Numerical simulation of the two-phase flows in a hydraulic coupling by solving VOF model

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Abstract. The flow in a partially filled hydraulic coupling is essentially a gas-liquid two-phase flow, in which the distribution of two phases has significant influence on its characteristics. The interfaces between the air and the liquid, and the circulating flows inside the hydraulic coupling can be simulated by solving the VOF two-phase model. In this paper, PISO algorithm and RNG $k$-$\varepsilon$ turbulence model were employed to simulate the phase distribution and the flow field in a hydraulic coupling with 80% liquid fill. The results indicate that the flow forms a circulating movement on the torus section with decreasing speed ratio. In the pump impeller, the air phase mostly accumulates on the suction side of the blades, while liquid on the pressure side; in turbine runner, air locates in the middle of the flow passage. Flow separations appear near the blades and the enclosing boundaries of the hydraulic coupling.

0. Introduction

Energy transmission by the hydraulic coupling is one of the energy-saving technologies [¹]. The hydraulic coupling can realize flexible connection by using fluid kinetic energy to transmit power, which has advantages such as good adaptability, shock absorption, etc. In practice, fluid couplings always operate with partial fills. The complex air/oil two-phase flows will influence the performance characteristics of the unit.

Both experimental and numerical studies have been carried out to analysis the flows in hydraulic couplings. Steven B. Ainley used Laser Velocimetry to measure the velocities in the pump impeller of a torque converter at four speed ratios, where strong erratic flows, e.g., secondary, jet and wake flows have been found [²]. In order to study the gas-liquid distribution, Hampel U et al. applied the technology of GT (Gamma Tomography) to obtain the phase interfaces on the axial surface of the hydraulic coupling in 2005 [³]. APAS (Autonomous Planner Array Sensor) technique was adopted to measure dynamic two-phase flow patterns on pressure and suction sides in a hydraulic coupling when input speed was 790rpm [⁴]. Fan Lidan et al. has observed the two-phase flow field in a model hydraulic coupling by PIV (Particle Image Velocimetry) [⁵].

Numerical studies based on CFD become more popular to study the internal flow in the hydraulic couplings. Until now, Full-filled flow field in the hydraulic coupling has been simulated extensively [⁶, ⁷], while few numerical research on real two-phase flow has been published. He Yandong et al. in Jilin University simulated the two-phase flow by solving a Mixture model, and obtained the two-phase
velocity field and pressure distribution \cite{8}. Since a mixture model comprised of gas and liquid phases was applied, the interface between the two phases couldn’t be simulated. In 2012, Zhao Jiyun et al. employed a VOF model to track the interfaces between gas and liquid in a single passage model \cite{9}. Analysis of the influence of the retainer on circulating pattern has been made based on the simulation results.

In this paper, VOF model was used to trace interfaces, and simulate flow patterns in the whole passage. It is believed that numerical simulation can provide details of the flow field, benefitting the design of the hydraulic coupling.

1. Modeling
UG software package was used to build the model of a hydraulic coupling with circular-shaped chamber. The main geometric parameters of this model were as followed: maximum diameter of flowpath $D=150$mm; blade thickness $b=1.5$mm; number of pump blades $B_1=23$; number of turbine blades $B_2=22$. The model of the flow passage of the hydraulic coupling was shown in Fig. 1.

![Flow passages](image1)

**Figure 1.** Flow passages.

2. Mesh Discretization
Due to the relatively large number of blades in the hydraulic coupling, structured hexahedral mesh was adopted by ICEM to ensure the quality of the mesh. In order to ensure the compatible $y^+$ values with the chosen turbulence model (the simulation results show that $y^+$ ranges between 0 and 46), grids were refined in the boundary layer on each blade surface. The minimum height of the first grid layer was 0.05. Total mesh number was about 3 million. The mesh in pump is shown in Fig. 2.

![Mesh in pump](image2)

**Figure 2.** Mesh in pump.

3. Control Equations and Boundary Conditions
3.1. VOF Model
The volume of fluid (VOF) model determines the shape and location of the free surface based on the concept of volume fraction of fluid. The value of volume fraction between zero and one indicates the type of the element, i.e., partial or surface element.

