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Drag torque modeling at high circumferential speed in open wet clutches considering plate wobble and mechanical contact

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ABSTRACT

A mathematical model is established to predict the drag torque increment due to the mechanical contact at high circumferential speed in open wet clutches. The fluid hydrodynamic actions are coupled simultaneously with the kinematic position of the tilted friction plate. The drag torque due to mechanical contact at high speed is estimated by presenting a rigid-body impact model. The calculation results are validated by experiments performed on an open multi-plate wet clutch test rig. The model calculation demonstrates that the drag torque also becomes higher with larger flow rates of lubricant at high circumferential speed.

1. Introduction

Multi-plate wet clutches are extensively employed in modern vehicle transmission systems. Typical applications of wet multi-plate clutches include a shifting clutch in an automatic transmission, a master clutch in dual clutch and continuously variable transmissions, a lock-up clutch in a torque converter, and a torque split device in a differential lock. A clutch pack alternates a number of friction plates (FPs) with separator plates (SPs), lubricated with a transmission fluid for cooling and protecting the device from thermal damage. The relative rotation between FPs and SPs leads to the viscous shear of lubricant, inevitably causing efficiency reduction and energy loss. The percentage of drag loss can reach approximately 20% in the overall transmission losses in conventional automatic transmissions [1]. Therefore, understanding the physical principles of the drag torque contributes to diminishing the undesirable effects of drag torque in real design practice.

Many studies have been conducted to study the drag torque behavior of wet clutches. Kitabayashi et al. [2] proposed a theoretical formula to calculate drag torque based on Newton inner friction law. This model is only applicable for a single-phase flow condition limited to low speed range. Kato et al. [3] extended the turbulence model of Hashimoto et al. [4] by including the rupture of the automatic transmission fluid (ATF) film. In his model, the rupture was assumed to start from the inner radius. On the basis of Kato et al.'s model, Yuan et al. [5] improved the model by adding a surface tension term to the pressure distribution supposing the oil film ruptures from the outer radius. The drag torque was calculated by integrating the shear stress within the shrunken equivalent radius of oil film after cavitation. Yuan et al. [6] also developed a CFD model to simulate the fluid flow in the gap using the volume of fluid (VOF) method. The simulation could capture the peak of drag torque, but the magnitude was much smaller compared with the test results unless the surface tension coefficient and the viscosity of ATF were magnified respectively. Aphale et al. [7] established an analytical model of drag torque based on asymptotic treatments of the Navier-Stokes (N-S) equations. The results of model was consistent with the experimental data in the single-phase regime since the aeration was not included in this model. Hu et al. [8] derived pressure and velocity distributions from three-dimensional N-S equations based on laminar flow assumption. The solution shows that the centrifugal oil flow acceleration dominated the shrinking of the oil film. Takagi et al. [9] demonstrated that the drag torque reached the peak value when air was entrained in the clearance. The presence of radial grooves enhanced the outward oil flow. Air entered the interface earlier and the drag peak could be reduced. Iqbal et al. [10] proposed an analytical model considering the viscous shearing effect of ATF and the ATF mist film by calculating the volume fraction of oil and predicting the evolution of oil film shape. Pahlovy et al. [11] studied air bubbles formation in the oil film by visualization test with a high speed camera and concluded that air bubbles were generated due to the pressure drop and cavitation at low speed. On this basis, an analytical drag torque model was developed by combining a new expression of volume friction of ATF. Wang et al. [12] presented a two-phase multiple reference frame (MRF) CFD model to simulate the drag torque, including the detailed design features in real transmissions. The aforementioned studies focus on one friction interface which consists of one FP and one SP. The clearance of the interface is fixed in all the above model. Recently, Wang et al. [13] described a statistical method to estimate the dynamically-changing plate distribution in axial direction in open multi-plate wet clutch. Though the prediction of drag torque is within a low speed range, the axial position distribution of multiple plates is studied for the first time and it

