A combined numerical and experimental investigation of disengaged wet clutch system power loss

A Combined Numerical and Experimental Investigation of Disengaged Wet Clutch System Power Loss

M. Leighton [1)]*, J. Davies [1)], N. Morris [1)], H. Rahnejat [1)], G. Trimmer [2)]
[1)] Wolfson School of Engineering, Loughborough University, Leicestershire, LE11 3TU, UK
[2)] J. C. Bamford Excavators, Rocester, Staffordshire, ST14 5JP, UK
*Corresponding author: M.Leighton@lboro.ac.uk

1. Introduction

Increased machine performance through reduction of drivetrain power losses is significant challenge for the powertrain engineer. It has been suggested that by reducing the losses from gears, bearings and lubrication systems by a half the fuel efficiency of a machine could improve by as much as 5% [1]. The continued trends of evermore stringent emissions standards, customer demands for increased fuel efficiency and machine life have resulted in great pressure on the manufacturers in both on- and off-highway vehicles.

A significant contributor to off-highway vehicle in-efficiency is the viscous drag caused during the rotation of disengaged wet clutches. These components can, by their design, experience periods of operation where there can be significant relative rotational plate velocities with only a thin lubricant film separating them. The resultant power loss, particularly under cold start conditions, can be significant. The addition of patterned grooves into clutch friction linings, to allow oil flow to dissipate heat during engagement, complicates the prediction of performance. Clutch plate design improvement is therefore a multi-variate problem.

Lloyd [2] conducted a series of experiments on the influence of wet clutches on losses to evaluate the parameters that influence the losses of a system. Lloyd used a numerical model to predict the increase in drag torque to evaluate to what extent each parameter varied the losses. The rotational speed, oil viscosity, plate flatness and separation were determined to be the most influential parameters effecting wet clutch performance. Lloyd compared the results against an experimental rig using a clutch pack assembly. A similar rig was used by Fish and Kitabayashi et al [3, 4]. Jibin et al. [5] used a single plate rig allowing an isolated study and a higher degree of experimental control.

The effect of cavitation on clutch performance can be significant. Pahlovy et al. [6, 7] designed a wet clutch rig which allows a high speed camera to see into the contact such that inter-plate cavitation can be recorded. This allowed them to evaluate the cavitation area and resultant reduction in viscous drag.

In this paper a Reynolds-based numerical model is presented with the inclusion of lubricant inertia. The numerical model allows rapid simulation studies without the need for remeshing of the computational domain.

A simplistic numerical model, applying an analytical flow equation [8] and cavitation model provided by Dowson [9], derived from the Navier-Stokes equation, was also used to provide very fast predictions and to analyse the effect that cavitation can have on wet clutch losses. For this model the clutch plate is split between regions that are either at on the flat 'lands' of the clutch surface or in the depths of the grooves cut for oil flow.

The current paper present an experimental, analytical and numerical approach to investigate the drag losses in partially immersed wet clutch systems with specific application to off-off high way transmission systems.

2. Test Rig

A single clutch test rig was designed and manufactured to investigate the drag torque produced between an isolated friction disc and stator pair. The plate clutch friction and stator plate are from the transmission system of an off-highway vehicle. The test rig can be thought of as three subsystems, the drive system, the shaft assembly and the tank assembly. The drive consists of a three-phase motor and external control box connected to a twin pulley system which rotates the drive shaft. The drive is then transmitted to the friction plate through the shaft assembly containing a torque limiting coupler and a Torqsense RWT421-EC-KG torque transducer. The transducer has a sample rate of 450Hz, an accuracy of ±0.125Nm and a sensitivity of 0.05Nm. The torque transducer can also operate in a torque range of 0-50Nm and a speed range of 0-15000rpm making it suitable for use in this study. The shaft system has a single friction plate mounted to it with two counter-plates held stationary at a fixed separation of 160µm.

The stator counter plates are mounted on a bolt with a smooth shoulder which allows them to slide freely with a fixed separation as would be possible in real-world applications. As such they can float to a balanced position through the mechanism of the inter-plate pressure. The clutch disc spins in a heated oil tank containing a heater and thermal control unit. The test rig can be seen in figure 1.

