Effect of different startup modes on the cavitation performance in Rotary lobe pump

D S Guo¹, Y B Li¹²

¹ College of Energy and Power Engineering, Lanzhou University of Technology, Lanzhou 730050, China
² Key Laboratory of Fluid machinery and Systems, Gansu Province, Lanzhou 730050, China

E-mail: liyibin58@163.com

Abstract. This present paper therefore used the dynamic mesh of FLUENT, RNG $k$-$\varepsilon$ turbulence model and Z-wart-Gerber-Belamri cavitation model are used to simulate a rotary lobe pump with unsteady method. Meanwhile the instantaneous flow field distribution and its internal cavitation of rotary lobe pump are analyzed. We also gave the judgment of incipient cavitation. At the same time the cavitation area and the inlet pressure of the lobe pump are determined under the conditions of incipient cavitation, critical cavitation, developing cavitation and complete cavitation. The results show that the bubble occurs firstly in the rotor meshing of the suction cavity, and with the rotation of the rotor, the cavitation is brought into the adjacent transition region, forming the attached cavitation bubble. In different linear startup modes, it can be seen that with the increase of rotational speed, it is easier to form a local low-pressure region at the meshing gaps between two rotors and the pump cavity. When the startup time is 0.1s, the diffusion of the low-pressure region of the inlet section is effectively suppressed, so the volume of the bubble is obviously reduced. Moreover, the critical NPSHc of the lobe pump is 3.59m and the critical cavitation point flow rate is 20.85m³/h, which is in good agreement with the experimental data.

1. Introduction

Rotary lobe pumps, such as vacuum pumps, blower pumps and hydraulic pumps as a kind of rotary volumetric pumps are widely used in industry. However, it is found that there is a problem of gas-liquid two-phase flow in the working process of the rotary lobe pump, which has a direct effect on the working performance of the lobe pumps. Cavitation is a specific flow phenomenon of liquid, when the local pressure in the liquid flow is lower than the saturated vapor pressure at the corresponding temperature, the vapor is developed rapidly under pressure, then the main flow is contracted and destroyed when flowing through the high-pressure region. Incipient, development and fracture are three important stages of cavitation. The cavitation phenomenon will lead to the decrease of the head and flow rate of the lobe pump, and then cause the changes of the performance of the pump, so improving the cavitation performance is very important for the lobe pumps, when it’s running in safely, steadily and efficiently.
In recent years, with the development and widely used of Computational Fluid Dynamics (CFD) technology, domestic and foreign scholars have introduced the dynamic meshes technology to solve the problems of computational domain change, realizing the CFD numerical calculation of the flow field inside the rotary lobe pumps [1-4]. Huang [5] used the k-ε model to simulate a three-lobe roots blower and compared the results of this simulation with those from semi-empirical formulas in terms of their non-uniformity of outlet flows. Hwang [6-7] proposed a new rotor profile with a variable trochoid ratio and investigated how to achieve higher volumetric efficiency and higher sealing performance. Y B Li et al. [8] taking the two-vane involute rotary lobe pump as the research object, using RNG k-ε turbulence model and dynamic grid technology, the internal flow field of the lobe pump is numerically simulated, and the influence of pressure angle on the transient characteristics of the flow field inside the rotary lobe pump is analyzed. Kethidi [9] conducted further studies on the influence of turbulence modelling on the CFD predictions of local velocity fields in twin screw compressors. Moreover, the transient behavior in centrifugal pumps have been researched experimentally by various methods: closure of valves or fast opening [10], fast startup sequences [11-13] and rotation speed fluctuations [14]. In previous researches, it has been found that the rapid transient results in obvious unsteady effects including the pressure pulsation and flow-rate fluctuations, which may be preponderant before the quasi-stable flow evolution. Arjeneh [15] measured the local pressure losses inside the suction plenum of a screw compressor, and then used the results to evaluate predictions of pressure losses in the suction plenums of compressors derived from 3-D CFD calculations. del Campo, D. et al. [16] proposed a simplified two-dimensional numerical method based on computational fluid dynamics (CFD) to study the cavitation effect in cavity. What the above literatures does make clear, however is that most of the research on cavitation flow are focused on centrifugal pumps [17-18]. The cavitation flow characteristics in rotary lobe pumps are rarely involved, so it is impossible to reveal more accurately the performance of the rotary lobe pump and the flow phenomenon inside the rotary lobe pump, meaning that topics related to the cavitation of lobe pumps is important.

