Numerical comparisons of the performance of a hydraulic coupling with different pump rotational speeds

Y Luo, L H Feng, S H Liu, T J Chen and H G Fan
State Key Laboratory of Hydro science and Engineering, Department of Thermal Engineering, Tsinghua University, Beijing, China, 100084
E-mail: liushuhong@mail.tsinghua.edu.cn

Abstract. A hydraulic coupling is a hydrodynamic device for transmitting rotating mechanical power. It is widely used in the machinery industry because of its advantages of high energy transmission efficiency, shock absorption and good adaptability, etc. In this paper, SIMPLEC algorithm and SST k-ω turbulence model were employed to simulate the steady state flows at operating conditions of two different rotational speeds (3000r/min and 7500 r/min) of the pump of a specified hydraulic coupling model. The results indicate the existence of similarity in the distributions of the flow fields between the two speeds, but the efficiency at the optimum condition is larger with higher rotational speed. It is concluded that the similarity principle of the efficiency of the hydraulic couplings does not apply in this case due to the relatively high rotating speed and small geometric specifications. It is also shown that the radially stratified pressure distribution on the torus section becomes more obvious with larger speed ratios, since the centrifugal movement plays more dominant roles over the circulating movement in these situations. When the speed ratio is small, with the completion of the circulating flow, the pressure distribution presents in a more circular pattern around the neutral zone of the torus section.

0. Introduction
Energy transmission by the utilization of the hydraulic coupling is one of the energy-saving technologies [1]. A hydraulic coupling can realize flexible connection by using fluid kinetic energy to transmitting power, which has some advantages in engineering applications, e.g., good adaptability and shock absorption, etc. Due to the complexity of the three-dimensional incompressible viscous flows, and the gas-liquid two-phase distribution variations at different working conditions in a partially filled hydraulic coupling, precise theoretical analysis of the flows and the hydraulic losses remains difficult.

The design of the hydraulic coupling has been evolved from the traditional one-dimensional beam method, to the optimization design based on three-dimensional CFD analysis [2]. CFD analysis of the flows in the hydraulic coupling has gain prevalence over the years, since it provides not only the external characteristic (e.g., transmitted torque, etc.), but also internal flow details, which are helpful in analysing the flow mechanism within the hydraulic coupling [3]. Fujitani Katsuro et al. firstly analyzed the internal flow conditions through numerically solving the 3D incompressible N-S equations in a hydraulic torque converter in 1988 [4]. Mitra N K et al. simulated the flow field in a hydraulic coupling in 2000, and summarized the influence of the coupling cavity types on its performance [5,6]. In 1993, Ma Wenxing and Zhang Bin tried 3D flow simulation of a torque converter
when input speed was 500r/min \[^7\]. In 2006 and 2007, Li Xuesong and Wu Kaifu, et al. simulated the fully-filled flows of a model hydraulic coupling with different input speeds (maximum speed was 3000r/min), whose computational results were in good agreement with the experimental results \[^8\,^9\]. It is believed that the analysis of the performance of the hydraulic couplings with full filling can provide some guidance for the design of the hydraulic couplings.

As stated above, the flows in the hydraulic coupling with input speed less than 3000r/min have been studied. However, few research has focused on the characteristics of hydraulic coupling with higher rotating speeds of the pump impeller. Nowadays, with the demand of higher ability of torque transmission, hydraulic couplings with higher rotating speeds are being developed. It is essential to have the knowledge of the variations of the characteristics of the hydraulic couplings with higher rotating speeds. In order to study the influence of the rotating speeds of the pump on the characteristics of a hydraulic coupling, a series of numerical calculations were performed with two different rotating speeds (namely 3000r/min and 7500r/min) of the pump impeller. Preliminary results are presented below.

1. Model and Mesh

1.1. Model

UG software package was used to build the geometric model of a regular hydraulic coupling without the auxiliary chamber. The main geometric parameters of this model were as followed: maximum diameter of flowpath \(D=150\text{mm}\); blade thickness \(b=1.5\text{mm}\); number of pump blades \(B_1=23\); number of turbine blades \(B_2=22\). The model of the flow passages of the hydraulic coupling and the schematic illustration of working mechanism was shown in Fig. 1.

![Figure 1. Flow passages and working mechanism.](image)

1.2. Mesh Discretization

Due to the relatively large number of blades, 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 120), grids were refined in the boundary layer on each blade surface. The minimum height of the first grid layer was 0.05. The mesh in pump is shown in Fig. 2.