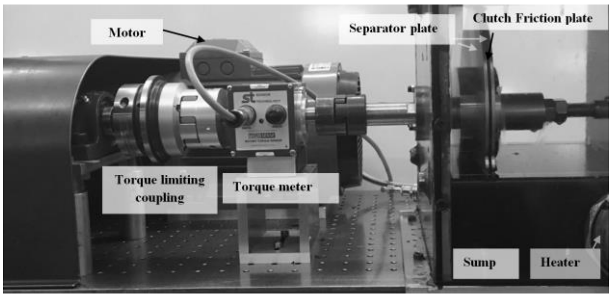

Figure 1 Test Rig

The clutch plates investigated in this study have simple radial grooves with a groove depth of 67 µm and width of 1.2 mm. An image of the clutch plate simulated and experimentally measured can be seen in figure 2.

Figure 2 Clutch Disc

3. Numerical Model

The numerical model developed applies a discretised Reynolds equation with inertial terms to determine the frictional power loss of the disengaged clutch plate.

Since the geometry of the patterned sections of the clutch plates have rotational symmetry every 6.4 degrees, computational time was reduced by applying a periodic boundary condition in the circumferential direction for the modelled area. The inner and outer radii of the meshed section are set to atmospheric pressure which is a close approximation of the real world conditions. A schematic of the boundary conditions can be seen in figure 3.

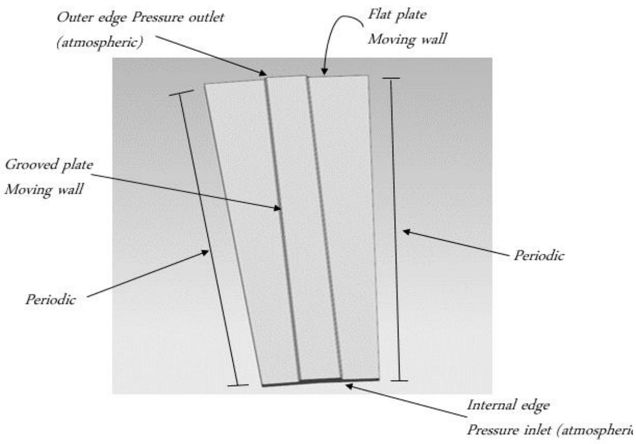

Figure 3 Boundary conditions

Since the geometry of the patterned sections of the clutch plate has rotational symmetry every 6.4 degrees, a periodic boundary condition in the circumferential direction for modelled area was applied as:

$$p(0, r) = p\left(\frac{\pi}{3}, r\right) \quad (1)$$

Where p is the nodal pressure and r denotes the radial direction.

The inner and outer radii boundary conditions are applied as:

$$p(c, r_i) = p(c, r_o) = p_{atm} \quad (2)$$

Where p_{atm} is the atmospheric pressure, c denotes the radial direction and i and o denote the inner and outer positions respectively.

Cavitation was accounted for by Reynolds cavitation and reformation boundary condition. Mass continuity was maintained in the cavitated region by estimating the lubricant volume fraction and assuming the remaining volume to be atmospheric pressure gas. Within the cavitated region, pressure induced flow is assumed to drop to zero and the pressure gradient at the reformation point is assumed to be zero such that:

$$\frac{dp}{dc}(c_{reform}, r) = 0 \quad (3)$$

The resistive frictional torque comprises pressure induced Poiseuille flow, Couette flow and also inertial terms. The nodal friction is calculated as:

$$F_N = \frac{\eta.dc.dr.U}{h} + \frac{dp}{dc}.h.\frac{dc.dr}{2} + \frac{dc.dr.h.\rho.dU}{dt} \quad (4)$$

Where η is the lubricant viscosity, U is the mean lubricant velocity, h is the nodal separation, ρ is the lubricant density and t denotes time.

Where a nodal area is determined to have potentially cavitated the pressure gradient is reduced to 0 and the lubricant volume is calculated by mass conservation. The nodal friction for cavitated nodes is calculated as:

$$F_N = \Pi_{cav}\frac{dc.dr.h.\rho.dU}{dt} \quad (5)$$

Where Π_{cav} is the ratio of fluid volume to total volume for the given node.