This paper therefore bases on the dynamic mesh techniques in the commercial CFD software ANSYS Fluent, RNG k-ε turbulence model and Z-wart-Gerber-Belamri cavitation model to simulate and analysis the transient behavior during different startup modes in a rotary lobe pump, especially focusing on the structure of the flow field inside the lobe pump and the effect of cavitation on the performance. This is the first effort to understand its effect on the performance of rotary lobe pump.

2. Rotor geometry mathematical model

2.1. Mathematical model of the curve

This present paper we introduce higher order curves to smooth transition between two curves. The pump rotor adopts three-leaf inner cycloid-higher order curves-cycloid-outer cycloid rotor profile. As shown in Figure 1, AB segment is an outer cycloid (coloured red), BC segment is higher order curves (coloured blue), CD segment is inner cycloid (coloured green).
Assumes that the line of the connect the base circle and the rolling round center and the x axis angle is $\theta$. The AB and CD segment profile equations are respectively (1) and (2).

\[
\begin{align*}
x &= \frac{(2z + 1)R_m\cos\theta + R_m}{2(z + 1)}
\end{align*}
\]
Definition: $\pi/(2z) \leq \theta \leq \pi/z$.

\[
\begin{align*}
x &= \frac{(2z - 1)R_m\cos\theta - R_m}{2(z + 1)}
\end{align*}
\]
Definition: $0 \leq \theta \leq \pi/(2z)$.

$R_m$: Rotor tip radius, $z$: Number of rotors.

Set the high-order transition curve equation as $\rho(\phi)$, and the equation can be expressed as:

\[
\rho(\phi) = a_0 + a_1\phi + a_2\phi^2 + a_3\phi^3 + a_4\phi^4 + a_5\phi^5 + a_6\phi^6
\]

(3)

Where $a_0$-$a_6$ are the design variable. In order to make the curve smooth, the acceleration and other indexes are taken into account, the following equations are optimized.

\[
\min W_1[J_{max}] + \max W_2[a_{max}]
\]

(4)

Where, $W_1$ and $W_2$ are weighted factor.

According to the geometric properties of Figure 1, the following edge constraints are given:

\[
\begin{align*}
\rho(0) &= \overline{OB} \\
\rho(\beta - \mu) &= \overline{OC} \\
v(0) &= v(\beta - \mu) = a(\beta - \mu) = a(0) = J(\beta - \mu) = J(0) = 0
\end{align*}
\]

(5)

According to the boundary constraint conditions of the equation (5), the optimal solution is obtained through the computer software programming calculation, the MATLAB software is used and finally a high-order curve expression is obtained.

2.2. Modeling and parameters

The numerical calculation is based on unsteady calculation and considering the symmetry of the three-leaf rotary lobe pump, the radial cross-section flow in the three-dimensional model is basically the same as that in the two-dimensional. The two-dimensional model can meet the demand of the flow field analysis. As shown in Figure 2, the two-dimensional model is used in the calculation in this paper.

![Figure 2. Mesh two-dimensional of lobe pump.](image)
Main structure and operating condition parameters of the rotor are shown in Table 1:

| Blade number | Tip diameter (D/mm) | Center distance (L/mm) | Rotor thickness (T/mm) | Inlet and outlet diameter (d/mm) | Speed n/(r·min⁻¹) |
|--------------|---------------------|------------------------|------------------------|----------------------------------|-------------------|
| 3            | 159.6               | 120                    | 100                    | 80                               | 350               |

### 3. Numerical analytical method

#### 3.1. Dynamic meshes model and mesh generation

In FLUENT, the dynamic mesh model can be used to simulate the flow of the basin shape with time due to the movement of the basin boundary, and the reverse rotation motion of the lobe pump blade rotor meshing. The dynamic mesh computing model is expressed as:

$$\frac{d}{dt} \int_V \rho \phi (u - u_s) dA = \int_V \Gamma \nabla \phi dA + \int_V S_\phi dV$$

