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## Numerical and Experimental Investigation of Drag Torque in a Two-speed Dual Clutch Transmission

by

\*Xingxing Zhou1, Paul Walker1, Nong Zhang1,2, Bo Zhu1,3, Jiageng Ruan1

*1 School of Electrical, Mechanical and Mechatronic Systems, University of Technology, Sydney (UTS), Sydney, NSW, Australia 2007*

*2 State Key Laboratory of Advanced Design and Manufacturing for Vehicle Design, Hunan University, Changsha, China 410082*

*3 BAIC Motor Electric Vehicle Co Ltd, BAIC Motor Electric Vehicle Co Ltd, Beijing, China102606*

*Xingxing Zhou*

*Tel: +61-2-9514 2517; Fax: +61-2-9514 2655.*

*E-mail address:* [*Xingxing.Zhou@student.uts.edu.au*](mailto:Xingxing.Zhou@student.uts.edu.au)

*Paul Walker*

*Tel: +62-2-9514 2412; Fax: +61-2-9514 2655.*

*E-mail address:* [*Paul.Walker@uts.edu.au*](mailto:Paul.Walker@uts.edu.au) *(Dr. Paul Walker)*

*Nong Zhang*

*Tel: +61-2-9514 2662; Fax: +61-2-9514 2655.*

*E-mail address:* [*Nong.Zhang@uts.edu.au*](mailto:Nong.Zhang@uts.edu.au) *(Prof. Nong Zhang)*

*Bo Zhu*

*Tel: +62-2-9514 2517; Fax: +61-2-9514 2655.*

*E-mail address:* [*zhubo@bjev.com.cn*](mailto:zhubo@bjev.com.cn)

*Jiageng Ruan*

*Tel: +62-2-9514 2412; Fax: +61-2-9514 2655.*

*E-mail address:* [*Jiageng.Ruan@student.uts.edu.au*](mailto:Jiageng.Ruan@student.uts.edu.au)

*Corresponding Author in Submission*

*Tel: +61-2-9514 2517; Fax: +61-2-9514 2655.*

*E-mail address:* [*Xingxing.Zhou@student.uts.edu.au*](mailto:Xingxing.Zhou@student.uts.edu.au) *(Xingxing Zhou)*

*Corresponding Author in Publication*

*Tel: +62-2-9514 2412; Fax: +61-2-9514 2655.*

*E-mail address:* [*Paul.Walker@uts.edu.au*](mailto:Paul.Walker@uts.edu.au) *(Dr. Paul Walker)*

\*Author to whom all further correspondence should be addressed

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# Numerical and Experimental Investigation of Drag Torques

# in a Two-speed Dual Clutch Transmission

## Abstract:

The theoretical analysis of drag torques within a two-speed dual clutch transmission is presented in this article. The numerical models are developed to study the different sources of drag torques in dual clutch transmission. Simulations are performed in Matlab/Simulink platform to investigate the variation of drag torques under different operating conditions. Then an experimental investigation is conducted to evaluate the proposed model using an electric vehicle powertrain test rig. Outcomes of experimentation confirm that simulation results agree well with test data. Therefore the proposed model performs well in the prediction of drag torque for the transmission, and can be applied to assess the efficiency of the transmission. Results demonstrate that the entire drag torque is dominated by the viscous shear in the wet clutch pack and gear churning losses. This lays a theoretical foundation to future research on reducing drag torque and applications of drag torque in powertrain system efficiency optimization.

Keywords: Drag torque; power loss; mathematical model; dual clutch transmission; experimental investigation.

## Introduction

In recent years there has been significant attention drawn towards reducing fossil fuel consumption and emissions in automotive industry. Improvement of the overall energy efficiency of existing technologies is one of the most important subjects for developing new vehicle technologies. As a consequence of this, the development of commercially viable hybrid electric vehicles (HEVs), fuel cell vehicles (FCVs) for using in the short to mid term, and pure electric vehicles (EVs) in the long term are one of the major contributions in the automotive industry to solve related issues[[1](#_ENREF_1)]. Pure EVs currently being used in the market are mainly equipped with single speed transmissions, with tradeoffs between dynamic (such as climbing ability, top speed, and acceleration) and economic performance (drive range). Nowadays, more and more EV researchers and designers are paying attention to application of multiple speed transmissions instead of traditional single speed transmissions, expecting to improve the EV performance. The usage of multispeed transmissions for electric vehicles is likely to improve average motor efficiency and range capacity, or even can reduce the required motor size. The detail advantages of two-speed transmission over single speed are demonstrated in previously reported work[[2](#_ENREF_2)].

As an important part of electric vehicle powertrain system design and optimization, it is of great importance to predict the transmission efficiency early in the design process, leading to improved powertrain efficiency as the system is refined. In addition to the design characteristics of the transmission, drag torque, acting as the sources of power losses within transmission, is affected by operating conditions, including variations in both speed and load. The total drag torque within a gear-train for any transmission is generally made up of several parts, including gear friction[[3-6](#_ENREF_3)], windage[[7-9](#_ENREF_7)]and oil churning[[10](#_ENREF_10), [11](#_ENREF_11)]. Other important drag torque sources have to be considered as well, comprising bearings and seal[[11-13](#_ENREF_11)], synchronizer and free-pinion losses[[14](#_ENREF_14), [15](#_ENREF_15)]. If the clutch is immersed in oil, torsional resistance and its influences caused by the viscous shear between wet clutch plates should be considered as well[[16](#_ENREF_16), [17](#_ENREF_17)]. Several researchers have done some specific studies regarding the drag torque within disengaged wet clutches [[18-21](#_ENREF_18)], which will be analysed in the 2nd section.

