Postprint: Strongly Confined Flow and Cavitation in Hydraulic Turbines

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Date: 2017-11-07T00:00:00+00:00

Abstract

To characterize the flow features within a hydraulic turbine brake and their influence on the brake' s torque performance, the flow structure and cavitation phenomena inside the brake were investigated using computational fluid dynamics methods. The distribution of flow parameters near the runner and baffle during transient startup and shutdown processes of the turbine brake was obtained, the location and morphology of cavitation zones within the brake' s enclosed cavity were captured, and the fluid loads acting on the baffle were solved. The results indicate that large-scale flow structures exist within the brake cavity, which develop axially as the axial distance between the baffle and runner increases; cavitation zones appear near the back surface and root of the runner blades, with the cavitation phase volume fraction increasing as the runner rotational speed increases, yet without exerting a significant influence on the brake' s torque coefficient; the resultant direction of the axial fluid force acting on the baffle always points toward the runner side, and its magnitude increases with both the runner rotational speed and the spacing between the impeller and baffle.

Full Text

Preamble

Highly Confined Flows and Cavitation Phenomenon in a Hydraulic Retarder

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Abstract: To depict the flow pattern and its influence on the torque performance of a hydraulic retarder, computational fluid dynamics technique is employed to explore the inner flow and cavitation phenomenon involved. Distributions of flow parameters in the vicinity of the rotor and the baffle during the start-up and stop process of the hydraulic retarder are obtained. Cavity position and morphology are virtually visualized in the closed retarder chamber. Hydraulic loads acting on the baffle are calculated as well. The results show that large-scale flow structures exist in the retarder chamber. As the axial distance between the baffle and the rotor increases, these large-scale flow structures develop in the axial direction. Cavitation occurs near the back surface of the rotor blades and near the rotor hub. Cavitation volume fraction rises as the rotational speed of the rotor increases but exerts a moderate influence on torque coefficient. The direction of the resultant hydraulic force acting on the baffle always points towards the rotor, and the force magnitude increases with the rotational speed and the axial distance between the rotor and the baffle.

Key words: Turbine retarder; Flow structure; Transient process; Cavitation; Torque

The advantages of hydraulic turbine retarders have been recognized in numerous engineering fields. In recent years, hydraulic turbine retarders have been tentatively applied to aircraft arresting systems with preliminary success. As a key component in the entire braking system, the retarder’ s performance determines both the braking effectiveness and system safety. Currently, the design of hydraulic turbine retarders still relies heavily on empirical methods, with corresponding design theories remaining inadequate. With advances in computational fluid dynamics (CFD) technology, the performance and internal flow characteristics of turbine retarders can be predicted during the design phase, which holds significant importance for optimizing retarder design.

From a structural perspective, optical measurement of the internal fluid flow in turbine retarders presents considerable challenges. If probe-based velocity measurement is employed, only the absolute velocity within the stator blade passages can be obtained, and the measurement apparatus and stepping mechanism are complex. Furthermore, the rotor speed in turbine retarders may exceed 3000 r/min, resulting in high tip speeds that generate hydrodynamic pressures posing serious threats to transparent experimental facilities.

To generate large braking torque, one can either increase the rotor speed or enlarge the retarder’ s outer diameter. Both approaches increase the likelihood of cavitation within the retarder. However, research on cavitation phenomena in turbine retarders has rarely been reported, with even basic speculation on the topic being scarce.

This study focuses on a hydraulic turbine retarder with straight blades on both the rotor and stator, featuring an axially adjustable baffle within the retarder chamber. The research aims to explain the large-scale flow structures within the turbine retarder and their influence on torque performance, with particular emphasis on investigating the morphology, location, and evolution of cavitation zones. Unsteady numerical simulation is employed to model the complete start-up and stop processes of the turbine retarder, capturing transient characteristics of flow parameters and cavitation phenomena. The effects of rotor speed and baffle position on internal flow characteristics and performance are considered, and the hydraulic loads acting on the baffle and their variation patterns are determined from flow parameter distributions.

1 Turbine Retarder Model

The internal structure of the hydraulic turbine retarder is shown in Figure 1: see original paper and (b). The retarder rotor contains 9 straight blades, while the stator contains 7 straight blades. A baffle is installed on the stator side, with its axial position being adjustable. The axial position of the baffle is expressed as the axial distance x between the baffle and the rotor blade side surface, which can be adjusted between 0.0–55.0 mm. The relative position between the rotor and stator remains constant. As evident from Figure 1: see original paper, different baffle axial positions result in different cavity volumes on the left side of the baffle, which can be inferred to produce different braking torques. This study investigates two baffle axial positions: $x = 25$ mm and $x = 45$ mm.