Where $u$ is speed, $u_s$ is the deformation velocity for dynamic mesh, $\Gamma$ is diffusion coefficient. $S_\phi$ is flux source term, $\partial V$ is the boundary of the control volume. The first term can be first order backward difference and the equation is expressed as:

$$\frac{d}{dt} \int_V \rho \phi dV = \frac{(\rho \phi V)^{n+1} - (\rho \phi V)^n}{\Delta t}$$

Where $n$ and $n+1$ are represents the current and immediate time steps, respectively, which are in order to achieve the progress of the time step.

For the two-dimensional computing region, the dynamic mesh technology combined with spring smoothing model and local remeshing model is adopted. The whole parts adopt triangle mesh with good adaptability (as shown in Figure 2). When the grids of rotor basin are maintained about 220,000. The cells skewness is controlled within 0.75, and the calculation results tend to be stable. The two rotors are set as moving boundary, the two ends of the pump body are set as rigid bodies, and the motion mode of the rotor is written by User Defined Function (UDF).

#### 3.2. Governing equation and cavitation model

FLUENT is used to calculate unsteady two-phase flow. The coupling model of pressure and velocity is PISO, and the two-phase flow is based on the mixture model including the cavitation model.

##### 3.2.1. Continuity equation

$$\frac{\partial}{\partial t} \rho_m + \nabla \cdot (\rho_m V_m) = 0$$

$$\frac{\partial}{\partial t} (\alpha_k \rho_k) + \nabla \cdot (\alpha_k \rho_k V_k) = \dot{m} \quad (k=I, v)$$

$$\rho_m = \alpha \rho_1 + (1-\alpha) \rho_v$$

Where $\dot{m}$ is the interphase mass exchange rate, subscript $m$ denotes mixed phase, $I$ denotes liquid phase, $v$ denotes vapor phase.

The equation for calculating the interphase mass exchange equation is expressed as:
\[\left\{ \begin{array}{l} m = \frac{C_e}{\mu} \frac{\sqrt{k}}{\sigma} \rho_i \sqrt{\frac{2(p'_e - p)}{\rho_i}} \\
- C_c \frac{\sqrt{k}}{\sigma} \rho_i \sqrt{\frac{2(p - p'_e)}{3\rho_i}} \end{array} \right. \tag{11}\]

Where \(\sigma\) is surface tension coefficient, \(k\) is turbulent kinetic energy, \(C_e, C_c\) are empirical coefficient. In calculation, \(C_e, C_c\) are regarded as constant: \(C_e = 0.02, C_c = 0.01\).

\[p'_e = \frac{1}{2}(p_v + 0.39 \rho_n k) \tag{12}\]

3.2.2. Cavitation model

Cavitation model adoption Z-wart-Gerber-Belamri model, the equation is expressed as:

\[R_e = F_{\text{vap}} \frac{3 \alpha_{\text{nd}} (1 - \alpha_r) \rho_c}{R_B} \sqrt{\frac{2(p_c - p)}{3\rho_i}} \tag{13}\]

where, \(p < p_v\).

\[R_e = F_{\text{cond}} \frac{3 \alpha_c}{R_B} \sqrt{\frac{2(p - p_c)}{3\rho_i}} \tag{14}\]

where, \(p > p_v\).

Where nucleation position volume fraction \(\alpha_{\text{nd}}\) is \(5 \times 10^{-4}\), \(R_B\) is diameter of the bubble, is \(1 \times 10^{-3}\) m, \(p\) and \(p_c\) are flow field pressure and vaporization pressure, respectively. \(F_{\text{vap}}\) and \(F_{\text{cond}}\) are empirical correction coefficients corresponding to the evaporation and condensation processes, the values are 50 and 0.01. The \(F_{\text{vap}}\) and \(F_{\text{cond}}\) are not equal because the condensation process is usually much slower than the evaporation process.

The calculation boundary conditions are given as follows: Under the design conditions in which cavitation calculation results are not as the initial conditions for the cavitation calculation. Then we can change the inlet pressure sequentially and make the lobe pump cavitation. The convergence conditions are as follows: the volume flow rate of inlet and outlet fluctuate periodically.