However, in order to comprehensively improve the whole automotive powertrain system efficiency, it is necessary to consider all aspect of the transmission power losses[[22](#_ENREF_22)]. There are only a few reported works on the entire transmission power losses[[23-27](#_ENREF_23)], which are mostly focus on manual transmission (MT) gear-train. There is limited, if any, published work to combine the gearbox components and wet clutch losses together to study. Especially, there is no published report on study of entire drag torque within wet dual clutch transmissions (DCT).

In this paper, drag torque within a two-speed dual clutch transmission is discussed. A general two-speed wet DCT is suggested to be equipped into a pure electric vehicle (EV), as shown in Figure 1. The system under consideration is modified from a 6-speed DCT (DQ250) into a two-speed DCT. The two-speed DCT housing is made from an aluminium alloy. And the DCT is made up of two clutches, the inner clutch (C1) and the outer clutch (C2). The two clutches have a common drum attached to the same input shaft from the electric motor, and the friction plates are independently connected to 1st or second gear. C1, shown in green, hereby connects the outer input shaft engaged with 1st gear, and C2, shown in red, connects the inner input shaft engaged with 2nd gear. In order to make transmission control system simpler and save manufacture fees, there are no synchronisers in this new type of two-speed DCT. Thus, the transmission can be looked at as two clutched gear pairs, and, in this sense, shifting is realised through the simultaneous shifting between these two halves of the transmission. For this special layout, vehicle equipped with a DCT can not only change speed smoothly with nearly no power hole, identified by Goetz[[28](#_ENREF_28)], but improve the EV efficiency as well.

Diff

C1

1st Gear

C2

S1 & S2

2nd Gear

S3

S1 & S2

S3

Diff

Electric Machine

Wheel

(1)

(5)

(4)

(7)

(6)

(9)

(8)

(2)

(3)

**Figure 1.** Schematic of a two-speed DCT powertrain system.

This paper is organised as follows. In the 2nd section, the different source of drag torque in the DCT is theoretically analysed and modelled including torsional resistance caused by viscous shear caused between wet clutch plates and concentrically aligned shafts, gear mesh friction and windage, oil churning, and bearing losses. Then experimental aspects comprising of the description of the UTS test rig, and the drag torque test procedures are presented and discussed in the 3rd section. In the 4th section, simulation and experimental results are compared and analysed to validate the effectiveness of the numerical model. Finally, some significant conclusions drawn from the study are summarized in the 5th section.

## Theoretical analysis of drag torque in wet dual clutch transmissions

The sources of power loss in a two-speed DCT, or drag torque, can be divided into five major origins: gears related meshing, windage and churning losses, bearings and oil seal related losses, concentric shaft related viscous shear loss, and disengaged wet clutch caused drag torque losses, as shown in Equation (1). For each individual power losses sources, the following formulae have been implemented in the simulation code.

(1)

Where is total power losses in DCT, is power losses caused by concentric shaft drag torque, is power losses caused by bearings drag torque, is power losses caused by gear meshing drag torque, is power losses caused by gear churning, is power losses caused by wet clutch plates drag torque.

If engaged with 1st gear, clutch 1 is closed, while clutch 2 will be open and subsequently develop a drag torque.

(2)

(3)

(4)

Where is motor output torque; is drag torque caused by concentric shafts viscous shear resistance; is drag torque caused by bearings; and is gear ratio,represents the output torque of the outer concentric shaft. is drag torque caused by gear pairs meshing, is the drag torque caused by wet clutch packs. is the drag torque caused by churning. And the individual power losses can be calculated in rotational speed multiplied by drag torque:

(5)

For the power losses caused by concentric shaft raised shear resistance is:

(6)

The total power loss caused by bearings and gear friction are:

(7)

(8)

The power losses resulting from gear windage and churning is:

(9)

The overall power loss caused by clutch package is:

(10)

Similarly, if engaged with 2nd gear, clutch 2 is closed, while clutch 1 open.

(11)

(12)

(13)

(14)

(15)

(16)

(17)

(18)

The overall efficiency of a DCT or any transmission in general can be obtained by monitoring the input and output speed and torque respectively.

(19)

### 2.1 Drag Torques Caused By Gears

Drag torques caused by gears can be generally divided into gear meshing loss, and gear windage and churning loss. Meshing losses are a result of rolling and sliding friction in mating gears and is, as such, a load dependent loss, whilst windage and churning losses are generated from the gears rotating in fluids, and are therefore speed dependent.

