Numerical and experimental studies on flow performances and hydraulic radial forces of an internal gear pump with a high pressure

Ying-Yuan Liu, Kang An, Hui Liu, Jian-Guo Gong and Le-Qin Wang

The College of Information, Mechanical and Electrical Engineering, Shanghai Normal University, Shanghai, People’s Republic of China; School of Mechanical and Power Engineering, East China University of Science and Technology, Shanghai, People’s Republic of China; Institute of Process Equipment, Zhejiang University, Hangzhou, People’s Republic of China

ABSTRACT
Unbalanced radial force is one of the most urgent issues for internal gear pumps (IGPs) working at high working pressures. A numerical model for the prediction of the unbalanced radial force is essential for the structural design of gear pumps, facing the challenges of gear engagement description, dynamic mesh method, and so on. In this work, a three-dimensional computational fluid dynamics model of an IGP with a high working pressure was established based on the 2.5-dimensional dynamic mesh method, with cavitation and gear engagements included. The flow behavior and the hydraulic radial force of the high-pressure IGP were studied, and experiments for pump performances were conducted for comparisons. Results indicated that the numerical model could provide a reasonable solution of the flow rate and pressure pulsation for the high-pressure IGP, verified by the experimental data. The circumferential pressure distribution of the pump does not increase linearly, but presents a stepped increase. The hydraulic radial force fluctuates with gear rotation, and its value increases with the working pressure.

Nomenclature

B Width of the gear, m
Cp Dimensionless coefficient
C₁ₑ RNG k–ε turbulence model coefficient
C₂ₑ RNG k–ε turbulence model coefficient
Cₘₑ RNG k–ε turbulence model coefficient
Dₑ Diameter of the addendum circle of the gear, m
f Hydraulic radial force of the gear, KN
fₓ X-axis component of the hydraulic radial force of the gear, KN
fᵧ Y-axis component of the hydraulic radial force of the gear, KN
F Empirical radial force of the gear, N
Fᵥap Evaporation coefficient
F_con Condensation coefficient
g Gravity acceleration, m/s²
i, j Number of two neighboring nodes
k Turbulent energy, J
kᵢⱼ Spring constant between node i and its neighbor j
m Iteration number
n The current time step
nᵢ The number of neighboring nodes connected to node i
p Pressure, Pa
pv Vapor pressure, Pa
pave Average pressure, Pa
Δp Pressure difference between the inlet and the outlet, Pa
Pᵥ Generation of turbulence kinetic energy, J
Rᵥ Bubble diameter, mm
Re Vaporization rate
Rₑ Condensation rate
t Time, s
x Node displacement
xi Displacements of node i
xⱼ Displacements of node j
Z Teeth number of the pinion
u Speed of the mixture flow, m/s
uₔ Speed of the gas phase, m/s
uᵢ Speed of the liquid phase, m/s
α Vapor fraction
αᵥ Nucleation site volume fraction
β Boundary node relaxation factor
σₖ Inverse effective Prandtl number for k
σₑ Inverse effective Prandtl number for ε
ε Turbulent dissipation rate
ρ Density of the mixture flow, kg/m³
ρₑ Density of the gas phase, kg/m³

© 2019 The Author(s). Published by Informa UK Limited, trading as Taylor & Francis Group. This is an Open Access article distributed under the terms of the Creative Commons Attribution License (http://creativecommons.org/licenses/by/4.0/), which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.
\begin{align*}
\rho_1 &= \text{Density of the liquid phase, kg/m}^3 \\
\mu &= \text{Viscosity of the medium, Pa S} \\
\mu_t &= \text{Turbulent viscosity, Pa S}
\end{align*}

\section{1. Introduction}

Internal gear pump (IGP) is a hydraulic pump and can convert the mechanical energy into the hydraulic energy, providing oil pressure for hydraulic system (Liu, Li, & Wang, 2019). Due to its advantages of compact structure, low noise, good inhalation performance, and small flow fluctuation, IGPs are widely employed in precision machine tools, injection molding machinery, agricultural machinery, forging machinery, marine equipment, automotive industry, and other fields (Bae & Kim, 2015; Paffoni, Progri, & Gras, 2004). With the development of the industrial technology, the IGP shows the trend of a higher pressure in recent years. The working pressure of the IGP has achieved approximately 40 MPa. High working pressure may lead to the increase of the unbalanced hydraulic radial forces, causing deformation of gears and shaft, and even inducing the rubbing or colliding of the adjacent metal parts. Meanwhile, the pulsation of radial forces caused by operating mechanism of the IGP (Li, Guo, & Li, 2018) may reduce lifespan of the shaft and induce serious damage for other components. Thus it is significant and necessary to investigate the radial forces of the IGP with a high working pressure for the adequate structural design and safe operation.

