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Numerical Study on Rotor Deformation of Multiphase Twin-Screw Pumps Under High Gas Volume Fraction Conditions

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ABSTRACT

Multiphase pumping with twin-screw pumps is a relatively new technology that has been proven successful in a variety of field applications. It has three advantages such as less environment pollution, few separation equipments and more convenient operation than the conventional system. Despite many advantages of this technology, some problems have been encountered when operating under conditions with high gas volume fractions (GVF). While twin-screw multiphase pump is operating under high GVF conditions, the inner temperature of the pump increases obviously. The clearances between rotors change greatly and influence the volumetric efficiency of the twin-screw multiphase pumps. In some severe conditions, it may cause the pump damage. In this paper, the actual force and thermal boundary conditions are proposed through further investigations of pressure distributions and heat transfer. And then the screw rotor deformation and temperature field are calculated under different GVF conditions with ANSYS software. The results indicate that the main deformation of screw rotors is thermal deformation and the maximum radial deformation occurs on the top of the rotor in the discharge port. Through analyzing the influence of screw rotor deformation on the clearance, it can be realized that the greatest clearance changes are in the root of the rotor, followed by circumferential clearance, and there is no changes in flank clearance.

1. INTRODUCTION

In petroleum industry, twin-screw pumps present a promising method to meet the requirements on pump performance and reliability onshore and offshore. Multiphase pumping provides a more efficient tool for transports of mixture include oil, water and gas, occasionally sand, natural gas hydrates and waxes (Cooper *et al*., 1999) . More and more petrochemical companies worldwide are attracted to the use of multiphase pumps for boosting multiphase flow. However, when handing gas-liquid mixture with very high gas volume fraction, twin-screw multiphase pumps have an obvious temperature increase. The clearances between rotors change greatly, and which causes serious influence on volumetric efficiency of the pumps. In some severe conditions, it may cause pumps damage.

Nakashima(2006) presents temperature distributions of the rotor and shell before and after the slug flow happened in twin-screw mixing pumps, based on the quality and energy conservation and the method of numerical simulation. Several former investigations perform the theoretical and experimental analyses of pumping behaviors of multiphase twin-screw pumps with very high gas volume fractions. Temperature distribution and pressure distribution rules are got by mathematical model as well as experimental investigation.

Tie-yu G. et al.(2011) presented a model to calculate deformation of a twin screw compressor, which provided a analysis for different materials under the same boundary conditions and a material for manufacturing rotors in a twin screw compressor with high gas volume fractions. Stosic et al. (2005) developed a numerical method to study the flow and deformation in the twin screw multiphase pump.

Most literature pay attention to the relationship of pressure distribution and operating performance, as well as the rotor deformation under the action of thermal effect alone or lower gas volume fractions, however, there is only few research about the rotor deformation under comprehensive effect of heat and pressure in high gas volume fractions. In addition, the medium temperature is directly loaded in the surfaces of screw rotor, which means the heat exchange between the screw rotor and medium is ignored when confirming the temperature boundary conditions. As a result, further theoretical analysis was made in this paper to investigate the effects of rotor deformation and the change of rotor gaps.

2. ANALYSIS MODELS

2.1 Three-Dimensional Model

To completely characterize a screw rotor of multiphase twin-screw pump when handling high GVF mixtures, three-dimensional model were carried out with SolidWorks or PRO/E. Then import the model into ANSYS software and make some appropriate modification to be a complete modeling. The three-dimensional model can be seen in Figure 1. The outer diameter and inner diameter are 128 mm and 82 mm, lead 53 mm, and total length is 868 mm. A double cycloidal-arc symmetrical profile, which combined with lines and meshing circular arc was used in research (Cao *et al*., 1999). The operating conditions of twin-screw pump are presented in Table 1.

