

Multiphase Drag Modeling for Prediction of the Drag Torque Characteristics in Disengaged Wet Clutches

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ABSTRACT

The undesired Drag Torque (DT) which is developed due to the shearing of fluid film in between the disk and separator plate reduces the efficiency of a transmission and increases the fuel consumption of a car. In order to minimize the transmission loss, the physics of the fluid flow mechanism inside the clutch should be understood well and the factors influencing the DT should be identified. In this paper, a model is proposed to predict the drag torque of a disengaged wet clutch at different rotation speeds, clearances, disk sizes and oil temperatures. The model explains well how the DT changes for the no groove disk, grooved disk and different ATF properties. The proposed model is validated by several experimental results conducted by a visualization tester and images of the fluid film taken during the test. Results show that there is a good degree of agreement between the DT trends derived from the proposed model and the test results for the same condition.

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INTRODUCTION

The demand of low fuel consumption car is increasing day by day due to the aspect of protection of the global environment. The regulations of fuel efficiencies of passenger cars in many countries have been established and become gradually stringent. Therefore, lowering fuel consumption is an urgent issue. Fuel efficiency can be improved by reducing the energy loss a power train and others. Some estimates shows that about 14%-30% of the energy from the fuel gets used to move a car down the road and 5-6% energy is lost due to drive train[1]. Therefore, a potential improvement of power train components can lower the fuel consumption significantly. Fig.1 shows an image of an automatic transmission. During the operation of a transmission, some of the clutches are engaged for power, while the rest of clutches remain disengaged. Due to the engagement and disengagement process, a frictional heat is generated which leads to some damage to clutches. Therefore, it is necessary to cool down the disks to increase the service life of clutch. An automatic transmission fluid (ATF) is delivered in between the friction disks and separator plate to cool them. Since the friction plates and separator plates are always in relative motion to each other, a shear force is generated on the fluid in the gap between the disks. This shear force generates a DT which is considered as a loss. This undesired DT reduces the efficiency of power transmission and increases the fuel consumption of a car. Therefore, it is of interest for transmission engineers to understand the physics

of this kind of flows and identify design variables and parameters that dictate the loss mechanism so as to minimize the loss in engineering practice.



Fig.1. An automatic Transmission which consists of multiple numbers of clutch disks and clutch plates

For more than several decades, a lot of researches have been done to understand the DT in disengaged wet clutches. In 1984, Hashimoto et al.[2] presented the governing equations for the flow in a thrust bearing subject to centrifugal force. These equations describe the flow between two adjacent flat rotating plates facing each other at an angle. Though this work is not directly applicable to clutches, it laid down a framework

for the modeling of DT for automatic transmission clutches operating in the turbulent regime. Later, in 1993, Kato et al. [3] estimated the DT characteristics utilizing Hashimoto's equations for wet clutches and brakes. In this model, cavitation is considered the only reason for the rupture of oil film at high speed and it starts from the inner radius. Subsequent studies suggest that laminar flow prevails in the gap and rupture of the oil film starts from the outer radius and progresses towards inner radius [4]. Later, Yuan et al. [5] introduced an additional term into Kato's pressure distribution equation to take into account ATF surface tension effect considering its relative importance over other forces in the low speed region. In both Kato's and Yuan's models, the major equations were derived considering turbulent flow, though their experimental data reflects that the flow would be laminar since the critical Reynold's number is far below 1400[6]. Kitabayashi et al. [7] presented a theoretical DT equation for laminar flow as shown in equation (1).

$$T = \frac{N\mu\pi(r_{out}^2 - r_{in}^2)\omega r_m^2}{h} \quad (1)$$

Clearly, the equation can only represent the rising portion of a typical DT curve at the low speed region and falls short to estimate the decline of the DT at higher speed. Recently, in 2013, Iqbal et al. [6] presented a DT model accounting the effect of cavitation in laminar flow. However, it is very unlikely for cavitation to occur in typical wet clutches when the speed is below 4000 rpm since the pressure in the clutch can hardly go down enough for the cavitation to trigger in which is radius dependent. Moreover, in their model, a variable flow rate is considered which is not evidenced by experimental data.