The unbalanced radial force acting on the bearing and two gears of the gear pump consists of the radial force generated by the liquid pressure along the circumference of the gear and that generated by the engagement of the gear. At present, the empirical and simplified formulas (Yan & Zhang, 1979) are widely adopted to estimate the average value of the radial force of the gear pump working at a low pressure. Among these formulas (see Equations 1–2), the subscripts \textit{driving} and \textit{driven} stand for the pinion and the annular gear, respectively. In order to simplify the calculation, it is assumed that the liquid pressure in the transition zone of the pump distributes linearly along the circumferential direction during the derivation of the empirical formulas. In recent years, a nonlinear lumped-parameter kinematic elastic dynamic model based on external gear pumps was proposed to predict the dynamic characteristics of gear pumps (Del Rincon & Dalpiaz, 2002). Later, this theoretical model was improved by considering the influences of various factors (e.g. gear eccentricity, clearance leakage and tooth profile error), and was adopted to study the pressure distribution in the inter teeth volume and characteristics of radial forces (Mucchi, Dalpiaz, & Del Rincon, 2010; Mucchi, Dalpiaz, & Rivola, 2011). It should be stated that some assumptions and simplifications were introduced inevitably for theoretical models, such as the calculation of geometric relations, transient positions of the gear teeth during the running process, and so on. Computational Fluid Dynamics (CFD) is a powerful approach to gain fluid flow and pressure distributions for various geometries and systems, which can reflect structural details of the pump model (Ghalandari, Mirzadeh Kooshshahi, Mohamadian, Shamshirband, & Chau, 2019; Mosavia, Shamshirband, Salwana, Chau, & Táh, 2019). Up to now, many researchers have focused on its applications in flow calculations of gear pumps. An accurate CFD model of gear pumps faces two main challenges due to its special working mechanism. One is the application of the dynamic mesh and the other is the simulation of the gear engagement between two gears. Regarding the dynamic mesh method, two-dimensional (2D) numerical models were widely adopted in previous works Campo Sud (2012) and Campo Sud et al. (2012). However, 2D model of gear pumps cannot fully reflect the actual structural details of the pump model. Recently, three-dimensional (3D) dynamic mesh method has been started to be explored to describe the motion of the gears, where the immersed solid method, the dynamic mesh method based on the commercial software package Pumplinx and the 2.5D dynamic mesh model are three potential options for flow analyses of gear pumps. For instance, Ham et al. (Ham, Kim, Oh, & Cho, 2018) and Singh et al. (Singh, Salutagi, Pieta, & Madhavan, 2019) conducted three-dimensional simulations of the flow performance, cavitation characteristics and pressure pulsation in gerotor pumps based on the dynamic mesh method in Pumplinx. Castilla and Gamez-Montero et al. (Castilla, Gamez-Montero, Raush, & Codina, 2017; Gamez-Montero, et al., 2012) and Yoon et al. (2017) investigated the flow performances of gerotor pumps and external gear pumps working at a low pressure by the 2.5D dynamic mesh model and the immersed solid method, respectively. They concluded that the three methods could provide adequate solutions for the flow behavior of the pump. However, it should be noted that the cavitation two-phase flow is not supported by the immersed solid method and Pumplinx has a high requirement on the target structure, which should be very similar to the computational module, and has less control over the computational accuracy. Thus the 2.5D dynamic mesh method is employed in this work, with cavitation behavior included.
Regarding the gear engagement, as we all know, the two gears cannot really contact with each other in the CFD models. Generally, the gear engagement was simplified as a small gap between two gears, which is due to the negligible leakage flow through the gap for pump models at a low pressure. However, the leakage flow increases with the rising working pressure and the clearance between the two gears cannot be set as a too small value due to the difficulty of generating high-quality grid. Thus it can be expected that this method is not suitable for the flow analysis of the pump working at high pressures. Based on this, many researchers have conducted a large amount of works for the simulation of the gear engagement (Houzeaux & Codina, 2007; Strasser, 2007; Castilla et al., 2010), and they concluded that the gear engagement has a remarkable effect on the flow performance of the gear pump. It should be stated that the above model for gear engagement are dependent on the mesh and the engagement point cannot be updated during the mesh motion. New grids and positions of the engagement point should be replaced after a smaller time step. Therefore, these grids of the pump model should be prepared in advance, which requires huge efforts in the modeling process. Then, Campo Sud (2012) and Campo Sud et al. (2012) proposed a new method to define the gear engagement in an external gear pump. Regarding this method, the viscosity of the medium at the engagement point is set as a much larger value than the normal viscosity of the medium, and the medium at the engagement point can be considered as solid. Then, the solidifying medium can be regarded as the gear engagement point. This method is independent on the grid and can be directly hooked in the 3D CFD model by a user defined function (UDF), which is also employed in the gerotor pump by Castilla et al. (2017).

Due to the above challenges, the current solutions for hydraulic radial forces are also limited to 2D model and simple geometries (Li et al., 2018). Based on this, a CFD model of an IGP was established based on the 2.5D dynamic mesh method and the solidifying medium method was introduced through a UDF for simulation of the gear engagement. Then, the hydraulic force performance of the pump with a high pressure can be investigated.

The main structure of this work is arranged as follows. The geometrical model of the pump and the test system are provided in Section 2. The numerical model based on the 2.5D dynamic mesh and solution strategies of the pump are reported in Section 3, with the gear engagement simulated by solidifying medium method. Then, the calculated pump performance and pressure pulsation are verified by experimental data. Later, analyses on hydraulic forces of the pump model are given in Section 4, including the circumferential pressure distributions and hydraulic forces on gears. Finally, conclusions drawn in this work are summarized in Section 5.