The torque is then calculated as the sum of the nodal torques multiplied by the radial position of the node. The torque can then be converted to a resultant power loss by:

$$P_{Loss} = 2\pi T N_{rps} \quad (6)$$

Where T is the total torque and N_{rps} is the number of revolutions per second.

Convergence of the iterative pressure loop is sort, such that the error between successive iterations has decreased by three orders of magnitude from the initial while applying a Gauss-Seidel point successive over relaxation. Areas that are determined to be in cavitation are considered to have no pressure gradient and lubricant transport is determined by applying mass conservation from the cavitation boundary. Where i and j terms are used to describe the discretised geometry in the radial and circumferential directions the error can be determined by:

$$Err = \frac{\sum_{i=0}^{I}\sum_{j=0}^{J}\left|p_{i,j}^{n} - p_{i,j}^{n-1}\right|}{\sum_{i=0}^{I}\sum_{j=0}^{J}p_{i,j}^{n}} \quad (7)$$

Where $n-1$ and n denote the previous iteration and current iteration respectively

4. Analytical Model

For comparison and improved computational times, a simplified model was developed for one specific clutch geometry with simple radial grooves as shown in figure 2. The simplified model uses an analytical approach;

$$\tau_l(R) = \frac{\omega R\eta}{h_l} \tag{8}$$

$$\tau_g(R) = \frac{\omega R\eta}{h_g} \tag{9}$$

Where ω is the rotational velocity, R is the local radius, l denotes the land regions and g denotes the groove regions.

Integrating between the inner and outer radii allows the calculation of torque;

$$T = \omega\eta\left(\frac{\pi}{2h_l}\left({R_o}^4 - {R_i}^4\right) - \frac{ND}{3h_l}\left({R_o}^4 - {R_i}^4\right) + \frac{ND}{3h_g}\left({R_o}^4 - {R_i}^4\right)\right) \tag{10}$$

Where N and D denotes the number of grooves and their width respectively.

Cavitation is accounted for through application of the equation given by Dowson [9] derived from cross film velocity for a rotating disc and stationary counter disc from the Navier-Stokes equation;

$$u(z) = -\frac{z(h-z)}{2\eta}\frac{\partial p}{\partial r} + \rho\frac{r\omega^2 z(h^3-z^3)}{12\eta h^2} \tag{11}$$

Where u is the lubricant velocity and z is height in the axial direction.

Implementing the known boundary conditions at the inner radius and assuming the pressure gradient in the radial direction is zero when the film cavities the following expression can be found.

$$\frac{3(p_{atm}-p_{cav})}{20\rho\omega^2} = R_i^2 + R_o^2(\ln R_o - 1 - 2\ln R_i) \tag{12}$$

Where p_{cav} is the cavitation pressure.

The expression can be solved to find the outer radius R_o. Equation 10 can then be solved between the inner radius and cavitation radius to give a prediction of drag torque.

5. Experimental investigation

The test rig was used to measure the drag torque between a rotating clutch plate and two static counter plates. The separation was fixed at 160 µm either side of the clutch plate and the temperature was held at 80 °C at the exit from the contact. The rotational speed was varied over a range of 1000 to 5000 rpm to in investigate the variation in resultant torque.

It was noted that the rotation resulted in a significant amount lubricant aeration which increased with rotational speed. As a result, it was necessary to measure the resultant lubricant viscosity after a representative running procedure. The lubricant aeration at low speed can be seen in figure 4 as an area of lighter coloured oil.

Figure 4 Rig running at 1000 rpm

The viscosity was measured after a running in procedure in which the test rig was run at constant rotational speed, temperature and separation for 2 minutes prior to the measurement. The measured viscosity can be seen in figure 5.