#### 2.1.1 Gear mesh

The gear meshing losses in gear pairs are dependent on both rolling and sliding friction. Methods for studying these losses reported in works [[3](#_ENREF_3), [4](#_ENREF_4), [10](#_ENREF_10)] on theoretically analysis and modelling the friction coefficient are divided into two methods. One majority of works study the friction coefficient variation over the gears geometry and the contact forces vector[[5](#_ENREF_5), [11](#_ENREF_11)], which seem impractical for wide application. Other works[[15](#_ENREF_15)] considers the gear pairs only loss which is more compact and applicable to general simulation and practical experiments. Consider the rigorous development of British Standards[[15](#_ENREF_15)], model presented in the standards is chosen to calculation the gear meshing caused drag torques, as shown below:

(20)

Where is meshing power loss, kw; is mesh coefficient of friction; is pinion torque, N.m; is pinion rotational speed, rpm; is operating helix angle, degree; is mesh mechanical advantage. Meshing mechanical advantage can be calculated using Equation (21). This equation is a function of the sliding ratios. For external gears, the sliding ratio at the start of approach action, Hs, is calculated using Equation (22), and the sliding ratio at the end of recess action, Ht, is obtained with Equation (23).

(21)

where is transverse operating pressure angle, degree; Hs is sliding ratio at start of approach; Ht is sliding ratio at end of recess.

(22)

(23)

where is gear ratio, which can be calculated by Equation (24); is gear outside radius, mm; is gear operating pitch radius, mm; is pinion outside radius, mm; is pinion operating pitch radius, mm.

(24)

where is number of gear teeth; is number of pinion teeth;

if the pitch line velocity, V, is 2m/s ≤ V ≤ 25 m/s and the load intensity (K-factor) is 1.4 N/mm2 <K≤ 14 N/mm2, then the gear meshing coefficient of friction, , can be expressed by Equation (25). Outside these limits, values for must be determined by experience. K-factor can be calculated using Equation (26). Exponents, j, g and h modify viscosity, , K-factor and tangential line velocity, V, respectively.

(25)

where is kinematic oil viscosity at operating sump temperature, cSt (mm2/s); K is K-factor, N/mm2; C1 is a constant; V is tangential pitch line velocity, m/s.

(26)

where is face width in contact, mm. Values to be used for exponents j, g and h and constant are as follows: J=-0.223, g=-0.40, h=0.70, . The coefficient varies from 0.07 to 0.03 during the simulation.

#### 2.1.2 Gear windage and churning

As alluded before, research on gear windage had been done by Dawson[[7](#_ENREF_7)] and Diab [[9](#_ENREF_9)] on larger high speed gears, and Eastwick[[8](#_ENREF_8)] concludes that the effects of gear windage loss in low and mid speed is not evident, while it will becomes gradually prominent in high speed. For the churning loss, both Changenet[[10](#_ENREF_10)] and British Standard[[15](#_ENREF_15)] present a study for a general gear box. However, the formulas presents in Changenet[[10](#_ENREF_10)] assumed that all the gears are submerged in the fluids, and too many parameters are required to validate to get a reasonable practical dimensionless drag torque coefficient Considers the rigours and universe of standards models, British Standard formulas are adopted here.

As shown in British Standards[[15](#_ENREF_15)], a gear dip factor, , must be considered before obtain gear windage and friction loss. This factor is based on the amount of dip that the element has in the oil. When the gear or pinion does not dip in the oil, . When the gear dips fully into the oil, . When the element is partly submerged in the oil, linearly interpolate between 1 and 0. The primary dip fact of the final gear is 0.21. The higher the speed, the smaller the dip factor becomes. When the speed increases, it will range from 0.21 to 0.1 according to the speed. The primary depth value with 0.21 is calculated via opening the transmission, and measuring the height of shaft, gear size and the oil tube. The final 0.1 is an estimated value after debugging the model parameters and referring from a Conference paper[[29](#_ENREF_29)]. The power loss equation for the tooth surface is named as a roughness factor, . Table 12.5 of reference[[30](#_ENREF_30)] presents several values based on gear tooth size. Equation (27) is a reasonable approximation of the values from Dudley’s model.

(27)

Where is roughness factor, is transverse tooth module, mm.

Gear windage and churning loss includes three kinds of loss. For those loss associated with a smooth outside diameter, such as the outside diameter of a shaft, use Equation (28). For those loss associated with the smooth sides of a disc, such as the faces of a gear, use Equation (29). It should be pointed out that Equation (29) includes both sides of the gear, so do not double the value. For those loss associated with the tooth surfaces, such as the outside diameter of a gear pinion, use Equation (30).

For smooth outside diameters,

(28)

For smooth sides of discus

(29)

For tooth surfaces

(30)

Where is power loss for each individual gear, kw; is gear dip factor; D is outside diameter of the gear, mm; A is arrangement constant, here set as 0.2; F is total face width, mm; L is length of the gear, mm; is generated helix angle, degrees. For helix angles less than 10o, use 10o in Equation (30).

For a common output shaft assembly, Equation (28) would be used for the OD of the shaft outside of the gear between the bearings, Equation (29) for the smooth sides of the gear, and Equation (30) for the tooth surfaces. After calculating the individual gears for each shaft assembly in a reducer, they must be added together for the total loss.

### 2.2 Bearing and Oil Seal Model

As an important part of the power losses in transmission, bearing loss have been analysed by Harris[[12](#_ENREF_12)] for a variety of bearing designs, considering both viscous friction caused torque and applied load generated torque. This work is widely accepted as the forefront on the topic, with similar results being applied in work[[4](#_ENREF_4), [11](#_ENREF_11)], And a similar bearing model is presented in reference[[15](#_ENREF_15)], which is widely used in German gear industry. Hence, the equations from the work[[15](#_ENREF_15)] are chosen and presented below.

