency of OS to FS at lower rotational speed.

- Additional axial gap is caused by the axial deformation of scroll wraps and discs, and its height decreases from the center region to the outer region of scroll wraps.

- In radial direction, the deformation of solid causes the ‘expanded scroll wraps’ and ‘shrinked FS inner wall’, which lead to the contact behavior even though OS follows the theoretical orbiting radius. In this case, the radial gap could be increased and there is a drastic decrease of contact force when \( \theta_s \) increase to 120°.

Moreover, several boundary conditions could be improved in current work to obtain more precise results, including the temperature distribution of outer wall of FS, the axial constrain of OS and radial motion of OS driven by crankshaft. Further research would be done in these aspects in the future. Firstly, the temperature distribution of outer wall of FS should be also obtained with fluid-solid-thermal CFD analysis or experiment. Besides, the OS is not constrained in Z-direction in reality but pushed by the back pressures of OS. The motion of OS in R-direction is coupled with the motion of crankshaft.

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