![Figure 2. Mesh in pump.](image)
2. Verification of Mesh Independence

Five different sets of mesh were chosen for the verification of mesh independence, under the same working condition (input rotational speed 3000rpm, speed ratio 0.95). The computed pump torque is shown in Fig. 3. It could be seen that as mesh number increases, the computed pump torque gradually converges. Considering the calculation accuracy and the computational time needed, the mesh system with about 10 million grids was chosen for the final calculations.

![Figure 3. Curve of pump torque in different mesh size.](image)

3. Turbulence Model and Boundary Conditions

3.1. Turbulence Model

The SST $k$-$\omega$ turbulence model was used in this paper. The equations are described as follow:

\[
\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho ku_j) = \frac{\partial}{\partial x_j} \left( \Gamma_k \frac{\partial k}{\partial x_j} \right) + G_k - Y_k + S_k
\]  
(1)

\[
\frac{\partial}{\partial t}(\rho \omega) + \frac{\partial}{\partial x_j}(\rho \omega u_j) = \frac{\partial}{\partial x_j} \left( \Gamma_\omega \frac{\partial \omega}{\partial x_j} \right) + G_\omega - Y_\omega + D_\omega + S_\omega
\]  
(2)

where $G_k$ represents the turbulence kinetic energy, $G_\omega$ represents $\omega$ equation, $\Gamma_k$ and $\Gamma_\omega$ are effective diffusion terms of $k$ and $\omega$, $Y_k$ and $Y_\omega$ are divergence terms of $k$ and $\omega$, $D_\omega$ is orthogonal divergence term, $S_k$ and $S_\omega$ are source terms.

3.2. Boundary Conditions

Fluent 13 was used for steady simulations. The rotational movements of pump and turbine were realized by using the Frame Motion method. No-slip boundary condition was adapted at the walls. Rotational speeds were specified with the pump impeller and the turbine runner. Data was transferred between the pump and the turbine through a prescribed interface. Other chosen solution specifications are shown in Table 1.

![Table 1. Solution specifications.](image)

4. Results and Discussions

In this paper, input rotating speeds of the pump for numerical calculations were set as 3000r/min and 7500r/min, respectively. Steady flow inside the hydraulic coupling at 42 operating points with speed...
ratios increasing from -2.0 to 2.0 were simulated with each rotating speed of the pump, covering the traction condition, the backward condition, and the reversing damped condition of the hydraulic coupling, as shown in Table 2.

**Table 2.** Operating points for calculations.

| Working condition       | Number of operating points studied | Corresponding speed ratios                                      |
|-------------------------|------------------------------------|----------------------------------------------------------------|
| Traction                | 17                                 | 0, 0.1, 0.3, 0.5, 0.6, 0.7, 0.8, 0.9, 0.91, 0.92, 0.93, 0.94, 0.95, 0.96, 0.97, 0.98, 0.99 |
| Backward                | 9                                  | 1.01, 1.03, 1.05, 1.08, 1.1, 1.2, 1.4, 1.6, 2.0                 |
| Reversing damped        | 16                                 | -0.2, -0.4, -0.5, -0.6, -0.7, -0.9, -0.95, -0.98, -0.99, -1, -1.01, -1.05, -1.1, -1.2, -1.5, -2.0 |

By deriving the input torque, the output torque and the efficiency of each operating point, characteristic curves could be drawn to compare the performance of the unit at different input speeds. By analyzing the computed flow field, the energy transfer process and loss source could be specified. Some parameters used for analyzing the performance of the hydraulic coupling are shown in Table 3.

**Table 3.** Parameters.

- $n_B$: Pump speed (r/min)
- $n_T$: Turbine speed (r/min)
- $M_B$: Pump torque (N·m)
- $M_T$: Turbine torque (N·m)
- $\lambda_B$: Torque factor of pump, $\lambda_B = \frac{M_B}{\rho gn_B^2 D^2}$ (min$^2$/m)
- $i$: Speed ratio, $i = \frac{n_T}{n_B}$
- $K$: Torque ratio, $K = -\frac{M_T}{M_B}$
- $\eta$: Efficiency, $\eta = -\frac{M_T}{n_T/M_B n_B} = Ki \approx i$

