Optimization of the tip profile of orbiting scroll in an asymmetry suction chamber scroll compressor

SUN Shuaihui1,2, Wang Zhe1,2, GUO Pengcheng1,2, ZHANG bo3, MAO Zhenkai3
1. Xi’an University of Technology, Xi’an, Shaanxi, 710048, China
2. State Key Laboratory of Eco-hydraulics in Northwest Arid Region of China, Xi’an, Shaanxi, 710048, China
3. Power China Northwest Engineering Corporation Limited, Xi’an, 710065, Shaanxi, PR China.
E-mail address: shs@xaut.edu.cn

Abstract: The scroll compressors with asymmetrical suction chamber have large discharge loss, as the pressure in outside chamber is far larger than the discharge pressure when the discharge process begins. It is necessary to optimize the scroll profile around the discharge port to reduce the discharge loss. The theoretical method is to redesign the tip profile to obtain the same volume ratio of outside working chamber with the inner working chamber. However, as it is difficult to predict the discharge loss through the flank gap in discharge chamber, the optimum design cannot be obtained. In this paper, we establish an optimization platform based on the CFD method. The profile of orbiting scroll tip is parametrized with CAD software, then the CFD calculation will be done and the performance parameters will be extracted. The MOST optimization algorithm is used to search the best performance parameters and control the parameterized value of scroll profile. After iterations, the best profile with the modification angle of 66.3° was obtained. Compared to the original model, the volumetric efficiency of the compressor increased by 4.11%; and the isentropic efficiency increased by 4.91%. Afterwards, the flow field are analysed to study the characteristics of the optimized scroll compressor. The results show that the backflow, the secondary vortex in discharge chamber and the discharge mass flow pulsation were reduced, which led to the decrease of the discharge loss and the improvement of the isentropic efficiency.

1. Introduction

Scroll compressors usually have several pairs of crescent-shaped working chambers which are formed by the engagement between the orbiting scroll and the fixed scroll. As the orbiting scroll orbits, the volume of each pair of working chamber changes, and a series of continuous processes of suction,
Compressors and discharge happen for pressurizing the gas. In a hermetrical scroll refrigeration compressor with a high-pressure shell, only one inlet port is designed to reduce the costs. As shown in Figure 1, during the suction process of symmetric suction chamber compressor (SSC), the fluid which flows into the suction chamber Suc_B has to bypass the suction flow passage. If the flow passage is not designed well, the flow loss in the passage is large, and even the fluid is blocked by the scroll vane of orbiting scroll[1], which results in the high pulsation of mass flow rate and the low volumetric efficiency.

In order to solve the problem, the asymmetric suction chamber compressors (ASC) were designed by extending the end angle of the fixed scroll profile by 180°. In this case, the original suction flow passage was changed into the suction chamber Suc_B, which was different from the suction chamber Suc_A in Figure 1[2]. The termination angle of fixed scroll profile in asymmetric suction chamber compressor (ASC) is larger than that of orbiting scroll, thereby the suction chamber Suc_A is asymmetrical with the Suc_B, as shown in Figure 1. Suefuji et al.[3] proposed the patent of ASC firstly, where the terminal angle of fixed scroll was about 60° to 120° larger than that of orbiting scroll. Qiao et al.[4] calculated the force and moment on the orbiting scroll in an asymmetric scroll compressor and pointed out the force and moment increased while their pulsation amplitudes decreased after the asymmetric structure was used. Shaffer et al.[5] developed a geometric model for an ASC using a control volume approach and proposed the calculation equations for the area of working chambers under different crank angles. The authors[6] established a transient numerical simulation model of scroll compressors with different flow passage and analyze the suction process of these compressors.
The results indicated that the suction mass flow rate increased and the suction pulsation was reduced significantly as the asymmetric suction chamber was used; but the discharge loss increased due to the asymmetric pressure distribution between working chambers. Cao et al.[7] modified the scroll profile around the discharge port of an ASC and calculated its performance; the results indicated that the pressure difference between two working chambers and the discharge losses were reduced after the modification. Wang[8] modified the orbiting scroll profile to ensure the volume ratio of the outside working chamber is equal to that of the inner working chamber. The isentropic efficiency increased but the under-compression and backflow happened. Therefore, the modification angle was not the optimum one. Recently, the optimization technology based on CFD method had been applied to modify the shape of blade of centrifugal pump and wind turbine[8-9], and obtained good modification schemes to improve the efficiency or reliability of these machines. Cavazzini combined the scroll compressor parameterized model, CFD method and optimization algorithm in the MATLAB to study the impact of different parameters on compressor efficiency [10]. Therefore, this paper established the optimization model based on the CFD method and ISIGHT software. By parameterizing the orbiting scroll tooth profile, the fluid domain of the discharge chamber was modified after the performance data was obtained by the CFD simulation. The MOST algorithm was used for searching the best modification angle of the profile of orbiting scroll. Finally, the flow field of the scroll compressor with the best modification angle was analyzed.

