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Research Article

Dynamics Models of Interacting Torques of Hydrodynamic Retarder Braking Process

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Abstract: Hydrodynamic retarder is a kind of assist braking device, which can transfer the vehicle kinetic energy into the heat energy of working medium. There are complicated three-dimensional viscous incompressible turbulent flows in hydrodynamic retarder, so that it is difficult to represent the parameters changing phenomenon and investigate the interactional law. In order to develop a kind of reliable theoretical model for internal flow field, in this study, the dynamics models of interacting torques between impellers and working fluid were constructed based on braking energy transfer principle by using Euler theory to describe the flow state in view of time scale. The model can truly represent the dynamic braking process.

Keywords: Brake systems, dimensionless characteristics, hydrodynamic retarder

INTRODUCTION

Hydrodynamic retarder is a kind of assist braking device, which can transfer the vehicle kinetic energy into the heat energy of working medium. The special features, that are large quantity torque, smooth braking effect and good radiator capability, draw the attentions of public. The energy transition process of hydrodynamic retarder is the coupling action result of different parameters in multi-physical fields, such as structure, flow, time and etc. The complexity results the heavy going of research on working mechanism and design methodology. So the basic theoretical research is quite meaningful to the development of automobile manufacturing industry (Limpert, 1975; Klaus, 1990a, b; Helmut *et al*., 1992).

There are complicated three-dimensional viscous incompressible turbulent flows in hydrodynamic retarder (Abe *et al*., 1991), so that it is difficult to represent the parameters changing phenomenon and investigate the interactional law. In order to develop a kind of reliable theoretical model for internal flow field, theoretical calculation of flow distribution was deduced to predict the output braking torque.

In this study, the dynamics models of interacting torques between impellers and working fluid were constructed based on braking energy transfer principle by using Euler theory to describe the flow state in view of time scale. The model can truly represent the dynamic braking process.

BASIC GOVERNING EQUATION

In fluid dynamics, the Euler equations are a set of equations governing in viscid flow. The equations represent conservation of mass (continuity), momentum and energy, corresponding to the Navier-Stokes equations with zero viscosity and heat conduction terms. The Euler equations can be applied to compressible as well as to incompressible flow-using either an appropriate equation of state or assuming that the divergence of the flow velocity field is zero, respectively.

In differential form, the equations are:

$$\begin{aligned} &\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0 \\ &\frac{\partial \rho \mathbf{u}}{\partial t} + \nabla \cdot (\mathbf{u} \otimes (\rho \mathbf{u})) + \nabla p = 0 \\ &\frac{\partial E}{\partial t} + \nabla \cdot (\mathbf{u}(E + p)) = 0 \end{aligned} \tag{1}$$

where,

ρ = The fluid mass density
u = The fluid velocity vector, with components u, v and w
E = $\rho e + ½ \rho (u^2 + v^2 + w^2)$ is the total energy per unit volume, with e being the internal energy per unit mass for the fluid
p = The pressure

In vector and conservation form, the Euler equations become:

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$$\frac{\partial \mathbf{m}}{\partial t} + \frac{\partial \mathbf{f}_x}{\partial x} + \frac{\partial \mathbf{f}_y}{\partial y} + \frac{\partial \mathbf{f}_z}{\partial z} = 0 \tag{2}$$

where,

$$\mathbf{m} = \begin{pmatrix} \rho \\ \rho u \\ \rho v \\ \rho w \\ E \end{pmatrix}; \quad \mathbf{f}_x = \begin{pmatrix} \rho u \\ p + \rho u^2 \\ \rho uv \\ \rho uw \\ u(E+p) \end{pmatrix};$$

$$\mathbf{f}_y = \begin{pmatrix} \rho v \\ \rho uv \\ p + \rho v^2 \\ \rho uw \\ v(E+p) \end{pmatrix}; \quad \mathbf{f}_z = \begin{pmatrix} \rho w \\ \rho uw \\ \rho vw \\ p + \rho w^2 \\ w(E+p) \end{pmatrix} \tag{3}$$

and f_x, f_y, f_z are all fluxes.

DESCRIPTION IN EULER EQUATION FOR ONE SPATIAL DIMENSION

For certain problems, especially when used to analyze compressible flow in a duct or in case the flow is cylindrically or spherically symmetric, the one-dimensional Euler equations are a useful first approximation. Generally, the Euler equations are solved by Riemann's method of characteristics. This involves finding curves in plane of independent variables along which partial differential equations degenerate into ordinary differential equations. Numerical solutions of the Euler equations rely heavily on the method of characteristics.

Following the Euler One Spatial Dimension Assumption, the flow can be described in circulating circle as Fig. 1.