4.1. Universal External Characteristic

The universal external characteristic curves show the relationship between efficiency $\eta$, torque $M$ and speed ratio $i$ at different pump speeds at traction condition. It reflects the maximum ability to transfer torque of the hydraulic coupling (Fig. 4 and Fig. 5).
In Fig. 4, the right curve shows the detail information of the left. According to the calculation results, when speed ratio $i$ is less than 0.9, the value of efficiency $\eta$ nearly equals to the value of $i$, and the value of the torque ratio $K$ is nearly 1. Efficiency $\eta$ becomes smaller than $i$, when $i$ is greater than 0.9. In the latter case, the torque can’t be seen as constant during torque transmission. When input rotational speed $n_B = 3000\text{r/min}$, the highest efficiency equals 0.925 appears at the operating point with speed ratio $i^* = 0.95$. With $n_B$ increasing to 7500r/min, the highest efficiency operating point doesn’t change, but $\eta$ increases to 0.929. As the left curve in Fig. 4 shows, the values of $\eta$ on each operating point with different $i$ roughly coincides, even though the input speed is different. This is also consistent with the similarity principle, i.e., ideally, efficiency doesn’t change when speed ratio is the same at similar working conditions. But the right curve in Fig. 4 shows that the value of $\eta$ is generally larger with higher input rotational speed ($n_B = 7500\text{r/min}$ ) when $i > 0.9$.

According to the $M-i$ and $K-i$ curves, the value of the pump torque $M_B$ nearly equals to the value of the turbine torque $M_T$ when $i < 0.9$, and $M_B > M_T$ when $i > 0.9$. As the right curve in Fig. 5 shows, the value of torque ratio $K$ is larger with higher $n_B$. The hydraulic coupling could hardly change the transferred torque, so the value of $\eta$ was closer to the value of $i$. That was why $\eta$ becomes larger with higher $n_B$. It shows that the torque and power which the hydraulic coupling could transfer is much larger with higher input speed. So for high-power working condition, the study of coupling with high input rotational speed will be very meaningful.

4.2. Primary Characteristic
The primary characteristic curve shows the relationship between the torque factor of the pump and the speed ratio, as shown in Fig. 6. The torque factor of the pump $\lambda_B = M_B/\rho g n_B^2 D^3$ is used to evaluate the energy capacity of the hydraulic coupling.

![Figure 5. $M-i$ and $K-i$ curves.](image)

![Figure 6. Primary characteristic curve.](image)
At the best efficiency point (BEP), the torque factor of the pump $\lambda_B$ is $1.36 \times 10^{-6}$ with input speed 3000r/min and $1.53 \times 10^{-6}$ with input speed 7500r/min. According to the similarity principle, if only the input speed changes, torque factor remains constant. But Fig. 6 shows that the value of the torque factor is larger when input speed is higher. A preliminary analysis of this discrepancy is as follow.

Ekman number $Ek = \nu / D^2 n_B$ defines the ratio between viscous forces to Coriolis forces, where $\nu$ is the dynamic viscosity of the fluid. As $D$ and $\nu$ remain constant, $Ek$ becomes different with changing $n_B$. Large difference in $Ek$ indicates the unsimilarity between the rotating flows, which in turn causes a variation in torque factor of the pump, due to the inapplicability of the similarity principle. So for general hydraulic couplings, care must be taken when considering the constant torque factor conversion to higher-speed working conditions.

4.3. Total External Characteristic

The total external characteristic is the characteristic describing the behavior of the coupling at the traction condition, the backward condition and the reversing damped condition in general, as shown in Fig. 7. Since the traction condition has been analyzed in the previous section, this section will focus on the characteristic on the backward condition and the reversing damped condition.

![Figure 7. Total External Characteristic.](image)

The backward condition refers to the condition when the turbine runner speed is greater than the pump impeller speed, the $i > 1$ part of the curve in Fig. 7. Due to the different input rotational speeds, when $n_B = 7500$ r/min, pump torque has greater changes than that with $n_B = 3000$ r/min, with the increase of the speed ratio. The higher input speed is, the greater the variations of the curve of backward characteristic is with increasing speed ratios.