The rotor diameter D_2 is 375.0 mm, and the blade width b_2 is 30.0 mm. At a rotor speed n of 3200 r/min, the maximum linear velocity within the turbine is approximately 63 m/s. For the stator portion, which is connected to the retarder housing as an integral unit, the inner surface diameter D of the retarder housing is 385 mm, and the stator blade width b_1 is 55.0 mm. The radial clearance s_1 between the baffle outer edge and the housing inner surface is 5.0 mm, and the circumferential clearance s_2 between the baffle and stator blades is 2.0 mm, as shown in Figure 1: see original paper. The presence of these clearances allows the working fluid to pass through at relatively high velocities, connecting the left and right cavities of the baffle while creating disturbances to the flow within the cavities. Figure 1: see original paper shows the physical prototype of the retarder baffle and stator.

2.1 Governing Equations and Numerical Model

The working medium enclosed within the turbine retarder cavity is water at room temperature with no inflow or outflow. The fluid is assumed to be incompressible, and the cavity flow dominated by rotor blade rotation is treated as a three-dimensional, unsteady turbulent flow governed by the Reynolds-averaged Navier-Stokes (RANS) equations. The renormalization group (RNG) k- turbulence model is employed to close the governing equation system. Compared with the standard k- turbulence model, the RNG k- model offers advantages in handling flow behavior induced by rotor rotation. Moreover, since the rotor blades studied here are straight and the cavity geometry is relatively regular, the RNG k- turbulence model can yield results that approach physical reality.

Cavitation is a transient phase-change phenomenon, and the Rayleigh-Plesset equation model provides a relatively detailed description of its mechanism. This model not only reflects cavitation characteristics at the fundamental level of bubble dynamics but also comprehensively considers various parameters affecting cavitation, such as initial gas nuclei content and nuclei diameter.

2.2 Initial and Boundary Conditions

Based on actual retarder operation, the cavity is initially filled with pure water at room temperature, with an initial static pressure of 0.3 MPa. Since the turbine retarder has no inlet or outlet, no boundary conditions are set for them. The rotor region is designated as a rotating domain, with rotor-stator interface conditions established between the rotating and stationary domains. These boundary conditions are configured in the CFD software ANSYS CFX, which also provides the solver. No-slip boundary conditions are applied at all solid walls, with scalable wall functions employed for near-wall treatment. The surface roughness of all walls in contact with the fluid is set to 0.015 mm.

2.3 Mesh Generation

The computational domain scale is 1:1 relative to the physical dimensions. To ensure simulation accuracy, high-precision meshes are used for spatial discretization of all computational domains. The computational domain accounts for radial clearances between rotor blades and housing, between stator blades and housing, and the circumferential clearance between the baffle and stator blades, with targeted meshing applied to these gap regions. Non-uniformly distributed meshes are established between gap region meshes and adjacent computational meshes to ensure effective data transfer at interfaces.

Mesh independence has been verified. The final adopted scheme employs approximately 2,200,000 total mesh cells. The near-wall mesh for the $x = 45$ mm case is shown in [Figure 2: see original paper], with y+ values ranging between 27–40 for this mesh scheme.

2.4 Torque Performance of the Retarder

During retarder operation, instantaneous changes in rotor speed inevitably cause instantaneous variations in retarder characteristic parameters. In the numerical simulation, the rotor speed variation curve is constructed based on actual operational data and applied as a boundary condition, as shown in [Figure 3: see original paper]. This curve encompasses the complete start-up and stop processes of the turbine retarder. The entire start-stop process lasts approximately 3.5 s, experiencing rapid rotor speed increase, gradual decrease, and rapid decline, directly reflecting the functional characteristics of the retarder.

The time step in the numerical simulation must match the rotor speed to capture instantaneous changes in retarder performance parameters, their distributions, and the evolution of flow structures within the retarder cavity. Different time steps are defined to correspond with various rotor rotational speeds, ensuring that the rotor rotation angle per time step is less than 1°.

Torque is the most direct parameter reflecting turbine retarder performance and is closely related to internal flow characteristics. When the turbine retarder is filled with pure water, torque T can be calculated using the following equation:

$$T = \lambda_T \rho D^5 n^2$$

where λ is the torque coefficient, is the density of pure water, D is the inner diameter of the turbine retarder cavity, and n is the rotational speed. The fluid dynamic loads on the rotor obtained from numerical simulation can be used to calculate the torque T acting on the rotor, thereby determining the dimensionless torque coefficient λ through the above equation. The calculated instantaneous torque coefficient variation throughout the retarder operation is shown in [Figure 4: see original paper].