2. Geometrical model and test system

2.1. Geometrical model

The structure of a typical high-pressure IGP is shown in Figure 1, including an annular gear, a pinion, a crescent-shaped baffle plate, and a pump case. The pump chamber is divided into two regions by the baffle: a high pressure chamber (outlet side, red zone) and a low pressure chamber (inlet side, blue zone). Two gears rotate eccentrically in same directions and accordingly, the fluid is transported from the low-pressure chamber to the high-pressure chamber. The main structural and operating parameters of the pump are displayed in Table 1.

The 3D hydraulic model of the IGP was established (see Figure 2), which is divided into three parts: the pump shell part (i.e. the pump shell, the inlet and outlet pipelines), the gear part, and the drainage holes on the annular gear. Meanwhile, the lengths of the inlet and outlet pipelines are extended to three times of their diameters for the steady flow and pressure. All radial clearances in the pump are set as 0.1 mm, while the axial clearance between gear part and the pump shell is not considered.

The medium conveyed by the pump is LHM-46 anti-wear hydraulic oil, and the corresponding properties are shown in Table 2. In general, it is unavoidable to dissolve a certain amount of air in hydraulic oil. When the pressure in the pump is lower than a certain value (i.e.

![Figure 1. Structure configurations of the IGP (partially from Liu, Li, & Wang, 2019).](image)

| Table 1. Structural and operating parameters of the IGP. |
|---------------------------------------------------------|
| Structural and operating parameters | Value |
| Tooth number of the pinion gear | 12 |
| Tooth number of the annular gear | 18 |
| Width of the gears (mm) | 48 |
| Rotational speed (r/min) | 1450 |
| Displacement (L/r) | 50 |
| Working pressure (MPa) | 0–25 |
Figure 2. Hydraulic model of the IGP (partially from Liu & Zhu, 2016).

Table 2. Properties of the medium in the pump at room temperature (20°C).

| Parameters         | Density (kg/m³) | Viscosity (Pa s) | Air separation pressure (Pa) | Saturated vapor pressure (Pa) |
|--------------------|-----------------|------------------|------------------------------|------------------------------|
| Oil                | 875             | 0.04025          | 4000                         | 0.03                         |
| Air                | 1.225           | 1.7894e–5        | /                            | /                            |

air separation pressure), the supersaturated air dissolved in the oil quickly separates from the oil and accordingly bubbles are produced. It should be noted that the saturated vapor pressure of the hydraulic oil is smaller than the air separation pressure, and the vaporization pressure $p_v$ in this case refers to air separation pressure.

2.2. Test system

A test system was built for investigating the external characteristics and pressure pulsations of the pump, composed of three parts: pump hydraulic circuit, instrument components and data acquisition equipment. The scheme of the hydraulic circuit of the test system is shown in Figure 3 and the parameters of the main instruments are given in Table 3. The IGP is driven by a four-stage three-phase asynchronous frequency conversion motor with the power of 110 kW. Then, hydraulic oil stored in the oil tank is pressurized to the pipeline by the pump and finally re-entered into the tank from the pipeline, forming a circulating circuit. The motor is controlled by a speed regulating potentiometer on the operating table and the rotational speed can change from 0 r/min to 3000 r/min. A speed & torque instrument is located at the pump shaft between the motor and the pump, facilitating real-time recording and monitoring of the speed and torque of the pump.

At the inlet of the pump, a vacuum pressure sensor with a range of 0–0.2 MPa (absolute pressure) is installed to monitor the inlet pressure. At the outlet of the pump, a hydraulic valve block containing a proportional valve is placed to regulate the working pressure and a high-frequency dynamic pressure sensor with a range of 0–25 MPa is accompanied for monitoring the working pressure of the pump. The two pressure sensors both have the accuracy levels of 2.5, corresponding to the error range of 0.005 MPa for the vacuum pressure sensor and 0.625 MPa for the high-frequency dynamic pressure sensor. The real-time data of the pressure transmitter are collected by AVANT series MI-7016 data acquisition device of ECON, with the sampling frequency of 2560 Hz. In addition, an elliptical gear flow meter is installed at the outlet pipeline to measure the flow rate of the pump, with the range from 2.5 to 15 m³/h. Moreover, to protect the pump, the valves and the flowmeter, an oil filter with a precision of 100 microns and an oil filter with a precision of 5 microns are installed at the inlet and the outlet of the oil pump, respectively.

3. Numerical method

3.1. Governing equations

3.1.1. Mixture model

Previous studies have demonstrated that cavitation has a great influence on the suction performance of gear pumps and the occurrence of the negative absolute pressure can be avoided by considering the cavitation behavior (Noorpoor, 2013). In this work, the cavitation flow is assumed as a homogeneous medium of sharing the same pressure and a mixture model is adopted for the simulation of the cavitation flow in the pump (Wu, Liu, Dou, & Zhang, 2011; Liu et al., 2012). The continuity equation and momentum equation of the mixture are displayed as follows:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{u}) = 0$$  \hspace{1cm} (3)

$$\frac{\partial (\rho \vec{u})}{\partial t} + \nabla \cdot (\rho \vec{u} \vec{u}) = -\nabla \cdot \left[ (\mu + \mu_t)(\nabla \vec{u} + \nabla \vec{u}^T) \right] - \nabla \cdot p + \rho \vec{g} + \nabla \cdot [\alpha \rho_g (\vec{u}_g - \vec{u})] + (1 - \alpha) \rho (\vec{u}_l - \vec{u})$$  \hspace{1cm} (4)