From the above review of the previous work, it is clear that an improved mathematical model should be developed to predict the DT characteristics in typical clutch scenarios[8],[9]. In this paper, a DT model considering laminar flow for the speed range 100~4000 rpm is presented. A customized equation of DT for the grooved disk is also presented. The velocity distribution equations in the gap are deduced by simplification of the continuity and Navier-Stokes momentum equations. In the pressure distribution equation, the effect of ATF surface tension is incorporated. The rupture of the oil film at high speed is described by the depletion of oil volume to counteract the effect of dominating centrifugal force working against the viscous and surface tension forces. The proposed model is validated by the experimental data and oil flow images taken from a visualization tester. The influence of ATF properties on the DT characteristics is also discussed in our paper. From the comparison of model results and experimental results, it shows that the estimated peak DT position is almost same as test results but the estimated magnitude of peak DT is lower than that of test results. Tentative reasons for such little discrepancy are discussed in the results and discussions section.

MODEL DEVELOPMENT

A basic diagram of typical clutch is shown in Fig. 2. As shown in the figure, the disks are separated by a uniform gap h . In the proposed model, the friction disks are rotating while the separator plates are kept stationary. Since our test facility permits only disk rotation, therefore the rotational state of both friction disk and separator was not considered this time. The oil enters the gap at a radius r_{in} and experiences a viscous shear force due to relative motion of the disks. The oil flow leaves the gap at a radius r_{out} driven by the centrifugal force. At the same time, the relative motion of the disks generates a viscous shear force and resists the radial flow of the oil [5]. The surface tension force of the oil also works against the radial flow.

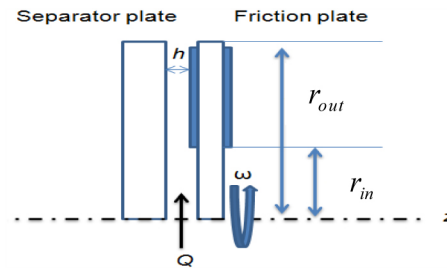


Fig. 2. Schematic diagram of a single pair of clutch disk.

The flow velocity fields and associated pressure distribution in the gap of the disks can be obtained by simplification of the continuity and Navier-Stokes momentum equations under certain assumptions. The assumptions are -

- i. The flow is Laminar on the basis of the Reynolds number (less than 1400)
- ii. The flow is treated as steady assuming that the flow rate, clutch speed, clutch configuration and boundary conditions do not change in time.
- iii. The flow velocity in the axial direction may be neglected considering thin film fluid ($h/r \ll 20$).
- iv. The effect of gravity is neglected.
- v. The fluid is incompressible.
- vi. The fluid is Newtonian and the coefficient of viscosity is constant considering small temperature variation.
- vii. Fluid pressure does not change across the film thickness
- viii. For parallel clutch plates with no grooves and waves, fluid pressure doesn't change in the tangential direction

After simplification, the governing Navier-Stokes equations can be expressed in the followings manner

$$V_r \frac{\partial V_r}{\partial r} - \frac{V_\theta^2}{r} + \frac{1}{\rho} \frac{\partial P}{\partial r} = \frac{\mu}{\rho} \frac{\partial^2 V_r}{\partial z^2} \quad (2)$$

$$V_r \frac{\partial V_\theta}{\partial r} - \frac{V_\theta V_r}{r} = \frac{\mu}{\rho} \frac{\partial^2 V_\theta}{\partial z^2}$$

$$\frac{1}{r} \frac{\partial P}{\partial \theta} = \mu \frac{\partial^2 V_\theta}{\partial z^2} \quad (4)$$

$$\frac{\partial P}{\partial \theta} = 0 \quad (5)$$

The boundary conditions for these equations are

$$\left. \begin{aligned} V_\theta(z=0) &= r\omega, V_\theta(z=h) = 0 \\ V_r(z=0) &= V_r(z=h) = 0 \\ V_z(z=0) &= V_z(z=h) = 0 \end{aligned} \right\} \quad (6)$$

The feeding flow rate through the gap between the disks can be expressed as follows-

$$Q = \int_0^{2\pi} \int_0^h V_r r dz d\theta = 2\pi r \int_0^h V_r dz \quad (7)$$

Integrating (7) with respect to z twice and applying the boundary conditions, the oil radial velocity distribution in the gap can be expressed as (referring to Chang [9])