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Nomenclature

c_{zi}, $c_{\alpha i}$, $c_{\beta i}$	damping coefficients, N/(m/s), N/(rad/s), N·m/(m/s), N·m/(rad/s); ($i = z, \alpha, \beta$)
d_x, d_y, d_z	displacements about X, Y, and Z axes, m
e_m	coefficient of restitution
f	flow factor
F_z	resultant force along Z axis, N
F_{fz}, F_{fz0}	transient and steady-state fluid force along Z axis, N
$\overline{F}_{ni}$	$\overline{F}_{ci}$ averaged normal contact force and friction force, N
G_r, G_θ	flow regime parameters
h_n	nominal clearance, m
h, h_{oth}	transient clearances of left- and right-side of the FP, m
h_0	steady-state clearance at quasi-equilibrium position, m
H	dimensionless clearance
I_x, I_y, I_z, I_t	moments of inertia about X, Y, and Z axes, and about FP's diameter, kg·m^2
k_{zi}, $k_{\alpha i}$, $k_{\beta i}$	stiffness coefficients, N/m, N/rad, N·m/m, N·m/rad; ($i = z, \alpha, \beta$)
K_i	the i^{th} elastic constraint, N/m; (i = 1, 2, … … N_c)
L	Lagrangian
M_x, M_y	resultant external torque about X and Y axes, N·m
M_{fx}, M_{fy}	transient fluid torque about X and Y axes, N·m
M_{fx0}, M_{fy0}	steady-state fluid torque about X and Y axes, N·m
M_p	mass of the FP, kg
n_1, n_2	rotational speeds of FP and SP, rpm
N_c	number of elastic constraints
N_f	number of friction plates
N_s	number of impacts during sampling time
N_t	step number of simulation
p	transient pressure, N/m^2
p_0	steady-state pressure, N/m^2
P_0	dimensionless steady-state pressure
P_a	angular impulse for one impact, kg·m^2·rad/s
p_{zr}, p_{xr}, p_{yr}, p_{zi}, p_{xi}, p_{yi}	perturbed pressure components, N/m^3, N/(m^2·rad), N/(m^3·s), N/(m^2·rad/s)
P_{zr}, P_{xr}, P_{yr}, P_{zi}, P_{xi}, P_{yi}	dimensionless perturbed pressure components
$\tilde{p}_z$, $\tilde{p}_x$, $\tilde{p}_y$	perturbed pressure components in complex variable form
Q_a	actual supplying flow rate of lubricants, L/min
Q_g, Q_l	flow rates of gas phase and liquid phase, L/min
$\overline{Q}_r$	theoretical required flow rate of lubricants, L/min
r_{out}, r_{in}, r^*	outer and inner radii of the plate, and equivalent radius of oil film, m
R_{out}, R_{in}	dimensionless outer and inner radii of the plate
Re_c, Re_p	Couette and Poiseuille flow Reynolds numbers
t_c	impact duration of one impact, s
t_s	simulation time, s
Δt	time step, s
T	total kinetic energy, J
T_{ci}	contact drag torque for each impact, N·m
T_c, T_f, T_o	contact, viscous, and overall drag torque, N·m
$\bar{u}_r$	average radial velocity of fluid, m/s
v_1, v_2	circumferential speeds of FP and SP, m/s
V	total potential energy, J
x	air mass quality
z	linear displacement of FP, m
z_0	initial linear displacement of FP, m

Greek Symbols

α, β	angular displacements of FP about X and Y axes, degree
α_0, β_0	initial angular displacements of FP about X and Y axes, degree
γ_f	frequency of perturbation, Hz
$\Delta\omega$	angular speed difference between FP and SP, rad/s
λ_i, Λ_i	wobbling angular speeds of FP before and after impact, rad/s
μ_e, μ_g, μ_l	viscosity of homogeneous flow, gas phase, and liquid phase, Pa·s
μ_k	dynamic coefficient of friction
ρ_e, ρ_g, ρ_l	density of homogeneous flow, gas phase, and liquid phase, kg/m^3
$\tau_{z\theta}$	viscous shear stress, N/m^2
ψ	average air volume fraction
ω_1, ω_2	rotational speeds of FP and SP, rad/s
ω_x, ω_y, ω_z	angular speeds of a rigid plate about X, Y, and Z axes, rad/s
Ω_e	dimensionless speed ratio

contributes to providing a mathematical insight into complicated plate behaviors.

In above mentioned literature, drag torque was calculated within a low and medium circumferential speed range. In general, drag torque increases proportionally with the speed of FP at low circumferential speed. After reaching the peak value, drag torque starts to drop to a low value at medium speed. As the speed continuously increases, reaching one critical speed, drag torque will increase to high values and keep rising. This is due to the unstable wobbling and consequent contact between FP and SP [1]. Compared with viscous shear torque, the large drag torque at high speed is more harmful to transmission systems since the mechanical contact between FPs and SPs leads to tremendous engine power consumption, undesired wear, and thermal damage of friction materials. However, studies on drag torque behavior at high speed are much fewer than those on viscous drag torque. Hilpert [14] firstly summarized the phenomenon that the drag torque increased suddenly to high values when the plates rotated in opposite directions. The burnt spot on the clutch plate indicated that there were mechanical contact which leaded to a thermal failure. Hilpert explained that the flutter of clutch plates in axial direction causes the failure and the flutter is gyroscopic in origin. In recent years, several efforts have been made to model the drag torque at high speed. Mahmud and Pahlovy [15] predicted drag torque at high speed based on the assumption of local vacuum formation due to negative pressure where the oil film ruptures. The local vacuum draws SPs close to each other and the drag torque rises as the clearance becomes smaller. With the same assumptions, Mahmud et al. [16] also conducted a numerical computation to simulate the movement of SPs. Nevertheless, the principle of drag torque rise is still due to the viscous shear effect instead of mechanical contact. The experiment results of Hou et al. [17] confirmed that the FP