Table.1: The operating conditions of twin-screw pump

Flow rate $/m^3 \cdot h^{-1}$	Inlet temperature /K	Inlet pressure $/MPa$	Outlet pressure $/MPa$	Rotate speed/ $r \cdot \min^{-1}$
50	308.15	0.2	1.2	1205

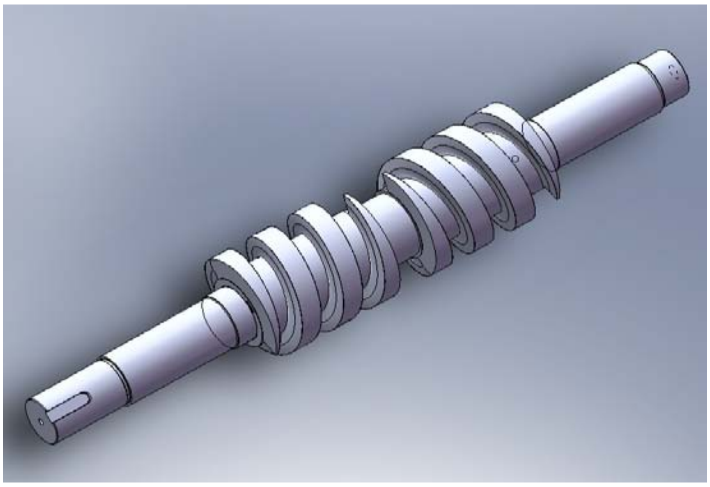


Fig. 1:Three-dimensional model

2.2 Finite Element Grid

A half of screw rotor was conducted with finite element mesh division in research as shown in figure 2. The same finite element grid was applied in both thermal analysis and structure analysis to get the temperature distribution and the thermal deformation. In finite element analysis for screw rotor, all chamfers and key grooves of both ends were removed as a simplified treatment, which has a small influence on the overall performance. On the one hand, it can reduce the workload of finite element analysis, on the other hand it can also reduce the deform possibility of the grid caused by the meshing, avoiding calculation results distortion.

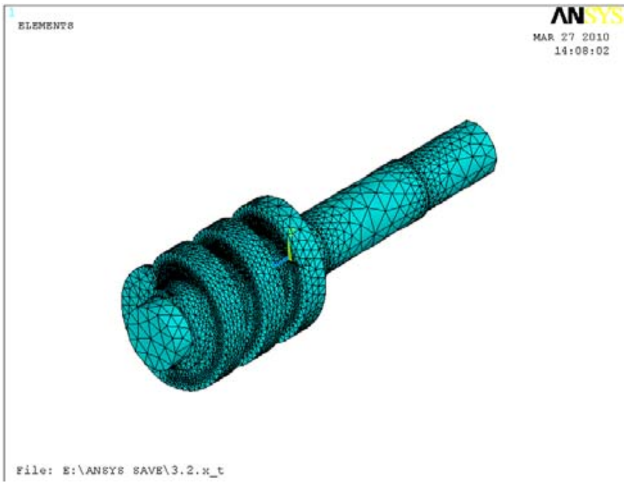


Fig. 2:The finite element grid of screw rotor

2.3 Leakage Models

Due to the complexity of spindle geometry, flow condition, leakage path and working fluids, which may be single liquid, low air rate mixed fluid, high air rate mixed fluid as well as gaseous phase in the screw pump then the leakage path may be single-phase fluid, single-phase gas or air mixture two-phase flow. Therefore different leakage mathematical models are required to describe the possibility at different situations. Those leakage models can be applied to two-phase flow inside the working space according to a method, which is explained in detail in C Feng et al.(2001) .

3. BOUNDARY CONDITIONS

3.1 Pressure

At the inlet of the domain, a total pressure condition was set. This condition is the most accurate due to the inflow energy is defined and the program is allowed to obtain gradients in velocity and pressure. According to the assumption, the pressure in the same screw rotor groove is uniformly distributed load. The pressure distribution in working space can be calculated by the program and the condition of circumferential surface is gradient pressure load of pressure difference between two chambers.

Pressure load conditions are as follows: the suction pressure: 0.2MPa; the pressure gradient of circumferential surface between the suction port and the first airtight chamber : 6.9434MPa m-1; The first airtight chamber pressure: 0.384MPa; the pressure gradient of circumferential surface between the first airtight chamber and the second airtight chamber for: 13.283MPa m-1; The second airtight chamber pressure: 0.736MPa; the pressure gradient of circumferential surface between the second airtight chamber and outlet pressure for: 17.5094MPa m-1; the outlet pressure: 1.2MPa.