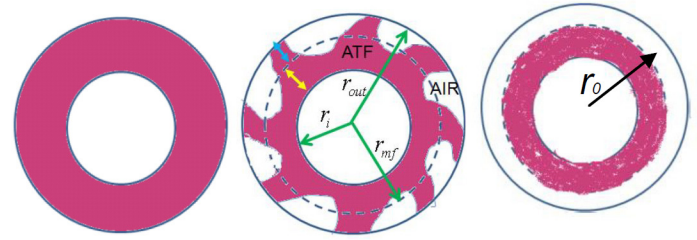
$$V_r = \frac{1}{2\mu} \frac{dP}{dr} z(z-h) - \frac{\rho\omega^2 r}{12\mu h^2} z(z^3 - h^3) - \frac{3\rho Q^2}{20\mu\pi^2 r^3 h^6} z(2z^5 - 6z^4 h + 5z^3 h^2 - h^5) \quad (8)$$

Substituting (8) in (7)

$$Q = -\frac{\pi h^3 r}{6\mu} \frac{dP}{dr} + \frac{\pi \rho \omega^2 r^2 h^3}{20\mu} + \frac{9\rho Q^2 h}{140\pi \mu r^2} \quad (9)$$

The first term on right hand side of (9) represents pressure induced flow or Poiseuille flow, the second term represents contribution due to centrifugal inertia and the third by the flow inertia.

For a constant flow rate, the radial extent of the continuous oil film r_o depends on the clutch speed. At lower rotating speed, the centrifugal force is small and there exists a continuous oil film between the plates as shown in Fig. 3(a). As the rotating speed increases, the centrifugal force becomes dominant and drives part of the oil outward in order to satisfy the conservation of mass. As a result, rivulets of oil start to take form at some outer radius r as shown in Fig. 3(b) and a volume mixture of oil and air takes place in between the plates. In order to find the volume fraction of oil in the gap, a hypothesized oil film is considered as shown in Fig. 3(c) where, the radial boundary r_o of the equivalent oil film lies at the oil-air interface.



(a) Full oil at low speed (b) Ruptured oil film (c) equivalent radius of the ruptured oil film

Fig. 3. Schematic diagram of the oil film at different clutch speed

When all the forces are balanced, the flow reaches a steady state condition and the volume of the oil film in the gap becomes stable. At this condition, the total pressure at the inner and outer radius of the oil film will be same.

Rearranging (9)

$$\frac{dP}{dr} = -\frac{6\mu Q}{\pi h^2 r} + \frac{3\rho\omega^2 r}{10} + \frac{27\rho Q^2}{70\pi^2 h^2 r^3} \quad (10)$$

Integrating (10) within the inner and outer boundary of the oil film

$$P(r_{o-}) - P(r_{in}) = -\frac{6\mu Q}{\pi h^2} \ln\left(\frac{r_{o-}}{r_{in}}\right) + \frac{3\rho\omega^2}{20} (r_{o-}^2 - r_{in}^2) - \frac{27\rho Q^2}{70\pi^2 h^2} \left(\frac{1}{r_{o-}^2} - \frac{1}{r_{in}^2}\right) \quad (11)$$

Due to surface tension on the oil film, there is a pressure jump ΔP across the boundary of r_o . Referring to Yuan et al. [4], the pressure jump can be approximated as

$$P(r_{o-}) - P(r_{o+}) = \frac{2\sigma \cos \phi}{h} \quad (12)$$

Substituting (12) in (11) and rearranging

$$P(r_{o+}) - P(r_{in}) = -\frac{6\mu Q}{\pi h^2} \ln\left(\frac{r_{o+}}{r_{in}}\right) + \frac{3\rho\omega^2}{20} (r_{o+}^2 - r_{in}^2) - \frac{27\rho Q^2}{70\pi^2 h^2} \left(\frac{1}{r_{o+}^2} - \frac{1}{r_{in}^2}\right) - \frac{2\sigma \cos \phi}{h} \quad (13)$$

At the steady state condition

$P(r_{o+}) - P(r_{in}) = 0$. If the feeding flow rate is known, the outer radius of the equivalent oil film can be solved from (13).

Fraction of the oil volume in the gap can be expressed as

$$\alpha = \frac{r_o^2 - r_{in}^2}{r_{out}^2 - r_{in}^2} \quad (14)$$