4. Results and Analysis

4.1. Calculation of NPSH

The equation of the NPSH is expressed as:

\[\text{NPSH} = H_1 + \frac{P_a}{\rho g} - \frac{P_v}{\rho g} \tag{15}\]

Where \(H_1\) is the gross head of inlet, \(P_a\) is the atmospheric pressure, \(P_v\) is the saturated vapor pressure of medium at certain temperature.

4.2. The different startup modes

The different linear start modes rotation are defined as:

\[N_L(t) = \begin{cases} \frac{N \times t}{T_p}, & t \leq T_p \\ N, & t > T_p \end{cases} \tag{16}\]
Where $N$ is the rated speed after startup, $t$ is the nominal acceleration time in this present paper, and is defined as the time it takes for the speed to rise from stop to rated speed. Figure 3 shows the variation of the rotational speed in different startup modes.

![Variation of the rotational speed in different startup modes.](image)

**Figure 3.** Variation of the rotational speed in different startup modes.

### 4.3. Comparison between numerical and test values

The non-cavitation test data for the lobe pump are shown in the following Table 2.

| Inlet (kPa) | Outlet (MPa) | Differential (MPa) |
|------------|--------------|--------------------|
| 1.6        | 0.106        | 0.104              |
| 2.7        | 0.199        | 0.196              |
| 3.5        | 0.308        | 0.305              |
| 4.0        | 0.416        | 0.412              |
| 4.6        | 0.501        | 0.495              |
| 4.6        | 0.598        | 0.593              |
| 5.6        | 0.701        | 0.695              |
| 5.5        | 0.797        | 0.792              |

**Table 2.** Test at different operating conditions.

Performance test of a lobe pump through a closed test bed are shown in Figure 4 below. The performance and cavitation test of lobe pump are carried out according to the GB/T12785 (Test methods for submersible pumps) under the condition of normal temperature and water. The inlet pressure is controlled by adjusting the manual ball valve so as to change the value of NPSH.

![Closed testing bed.](image)

**Figure 4** Closed testing bed.

In numerical calculation, by giving the corresponding inlet pressure of the rotor pump to reduce the NPSH while keeping the outlet pressure constant as 0.4MPa. The cavitation flow in the pump under different $P_{\text{inlet}}$ of -70kPa, -80kPa, -90kPa and -100kPa respectively has been calculated.
Figure 5 shows the efficiency/volume flow rate-pressure difference curves obtained from numerical prediction and experiment of the rotary lobe pump under no cavitartion conditions. The results show that the efficiency of the numerical calculation under different pressure differential conditions is higher than that of the test results. The error is less than 3%, the main reason is that the pressure difference has a great influence on its volumetric efficiency, and with the pressure difference increasing, the volume loss increases sharply. It can be seen from Figure 6 that the flow rate obtained by the numerical calculation is in good agreement with the test data, and it can be concluded that the unsteady numerical results of single-phase unsteady calculation using RNG $k-\varepsilon$ turbulence model are acceptable. It can also be used as the initial flow field of cavitation calculation.

The flow attenuation characteristics, such as Figure 7, that is the numerical and experimental flow characteristic curves of the lobe pump under differential conditions. When the inlet NPSHr of the pump is 4.5m, the flow rate does not decrease obviously, it is said that there is no obvious cavitation phenomenon in the pump at this time, however when the NPSHr at the inlet of the pump drops to 3.59m, the flow rate decreases obviously, and the flow rate drops by about 17%. At this time, there is obvious cavitation in the pump, which is generally considered as the NPSHc point. When the NPSHr at the inlet of the pump continues to decrease, the flow rate is declining until the pump inlet cannot suction liquid. At very this time the cavitation in the pump has been very serious.