2. Mathematical Model

2.1. Physical Model

A scroll refrigeration compressor was taken as the research object in this paper. The main parameters of the scroll compressor are shown in Table 1. Figure 2. shows the modification angle of the orbiting scroll according to the equal built-in volume ratio[11]. In order to eliminate the working fluid backflow and improve the efficiency, the modification angle would be optimized based on CFD and optimization model.

| Parameters                           | Value |
|--------------------------------------|-------|
| Thickness of scroll vane /mm         | 3.6   |
| Profile height /mm                   | 4     |
| Orbiting radius ρ /mm                | 4.5   |
| Scroll warp height H/mm              | 36    |
| Base circle radius a /mm             | 2.578 |
| Initial angle of involute α /rad     | 0.698 |
| Determination angle of fixed scroll /rad | 21.99 |
| Determination angle of orbiting scroll /rad | 18.85 |
| flank clearance /mm                  | 0.030 |
| Axial clearance /mm                  | 0.015 |

2.2. Numerical simulation model

The numerical simulation model was established using commercial software PumpLinx[6]. The hexahedral grids were generated in working chambers. Ten layers of grids were generated in the flank
clearance and five layers were generated in the axial clearance. As the orbiting scroll rotated, all grids were regenerated at every timestep during the numerical simulation. The turbulence model was adopted RNG k-e model. The adiabatic wall was used for wall boundary conditions. Refrigerant R22 was used as the working fluid and its real properties were obtained from NIST Chemistry WebBook[12]. The R22 property tables including density, enthalpy, thermal conductivity and viscosity were added into PumpLinx.

The pressures of inlet and outlet were set to be 0.627MPa and 2.146MPa, respectively. The inlet temperature was 307.7K. The rotational speed was set to be 2880 r·min⁻¹. When the averaged mass flow rate difference between the inlet and outlet less than 1% was considered convergence. The accuracy of the numerical model was verified by the experimental results. The comparisons results can be found in Ref. [6].

![Diagram](image)

**Figure 2.** Schematic diagram of modified profile of orbiting scroll tip

### 2.3. Optimization procedure

The optimization program was built through the isight optimization platform which contained the following parts[13], as shown in Figure 3. Firstly, the modified angle of the profile of orbiting scroll tip was considered as the input parameter of the optimized model through the modelling software, which ranged from 65 to 75°. By modifying this parameter, the original model can be modified and saved as an STL file. Secondly, the CFD software PumpLinx was integrated into the ISIGHT platform to import the STL file and create the rotor fluid domain grid which was added to the case file of the original model. The inlet and outlet fluid domain grids were not modified, and the calculation of the
new case file was controlled through the batch file. Afterwards, the CFD calculation result file was imported into EXCEL to obtain the parameters required to calculate the objective function value. Then, the optimization algorithm determines the input parameter value for the next calculation, and the iteration started again. Until the best isentropic efficiency was obtained, the iteration stopped and the optimum results were recorded.

![Figure 3](https://example.com/figure3.png)

**Figure 3.** Schematic diagram of optimization procedure

3. Results and Discussion

In this optimization, the modification angle of the profile of orbiting scroll tip ranged from 65° to 75°. The initial correction angle was 75°. The isentropic efficiency was the target of optimization. After 9 iterations of calculation, the optimal modification angle was determined to be 66.3°. Table2 shows the time-average performance parameters of the compressor before and after optimization. The volumetric efficiency of the optimized model was improved, and the exhaust temperature is reduced by 6.57K. The adiabatic efficiency of the optimized model is 60.43%, which is 4.91% higher than the original model. This is because the optimized model changed the under-compression state of the prototype compressor's working chamber and improved the compressor's volumetric efficiency and adiabatic efficiency.

| Model                  | Suction mass flow rate [kg·s⁻¹] | Volumetric efficiency [%] | Isentropic efficiency [%] | Discharge temperature [K] |
|------------------------|---------------------------------|---------------------------|---------------------------|---------------------------|
| Original model(75°)    | 0.08854                         | 92.35                     | 55.52                     | 384.36                    |
| Optimized model(66.3°) | 0.09354                         | 96.56                     | 60.43                     | 377.79                    |