For every flow particle, its velocity can be projected on Axial Plane of Blade passing through as Fig. 2. In order to show the relationship of velocity vectors in the inlet and outlet of blade, the axial projection plane was draw for the central stream surface. In Fig. 3, the velocity vector triangles were formatted at different positions in the inlet and outlet.

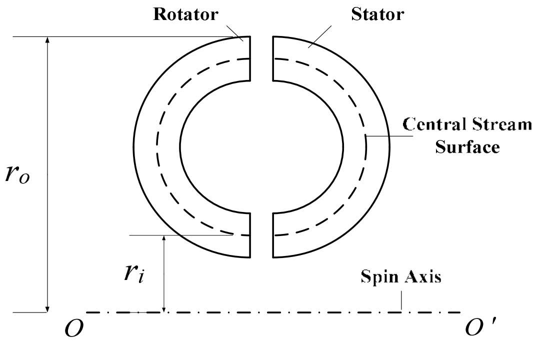

Fig. 1: Flow passage on axial plane of hydrodynamics retarder (circulating circle)

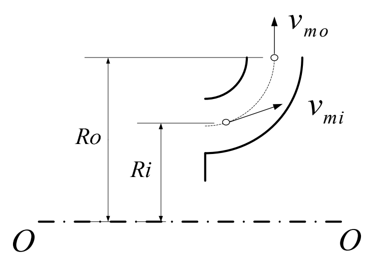

Fig. 2: Velocity projection on axial plane of blade

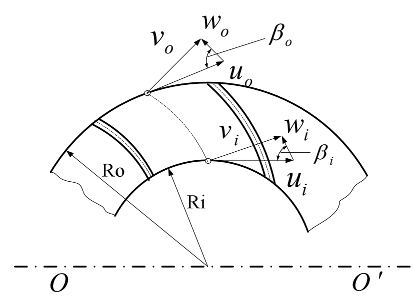

Fig. 3: Projection of blade middle rotating face on axial direction

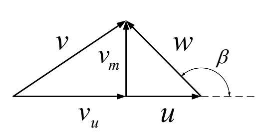

Fig. 4: Velocity triangle of fluid particle

In analysis, the absolute velocity was expressed in two orthogonal sub-vectors with v_m and v_u:

$$v = v_m + v_u \tag{4}$$

where,

- v_m : Meridional component of velocity, which is tangent to the flow line in axial plane
- v_u : Peripheral component of velocity

In the Velocity Triangle of Fluid Particle that was showed in Fig. 4, the absolute velocity, relative velocity, following velocity, meridional component of velocity and peripheral component of velocity were described to express the relationship with each other. The following velocity of each Fluid Particle on central surface of revolution between blades can be denoted as following and the meridional component of velocity is the same:

$$u = r\omega = \frac{2\pi Rn}{60} \tag{5}$$

$$v_m = \frac{Q}{A_m} \tag{6}$$

where,

r = The radius of fluid particle to the revolution axis (m)
ω = Rotation angular speed of blade (rad/s)
n = Rotation speed of blade (r/min)
Q = Circulating flux through blade flow passage (m^3/s)
A_m = The cross-sectional area of flow passage tangent to meridional component of velocity (m^2)

So, it's deduced as:

$$\begin{aligned} v_u &= u - v_m ctg(\pi - \beta) \\ &= u + v_m ctg\beta \\ &= \omega r + v_m ctg\beta \end{aligned} \tag{7}$$

DYNAMIC TORQUE MODEL

In order to control the brake torque of hydrodynamic retarder accurately, it's necessary to build a practical mathematical model to describe the real working situation of hydrodynamic retarder. The most existing models are all static models which just considered the relationship of brake torque with rotation speed or effective diameter of circulating circle (Abe *et al*., 1991; Klaus and Flack, 1996; Joel, 2002; Lasse, 2002), so they were not overall enough to reflect the all different factors influencing the brake performance, moreover they could not provide the effective feedback quantity to accuracy control. A kind of practical dynamic mathematical model was developed in following. Take a fluid unit element as control body showed in Fig. 5.

The theorem of moment of momentum for hydromechanics can be expressed as follows. And then the forces on the control body can be calculated by resolving the change rate of moment of momentum, rather than integrating the stress distribution:

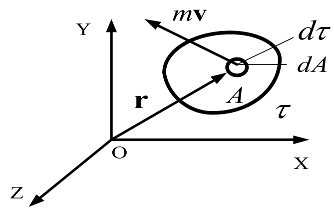

Fig. 5: Control body in flow field

$$\begin{aligned} &\sum \mathbf{F} \times \mathbf{r} - \oiint_A \rho(\mathbf{n} \cdot \mathbf{v})(\mathbf{r} \times \mathbf{v})\, dA \\ &= \frac{\partial}{\partial t} \iiint_\tau \mathbf{r} \times \rho \mathbf{v}\, d\tau \end{aligned} \tag{8}$$