3.2 Temperature

The temperature of screw rotor surfaces is not equal to the surrounding medium temperature. Also there is temperature difference between the end of rotor and the medium inlet and outlet, especially under higher gas volume fraction conditions. So convection heat transfer model is applied as the temperature boundary conditions.

The temperature and pressure inside pumps are different every moment in the operating process and the medium properties are changing as the temperature and pressure changes, so the convection heat transfer coefficient is calculated with the relevant pressure and temperature in each time. In this paper the temperature difference between adjacent chambers are regarded as the same and circumferential surface are covered by liquid phase medium (water) leaking from high pressure chamber to low one. According to properties of all states, the heat transfer coefficient of each surface can be calculated by the program mentioned above. When the gas volume fraction is 94%, the property and convection heat transfer coefficient as shown in Table 2.

Table.2:The property and heat transfer coefficient

Items	$\frac{T}{\mathrm{K}}$	$\frac{\lambda\times10^{2}}{\mathrm{W\cdot m^{-1}\cdot K^{-1}}}$	$\frac{\nu\times10^{7}}{\mathrm{m^{2}\cdot s^{-1}}}$	Pr	$\mathrm{Re}\times10^{-4}$	$\frac{h}{\mathrm{W\cdot m^{-2}\cdot K^{-1}}}$
Entrance	308.15	6.2	40.4	4.28	4.75	623
Suction	308.15	2.61	79.4	0.704	1.94	68.8
Inlet	308.15	62.4	7.24	4.8	0.446	247
The top of chamber	312.12	63.1	6.76	4.4	0.478	251
Chamber 1	312.12	2.64	42.2	0.704	3.66	110
The top of chamber	316.09	63.71	6.33	4.065	0.511	255
Chamber 2	316.09	2.66	22.5	0.704	6.85	176
Outlet	320.06	64.27	5.92	3.77	0.545	259
Exit	320.06	6.38	9.2	3.45	20.8	1760

4. RESULTS AND DISCUSSION

4.1 The Deformation of Screw Rotor

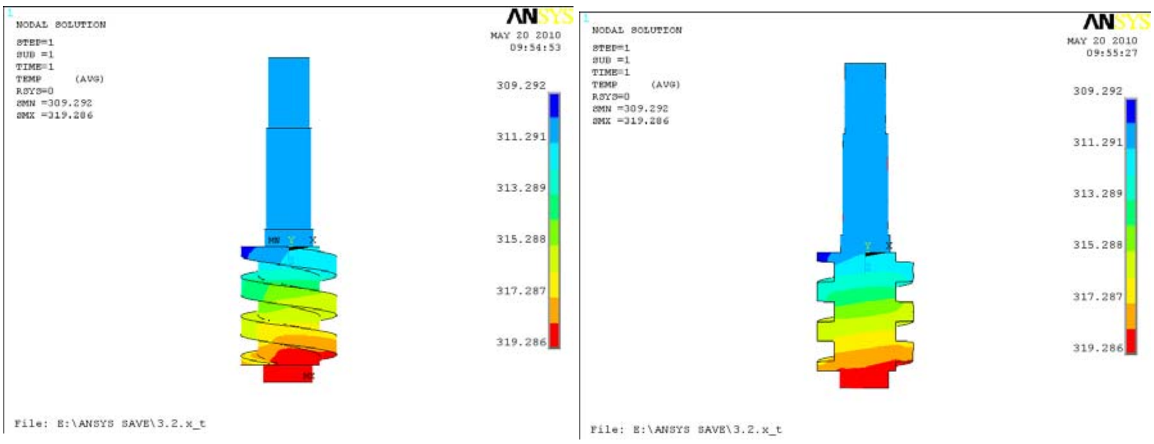


Fig. 3:Temperature distributions of the screw rotor

The temperature distributions of the screw rotor are presented in figure 3, where the highest temperature happens near the outlet of the rotor and the minimum temperature happens near the inlet. With the GVF increases, the convection heat transfer becomes poor between the medium and screw surfaces, so the temperature difference is becoming much bigger. In this paper, it is reasonable to adopt convection heat transfer instead of the traditional method that the temperature of the medium is applied directly to the end of screw rotor.