As seen in [Figure 4: see original paper], the torque coefficients for both baffle positions remain relatively stable throughout the operation, with no significant fluctuations except during start-up and stop phases. The torque coefficient value at $x = 45$ mm is significantly higher than that at $x = 25$ mm, indicating from the retarder' s external characteristics that increasing the volume of the cavity left of the baffle helps improve torque performance. Notably, while rotor speed continuously changes during retarder operation, the torque coefficient remains relatively stable, suggesting that temporal variations in cavity flow parameters do not significantly affect overall torque performance. Additionally, the magnitude of λ is influenced by factors such as retarder size and rotor blade shape.

2.5 Internal Flow Characteristics in the Turbine Retarder

The influence of baffle position on torque coefficient reflected in [Figure 4: see original paper] is closely related to internal flow characteristics. From the component configuration within the turbine retarder cavity, the flow exhibits distinct confined flow features, primarily influenced by boundary conditions and rotor rotation.

Without a stator and with a sufficiently large axial distance x between the baffle and rotor blade side surface, a large-scale induced vortex appears in the left cavity of the baffle, with its axis coinciding with the rotor axis. When the volume of the right cavity of the baffle changes, its internal flow structure also changes accordingly. [Figure 5: see original paper] shows the instantaneous flow structure on the meridional plane through the rotor axis for $x = 25$ mm, presenting four rotor speeds—3200, 2500, 2000, and 1500 r/min—for comparison.

Despite different rotor speeds, the flow structures on the meridional plane are similar. Between rotor blades, vortices with scales comparable to the blade radius appear, extending into the stator region as shown in [Figure 5: see original paper]. However, the axial development of these vortices is significantly suppressed by the baffle. A vortex pair emerges in the right cavity of the baffle, insensitive to rotational speed changes. Both vortices in the pair have comparable scales and intensities. Notably, the driving force for this vortex pair formation primarily originates from gap flows between the baffle outer edge and housing, between the baffle and stator blades, and from wall effects in the right cavity of the baffle. Removed from the influence of rotor blade rotation, these two vortices maintain relatively regular and stable shapes.

Using a similar presentation method as [Figure 5: see original paper], the flow structure morphology for $x = 45$ mm is shown in [Figure 6: see original paper]. For the four rotational speed conditions, the vortex structures are also similar, and a large-scale vortex appears within the rotor blade passage due to the combined effects of the stator blades and baffle. Compared with [Figure 5: see original paper], the increased cavity volume between the rotor and baffle provides space for axial vortex development. Meanwhile, the vortices in the right cavity of the baffle are suppressed due to cavity reduction, particularly those near the hub. From [Figure 6: see original paper], it can be inferred that fluid flowing through the gap between the baffle and retarder housing inner wall continuously supplies energy to the medium in the right cavity, while the shape of the right cavity wall suppresses vortex development caused by gap flow within the right cavity.

The flow state within rotor-stator gaps serves as an important indicator of rotor-stator interaction phenomena. To describe this interaction, the axial mid-plane between rotor and stator blades is defined as the rotor-stator interface, from which absolute velocity distributions are extracted. [Figure 7: see original paper] shows velocity distributions for both $x = 25$ mm and $x = 45$ mm cases, considering two rotational speeds: $n = 3200$ and 1500 r/min. The rotor rotation direction in [Figure 7: see original paper] is clockwise.

For the two baffle positions considered, the high-velocity zone morphology at the rotor blade outer edge differs significantly. At $x = 25$ mm, the circumferential continuity of the blade outer edge velocity region is strong, with medium at higher circumferential velocity squeezing into the right cavity through the radial gap between the baffle and housing inner wall. This situation alleviates as the axial distance between rotor blades and baffle increases at $x = 45$ mm.

As rotor speed decreases, the overall fluid velocity shows a decaying trend. Compared with Figure 7: see original paper, dispersed small-area high-velocity zones appear near the rotor blade outer edge in Figure 7: see original paper, though the velocity distribution characteristics remain similar between the two figures, differing only in overall magnitude. Compared with Figure 7: see original paper, the velocity distribution in Figure 7: see original paper tends toward greater uniformity.

In [Figure 7: see original paper], rotor and stator blade positions can be identified from the velocity distributions. It can be predicted that changes in either rotor blade count or stator blade count will alter the velocity distribution morphology on this plane. The rotor-stator interaction region here differs from the general concept of rotor-stator interference in turbomachinery, as the latter involves inflow and outflow. The influence of large-scale flow structures in the left cavity of the baffle on the velocity distribution at the rotor-stator interface cannot be neglected, as evidenced by the dispersed velocity cells in the radial mid-section shown in [Figure 7: see original paper].