For hydrodynamic retarder, by taking the working medium particle as control body, the formula was turned to calculate the torque between impeller and liquid:

$$\mathbf{T} = \frac{\partial}{\partial t} \iiint_V \mathbf{r} \times \rho \mathbf{v}\, dV + \oiint_A \rho(\mathbf{n} \cdot \mathbf{v})\mathbf{r} \times \mathbf{v}\, dA \tag{9}$$

where,

r = Position vector of fluid particle unit element (m)
v = Velocity vector of fluid particle unit element (m/s)
dV = Volume of fluid particle unit element (m^3)
dA = Superficial area of fluid particle unit element (m^2)
V = Volume of control body (m^3)
A = Superficial area of control body (m^2)
ρ = Fluid mass density (kg/m^3)

The algebra representation of the above vector representation can be deduced as:

$$\begin{aligned} T &= \int_i^o \frac{\partial}{\partial t}(r\rho v_u)A_m dr + (\rho_o v_{uo} v_{mo} r_o A_{mo} \\ &\quad - \rho_i v_{ui} v_{mi} r_i A_{mi}) \\ &= \int_i^o \frac{\partial}{\partial t}[r\rho(\omega r + v_m ctg\beta)]A_m dr \\ &\quad + [\rho_o(\omega r_o + v_{mo} ctg\beta_o)v_{mo} r_o A_{mo} \\ &\quad - \rho_i(\omega r_i + v_{mi} ctg\beta_i)v_{mi} r_i A_{mi}] \\ &= \rho \int_i^o [r(\omega' r + v'_m ctg\beta)]A_m dr \\ &\quad + [\rho_o(\omega r_o + v_{mo} ctg\beta_o)v_{mo} r_o A_{mo} \\ &\quad - \rho_i(\omega r_i + v_{mi} ctg\beta_i)v_{mi} r_i A_{mi}] \end{aligned} \tag{10}$$

Because of the flow consecutiveness of fluid in canned pipe, the properties can be expressed as:

$$v_{mi} = v_{mo} = v_m ,\quad A_{mi} = A_{mo} = A_m ,\quad \rho_i = \rho_o = \rho \tag{11}$$

And then the formula can be simplified as:

$$T = \rho A_m \int_i^o (\omega' r + v'_m ctg\,\beta) r dr + \rho A_m v_m [(\omega r_o + v_m ctg\,\beta_o) r_o - (\omega r_i + v_m ctg\,\beta_i) r_i] \tag{12}$$

In this formula, the impeller torque consists of two parts and the integration part $\rho A_m \int_i^o (\omega' r + v'_m ctg\beta) r dr$ indicates the unsteady state factor. When the hydrodynamic retarder drives to the steady state, $\omega' = 0$ and $v'_m = 0$, the integration unsteady state factor part turns to 0. So the steady state torque can be expressed without the integration unsteady state factor part:

$$T = \rho A_m v_m [(\omega r_o + v_m ctg\beta_o) r_o - (\omega r_i + v_m ctg\beta_i) r_i] \tag{13}$$

And this formula is equivalent with the existing ones.

For the torque between rotator and liquid, the formula is:

$$T_{R-F} = \rho A_m \int_i^o (\omega'_R r + v'_m ctg\beta) r dr + \rho A_m v_m [(\omega_R r_{Ro} + v_m ctg\beta_{Ro}) r_{Ro} - (\omega_R r_{Ri} + v_m ctg\beta_{Ri}) r_{Ri}] \tag{14}$$

For the torque between stator and liquid, the formula is:

$$T_{S-F} = \rho A_m \int_i^o (\omega'_S r + v'_m ctg\,\beta) r dr + \rho A_m v_m [(\omega_S r_{So} + v_m ctg\,\beta_{So}) r_{So} - (\omega_S r_{Si} + v_m ctg\,\beta_{Si}) r_{Si}] \tag{15}$$

In Eq. (14) and (15), the integration unsteady state factor parts are meaningful to the real braking process of hydrodynamic retarder. In the short-term oil feeding and oil draining process, the output torque of hydrodynamic retarder is unsteady, so the two equations are more objective to reflect the true situation occurring in the internal space of impellers. And the equations are universal by consisting of the integration unsteady state factor part. The unsteady output torque calculation can provide reference for the structure and control design.

CONCLUSION

In order to control the brake torque of hydrodynamic retarder accurately, it's necessary to build a practical mathematical model to describe the real working situation of hydrodynamic retarder, especially the short-term oil feeding and oil draining process. The most existing models are not suitable enough to reflect the all different factors influencing the brake performance. Based on the theorem of moment of momentum for hydromechanics, a kind of practical dynamic mathematical model was developed. The deduced equations are more objective to reflect the true situation occurring in the internal space of impellers. And it can provide reference for the structure and control design.

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