The reversing damped condition refers to the working conditions when the rotation of the turbine runner reverse, which means that the turbine runner is driven by working machine, i.e., the $i < 0$ part of the curve in Fig. 7. Numerical results show that the torque becomes minimum when speed ratio is -1.01 on reversing damped condition, which means circulation flow rate became zero. Due to the differences between the shape and the number of the blades of the pump and turbine, minor differences exist between the characteristic of the two sides. When the reverse turbine speed becomes greater than the pump, the flow rate and the torque increase, and the characteristic stays stable. Minimum pump torque is larger when $n_B = 7500$ r/min than when $n_B = 3000$ r/min, so is the variation range. At reversing damped condition, the amplitude of the turbine torque caused by the forces by the liquid equals to the loading torque, only in the opposite direction, generating a braking resistance. The transmitted torque is larger when $n_B = 7500$ r/min.

4.4. Internal Flow Analysis
Three operating points (small speed ratio point, best efficiency point, i.e., BEP, and large speed ratio point) were chosen to analyze the computed pressure distribution and streamlines under tractive working condition with different input rotational speeds (3000r/min and 7500r/min). The location of the angular plane on which show the pressure and streamlines distributions is illustrated in Fig. 8.

The angular plane is a surface of torus section, with the pump on the left, and the turbine on the right. The pressure distributions and streamlines are shown in Fig. 9 and Fig. 10 under different speed ratios.

Figure 8. Angular plane location.

Figure 9. Pressure distributions and streamlines (\(n_B = 3000\text{r/min}\)).

Figure 10. Pressure distributions and streamlines (\(n_B = 7500\text{r/min}\)).
By analyzing the pressure distribution, it can be easily found that the pressure becomes larger with increasing input speed. But whatever the input speed is, when the speed ratio is small, the pressure presents in a radial distribution from the center of torus section and the low pressure area appeared on the center; when the speed ratio increases, a radially stratified pressure distribution develops gradually.

Fluid mainly does centrifugal movement together with the coupling, and the hydraulic head difference caused by the slip is small when the speed ratio is large, causing a weak circulation movement. Centrifugal force plays a dominant role in this case, and a radially stratified pressure distribution around the rotating axis of the hydraulic coupling can be observed in the hydraulic coupling. When the speed ratio is small, circulating movement on the torus section becomes stronger, and the fluid completes a full circulation, so the pressure presents in a circular distribution around the neutral zone of the torus section.

By analyzing the streamlines, it can be seen that when the speed ratio is small, the slip of pump and turbine is large, generating a large centripetal movement, resulting in a circular structure of streamlines; with increasing speed ratio, circulation flow rate drops significantly, and the flow separations and local vortices appear near the wall, causing an energy loss. Similarities can be observed in the flow conditions and the loss sources at different conditions of different input speeds.

5. Concluding Remarks
It is proved that SST $k-\omega$ turbulence model is capable to predict the characteristic of the hydraulic coupling and its internal flows. Meanwhile, the characteristics with high rotating speed of the pump can’t be accurately predicted by conversion from low-speed characteristic using similarity principle directly. It can be observed that flow condition and loss source have some similarities at different speed conditions.

Acknowledgment
The authors would like to thank the National Basic Research Program of China (NO.2011CB707204) for its financial support.

References
[1] Liu Y C and Yang N Q 2006 Application and Energy Saving Technology of Hydraulic Coupling (Beijing: Chemical Industry Press) (In Chinese)
[2] Wu X F 2004 The CAD/CFD Research on the Modern Design Method of Multi-blade Centrifugal Fan for Air Conditioning System (Wuhan: Huazhong University of Science and Technology) (In Chinese)
[3] Cai Z L and Luo S 2001 Journal of Huazhong University of Science and Technology 29 7 (In Chinese)
[4] Fujitani K, Himeno R and Takagi M 1988 SAE Transactions 97 1366
[5] Huitenga H and Mitra N K 2000 ASME Journal of Fluids Engineering 122 683
[6] Huitenga H and Mitra N K 2000 ASME Journal of Fluids Engineering 122 689
[7] Ma W X, Zhang Bi and Luo B 1992 Journal of Jilin University of Technology 22 52 (In Chinese)
[8] Li X S, Ma W X, Wu Y Z, Liu C B and Cai W 2006 3D Numerical Simulation on Flow Field in Coupling during Braking and Braking Torque Calculation 4th National Conf. of Fluid Power Transmission & Control (Dalian, China, 2-5 August 2006) p 659 (In Chinese)
[9] Wu K F 2007 Numerical Simulation of the Flow Field and Research on the Control of Flow Separation in Hydrodynamic Coupling (Jilin: College of Mechanical Science and Engineering, Jilin University) (In Chinese)