Numerical simulation of three-dimensional unsteady flow in a scroll expander applied in waste heat recovery

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Abstract. Three-dimensional numerical simulations of a scroll expander were performed with dynamic mesh technology. R245fa was selected as the working fluid in the simulations. The PISO algorithm was applied to solve the governing equations with RNG $k-\varepsilon$ turbulent model. The distribution and variation of three-dimensional flow field inside the expander were obtained. The research indicates that the flow field is nonuniform and asymmetrical distributions exist inside the expander. Vortex flows also exist in some working chambers. Dynamic clearance leakage flows and inlet orifice throttling have great effects on the flow field distribution. Transient output torque and the mass flux have periodic fluctuations during the working cycles.

1. Introduction
The development of automotive industry has been subject to the global energy and environmental crises in recent years. The typical thermal efficiency of vehicle engine is only about 20% to 45%. The remaining energy is dissipated into atmosphere through exhaust, the cooling system and lubrication system [1]. Currently, the Organic Rankine Cycle (ORC) is an effective method for low grade heat recovery of vehicle engines. Its advantages are low working fluid evaporating temperature, structural simplicity, and high cycle efficiency [2] [3].

As the core component of ORC system, the expander’s performances have great influences on the system cycle efficiency. Among positive displacement machines, the scroll machine is a good candidate for ORC, because of its reduced number of moving parts, reliability, wide output power range, and broad availability [4]. In recent years, some researches on the performances of the scroll expander have been carried out. However, the working process simulation of the scroll expander is limited to zero or one dimensional thermodynamic model [4-6]. In some one dimensional theoretical models, the flow field variables such as pressure, temperature, and density are calculated as the function of the volume [5] [6]. Due to the flow field is assumed to be at equilibrium state, the non-uniformity and asymmetry of the flow field are neglected. The scroll expanders applied in the ORC experiments are mostly modified from the scroll compressor and have low efficiencies [7-9]. It is difficult to obtain more flow field information by experiments due to the structural complexity of the scroll geometry. The two dimensional numerical models with simplified inlet boundary also could not reflect clearance leakages, inlet orifice throttling, vortex flows in the working chambers, etc [10].
This paper presents a three dimensional numerical simulation of a scroll expander applied in ORC system with dynamic mesh technology. The variations of three dimensional unsteady flows and the thermodynamic performances are shown.

2. Methods
The scroll expander consists of a pair of scrolls (a fixed scroll and an orbiting scroll), shell, anti-rotation mechanism, counterbalance, and other devices. The scroll wraps are assembled with a relative angle of $\pi$ to form crescent shaped closed chambers. As shown in figure 1, the high-pressure organic working fluid vapours deliver the output work through eccentric crankshaft after the processes of suction, expansion, and discharge. In this paper, a scroll compressor used in vehicle air conditioning system was modified to operate as scroll expander. The main scroll profiles geometric parameters are shown in table 1. The Perfect Meshing Profile (PMP) composed of two arcs and a single line curves was applied to modify the profiles of the central scroll wraps in order to improve the performance and the scroll wrap tip’s structural strength [11].

![Scroll Expander Diagram](image)

Figure 1. The working principle of a scroll expander.

![Grid Model](image)

Figure 2. Grid model of the scroll expander:(a) $-z$ view, (b) $+z$ view, (c) $-x$ view, (d) full view

Table 1. Scroll profiles geometric parameters

| Parameters | Value       |
|------------|-------------|
| $r_b$ (m)  | 3.2×10^{-3}|
| $r_o$ (m)  | 5.58×10^{-3}|
| $\phi_{i0}$ (rad) | 0.1745 |
| $\phi_{o0}$ (rad) | 5.236 |
| $\phi_{i1}$ (rad) | 16.556 |
| $\phi_{o1}$ (rad) | -1.2217 |
| $\phi_{i2}$ (rad) | 2.094 |
| $\phi_{o2}$ (rad) | 16.556 |
| $H$ (m)    | 3.3×10^{-2} |

Where $r_b$ is the base circle radius; $r_o$ is the orbiting radius; $\phi_{i0}$ and $\phi_{o0}$ are the inner and outer involute initial angles, respectively. $\phi_{i1}$ and $\phi_{o1}$ are inner and outer involute start angles. $\phi_{i2}$ and $\phi_{o2}$ are the inner and outer involute end angles, $H$ is the scroll wrap height.