3.1.2. Cavitation model

During the generation and evolution stages of the cavitation, the gas phase transport equation (see Equation (5)) is employed to describe the mass exchange between the gas phase and the liquid phase, including the vaporization of the liquid phase and condensation of the bubble phase (ANSYS FLUENT Theory Guide, 2014). Zwart–Gerber–Belamri model is taken to describe the mass exchange of the two phases, where the vaporization rate $R_v$ and the condensation rate $R_c$ are shown in Equations (6) and (7), respectively. It can be observed
that four empirical parameters may affect the cavitation flow: the bubble diameter $R_b$, the nucleation site volume fraction $\alpha_{\text{nuc}}$, the evaporation coefficient $F_{\text{vap}}$ and the condensation coefficient $F_{\text{con}}$. In general, $\alpha_{\text{nuc}}$ is given as the intrinsic coefficients of the liquid and is set as 0.09 in hydraulic oil; $F_{\text{con}}$ is set as the default value of 0.01; $R_b$ and $F_{\text{vap}}$ are set as 0.01 mm and 0.4, respectively.

\[
\frac{\partial (\alpha \rho_g)}{\partial t} + \nabla \cdot (\alpha \rho_g \bar{u}_g) = R_e - R_c 
\]

\[
R_e = F_{\text{vap}} \frac{3 \alpha_{\text{nuc}} (1 - \alpha) \rho_k}{R_b} \sqrt{\frac{2 p_v - p}{3 \rho_l}} 
\]

\[
R_c = F_{\text{con}} \frac{3 \alpha_{\text{nuc}} \rho_g}{R_b} \sqrt{\frac{2 p_v - p}{3 \rho_l}} 
\]

3.1.3. Turbulence model

The RNG $k-\epsilon$ model is introduced to close the above equations herein. And the turbulent kinetic energy and turbulent kinetic energy dissipation rate are presented as Equations (8) and (9), where the turbulent viscosity is shown in Equation (10). There are five empirical constants in this model, i.e. $C_{1\epsilon}, C_{2\epsilon}, C_{\mu}, \sigma_k$ and $\sigma_\epsilon$, with the default values of 0.0845, 1.45, 1.68, 1.0 and 1.3, respectively.

\[
\frac{\partial (\rho k)}{\partial t} + \nabla \cdot (\rho \bar{u} k) = P_k + \nabla \cdot \left( \left( \mu + \mu_t \sigma_k \right) \nabla \cdot k \right) - \rho \epsilon 
\]

\[
\frac{\partial (\rho \epsilon)}{\partial t} + \nabla \cdot (\rho \bar{u} \epsilon) = C_{1\epsilon} \frac{\epsilon}{k} P_k - C_{2\epsilon} \rho \left( \frac{\epsilon^2}{k} \right) + \nabla \cdot \left( \left( \mu + \mu_t \sigma_\epsilon \right) \nabla \cdot \epsilon \right) 
\]

\[
\mu_t = \rho C_{\mu} \frac{k^2}{\epsilon} 
\]
3.2. Mesh and mesh motion

The governing equations of the cavitation flow in the pump are discretized by finite volume approach and the unstructured mesh is used for the domains (Ramezanizadeh, Alhuyi Nazari, Ahmadi, & Chau, 2019; Akbarian et al., 2018). The meshes of the gear are obtained by extending or stretching 2D triangular mesh along the axial direction of gears. The motion of gears are simulated by 2.5D dynamic mesh and the rotation speeds of the two gears are implemented by a UDF. The end faces of gears are set as deforming mesh and the gear surfaces are treated as moving mesh. Both the smoothing and the re-meshing methods are used to describe the mesh updating. There are two empirical parameters for the mesh smoothing. One is the boundary node relaxation (see Equation 11), controlling the updating of the node position. The range of the value is [0, 1], where the value of 1.0 presents that the nodes are not relaxation processed, and the value of 0 means that the nodes are kept unreformed. In this work, the value is chosen as 0.15 to guarantee the mesh quality during the process of mesh deformation. The other empirical parameter is the number of iterations, governing the resolution of the balanced equation (see Equation 12) and the value is taken as 5. In addition, a small time step of $5.75 \times 10^{-6}$ s is chosen for mesh updating, which is just the time for the pinion rotating one tooth.

$$\bar{x}_n^{n+1} = \bar{x}_n^n + \beta \Delta \bar{x}$$  \hspace{1cm} \hspace{1cm} (11)

$$\Delta \bar{x}_i^{n+1} = \frac{\sum_{j}^{n} k_{ij} \Delta \bar{x}_j^n}{\sum_{j}^{n} k_{ij}}$$  \hspace{1cm} \hspace{1cm} (12)

Two mesh models with different cell densities are carried out to exclude the influence of grid on calculation accuracy: mesh model A is with the element number of 3,775,971 and mesh model B is with the element number of 5,090,923. The two mesh models contain the same element number at the zone of the inlet and outlet pipes, but the element numbers at the gear zone are different from each other. Taking the working condition of the speed of 1450 r/min and the working pressure of 2.88 MPa as an example, the flow pulsation of the pump and the change of mesh equi-volume skewness are monitored during the process of the dynamic mesh updating (see Figure 4). It can be seen in Figure 4(a) that both flow rates calculated by the two mesh models nearly present the same amplitudes and periods. Meanwhile, the average values of the calculated flow rates are compared with the experimental results (see Table 4). It can be seen that the errors of the calculated flow rates are around 3% for both models and the error obtained by mesh B is slightly smaller than that by mesh A. In addition, it can be observed in Figure 4(b) that although the cell equi-volume skewness has a little decrease with the increasing mesh number, the cell equi-volume skewness of the two mesh models are quite small. Herein, the mesh A is selected for the following calculations to reduce the computational efforts. The selected mesh model is displayed in Figure 5, with the element and node numbers of 3,775,971 and 1,835,197, respectively.