Load dependent bearing loss is:

(31)

Speed dependent loss is:

(32)

Where is no-load torque of the bearing, only speed independent, Nm; is bearing dip factor. Factor adjusts the torque based on the amount that the bearing dips in the oil and varies from to . Use if the rolling elements do not dip into the oil and if the rolling elements are completely submerged in the oil, linearly interpolate between and . Values for the dip factor range can be found in reference[[15](#_ENREF_15)].

The oil seal power losses is calculated from reference[[15](#_ENREF_15)].

(33)

Where is the velocity, and is the diameter. The total oil seal losses and bearing losses are expressed in.

(34)

### 2.3 Concentric Shaft Drag Torque

The DCT has two concentric arranged shafts which connect the gears and clutches respectively. When one of the concentric shaft running, the other concentric one will run in a different speed due to the different engaged gear ratios. Therefore, there exists relative speed between two concentric shafts, which cause viscous shear resistance. This concentric shafts shear torque phenomenon has been studied by Schlichting presented in reference[[31](#_ENREF_31)] acting as an example of Couette flow. When the DCT input speed is at 4000 rpm (test speed maximum range), the Taylor’s number is 779 which is smaller than critical one 1708. Hence the assumption for Couette flow is reasonable. Therefore, the drag torque caused by concentric shafts can be expressed as:

(35)

where , are the inner radius of the outer shaft and out radius of the inner shaft. is the relative speed between two concentric shafts. The concentric length is expressed with h. It is assumed here that the annual area is lubricated with continuous flow.

### 2.4 Drag Torque within Multi-plates Wet Clutches

The theories for drag torque within wet multi-plate clutches has been discussed by several authors in the past decades. In 1984, the governing equations presented by Hashimoto[[13](#_ENREF_13)] describe the flow between adjacent flat rotating plates which laid down a frame work for subsequent clutch studies. Then Kato et al.[[16](#_ENREF_16)] explained oil film shrinking between two adjacent clutch plates due to centrifugal force which is widely accepted and referred. Kitabayashi et al.[[18](#_ENREF_18)]provides demonstration of the drag torque which is accurate in low speed ranges, but it is poor with predicting drag torque at high speed region as it can only show the rising portion of a typical drag torque curve at the low speed region when the clearance if full of oil film. It was demonstrated by Yuan et al. [[19](#_ENREF_19)] using the commercial computational fluid dynamics (CFD) code FLUENT. As Yuan et al.[[21](#_ENREF_21)] then provides an improved model which has a reasonable accurate at both low and high speeds, this drag torque model will be compared here with a new shrinking model, which is conducted by Li et al. [[32](#_ENREF_32)]based on analytical and experimental investigation.

**2.4.1 Surface Tension Model**

Yuan et al. [[19](#_ENREF_19)] has presented a reasonable logic to describe the phenomenon in wet clutch plates. When the wet DCT clutch packages is full of dual clutch transmission fluid (DCTF) with viscous, a schematic of a DCT clutch pack is shown in Figure 2. Within the clutch pack, there feeding pressure (inner diameter) is approximately the same as the exiting pressure (outer diameter). The fluid is continually pumped through the clutch pack and flow between two clutches plates is essentially driven by the centrifugal force, while viscous shear force and surface tension forces tend to resist this motion.

**h**

**Separate Plate**

**Friction Plate**

Q

r1

r2

**Figure 2.** Schematic of an open wet clutch.

The effective radius within the clutches plates is determined by the centrifugal force, viscous force and surface tension forces. When rotational speed is low, the viscous and surface tension forces are larger than that of centrifugal force and full immersion is maintained. Thus, clutches plates are full of fluid with as effective radius. With the rotational speed increasing, the centrifugal force will increase. Once the centrifugal force is larger than viscous and surfaces tensions forces, the effective radius will become smaller.

Via analysis of the balancing of centrifugal, viscous and surface tension forces, Yuan[[21](#_ENREF_21)] continues solving the Reynolds equation, which can be expressed in polynomial form. The effective clutch plate outer radius can be achieved as the root of the Equation (35) if less than the existing outer radius.

(36)

The Reynolds number is used to determine the turbulent flow efficiency f shown in Eq. (36), as shown in reference Yuan[[21](#_ENREF_21)]. According to reference[[18](#_ENREF_18)], the effective drag torque within the clutch packs then can be expressed as follows:

(37)

Where represents the relative speed between two adjacent clutch plates. From Figure 2, can also means the relative speed between the input clutch plates and the output clutch plates.

**2.4.2 Alternative Shrinking Model**

After studying the numerical and experimental investigated wet clutch drag torque model conducted by Yuan et al.[[21](#_ENREF_21)], Li et al.[[32](#_ENREF_32)] improved the model with a new way to obtain the equivalent radius considering oil film shrinking phenomenon, which will be discussed as below.

When the clearance is full of oil film, i.e., , then the required input flow rate, can be obtained

(38)

Therefore Equation (38) can be used to calculate the required input flow rate enable the clearance being full of oil film. It shows that the demanded input flow is influenced by the angular speed of the clutches. A trending figure to show the needed flow rate can be plotted by simulation shown in Figure 3.