![Mesh Details](image)

Figure 2 shows the computational unstructured mesh which was generated in pre-processing software GAMBIT. Compared with different numbers of mesh under the same topology, the mesh shown in figure 2 meets the simulation accuracy requirement. Dynamic mesh technology was applied in this simulation. With the eccentric rotation of the orbiting scroll, the working chambers’ volumes change gradually. The unstructured mesh was updated by the functions of smoothing and remeshing at appropriate time step to control the mesh skewness. The movement of orbiting scroll was controlled by user defined function (UDF). The flow inside the expander is developed turbulence with curved
walls. The $k-\varepsilon$ turbulent model with renormalization group (RNG) was selected as the fluid model \[12\]. Non-slip boundary condition was employed on the wall. Near wall regions were analysed using the standard wall functions method. The convection term in equations was discretized using second-order upwind scheme. The central difference method was adopted to deal with the diffusion term. PISO algorithm of the pressure based segregated solver was applied to solve the discrete governing equations. Pressure boundary conditions were adopted for inlet and outlet. According to the actual working condition, the inlet pressure and temperature were 0.9MPa and 405K, respectively. The outlet pressure was 0.3MPa. R245fa was selected as the working fluid from NIST (National Institute of Standards and Technology) real gas model.

### 3. Results and Discussion

In this section, the transient performances were analyzed at two different rotation speeds, 1000 and 1500 r/min. Internal unsteady flow field of the scroll expander at 1000 r/min was also analyzed.

#### 3.1. Performance analysis

Figure 3 shows the variations of inlet and outlet mass flux with time in four working cycles. The inlet and outlet mass flux presents relatively stable periodic fluctuation from the second cycle. Due to the periodic change of suction orifice flow area, the gas velocity and density in inlet pipe also change with time. Moreover, the mass flux of the 1500 r/min case is obviously greater than the 1000 r/min case. According to the equation (1), the mass flux average value at inlet and outlet can be calculated using time-integration algorithm.

\[
\overline{q_m} = \frac{1}{T} \int_{t}^{t+T} q_m(t) dt
\]

\[
\overline{T_e} = \frac{1}{2\pi} \int_{0}^{2\pi} T_e(\theta) d\theta
\]

\[
P = 2\pi \cdot T_e \cdot n / 60
\]

Where $\overline{q_m}$ and $q_m(t)$ are the average and transient mass fluxes in one working cycle, respectively. $\overline{T_e}$ and $T_e(\theta)$ are the average and transient torque. $P$ is the output shaft power. $n$ is the expander rotation speed.

The scroll expander with adequate lubrication was assumed. The friction torque had no affect on the output torque generated from the gas force driving the orbiting scroll. Figure 4 shows the variation of transient torque with the crankshaft angle in one working cycle. The transient torque has a nonlinear increase between $0^\circ$ and $300^\circ$, where the torque of the 1500 r/min case is obviously lower than the 1000 r/min case. However, the torque decreasing rate is less than the rotation speed growth rate.

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**Figure 3.** Variation of inlet and outlet mass flux with time.

**Figure 4.** Variation of torque with the crankshaft angle.
Hence, according to equation (3), the output power of 1500 r/min case is much greater. The torque peak value occurs at about 300° where the suction orifice still connects with one of the expansion chambers and obviously asymmetric pressure distribution exists between the two expansion chambers. As is shown in figure 4, the maximum torque occurred at 300°. And with the increase of crankshaft angle, the torque shows continuous decrease between 300° and 360°. Consequently, the great fluctuation of transient torque exists in one working cycle, which is detrimental to the working stability. Meanwhile, great fluctuation even increases the vibration and noise intensity of the scroll expander.

**Flow field analysis**

Due to the conjunction of suction, expansion, and discharge, the crankshaft angle between 0 and 2π was taken to describe the flow field distribution and variation. The flow fields at four typical positions of 0, 0.5π, π and 1.5π were analysed at the speed of 1000r/min. As shown in figure 5(a), the crankshaft angle is zero, where the suction orifice only connects with central suction chamber.