**Figure 4.** Mesh independence study (1450 r/min, 2.88 MPa): (a) flow rates and (b) mesh equi-volume skew.

**Table 4.** Comparison of the calculated flow rates with the experimental values.

| Mesh number | Calculated flow rate (L/min) | Experimental flow rate (L/min) | Error |
|-------------|-----------------------------|-------------------------------|-------|
| A           | 67.895                      | 70.2                          | −3.28%|
| B           | 67.945                      | 70.2                          | −3.21%|
3.3. Solution method

The numerical method is based on a conservative finite-volume formulation and is conducted based on the commercial software package FLUENT (Ding & Liu, 2017). Boundary conditions of the numerical model are determined based on the operating parameters of the tested pump. The inlet pressure is the atmospheric pressure, and the outlet pressure is loaded to a given pressure value, ranging from 0 to 25 MPa. The rotational speeds of the pinion and the annual gear are set as 1450 and 966.67 r/min respectively by a UDF of ‘DEFINE_CG_MOTION’. Then, the pinion and the annual gear have the same circumferential velocity and run synchronously in the same direction.

To ensure the convergence of calculation and obtain more accurate results, the calculation is carried out in three steps: first, the steady calculation of the IGP without cavitation is conducted, and then the calculation considering cavitation flow is continued with the above result as an initial condition; third, the gear engagement is included for a final solution. For the above calculation, pressure-based solver suitable for low-speed problems is employed. The diffusion term of governing equations is calculated by the Green–Gauss node-based scheme with a high precision. The momentum, turbulent kinetic energy, and turbulence dissipation rate (Mou, He, Zhao, & Chau, 2017) are all discretized by the second-order upwind scheme for a higher accuracy computational results. Besides, the calculation is considered to be converged when the residuals of all variables are less than $10^{-5}$ and the monitored parameters become stable.

3.4. Simulation method of the gear engagement

The solidifying medium method (or contact point viscosity method) is taken to simulate the solid–solid contact behavior of the engagement point, which is closer to the practical running situation of the gear pump. In this method, firstly, a UDF of ‘LOOKUP_THREAD’ is employed to search the nearest two nodes between the two gears and the position of the two nodes are stored in a user-defined memory by the UDF of ‘C_UDMI’. Then, the two nearest nodes can be seen as the engagement points and the viscosity of the medium near the position of the engagement points are set as a larger value using a UDF of ‘DEFINE_PROPERTY’. In this way, the medium with a large viscosity near the engagement point can be seen as solid wall, acting as a solid to solid contact between the two gears and separating the low-pressure chamber and the high-pressure chamber from each other.

In this paper, the medium viscosity at the engagement point is chosen as 40 times of the viscosity of the hydraulic oil ($40 \times 0.04025$ Pa S). Meanwhile, the viscosity of the medium near the engagement point changes linearly, from the viscosity $\mu$ ($0.04025$ Pa S) to $40\mu$ ($1.61$ Pa S) of the medium, to avoid the calculation divergence by the sudden change of the viscosity. The viscosity distribution of the medium around the engagement point is shown in Figure 6.

4. Results and discussion

4.1. Pump performance and verification of the numerical method

4.1.1. Performance curve of the pump

The tested and calculated flow rates at various working pressures for the rated speed (1450 r/min) of the pump are given in Figure 7. It can be seen that the tested flow rates do not present significant decrease for the working pressure considered, while the calculated flow rates decrease with the increase of the working pressure. And the calculated flow rates are smaller than the tested flow rates. This is because for the test pump, the radial clearance compensation device is set in the crescent-shaped baffle plate, minimizing the leakage flow through the radial clearance. However, the compensation device is not considered in the numerical model due to its complex geometry, resulting in a larger leakage flow.

Meanwhile, the calculated flow rates by two models with and without consideration of the gear engagement are displayed for comparisons in Figure 7. Results display that the calculated flow rates are more consistent with the tested flow rates when the engagement of gears is taken into consideration, improving the calculation accuracy of the pump under medium and high pressure conditions. This is because that the larger medium viscosity at the engagement point hinders the leakage flow through the clearance between the two gears and results in the decrease of the leakage flow rate. Moreover, it can be observed that the errors between the calculated and
the tested flow rates increase with the working pressure due to the leakage flow. The maximum error at the working pressure of 16.67 MPa is about 10% for the model with the consideration of the gear engagement, while it is up to 29% for that without the consideration of the gear engagement. This presents that the consideration of gear engagement could provide a more accurate solution for flow performances of the pump. Therefore, the gear engagement should be included for flow calculations at medium and high working pressures. In addition, it should be noted that the corresponding error for the model with the gear engagement could be acceptable in practical engineering.