As shown in Figure 3, the required input flow for full oil film between clutch plates grows with the increasing of the clutch plate’s speed. However, in the real practice, particular in the two-speed DCT, the actual flow rate is constant. Therefore, the oil film will shrink when the flow rate can not meet the required flow rate with the increasing of the speed. Then define the required flow rate as , the actual flow rate is , and the effective outer oil film radius as .

When ,

(39)

When , the relationship of oil flow rate and volume between import and export, the equivalent radius of oil can be presented as:

(40)

Equation (40) can be rearranged as

(41)



**Figure 3.** Relationship of ideal and actual required flow rate.

With the obtained equivalent effective radius from Equation (41), the drag torque can be achieved as follows:

(42)

(43)

Comparisons are made between the two kinds of wet clutch drag torque model, and the simulation results are shown in Figure 4.



**Figure 4.** Relationship of drag torque and angular velocity (2nd model) with clearance 0.20 mm.

Figure 4 shows the simulation results for the first surface tension and the second alternative shrinking model for predicting wet clutch drag torque. It shows the relationship between drag torque and angular velocity. The sold line shows the drag torque for the 1st surface tension model, and the dashed line for the 2nd new shrinking model. Both of the line show that the drag torque is linearly increasing with the angular velocity in the first part from 0 to the peak value. However, after the speed with the peak drag torque, the drag torque reaches almost zero for the 1st surface tension model. And the drag torque for the second alternative shrinking model doesn’t decrease so dramatically. In real practice, in order to continue cool the clutch packages, the follow rates should not be diminished, therefore, the drag torque should not be drop down to zero in high speed, i.e., higher than 3000 rpm, consequently. The second model is more likely to present actual conditions. Therefore, the second shrinking model will be chosen for further modelling and simulation within this work.

**2.4.3 Drag Torque within two-speed DCT**

**C1**

**C1**

**Motor output speed**

**Differential**

**ω0**

**ω1**

**ω2**

**Δω1**

**C2**

**(*a*)**

**Differential**

**Motor output speed**

**ω0**

**ω1**

**ω2**

**C2**

**Δω2**

**(*b*)**

S3

S1

S2

S1

S2

S3

**Figure 5.** Schematics of powertrain equipped with two-speed DCT,

(*a*) clutch 1 closed, clutch 2 open; (*b*) clutch 2 closed, clutch 1 open.

For steady state conditions, if the Clutch 1 (C1) is engaged, as shown in Figure 5 (a), and the C2 is in open state, the vehicle is running with 1st gear. The clutch 1 input shaft (S1) is rotating via mating of the input clutch, and its speed is the same as the output speed of EM (). Then the 1st gear is engaged with the pinions in the layshaft (S3). The layshaft speed is /. Then the layshaft (S3) will rotate with the 1st gear, causing the inner shaft (S2) rotating as well. The inner shaft (S2) speed becomes \*/. Therefore, the C2 output clutch plates will rotate freely caused by the engaged 2nd gear, with speed \*/. The relative speed between the input clutch plates and the 2nd output clutches plates is:

(44)

Similarly, if the C2 is engaged, as shown in Figure 5(b), and the vehicle runs with the 2nd gear, the relative speed between the input clutch plates and the 1st output clutch plates is \*(1-/).

(45)

Equation (43) shows that the drag torque is directly affected by the relative rotational speed between the input and output clutch plates. Substituting of Equation (44) and Equation (45) into Equation (43) respectively, the wet clutch drag torque

(46)

(47)

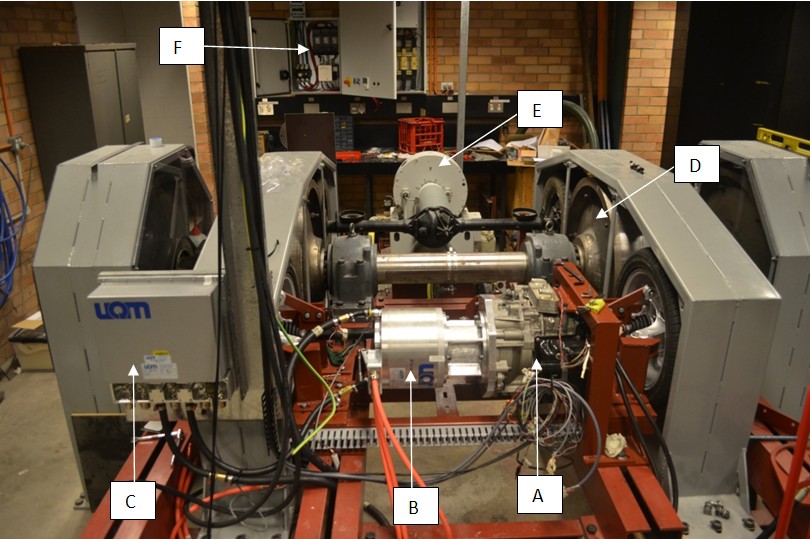
In transmissions, if the input speed from the motor is the same, as is larger than that of G2, it comes out. Thus the relative rotational speed for clutch 2 (run with 2nd gear) is always larger than that of C1 (1st gear). If the effective radius of the clutch 1 and clutch 2 are the same, the drag torque engaged 2nd gear should be larger than that of engaged with 1st gear. However, the effective radius of plates in clutch 2 (C2) of this particular DCT are larger than that of C1. Therefore, the finally drag torque will be determined by both the differences in effective radius and gear ratios.