3 Cavitation Phenomena

The impact of cavitation on hydraulic turbine retarders has not yet been systematically explained. Existing reports confirm that cavitation may reduce retarder torque performance by 3%. Noise induced by cavitation bubble collapse can also be perceived during actual retarder operation. For the four rotational speeds studied, isosurfaces of cavitation phase volume fraction within the retarder cavity are shown in [Figure 8: see original paper] and [Figure 9: see original paper].

Cavitation location, cavity morphology, and cavitation phase volume fraction are commonly used to describe cavitation phenomena. Describing cavitation formation and development is not only necessary for understanding cavitation itself but also helps identify locations where components may undergo cavitation erosion.

As seen in [Figure 8: see original paper], cavitation zones are dispersedly distributed near rotor blades. No significant cavitation zones appear between stator blades, which is related to the straight blade configuration. Larger cavitation zones appear near the rotor hub and blade suction surface, occupying a substantial portion of the total cavitation volume. Some cavitation zones are located near the rotor-stator interface at a radial distance of approximately 1/3 D_2 from the rotor axis. In [Figure 8: see original paper], although cavitation is significantly suppressed at the lowest rotational speed of $n = 1500$ r/min, the relationship between cavitation intensity and rotor speed does not exhibit a clear monotonic trend.

When $x = 45$ mm, axial flow development is more pronounced, but the overall cavitation intensity is weaker than at $x = 25$ mm, as shown in [Figure 9: see original paper]. At $n = 3200$ r/min, the total cavitation phase volume fraction is significantly lower than in Figure 8: see original paper, though the cavitation locations are similar. In Figure 9: see original paper and (c), cavitation concentrates in several radially limited strips with poor one-to-one correspondence to rotor blades. In Figure 9: see original paper, cavitation zones are more concentrated with more strip-like morphology, while their locations differ from other operating conditions.

4 Axial Force on the Baffle

Baffle axial position adjustment is achieved through a hydraulic booster connected to the baffle. The axial force acting on the baffle serves as an important reference for hydraulic booster selection. Meanwhile, since the axial force on the baffle originates from fluid pressure on the baffle surface, the resultant force may continuously vary during retarder operation, significantly affecting retarder vibration characteristics.

Based on the instantaneous pressure distribution on the baffle surface obtained from numerical calculation, the variation of the axial force component F in the x-direction with rotor rotation can be calculated, as shown in [Figure 10: see original paper]. For both baffle positions ($x = 25$ mm and $x = 45$ mm), F always points toward the rotor, evident from the consistently positive F values in [Figure 10: see original paper]. Additionally, the two curves in [Figure 10: see original paper] follow the shape of the speed variation curve shown in [Figure 3: see original paper]. It can be concluded that the resultant axial hydraulic force on the baffle increases with rotor speed. For the two baffle positions studied, F variation is relatively smooth for $x = 45$ mm, while the axial force curve for $x = 25$ mm contains several segments with sharp F drops. Since the axial force is obtained by integrating pressure over the baffle surface, it is difficult to evaluate which baffle surface contributes more to these fluctuations. However, comparison of axial force magnitudes between $x = 25$ mm and 45 mm cases clearly shows larger axial forces under $x = 45$ mm conditions. Furthermore, from [Figure 5: see original paper] and [Figure 6: see original paper], a single large-scale vortex contributes more to increasing the resultant axial force on the baffle than the vortex pair appearing in the right cavity.

The axial force on the baffle is related to multiple factors, among which the clearance dimensions between the baffle and housing and between the baffle and stator blades cannot be ignored. Since the medium in the right cavity of the baffle is sensitive to fluid entering through the clearances, clearance dimensions inevitably affect the static pressure distribution in the right cavity and on the baffle surface, necessitating comprehensive consideration of both clearance size and baffle axial position contributions to F .

Conclusions

(1) Large-scale flow structures exist within the hydraulic turbine retarder cavity. As the axial distance between the baffle and rotor increases, the flow structure in the cavity between them develops axially, while the vortex pair in the right cavity of the baffle is suppressed. Rotational speed variation has no significant effect on flow structure characteristics in either side of the cavity.

(2) During hydraulic turbine retarder operation, cavitation zones appear near the rotor hub and blade suction surface, with different cavitation zone morphologies associated with each blade. No monotonic relationship exists

between cavitation phase volume fraction and rotor speed. As the cavity volume between the baffle and rotor increases, the total volume fraction occupied by cavitation zones decreases.

(3) Throughout the entire hydraulic turbine retarder operation, the torque coefficient remains essentially stable despite rotor speed variations. The axial component of the fluid force acting on the baffle increases with rotor rotational speed and increases when the cavity volume between the rotor and baffle becomes larger.

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Note: Figure translations are in progress. See original paper for figures.