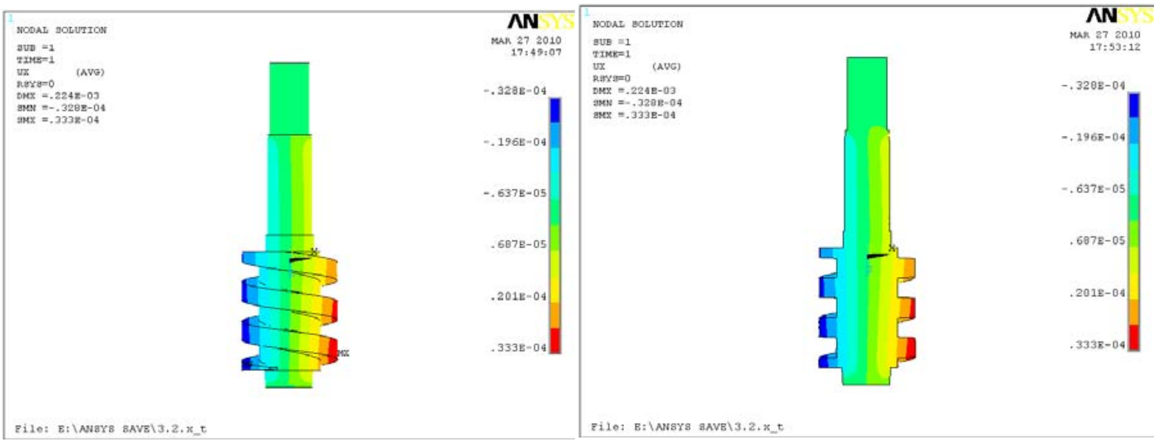


Fig. 4:The deformation of screw rotor in the x direction

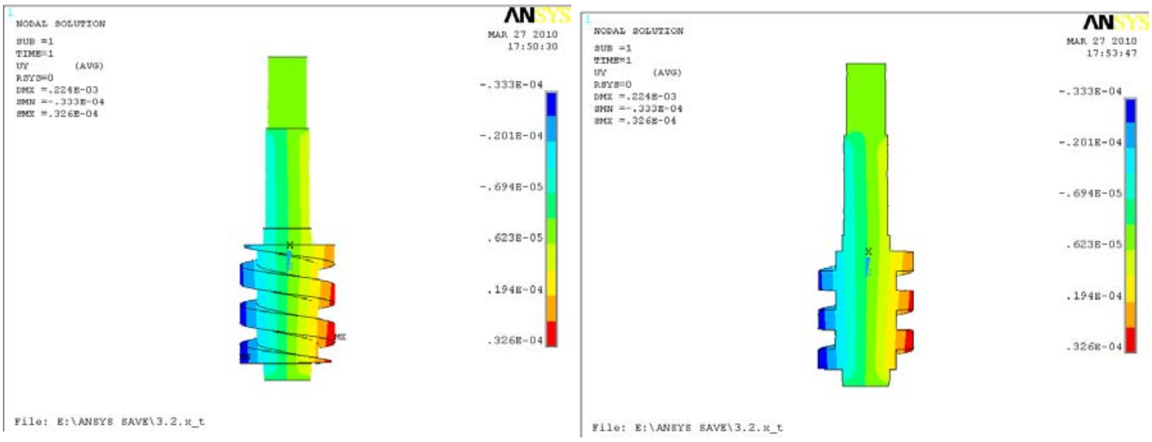


Fig. 5:The deformation of screw rotor in the y direction

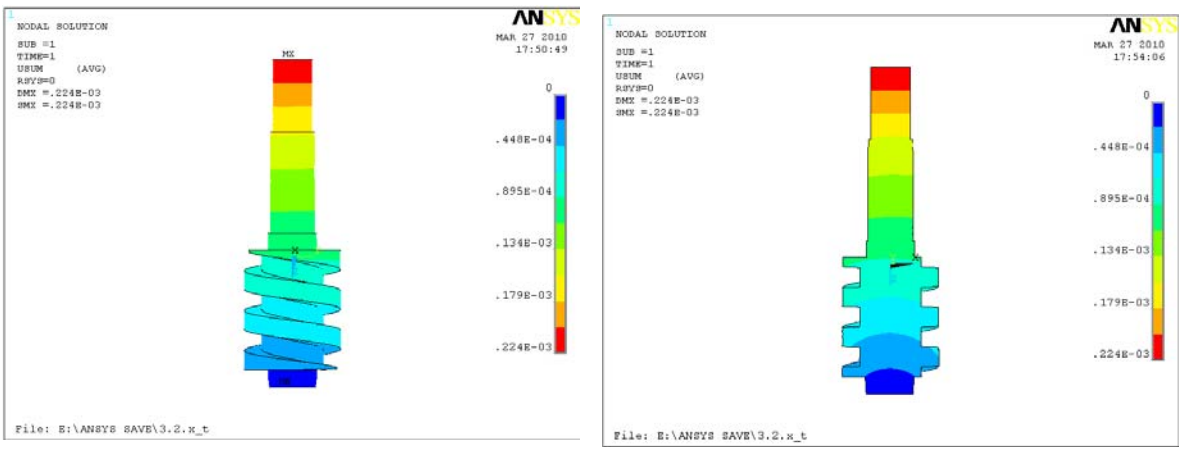


Fig. 6:The comprehensive deformation of screw rotor

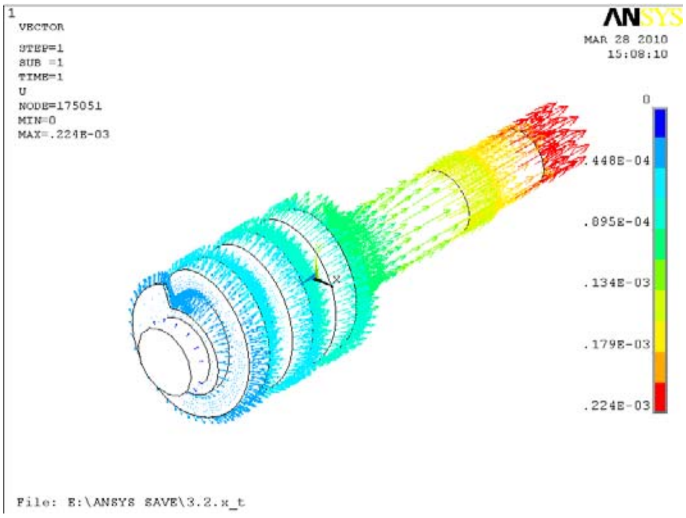


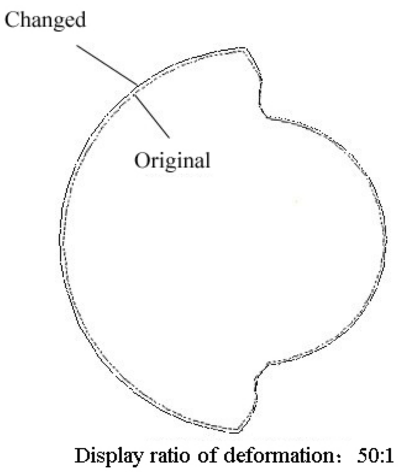
Fig. 7:Vector representation of screw rotor deformation

Figure 4 and figure 5 present the amount of screw rotor deformation in the x, y direction, and the comprehensive deformation of screw rotor is shown in figure 6. The maximum radial deformation, which occurs in the top of the gear near the outlet of rotor, are 0.333×10-4m and -0.328×10-4m in the x direction and -0.333×10-4m and 0.326×10-4m in the y direction. The data above show that the radial deformation of screw rotor is unequal in each direction, this is because the screw rotor temperature distribution is uneven, rotor surface pressure is different and the rotor position is not correct caused by the pressure distribution.