## Experimental Apparatus

In this section, test facility and hardware, test instrumentation, data acquisition, and test operation will be briefly presented.

### 3.1 Test Facility and Hardware

The test facility used for this study is shown in Figure 6 with several changes to the original UTS powertrain system test rig. The new test rig is based on an electric vehicle power train system. The original Engine is replaced with a 330V DC drive permanent magnet Motor which is converted from AC high voltage electric power; the original AT transmission is replaced with a modified two-speed DCT, and the detail transmission parameters are shown from Table 1 to Table 4. A group of flywheels and tires are used to simulate the vehicle inertia. The dynamometer, HPA engine stand 203, a load drag on the flywheels. This part of test rig consists of two wheels contacted with flywheels, another final drive connected with shaft before dynamometer. The DCT has a separate supply pump. The lubricate fluid will be instead with DCTF rather ATF. And the vehicle control unit (VCU) and transmission control unit (TCU) will be built based on dSPACE Micro-Auto Box.



**Figure 6.** Modified test rig of two-speed DCT powertrain, A) DCT; B) Electric motor; C) Motor controller; D) Groups of flywheels; E) Dynamometer; F) High voltage power supply.

Table 1. DCT clutch plates geometric parameters

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Clutch 1 plates | | Clutch 2 plates | | Clutch Clearance | Plates Number |
| Inner Radius | Outer Radius | Inner Radius | Outer Radius |
| 57.5 mm | 69.5 mm | 81.5 mm | 96.5 mm | 0.5 mm | 8 |

Table 2. DCT gears geometrical data

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| Gears | 1st gear | | 2nd gear | | Differential | |
| Gear | Pinion | Gear | Pinion | Gear | Pinion |
| Ratio |  | 2.0455 |  | 1.452 |  | 4.058 |
| Pitch circle diameter | 108.25 | 53.25 | 95.5 | 66 | 212.5 | 52.5 |
| Tooth depth | 2.875 | 2.875 | 2.75 | 2.75 | 3.75 | 3.75 |
| Centre distance | 80.75 | 80.75 | 80.75 | 80.75 | 132.5 | 132.5 |
| Outside Diameter | 114 | 59 | 101 | 71.5 | 220 | 60 |
| Face width | 18 | 16 | 15 | 15 | 35 | 37 |
| Number of teeth | 45 | 22 | 45 | 31 | 69 | 17 |
| Helix angle | 30 | 30 | 32 | 32 | 32 | 32 |
| Transverse operating  pressure angle | 0.398 | 0.398 | 0.405 | 0.405 | 0.405 | 0.405 |
| 22.796 | 22.796 | 23.228 | 23.228 | 23.228 | 23.228 |
| Module | 2.405 | 2.420 | 2.122 | 2.129 |  |  |

Table 3. DCT Bearings Information

|  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- |
| Bearings | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
| Type | Needle | Needle | Needle | Deep groove  ball | Roller,  angular contact | Roller,  angular contact | Roller,  angular contact | Roller,  angular contact |
| Seals | no | no | no | Non-  contact | no | no | no | no |
| Casing | external | no | no | all | all | all | all | all |
| ID | 34.00 | 24.00 | 35.00 | 28.00 | 30.00 | 38.00 | 28.00 | 40.00 |
| OD | 56.00 | 28.00 | 42.00 | 68.00 | 50.00 | 83.00 | 62.00 | 72.00 |
| Thickness | 14.00 | 24.00 | 26.00 | 18.00 | 18.00 | 18.00 | 13.00 | 19.00 |
| dm | 45.90 | 26.05 | 38.61 | 50.78 | 40.83 | 63.29 | 47.14 | 57.52 |

Table 4. Dual Clutch Transmission Fluid (DCTF) Properties and Test conditions

|  |  |  |  |
| --- | --- | --- | --- |
| Flow rate (m^3/s) | Oil density (kg/m3) | Oil viscosity (Ns/m2) | Temperature (oC) |
| 0.00005 | 853.4 | 0.0290 | 40 |

During the test, the temperature is between 35oC and 45oC, normally approximate 40 oC. Because every test point is running about 2 - 3 minutes then taking a breaking. Hence it is assumed that the oil temperature does not change too much during this period.

### 3.2 Instrumentation and Data Acquisition

The test instrumentation used in this study only required measurement of the input and output speed and torque of the DCT respectively, and the DCT inside oil case temperature. Two wireless torque sensors are installed in the front half drive shafts, shown in Figure 7, which are used to acquire the output torque data of the DCT. The wireless torque sensor was bought from ATI Technologies. And the oil temperature is tested by the temperature sensor embedded in the transmission itself. The torque sensors are calibrated before testing. And the DCT input torque is equal to the output torque of electric motor, which can be collected by the motor feedback torque. This data was collected at 1000 per second throughout the test via dSPACE Control Desk Recorder, and then processed in computer.

Electric Motor

Wireless Torque sensor



DCT

Flywheels

Motor

Controller

Torque Sensors Data Recorder

Electrical Connection

Mechanical Connection

**Figure 7.** Schematic test rig system

### 3.3 Test Operation

The test procedure used for collecting the data was the following. For a given group of operating conditions (gear, speed, motor torque, and lubricating oil temperature) the test rig was operated and stabilized last for at least 2 minutes. Then average one is chosen. And the error of efficiency is ±1%. Via comparing the input and output torque of transmission under different driving conditions, the difference of total drag torque power loss can be measured by torque sensor.