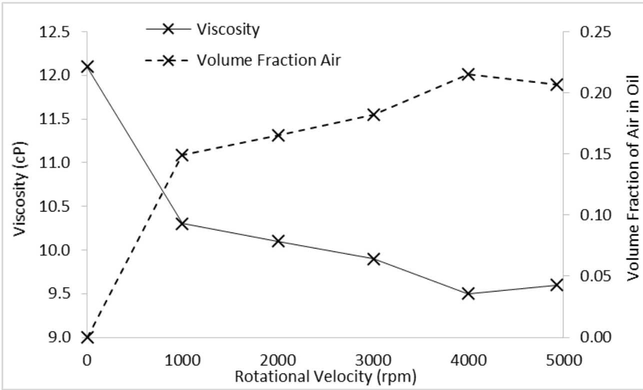

Figure 5 Experimental and numerical results

When η_T is the viscosity of the mixture and η_L and η_G are the viscosities of the gaseous and liquid phases respectively. The air volume fraction can calculated as:

$$VF = \frac{\eta_m-\eta_L}{\eta_G-\eta_L} \tag{13}$$

Where η_m, η_L and η_g, are the measured, liquid and gas viscosities respectively.

It can be seen from figure 5 that viscosity in operational conditions drops quickly as the amount of air in suspension increases. There is then a steady reduction viscosity as rotational speed increases.

6. Results

Simulations of the clutch geometry were run for a variety of rotational velocities with both the numerical Reynolds based code with inertia effects and the analytical lubricant flow model with cavitation effects. The cases simulated were for the clutch geometry already described, with simple radial grooves and a separation of

160 µm between the clutch and counter plate. An operationally representative oil temperature of 80 °C was simulated giving an oil viscosity of 9.8 Pa.s at 0 rpm. The resultant torques were doubled to represent the conditions of the component level test rig where the clutch plate spins between two counter plates.

Table 1 Test parameters

Inner diameter	100 mm
Outer diameter	140 mm
Viscosity at 80°C and 0 rpm	9.8 Pa.s
Plate separation	160 µm

Experimental results were obtained using the rig previously described for validation of the models. The results can be seen in figure 6.

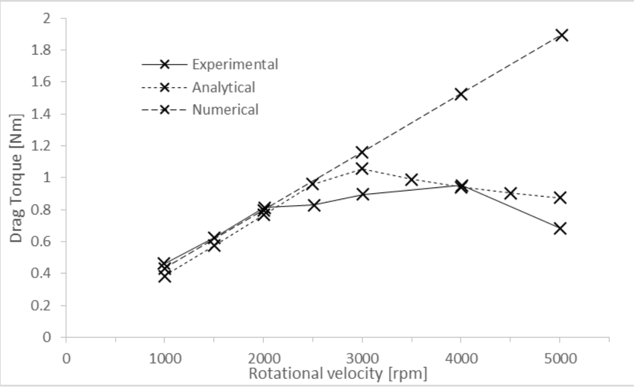

Figure 6 Experimental and numerical results

Figure 6 shows that the experimental and numerical results have a strong agreement at speeds up to 2000rpm. In this region the Reynolds based code with inertial effect fit the measured data best. At speeds over 2000rpm the experimental data diverges from the models and no longer displays a nearly linear increase with rotational speed suggesting that cavitation has begun to have a substantial effect on the viscous drag torque.

The cavitation model using simplified fluid flow model follows the trend of the experimental data well although the experimental results show a peak torque at lower rotational speeds than the model predicts. After 2000 rpm the model fit to the data is not as strong although it continues to follow the trend.

It is clear from the results that cavitation is a significant factor effecting the resultant torque experienced by clutches at high rotational speeds. The data suggests that even a basic lubricant flow model can give a reasonable prediction of resultant torque across a significant range of rotational speeds.

The experimental results peak at a lower torque and the torques experienced above that speed suggest that the predicted cavitation effect underestimated the real world conditions. It may be that this is due to the full flooded inlet condition which may not be met where highly aerated oils are in use. If the pressure at the inlet is not sufficient to ensure only lubricant enters the contact then the cavitation conditions change allowing cavitated regions to form more easily

7. Conclusions

The current paper demonstrates the applicability of two models for the prediction of partial immersed wet clutch systems specific to off highway application.

At low speeds, where a continuous lubricant film exist between the plate, little variation between either the model and experiment is observed. Once the lubricant film begins to cavitate a significant difference in the models ability to predict the drag torque occurs.

It has been demonstrated that cavitation is a critical factor in predicting the drag losses in off highway wet clutch systems specifically at high rotational speeds.

Lubricant aeration generated by churning of oil in the sump has a significant effect on the lubricant viscosity and this is suggested to play a significant role in determining rotational speed at which cavitation become evident.

8. References

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