The temperature distribution and pressure distribution make the screw rotor shape changed, but the temperature distribution is the main factors of rotor deformation. The maximum axial deformation occurs at the inlet of the screw rotor, because the screw rotor can only expand free in the entrance and the accumulative effect of the deformation leads the rotor deformation most obvious happened in the face of suction side. The axial dimensions of the screw rotor is far longer than the radial dimensions, so the biggest axial deformation is far bigger than the biggest radial deformation and the biggest deformation is 0.224 x 10-3m. Beside the inherent characteristics of the materials decide the magnitude of rotor deformation, like the modulus of elasticity, poisson's ratio as well as coefficient of thermal expansion, the rotor geometric parameters also play a very significant role in the deformation. As accumulative effect of thermal expansion, when screw rotor with different screw rotor diameter, its biggest radial deformation will be different and axial deformation will also be different with the screw rotor length different. It can be observed from the figure 7 that the deformation in each direction of screw rotor under the gas volume fraction 94%.

4.2 The Deformation of Section

Figure 8 and figure 9 exhibit the rotor profiles change near the outlet side of pump with the gas volume fraction 80% and 99%. The rotor profiles, which expand outward with the comprehensive force caused temperature and pressure, are similar to the results under the GVF increasing from 80% to 99%. The temperature near the outlet is increasing gradually with the increase of the GVF, which lead to the expansion of screw rotor more obviously at the corresponding location. As can be seen from the figures that the rotor deformation is the widest in the 99% GVF condition. The maximum radial deformation, which occurs in the top of the gear near the outlet of rotor, is about 0.30×10-4m in average. The deformation around the big side is bigger than the deformation around small side, which show that the circumferential side of rotor changes more obviously compared with the radial side in agreement with the theoretical analysis.



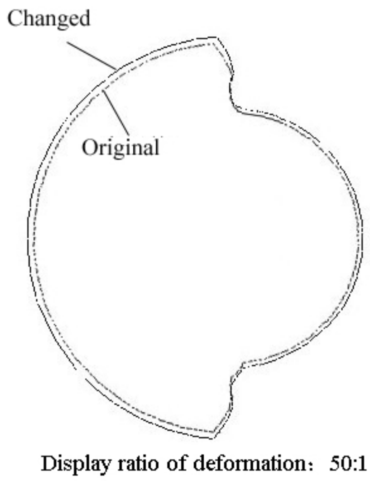


Fig. 8:Rotor profiles change with GVF 80% **Fig. 9**:Rotor profiles change with GVF 99%

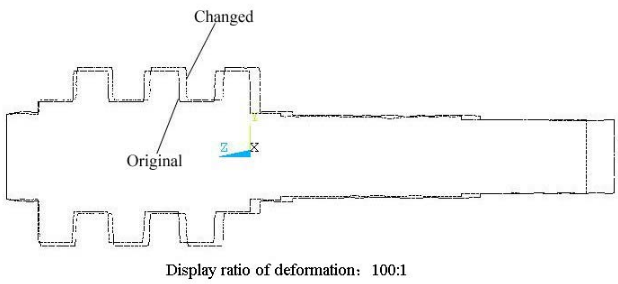


Fig. 10:Rotor axial cross section change with GVF 80%

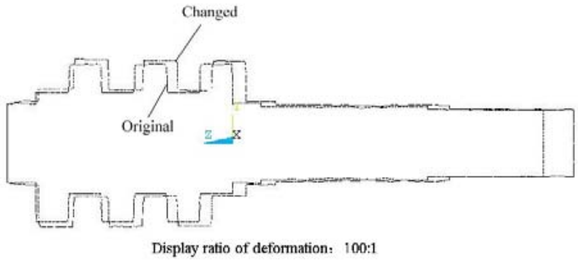


Fig. 11:Rotor axial cross section change with GVF 99%

The rotor axial cross section change with GVF of 80% and 99% are showed in figure 10 and figure 11. The magnitude of deformation aren't the same in radial and axial direction and rotor axis deformation is more obviously near the import, which shows the accuracy of the theoretical conclusion mentioned.