4.4. Cavitation Evolution and Flow Field Analysis of the normally startup

In order to analyze the cavitation evolution process in the lobe pump. When the fluctuation of outlet and inlet flow rate is stable, which indicates that the pump has been running smoothly. Take the inlet pressure at -70kPa, -80kPa, -90kPa and -100kPa respectively, under different conditions at the same time (the outlet flow rate reaches the maximum), when the internal pressure and bubble distribute. As shown in Figure 8(a), with the decrease of the inlet pressure of the pump, the low-pressure region first appears in the rotor meshing and the gaps with the pump cavity, and the pressure in the transition cavity decreases gradually. As the inlet pressure continues to decrease, the low-pressure region begins to move from the inlet to the pump cavity. Then it occupies the whole suction cavity and the entire transition cavity.
As shown in Figure 8(b), the cavitation process can be divided into: (1) The bubble first occurs in the suction cavity rotor meshing and the gap between the rotor and the pump cavity. (2) With the inlet pressure decreasing, the bubble region expands outward with the outer edge of the rotor, meanwhile near the front end of rotor meshing there is also bubble area. At the same time the flow rate of the pump decreased significantly. (3) As the inlet pressure continues to decrease, the bubble area has spread to the transition cavity, forming an attached bubble, which accounts for about 1/5 of the suction cavity. (4) When the pressure of inlet drops to -100kPa, the bubble almost occupies the whole suction cavity, and the cavitation wake has completely blocked the suction section of the lobe pump.

4.5. Cavitation Evolution in lobe pump under different startup modes

Figure 9 shows the distribution of bubbles inside the lobe pump in different startup modes. It can be seen that with the increase of rotational speed, it is easier to form a local low-pressure region at the meshing gaps between two rotors and the pump cavity. At the same time, the range of low-pressure region shows a tendency of gradual diffusion, and the bubble at the gaps increase gradually with the diffusion of low-pressure region. The diffusion of low-pressure region is more pronounced at an inlet pressure of -80kPa and a startup time of 0.05s (As shown in Figure 9 (a)). When the startup time is 0.1s, the diffusion of the low-pressure region of the inlet section is effectively suppressed, so the volume of the bubble is obviously reduced (As shown in Figure 9 (b)). As can be seen in Figure 10(a), the bubble volume fraction decreases significantly with the prolonged the startup time.

Figure 9. Cavitation process under different startup modes of $P_{inlet}$ is -80kPa.

Figure 10. Vapor volume fraction (Volume Average) of different startup modes.
As shown in Figure 11, when the pressure of inlet is further reduced to -90kPa, the distribution range of the bubbles further expands as the range of the low-pressure region further expands. It can be seen that when the startup time is 0.05s, the bubble diffuses into inlet section, and with the increase of rotational speed, the bubble volume fraction also increases (Figure 11(a)). When the startup time is 0.1s, the growth rate of bubble volume fraction is obviously slowing down (Figure 10(b)), and with the increase of rotational speed, there is no bubble blockage in the inlet section (Figure 11(b)).

The reason for this phenomenon is that because the increase of the rotational speed, a local low-pressure is easily formed at the suction end of the lobe pump, so that the bubble increase. Meanwhile, the inlet flow is more easily disturbed. However, with the prolongation of startup time, the increase rate of rotational speed is decreases, which can effectively avoid the formation of local low-pressure at the suction end, inhibiting the bubble formation rate and the diffusion range.

![Figure 11. Cavitation process under different startup modes of P_{inlet} is -90kPa.](image)

5. Conclusion
RNG $k-\varepsilon$ turbulence model and the dynamic meshes have been used to compute the two-dimensional turbulence flow field distribution and analysis its internal cavitation of rotary lobe pump. The conclusions as follows:

1) The CFD calculation and experimental is in good agreement, which verifies the reliability of the numerical model. Based on this, using the RNG $k-\varepsilon$ turbulence model to predict the performance of the rotary lobe pump has good accuracy and reference value. And the cavitation process in lobe pump can be divided into incipient cavitation, critical cavitation, developing cavitation and complete cavitation.

2) The NPSHc of the lobe pump is 3.59m and the critical cavitation point flow rate is 20.85m³/h by numerical calculation and test. The flow rate drops by about 17%. The bubble first occurs in the gaps between the two rotors and between the rotors and pump cavity. Then the bubble area has spread to the transition cavity forming an attached bubble.