3.1.1. Pressure distribution and variation. Figure 5 shows the pressure distribution and variation in the working chambers on a cross section area perpendicular to the z axis where z is equal to 16.5mm. The pressure inside the suction chamber increases first and then decreases with the change of the suction orifice effective flow area and the suction chamber volume. As figure 5(d) shows, in the later stage of suction process, the central suction chamber changes into a couple of crescent shaped closed expansion chambers gradually. However, the suction orifice still connects with one of the expansion chambers. As a result, obviously asymmetric pressure distribution exists between the expansion chambers, which could benefits for driving the orbiting scroll. The pressure in expansion chambers decreases gradually with the crankshaft rotation.

![Figure 5](image1.png) ![Figure 6](image2.png)

**Figure 5.** Pressure distribution on a cross section area perpendicular to z axis: (a) θ=0, (b) θ=0.5π, (c) θ=π, (d) θ=1.5π

**Figure 6.** Pressure distribution on a cross section area perpendicular to x axis: (a) θ=0, (b) θ=0.5π, (c) θ=π, (d) θ=1.5π

Figure 5(a) and (b) show that the pressure inside the discharge chambers suddenly drops due to the connection between discharge chambers and outboard chamber. However, as shown in figure 5(c) and (d), the pressure rises with the reduction of the discharge chamber volume under a certain back pressure. Figure 6 shows the pressure distribution and variation in the working chambers on a cross section area perpendicular to x axis where x is equal to 0 mm. As shown in figure 6(a), the pressure...
inside the suction chamber presents obviously non-uniform distribution due to the suction throttling. Also, the pressure rises along positive z axis. Figure 6(b) shows that the pressure difference between suction chamber and expansion chamber increases and local pressure losses inside the expansion chambers exist near the axial clearance leakages. The pressure non-uniform distribution in the expansion chambers are shown in figure 6(d). As a result, the driving forces are different on different height of scroll wraps, which has great influence on the overturning moment.

3.1.2. Velocity vector distribution and variation. The velocity distribution and variation are influenced by the rotation of the orbiting scroll and the leakage flows. The suction throttling and clearance leakages flows are observable in figure 7 and figure 8. The radial clearance leakage flows are observed in figure 7. The leakage flow intensity changes dynamically between working chambers with the variation of pressure difference. The leakage flows would reduce the expander volume efficiency obviously and induce eddies to form. In addition, the movement of the orbiting scroll has great influence on the flow at the discharging port where large flow resistance exists. As figure 8 shows, large scale eddies exist in working chambers and change continuously. The eddies could cause the increase of the fluid internal friction losses. Velocity vector along positive z axis changes obviously with the variation of the suction orifice effective flow area.

![Velocity vector distribution on a cross section area perpendicular to z axis: (a) \( \theta=0 \), (b) \( \theta=0.5\pi \), (c) \( \theta=\pi \), (d) \( \theta=1.5\pi \)](image-a)

![Velocity vector distribution on a cross section area perpendicular to x axis: (a) \( \theta=0 \), (b) \( \theta=0.5\pi \), (c) \( \theta=\pi \), (d) \( \theta=1.5\pi \)](image-b)

3.2.3. Temperature distribution and variation. As shown in figure 9 and figure 10, the temperature drop is about 8K to 15K in one working cycle. Temperature distribution inside the working chambers is obviously non-uniform.
Figure 9. Temperature distribution on a cross section area perpendicular to z axis: (a) $\theta=0$, (b) $\theta=0.5\pi$, (c) $\theta=\pi$, (d) $\theta=1.5\pi$

Figure 10. Temperature distribution on a cross section area perpendicular to x axis: (a) $\theta=0$, (b) $\theta=0.5\pi$, (c) $\theta=\pi$, (d) $\theta=1.5\pi$

4. Conclusion

The three-dimensional numerical simulation of a scroll expander applied in organic rankine cycle waste heat recovery system has been accomplished by dynamic mesh technology. The results provide a fundamental understanding about the unsteady flow inside the scroll expander and the variation of the expander transient performances. Transient output torque and the mass flux have periodic fluctuations during the working cycles. Asymmetric and non-uniform distributions of pressure, velocity, and temperature exist inside the scroll expander. Dynamic clearance leakage flows and inlet orifice throttling exist during the working cycles.

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