4.1.2. Transient flow and pressure fluctuations

In general, the displacement of the IGP is determined by the area swept by the oil chamber of the gear pump in the tooth cavity, changing with the rotation of the gears. It can be expected that the instantaneous flow and pressure of the IGP show pulsating signals. Taking the instantaneous flow rate at 8.33 MPa as an example, the instantaneous flow rate for the pinion rotating two teeth is given in Figure 8. It can be seen that the calculated flow rates considering gear engagement are greater than that without considering the gear engagement, which is consistent with the conclusion obtained in Figure 6. Meanwhile, the flow periodicity and displacement behavior of the two
calculated flow rates are similar to each other. However, it should be noted that considering the engagement point by the solidifying medium method may bring some small changes of the flow pulsations. Exactly speaking, when the gear engagement is not taken into consideration, the calculated flow rate increases to a steady peak value first and then decreases during one period of the flow. However, the flow rate of the IGP has several peaks in the process of rotating one tooth when the gear engagement is considered.

Figure 9 shows the outlet pressure pulsation of the IGP at the rated speed of 1450 r/min and the working pressure of 16.67 MPa, including the time domain and the frequency spectrum obtained by FFT analysis. Herein, a dimensionless coefficient $C_p$ (see Equation 13) is introduced to describe the magnitude of the pressure pulsation. It can be observed that although the calculated pressure pulsation is slightly smaller than the tested value (see Figure 9a), the main frequencies of both pressure signals are equal to the frequency of $12f_n$ (see Figure 9b). The deviation of the calculated amplitudes from that of tested pressure pulsations may be due to the influence of external disturbances during the experiment, such as the elements of the hydraulic circuit on the pump output (e.g. diameter and length of the pipe element). The same main frequency is the teeth number of the pinion multiplied by its rotational frequency ($Zf_n$), which means a period equal to the time required for the rotating one teeth of the pinion. It is usually thought that this frequency results from the inevitable unsteady discharge process of gear pumps (Schiffer, Benigni, & Jaberg, 2013; Liu et al., 2019).

$$C_p = \frac{p - p_{ave}}{p_{ave}}.$$  \hspace{1cm} (13)

As mentioned in Sections 4.1.1 and 4.1.2, it can be concluded that the above 3D model considering the engagement of gears could provide an adequate solution of the pump performance, flow rates and pressure pulsations. Based on this, this model is adopted for flow and hydraulic forces analyses of the pump model in this paper.

### 4.2. Distributions of the circumferential pressure

The fluid pressure in the pump has a great impact on the working condition of the gear (Shen, Li, Qi, & Qiao, 2018), especially for the hydraulic radial forces on gears. As mentioned in the section of Introduction, the circumferential pressure distribution of the gear pump is always

![Figure 10. Circumferential pressure distribution of the IGP for various internal pressures (1450 r/min).](image)

| Working pressures (MPa) | Pinion | Annular gear |
|-------------------------|--------|--------------|
| 2.88                    | 6.11   | 8.18         |
| 8.33                    | 17.25  | 23.15        |
| 16.67                   | 34.29  | 46.05        |
| 25                      | 51.29  | 68.92        |

**Table 5.** Hydraulic radial forces of the gears.
assumed to be a linear distribution for simplifying the calculation of the hydraulic radial forces, which is a main reason leading to the remarkable error of the radial force. Thus, accurate prediction of the pressure distribution, especially in the transition zone of the IGP is of great significance for the solution of radial force. Based on this, the pressure of a rotating point between the crescent-shaped baffle plate and the annular gear rotating with the annular gear synchronously was monitored by a UDF of ‘DEFINE_EXECUTE_AT_END’ and ‘C_P’. The monitored pressure is the circumferential pressure between the crescent-shaped baffle plate and the annular gear.

Figure 10 shows the circumferential pressure distribution of IGP at the rated speed of 1450 r/min for various internal pressures (i.e. 8.33 and 16.67 MPa). It can be observed that the circumferential pressure of

Figure 11. Hydraulic radial forces of the pinion at the working pressure of 8.33 MPa: (a) X-axis and Y-axis components of the hydraulic radial forces of the pinion and (b) Values and directions of the hydraulic radial forces of the pinion.
the gear pump does not increase linearly but presents stepped increases. Meanwhile, the circumferential pressure only rises at the sealing gap between the addendum of the annular gear and the crescent-shaped separator, and it is remained unchanged when a large chamber is formed between the addendum of the annular gear and the crescent-shaped separator. Moreover, it can be observed that the step number for the pressure increase is determined by the number of the sealing chamber formed by the annular gear and the crescent-shaped separator. In this case, the step numbers for the increases of the pressure are four for the three working pressures mentioned. In addition, the growth rate of the pressure in the sealing area of the crescent baffle and the annular gear increases with the working pressure.

4.3. Hydraulic radial force on gears

The mean values of the calculated hydraulic radial forces of the two gears in the IGP with various working pressures are displayed in Table 5. It can be seen that the hydraulic radial forces of the pinion and the annular gear
increase with the working pressure linearly. Regarding the pinion, the hydraulic radial force is 51.29 kN when the pump works at the pressure of 25 MPa, which is over 8 times of that at the pressure of 2.88 MPa. The same conclusion can also be found for the annular gear. In summary, it can be concluded that the radial force of the gears at a high pressure is much higher than that at a small pressure value. In addition, it should be stated that the calculated hydraulic radial force of the annular gear may be smaller than the actual value, due to the leakage flow in the numerical model without the circumferential seal between the annular gear and the pump case.