3) Compare with the two different linear startup modes, when the startup time is 0.1s, the diffusion of the low-pressure region of the inlet section is effectively suppressed, so the volume of the bubble is obviously reduced. Moreover, the growth rate of bubble volume fraction obviously slows down.
Acknowledgements

The authors are grateful to the financial support from the National Natural Science Foundation of China (Research Project No.51369015).

References

[1] Vogelsang H, Verhijsldonk B, Tiirk M, and Hornig G Pulsation problems in rotary lobe pumps 1999 World Pumps, 389, pp 45-52.
[2] Močilana M, Husárb Š, Labajc J, and Žmindáka Milan 2017 Non-stationary CFD simulation of a gear pump. Procedia Engineering 177 pp 532-539.
[3] Valdès L C, Barthod B, and Perron Y L 1999 Accurate prediction of internal leaks in stationary dry roots vacuum pumps. Vacuum 52 (4) pp 451-459.
[4] Mimmi G and Pennacchi P 1999 Analytical model of a particular type of positive displacement blower. Proc. Inst. Mech. Eng., Part C J. Eng. Mech. Eng. Sci., 213 (C5) 517-526.
[5] Huang Z F, and Liu Z X 2009 Numerical study of a positive displacement blower, ARCHIVE Proc. Institution Mech. Eng. Part C J. Mech. Eng. Sci., 223 (10) pp 2309-2316.
[6] Hwang Y W and Hsieh C F 2006 Study on high volumetric efficiency of the roots rotor profile with variable trochoid ratio. Proc. Institution Mech. Eng. Part C J. Mech. Eng. Sci., 220 (9) pp 1375-1384.
[7] Hsieh C F and Hwang Y W 2007 Study on the high-sealing of Roots rotor with variable trochoid ratio. J. Mech. Des., 129 (12) pp 1278-1284.
[8] Li Y B, Jia K, Meng Q W and Shen H 2013 Flow simulation of the effects of pressure angle to lobe pump rotor meshing characteristics. IOP Conf. Series: Materials Science and Engineering, 52 032022.
[9] Kethidi M, Kovacevic A, Stosic N and Smith I K 2011 Evaluation of various turbulence models in predicting screw compressor flow processes by CFD. 7th International Conference on Compressors and their Systems, pp 347-357.
[10] Tanaka T and Tsukamoto H 1999 “Transient behavior of a cavitating centrifugal pump at rapid change in operating conditions—Part 1: Transient Phenomena at Opening/Closure of Discharge Valve,” ASME J. Fluids Eng., 121 pp 841-849.
[11] Tanaka T and Tsukamoto H 1999 “Transient behavior of a cavitating centrifugal pump at rapid change in operating conditions—Part 2: Transient Phenomena at Pump Startup/Shutdown,” ASME J. Fluids Eng., 121 pp 850-856.
[12] Tanaka T and Tsukamoto H 1999 “Transient behavior of a cavitating centrifugal pump at rapid change in operating conditions—Part 3: Classifications of Transient Phenomena,” ASME J. Fluids Eng., 121 pp 857-865.
[13] Picavet A and Barrand J P 1996 “Fast start-up of a centrifugal pump experimental study,” Proceedings of the Pump Congress, Karlsruhe, Deutschland.
[14] Tsukamoto H, Yoneda H and Sagara K 1995 “The response of a centrifugal pump to fluctuating rotational speed,” ASME J. Fluids Eng., 117 pp 479-484.
[15] Arjeneh M, Kovacevic A and Rane S, 2015 Numerical and experimental investigation of pressure losses at suction of a twin-screw compressor. IOP Conf. Series: Materials Science and Engineering, 90 012006.
[16] del C D, Castilla R and Raush G A 2012 Numerical analysis of external gear pumps including cavitiation. Journal of Fluids Engineering, 134 (8) 11051.
[17] Dupont P, Okamura T and Dupont P 2003 Cavitating flow calculation in industry. International Journal of Rotating Machinery, 9(3) pp 163-170.
[18] Zwart P, Gerber A G and Belamri T 2004 A two-phase model for predicting cavitacion dynamics[C]// Proceedings of ICMF 2004 International Conference on Multiphase Flow, Yokohama, Japan, pp 1-11.