## Results and analysis

In order to validate the proposed model under the conditions of Table 1 and Table 2, simulation and experimental tests are made. The following contents are results and analysis regarding experimental and simulation results.

Figure 8 and Figure 9 show the DCT efficiency for 1st gear (a) and 2nd gear (b) from simulation prediction and during testing using constant input torque and constant input speed respectively. In Figure 8, constant input torque (60 Nm) and various input speed from electric motor is set in this group of study. The DCT average efficiency is higher than 95% in both gears. And the efficiency in the 2nd gear shown in Figure 8 (b) is slightly higher than that of in the 1st gear shown in Figure 8 (a). Both of the efficiency decreases until reach their bottom values respectively. After a certain speed, i.e. critical speed, the drag torque will increase gradually with the speed continually increasing in high speed. The predicted critical speeds for 1st and 2nd gear are 2950 rpm and 2530 rpm respectively, which are very close to the speeds obtained from test results. The differences between simulation and test results are smaller, less than 0.5%. It appears that the simulation results are always a little higher than test results by 0.2% in efficiency. It is likely there are some other minor sources affecting the test final results, such as the windage loss from the shaft and 1st and 2nd gear pairs by oil-air mixture resistance.



(a) 1st gear

****

1. 2nd gear

**Figure 8.** Comparison of DCT efficiency between simulation and test results with input torque 60 Nm

In Figure 9, results for constant input speed (3000 rpm) and variation of input torque from electric motor is presented. The input torque for first gear (a) changes from 20 Nm to 60 Nm, whilst for the second ranges from 30 Nm to 60 Nm, because second gear ratio (G2=5.36) is smaller than first one (G1=8.45), i.e., smaller gear ratio requiring larger torque to start up or maintaining stable speed. Both Figure 9 (a) and (b) show that the DCT efficiency increases with the input torque continually increasing. And the efficiency in the second gear shown in Figure 9 (b) is slightly higher than that of in the first gear shown in Figure 9 (a). The differences between simulation and test results are smaller, especially in higher input torque with less than 0.5%.

From Figure 8 and Figure 9, it can be concluded that the results from the predicted drag torque model agrees well with that from experimental test.



1. 1st gear



1. 2nd gear

**Figure 9.** Comparison of DCT efficiency between simulation and test results with input speed 3000 rpm

****

(a) 1st gear ****

(b) 2nd gear

**Figure 10.** Individual drag torque power loss from simulation result with input torque 60 Nm.

Figure 10 shows the individual drag torque losses from simulation results at stable statue. The clutch drag torque is the major loss in both gears, followed by gear churning and windage losses and bearing losses. And the clutch drag torque loss in 1st gear is larger than that of 2nd gear if under the same input speed. The reason for that is when 1st gear engaged, the larger clutch (C2) will open, which generates higher torque than that of the smaller clutch (C1) when the 2nd gear is engaged.

Figure 11 shows the percentage of losses by individual drag torque sources. It shows that the percent of loss generated by clutch drag increases gradually before reach the critical speed. After the critical speed, the percent of clutch drag loss drops down slightly. It is closely corresponding with the shape in Figure 8 and Figure 10. The percent of power loss in 1st gear by clutch drag is higher than that of in 2nd gear. And gear churning and windage loss holds the reverse condition, that is, the percent of power losses in 1st gear by gear churning and windage losses is lower than that of in 2nd gear.



1. 1st gear



1. 2nd gear

**Figure 11.** Percent of Individual drag torque power loss from simulation results with input torque 60 Nm.

****

1. 1st gear



1. 2nd gear

**Figure 12.** Individual drag torque power loss from simulation result with input speed 3000 rpm.

Figure 12 shows the individual drag torque losses from simulation results with constant input speed 3000 rpm. Input torque from electric motor changes from 20 Nm to 70 Nm. Both Figure 12 (a) and (b) demonstrate that the clutch drag torque is the major loss in both gears, followed by gear churning and windage losses and bearing losses. And the gear meshing loss shares only a small part of the global resisting torque. Also the clutch drag torque loss in 1st gear is larger than that of 2nd gear, which can explain why the efficiency of DCT first gear is smaller than that of second gear. Moreover, only bearing and oil seal, and gear meshing loss increase slightly with the increasing of input torque, as both represent only a small part of the total drag torque losses, which can explain why the DCT efficiency will increase with increasing of input torque at constant input speed.

## 5 Conclusions

The purpose of this paper is to develop a model of drag torque in a dual clutch transmission to predict its operating efficiency. After theoretical analysis of five major sources of drag torque within a two-speed DCT, this article then implemented the developed mathematical model of the transmission and applied it to steady state simulation. To support the theoretical study an experimental investigation is conducted to compare the physical results with that of obtained from simulation under various operating conditions.