In general, both flow and pressure pulsations can cause fluctuations of the hydraulic radial force with the frequency of $12f_n$. The frequency means that the period of the hydraulic radial forces is equal to the time required for the pinion rotating one tooth ($30^\circ$). In this paper, taking the pump working at the rotation speed of 1450 r/min and the working pressure of 8.33 MPa as an example, the pulsating hydraulic radial forces of the pinion and the annular gear within two periods (the pinion rotating $60^\circ$) are shown in Figures 11 and 12, respectively.

In these figures, the value and the direction of the hydraulic radial forces are included, where the direction of the hydraulic radial force is described by the angle between the vector force and the X-axis. It can be found that both the value and the direction of the hydraulic radial forces on the gears pulsate at the same time. This is because that the sealing position between the gear and crescent-shaped baffle plate changes with the rotation of the gear. The number of teeth chamber entering sealing between the gear and the crescent-shaped baffle plate in the transition zone sometimes changes with the rotation of gears. For example, the number of teeth chamber in the transition zone changes from 3 to 2 due to the change of the sealing position (see Figure 13), leading to a large fluctuation of the radial force.

In addition, the direction of the hydraulic radial force for the pinion points toward the upper left direction from the rotation center of the pinion, while that for the
Y.-Y. LIU ET AL.

Figure 14. Directions of the hydraulic radial forces of the pinion and the annular gear.

annular gear points toward the lower right direction from the center of the annular gear (see Figure 14).

Based on the above analyses, it is found that the radial force of the gears at a high pressure is much higher than that at a small value, and the radial force of the pump should be compensated. Generally, a pressure compensating groove (see Figure 15) of the pump case is employed to compensate large radial forces of the gear pump working at a high pressure, and the structural design of the compensation device is also closely related to the radial forces calculated. The hydraulic radial forces have been investigated by CFD calculations and the structural design of the compensation device of the pump at a high pressure will be a main topic in future studies.

5. Conclusion

In this work, a three-dimensional numerical model of an IGP was established based on 2.5D dynamic mesh method. User-defined functions were introduced to define the gear engagement and the cavitation flow was considered to avoid the occurrence of the negative pressure. Flow behavior and hydraulic radial forces of the pump at a high working pressure were analyzed, and experiments for external characteristics and pressure pulsations of the pump were conducted for comparisons. Main conclusions drawn through numerical investigations of this work are provided below:

(1) The 3D numerical model with gear engagement and cavitation could provide a fairly good solution of the flow rate and pressure pulsation of the high-pressure IGP, verified by the experimental data.

(2) The calculated pressure pulsations of the pump model are slightly smaller than the tested value. The main frequency of the pressure pulsation is \(12f_0\), which is just the teeth number of the pinion multiplied by its rotational frequency \((Z^*f_0)\), resulting from the inevitable unsteady discharge process of gear pumps.

(3) The circumferential pressure distribution of the pump does not increase linearly, but presents a stepped increase. The step number of the pressure growth is determined by the number of the sealing chamber formed by the annular gear and the crescent-shaped separator, and the pressure in the sealing area of the crescent baffle and the annular gear increases with the working pressure.

(4) The hydraulic radial force of the pump model fluctuates with gear rotation, and its value increases with the working pressure. The hydraulic radial force at a high pressure is much higher than that at a small value, and the compensation device should be set up in the pump.

(5) The structural design of the compensation device is based on the predicted radial forces, which should be addressed in future. Due to the complexity of structural configurations and working principles of the compensation device, the fluid-solid interaction may be a possible option, and further investigations are still needed.

Disclosure statement

No potential conflict of interest was reported by the authors.

Funding

This work was supported by National Natural Science Foundation of China: [Grant Numbers 51608325, 51806145].

References

Akbarian, E., Najafi, B., Jafari, M., Faizollahzadeh Ardabili, S., Shamshirband, S., & Chau, K. W. (2018). Experimental and computational fluid dynamics-based numerical simulation of using natural gas in a dual-fueled diesel engine. *Engineering Applications of Computational Fluid Mechanics, 12*(1), 517–534.

ANSYS FLUENT Theory Guide. (2014). Release 16.0.0. Pittsburgh, USA

Bae, J. H., & Kim, C. (2015). Design of rotor profile of internal gear pump for improving fuel efficiency. *International Journal of Precision Engineering and Manufacturing, 16*(1), 113–120.

Campo Sud, D. D. (2012). Analysis of the suction chamber of external gear pumps and their influence on cavitation
and volumetric efficiency (Doctoral dissertation). Universitat Politècnica de Catalunya, Catalunya. Retrieved from https://upcommons.upc.edu/handle/2117/94595

Campos Sud, D. D., Castilla, R., Raush, G. A., Montero, P. G., & Codina, E. (2012). Numerical analysis of external gear pumps including cavitation. *Journal of Fluids Engineering, 134*(8), 081105.

Castilla, R., Gamez-Montero, P. J., Ertrürk, N., Vernet, A., Cousirat, M., & Codina, E. (2010). Numerical simulation of turbulent flow in the suction chamber of a gear pump using deforming mesh and mesh replacement. *International Journal of Mechanical Sciences, 52*(10), 1334–1342.

Castilla, R., Gamez-Montero, P. J., Raush, G., & Codina, E. (2017). Method for fluid flow simulation of a gerotor pump using OpenFOAM. *Journal of Fluids Engineering, 139*(11), 111101.