The tracking of the interfaces between the two phases is accomplished by the solution of a
continuity equation for the volume fraction of one of the phases. For the two-phase flow where primary phase is liquid, the continuity equation of the gas can be written as:

$$\frac{\partial}{\partial t} (\alpha_g \rho_g) + \nabla \cdot (\alpha_g \rho_g \mathbf{v}_g) = S_{\alpha_g} + \left( \dot{m}_{lg} - \dot{m}_{gl} \right)$$

(1)

where $\dot{m}_{lg}$ is the mass transfer from liquid to gas, and $\dot{m}_{gl}$ is the mass transfer from gas to liquid. $\alpha_g$ is the volume fraction of gas. $S_{\alpha_g}$ is the source term.

The volume fraction $\alpha_l$ of the liquid phase can be computed based on the following constraint:

$$\alpha_l + \alpha_g = 1$$

(2)

The momentum equation, shown below, is dependent on the volume fractions of all phases through the properties $\rho$ and $\mu$.

$$\frac{\partial (\rho \mathbf{v})}{\partial t} + \nabla \cdot (\rho \mathbf{v} \mathbf{v}) = -\nabla p + \nabla \cdot \left[ \mu (\nabla \mathbf{v} + \nabla \mathbf{v}^T) \right] + \rho \mathbf{g} + \mathbf{F}$$

(3)

The volume-fraction-averaged density takes on the following form:

$$\rho = \alpha_l \rho_l + \alpha_g \rho_g$$

(4)

where $\rho_l$ is the density of liquid, and $\rho_g$ is the density of gas.

The resulting velocity field is shared among the phases.

3.2. Turbulence Model

The equations of RNG $k-\varepsilon$ model are described as followed.

$$\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_i} (\rho k u_i) = \frac{\partial}{\partial x_j} \left( \alpha_{\mu_{ef}} \frac{\partial k}{\partial x_j} \right) + G_k + G_b - \rho \varepsilon - Y_{\varepsilon} + S_k$$

(5)

$$\frac{\partial}{\partial t} (\rho \varepsilon) + \frac{\partial}{\partial x_i} (\rho \varepsilon u_i) = \frac{\partial}{\partial x_j} \left( \alpha_{\mu_{ef}} \frac{\partial \varepsilon}{\partial x_j} \right) + C_{\varepsilon} \frac{\varepsilon}{k} (G_k + C_{\varepsilon \varepsilon} G_{\varepsilon}) - C_{\varepsilon \varepsilon} \rho \frac{\varepsilon^2}{k} - R_{\varepsilon} + S_{\varepsilon}$$

(6)

In these equations, $G_k$ represents the generation of turbulence kinetic energy due to the mean velocity gradients. $G_b$ is the generation of turbulence kinetic energy due to buoyancy. $Y_{\varepsilon}$ represents the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate. The quantities $\alpha_k$ and $\alpha_{\varepsilon}$ are the inverse effective Prandtl numbers for $k$ and $\varepsilon$, respectively. $S_k$ and $S_{\varepsilon}$ are source terms.

3.3. Boundary Conditions

Fluent 13 was used for unsteady simulations. The rotational movements of pump impeller and turbine runner were realized by using the Multiple Reference Frame method. Rotational speeds were specified with the pump impeller and the turbine runner. No-slip boundary condition was adapted at the walls. Data was transferred between the pump and the turbine through a prescribed interface. Residuals were set to $1 \times 10^{-5}$. Other chosen solution methods are shown in Table 1.

**Table 1. Solution methods.**

| Pressure-Velocity Coupling Scheme | PISO |
|----------------------------------|------|
| Spatial Discretization (Pressure) | PRESTO! |
| Spatial Discretization (Volume Fraction) | Geo-Reconstruct |
| Spatial Discretization (Momentum, $k$, $\varepsilon$) | Second Order Upwind |

In this paper, the rotational speed of the pump was set to 3000rpm, with a 80% liquid fill. The liquid phase was defined as a transmission oil. 20% volume of the fluid in the passage was defined as
air-phase, and remaining 80% volume was oil. The initial distribution of the two phases is shown in Fig. 3, where the red part indicates the air-phase.