Table 2. Performance of two compressors

Figure 4 shows the variation of discharge mass flow rate of the original model and the optimized model in one period. The 0 degree was defined when the suction process in Suc_A was finished. Both diagrams fluctuated with the crank angle and reach the minimum value at the crank angle of 54° and 46°, respectively. As the profile of orbiting scroll was cut, the outer working chamber Com_2B(Figure 5) firstly connected to the discharge chamber. At this moment, as the pressure in working chamber was lower than discharge pressure, some gas flowed back into the working chamber, which led to the decreases of discharge mass flow rate. In the original model, the mass flow rate became minus at the crank angle of 54°, which meant that the backflow was severe and led to large flow loss. After
optimization, the minimum mass flow rate happened at about 46° and kept positive. Therefore, the discharge mass flow rate of the optimized model was larger than the original one, which led to the improvement of the volumetric efficiency.

Figure 4. Discharge mass flow rate of different models

Figure 5. Pressure distribution (a) and p-θ diagrams with crank angle (b) in different compressor

Figure 5 shows the pressure distribution of the optimized model and the p-θ diagrams for both models during discharge process. The simulation results show that the pressure in one working chamber is almost uniform, but it is different from the other corresponding chambers, such as the Com_1A and Com_1B. This is caused by the asymmetric suction chamber Suc_B which is 180° earlier than the Suc_A; the gas in Suc_B and Com_B was compressed 180° earlier and their pressure was higher. In Figure 5(b), the 0 degree was defined when the suction process in Suc_A was finished. Because the
process in Suc_B was 180 degree earlier than that in Suc_A, the compression chamber Com_1B become Com_2B at 180degree in Figure5(b). Hence, the Figure5(b) actually shows the p-θ diagrams in the working chambers Com_2A, Com_2B and Com_2B’ (optimized model) in a period. In the optimized model, the modification angle changed from 75° to 66.3°, which led to the delay of the discharge angle of Com_2B’. As shown in Figure5(b), the discharge angle of Com_2B’ was 315° and the pressure in Com_2B’ reached the discharge pressure at the crank angle. However, the discharge angle of Com_2B (the original model) was 306° when the pressure in Com_2B was lower than the discharge pressure, which resulted in the under-compression and the backflow. And the discharge loss of the optimized model was lower than that of the original model. The pressure in Com_2B was higher than that of Com_2B’, so the leakage mass flow rate from the Com_2B was larger than that in Com_2B’, and the volumetric efficiency of the original model was lower.

**Figure 6.** Pressure distribution of discharge and compression chamber in different model

**Figure 7.** Velocity field of discharge chambers (a) original model (b) optimized model
Figure 6 shows the pressure distribution in the discharge chamber of the original model (a) and the optimized model (b). When the crank angle is 126°, the compression chamber Com_2B of the original model begins to connect with the discharge chamber. The compression process of the compression chamber Com_2A has not yet finished and its pressure is far lower than the discharge pressure. After optimization, the pressure in Com_2B increases due to longer compression time.

Figure 7 shows the velocity field of discharge chamber of the two models at the beginning of the discharge process. Because the pressure in the discharge chamber is is higher than the compression chamber, the working fluid flows back into the compression chamber in the form of gap jets. The backflow has high velocity and interacts with the main flow in the working chamber to form a complex secondary vortex. As shown in Figure 7(b), the velocity of backflow in the optimized compressor is lower than that of the original flow, and the flow loss is also lower.

4. Conclusions
In order to reduce the discharge loss of scroll compressor with the asymmetric suction chamber and improve its isentropic efficiency, this paper established the optimization model based on the CFD method using the ISIGHT software, and optimized the orbiting scroll tip profile by modifying the initial angle. The best modification angle was obtained and the results were analysed and compared. The following conclusions were drawn:

1. After optimization, the modification angle of the tip profile of orbiting scroll was reduced from 75° to 66.3°, the volumetric efficiency of the compressor increased by 4.11%; and the isentropic efficiency increased by 4.91%; the optimization effect was obvious.
2. After optimization, the pulsation of discharge mass flow was significantly reduced, and the backflow in discharge pipe was reduced.
3. In optimized compressor, the pressure in compression chamber Com_2B at the beginning of discharge process was basically equal to the discharge pressure, which lead to the lower backflow rate and discharge loss.

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