Ding, X. S., & Liu, B. (2017). *CAX Engineering application series: Fluent 17.0 fluid simulation from Introduction to Proficiency*. Beijing, BJ: Tsinghua University Press.

Fernandez Del Rincon, A., & Dalpiaz, G. (2002). A model for the elastodynamic analysis of external gear pumps. *International conference on noise & vibration engineering*. Sas, Van Hal.

Ghalandari, M., Mirzadeh Koohshahi, E., Mohamadian, F., Shamshirband, S., & Chau, K. W. (2019). Numerical simulation of nanofluid flow inside a root canal. *Engineering Applications of Computational Fluid Mechanics, 13*(1), 254–264.

Ham, J., Kim, S., Oh, J., & Cho, H. (2018). Theoretical investigation of the effect of a relief groove on the performance of a gerotor oil pump. *Journal of Mechanical Science and Technology, 32*(8), 3687–3698.

Houzeaux, G., & Codina, R. (2007). A finite element method for the solution of rotary pumps. *Computers & Fluids, 36*(4), 667–679.

Li, Y. B., Guo, D. S., & Li, X. B. (2018). Mitigation of radial exciting force of rotary lobe pump by gradually varied gap. *Engineering Applications of Computational Fluid Mechanics, 12*(1), 711–723.

Liu, Y. Y., Li, Y. R., & Wang, L. Q. (2019). Experimental and theoretical studies on the pressure fluctuation of an internal gear pump with a high pressure. *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science, 223*(3), 987–996.

Liu, J. T., Liu, S. H., Wu, Y. L., Jiao, L., Wang, L. Q., & Sun, Y. K. (2012). Numerical investigation of the hump characteristic of a pump–turbine based on an improved cavitation model. *Computers & Fluids, 68*, 105–111.

Liu, Y. Y., & Zhu, Z. C. (2016). 高压内啮合齿轮泵的三维CFD数值计算与试验研究 [3D Numerical and Experimental studies on Hydraulic performances of an Internal Gear Pump with a High Pressure]. *流体机械, 44*(12), 5–10.

Mosavi, A., Shamshirband, S., Salwana, E., Chau, K. W., & Tah, J. H. (2019). Prediction of multi-inputs bubble column reactor using a novel hybrid model of computational fluid dynamics and machine learning. *Engineering Applications of Computational Fluid Mechanics, 13*(1), 482–492.

Mou, B., He, B. J., Zhao, D. X., & Chau, K. W. (2017). Numerical simulation of the effects of building dimensional variation on wind pressure distribution. *Engineering Applications of Computational Fluid Mechanics, 11*(1), 293–309.

Mucchi, E., Dalpiaz, G., & Del Rincon, A. F. (2010). Elastodynamic analysis of a gear pump. Part I: Pressure distribution and gear eccentricity. *Mechanical Systems and Signal Processing, 24*(7), 2160–2179.

Mucchi, E., Dalpiaz, G., & Rivola, A. (2011). Dynamic behavior of gear pumps: Effect of variations in operational and design parameters. *Meccanica, 46*(6), 1191–1212.

Noorpoor, A. R. (2013). Optimization gear oil pump in order to energy saving and environmental impact in a diesel engine. *International Journal of Automotive Engineering, 3*, 496–507.

Paffoni, B., Progrì, R., & Gras, R. (2004). Teeth clearance effects upon pressure and film thickness in a trochoidal hydrostatic gear pump. *Proceedings of the Institution of Mechanical Engineers, Part G: Journal of Aerospace Engineering, 221*(4), 247–256.

Ramezanizadeh, M., AlhuyiNazari, M., Ahmadi, M. H., & Chau, K. W. (2019). Experimental and numerical analysis of a nanoluidic thermosyphon heat exchanger. *Engineering Applications of Computational Fluid Mechanics, 13*(1), 40–47.

Schiffer, J., Benigni, H., & Jaberg, H. (2013). Development of a novel miniature high-pressure fuel pump with a low specific speed. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, 227*(7), 997–1006.

Shen, H. D., Li, Z. Q., Qi, L. L., & Qiao, L. (2018). A method for gear fatigue life prediction considering the internal flow field of the gear pump. *Mechanical Systems and Signal Processing, 99*, 921–929.

Singh, R., Salutagi, S. S., Pieta, P., & Madhavan, J. (2019). Study of effect of air content in lubrication oil on gerotor pump performance using CFD simulations. *SAE Technical Paper, No.* 2019-06-0300.

Strasser, W. (2007). CFD investigation of gear pump mixing using deforming/agglomerating mesh. *Journal of Fluids Engineering, 129*(4), 476–484.

Wu, Y. L., Liu, S. H., Dou, S. H., & Zhang, L. (2011). Simulations of unsteady cavitating turbulent flow in a Francis turbine using the RANS method and the improved mixture model of two phase flows. *Engineering with Computers, 27*(3), 235–250.

Yan, J. K., & Zhang, P. S. (1979). 液压传动 [hydraulic transmission]. Beijing: National Defense Industry Press.

Yoon, Y., Park, B. H., Shin, J., Han, Y. O., Hong, B. J., & Yun, S. H. (2017). Numerical simulation of three-dimensional external gear pump using immersed solid method. *Applied Thermal Engineering, 118*, 539–550.