![Phase distribution initialization](image)

**Figure 3.** Phase distribution initialization.

4. **Results and Discussions**

Flows with speed ratios $i$ of 0.95, 0.9, 0.8 and 0.5 were simulated. It can be seen that the zones of air phase were close to the rotation axis. By analyzing the computed flow field and the phase distributions, the energy transfer process and the flow mechanism could be figured out.

4.1. Air-phase Distribution in the Whole Flow Passage

Internal fluid flow of the hydraulic coupling is considered to have two components: one circumferentially about the coupling axis, and the other circulating fluid between the pump impeller and the turbine runner in the plane of the coupling axis. Flow analysis based on this assumption was carried out as below.

![Air distribution in the whole flow passage](image)

**Figure 4.** Air distribution in the whole flow passage.

Air-phase distributions inside the passage enclosed by the turbine runner and the pump impeller at working conditions with different speed ratios are shown in Fig. 4. The blue color represents the gas phase. It can be seen that, at operating point with $i = 0.95$, the fluid mainly moves circumferentially about the coupling axis. The interfaces between air and oil form a cylinder surface whose axis is the rotational axis of hydraulic coupling. When $i = 0.9$, the fluid begins to circulate between the pump impeller and the turbine runner. When moving circumferentially about the coupling axis in the turbine runner, the oil circulates to the pump before reaching the inner side of torus section. In this condition, air accumulates mainly in the pump side. When $i = 0.8$, the circumferential movement around the coupling axis weakens further, and circulation between pump impeller and turbine runner increases. In this condition, air accumulates mainly in the pump side. When $i = 0.5$, strong circulation between pump impeller and turbine runner occurs. In this condition, the air phase is concentrated in the center of torus section, with a number of volumes of air separated from the main volume of air.
4.2. Streamlines in the Whole Flow Passage
Streamlines inside the passage enclosed by the pump and the turbine at working conditions with different speed ratios are shown in Fig. 5.

![Figure 5. Streamlines in the whole flow passage.](image)

By comparing Fig. 4 and Fig. 5, it can be seen that velocity in air zones is smaller than in liquid zones, with more backflows within.

4.3. Flows on Auxiliary Surfaces
To analyze the detail flow structures in the fluid coupling, two auxiliary planes, namely an angular plane A-A and a cylindrical surface B-B (radius $R=53\text{mm}$), are specified, as shown in Fig. 6.

![Figure 6. Surface locations.](image)

4.3.1. A-A plane. The streamlines and the phase distributions on A-A plane are shown in Fig. 7. The pressure distributions are shown in Fig. 8. In these figures, pump impeller lies on the left, and the turbine runner lies on the right.

![Figure 7. Streamlines and phase distributions on A-A.](image)

The previously described flow fields and phase distributions at different speed ratios can also be seen in Fig. 7. It can also be observed that the center of the circulating flow between the pump impeller and the turbine runner shifts with different speed ratios.
By comparing Figures 7 and 8, it can be seen that the areas of low pressure coincide with the existence of the air phase.

4.3.2. B-B surface. Streamlines and the phase distributions on B-B surface are shown in Fig. 9. Pressure distributions are shown in Fig. 10.

The surface cuts through the center of the torus section where no air exists with high speed ratios. When \( i = 0.8 \), accumulations of air on the suction sides of the impeller blades can be observed. Same phase distributions in pump can be seen when \( i = 0.5 \), while in runner the air volumes stay in the center between the blades. Flow separations appear near the blades and the enclosing boundaries of the hydraulic coupling.

5. Concluding Remarks
In this study, a VOF two-phase model was applied to track phase interfaces of the flow inside the hydraulic coupling of partial filling.
It can be observed from the simulation results that circulation between the pump impeller and the turbine runner strengthens with decreasing speed ratio. When the speed ratio decreases, the phase interfaces incline to the pump side near the center of the rotating axis of the coupling until large circulation forms, where air volumes move to the center of torus section. Volumes of air tend to accumulate near the suction sides of the pump impeller blades, and the center between the turbine runner blades. In air zones, velocity is low and backflow occurs because of the low pressure. Flow separations appear near the blades and the enclosing boundaries of the hydraulic coupling.

It is believed that further simulations and analysis on the two-phase flows in hydraulic couplings with different partial fills are needed, which will benefit their designs and operations.

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