4.3 The Rotor Gaps Change

The screw rotor leakage gaps, which can be classified as radial gap, flank gap and circumferential gap, play an important role in the performance of twin-screw pump. Determining the rotor gaps should consider the various factors, which mainly includes the rotors deformation and expand when they are heated in the operation process, the clearance uniformity of different tooth, the relative speed of different surfaces for different requirements, and the driver domain between rotors.

Due to the expansion of the heated screw rotor, leakage gaps influence will occur, the following changes:

1) Radial gap: The mesh clearance between the top and the root changes obviously, for the deformation direction of the expansion is opposite in the operating process. This mesh clearance change is the superposition of two deformations, as shown in figure 12 below.

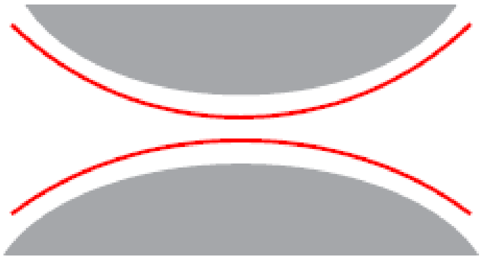


Fig. 12:Radial gap changes (the shaded shows change before, red line shows changes)

2) Flank gap: The deformation in axial direction of two screw rotors is similar when they are heated. Although the thickness of the teeth increases under the action of thermal expansion, the distance of the two teeth also becomes bigger. So flank gap do't have change apparently.

3) Circumferential gap: The gap between surrounding housing and the top of tooth reduces, due to the thermal expansion of screw rotor when they are heated in running. The outlet decrease is bigger than the inlet decrease of circumferential gap for the temperature distribution difference, as shown in figure 13.

Fig. 13:Circumferential gap changes (the shaded shows change before, red line shows changes)

The curves of three different gaps described above are presented in figure 14 below. Flank gap is almost the same with GVF changes, while radial gap and circumferential gap are getting smaller with the increment of GVF. This is mainly because different leakage clearance not has the same reaction to thermal expansion of screw rotors.

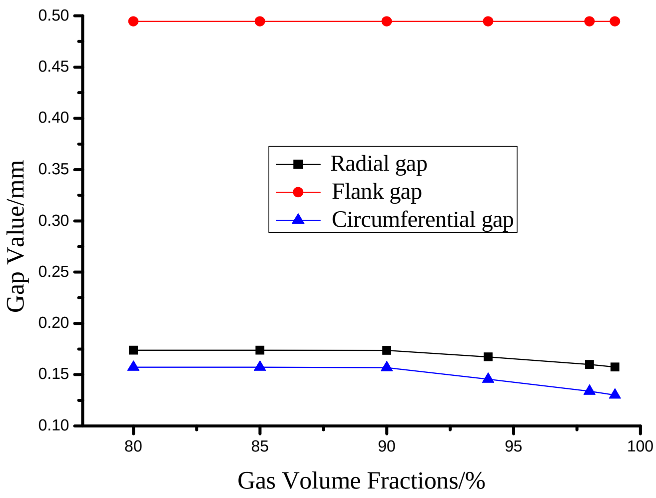


Fig. 14:Gap changes with different GVFs

5. CONCLUSION

With the FEM simulation software, the theoretical model of multiphase twin-screw pump was calculated to get the operating performance and comprehensive deformation as well as rotor profiles change and the rotor gaps change with a very high gas volume fraction. The following conclusions are obtained:

1) The maximum radial deformation of screw rotor, occurred in the top of the gear near the export, are 0.333×10-4m and -0.328×10-4m in the x direction and -0.333×10-4m and 0.326×10-4m in the y direction. When the GVF is greater than 90%, the rotor deformation obviously increases as temperature rise. The heat distortion is more serious than force deformation, so the pressure distribution of screw rotor can be ignored compared with temperature distribution.

2) By comparing the rotor profiles change and axial cross section change under different GVFs, the deformation rule is analyzed, for the improvement of profiles design and optimization of rotor clearance.

3) The clearances characteristic and the rotor gaps change are analyzed in different working conditions, especially at very high gas volume fraction. Radial gap and circumferential gap are decreasing with the influence of thermal expansion, but the deformation of radial gap is more obvious than the deformation of circumferential gap. The flank gap doesn't have any apparent change.

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