The test results for DCT efficiency agrees reasonable accurately with the simulation results, especially the critical speed prediction. The average error in DCT efficiency is less than 0.5%. It can be concluded that the proposed model performs well in the prediction of drag torques for the transmission, and is can be applied to assess the efficiency of the transmission. Results demonstrate that (1) the mean two-speed DCT efficiency can be reach 95%. (2)And the efficiency of DCT grows with the input torque increasing, and decreases with the rise of input speed. Additionally, (3) the entire drag torque is dominated by the viscous shear in the wet clutch pack, followed by the differential gear churning and windage losses. And the gear meshing loss holds only a small contribution of the global resisting torque. (4) Moreover, when the vehicle run in 2nd gear, the drag torque generated by clutch shear stress is smaller than that of run in 1st gear in the same input speed. (5) The gear churning and windage losses for 2nd gear, however, are larger than that of in 1st gear.

This work can be looked as a reference to future research on reducing drag torque, applications of drag torque in two-speed or multi-speed electric vehicle powertrain system efficiency optimization and torque estimated shift control. With some slight adjustments according to the detail transmissions layout, this numerical method for predicting the two-speed DCT drag torque can also be applied to other transmissions equipped with wet clutch packages, let alone ordinary manual transmission. But the limitation of this work does not consider the influences of temperature on drag torque, which will be analysed on our future work regarding transmission thermal behaviours analysis.

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**Nomenclature**

|  |  |  |
| --- | --- | --- |
|  | = | arrangement constant for gearing |
|  | = | face width in contact |
| C1 | = | clutch 1 |
| C2 | = | clutch 2 |
| D | = | outside diameter of the gear |
|  | = | diameter |
|  | = | efficiency of DCT |
| F | = | total face width |
|  | = | applied load |
| f | = | turbulent flow coefficients |
|  | = | bearing dip factor |
|  | = | gear dip factor |
|  | = | mesh coefficient of friction |
|  | = | bearing load empirical factor |
| Hs | = | sliding ratio at the start of approach action |
| Ht | = | sliding ratio at the end of recess action |
| h | = | tangential line velocity modifying exponent |
|  | = | clutch plates clearance |
|  | = | length of concentric shaft |
| j | = | Viscosity modifying exponent |
| K | = | load intensity |
| L | = | length of the gear |
|  | = | mesh mechanical advantage |
|  | = | transverse tooth module |
|  | = | number of frictional surface |
|  | = | pinion rotational speed |
|  | = | motor speed |
|  | = | power losses caused by bearings drag torque |
|  | = | power losses caused by concentric shaft drag torque, |
|  | = | load independent power loss |
|  | = | speed independent power loss |
|  | = | power losses caused by gear churning |
|  | = | power losses caused by wet clutch plates drag torque |
|  | = | power losses caused by gear meshing drag torque |
|  | = | individual gear windage and churning loss |
|  | = | total power losses in DCT |
|  | = | oil seal power losses |
|  | = | flow rate |
|  | = | pinion torque |
|  | = | drag torque caused by bearings |
|  | = | drag torque caused by bearing (1) and (2) |
|  | = | drag torque caused by wet clutch packs |
|  | = | drag torque caused by churning |
|  | = | drag torque caused by concentric shafts viscous shear resistance |
|  | = | output torque of the outer concentric shaft |
|  | = | output torque of the inner concentric shaft |
|  | = | final output torque from DCT |
|  | = | drag torque caused by gear pairs meshing |
|  | = | drag torque caused by 1st gear pair meshing |
|  | = | motor output torque |
| V | = | pitch line velocity |
|  | = | velocity for oil seal |
|  | = | outer radius of the inner shaft |
|  | = | inner radius of the outer shaft |
|  | = | roughness factor |
|  | = | gear outside radius |
|  | = | gear operating pitch radius |
|  | = | pinion outside radius |
|  | = | pinion operating pitch radius |
|  | = | gear ratio |
|  | = | mean radius |
|  | = | outer radius of the clutch |
|  | = | 1st gear ratio |
|  | = | 2nd gear ratio |
|  | = | number of pinion teeth |
|  | = | number of gear teeth |
|  | = | transverse operating pressure angle |
|  | = | generated helix angle |
|  | = | operating helix angle |
|  | = |  |
|  | = | kinematic oil viscosity |
|  | = | density of oil |
|  | = | output speed of the electric motor |
|  | = | relative speed between two concentric shafts |
|  | = | relative speed within clutch 1 |
|  | = | relative speed within clutch 2 |
|  | = | pressure difference between the input and output of clutch pair |

## References

1. Chan CC. Electric, Hybrid & Fuel Cell Vehicles: Overview, State-of-the-art, Key Technologies & Issues*.* *Proceedings of the IEEE* 2007; 95(4): 704-718.

2. Zhou X, Walker P, Zhang N, and Zhu B. Performance Improvement of a Two Speed EV through Combined Gear Ratio and Shift Schedule Optimization*.* *SAE Technical Paper 2013-01-1477* 2013.

3. Diab Y, Ville F and Velex P. Investigations on Power Losses in high speed gears*.* *Proc. Inst. Mech. Eng. Part J: J. Eng. Tribol.* 2006; 220: 191-198.

4. Heingartner P and Mba D. Determining Power Losses in the Helical Gear Mesh*.* *Gear Technol.* 2005: 32-37.

5. Li S, Vaidyanathan A, Harianto J, and Kahraman A. Influence of Design Parameters on Mechanical Power Losses of Helical Gear Pairs*.* *Journal of Advanced Mechanical Design, Systems, and Manufacturing* 2009; 3(2): 13.

6. Xu H, Kahraman A, Anderson NE, and Maddock DG. Prediction of mechanical efficiency of parallel-axis gear pairs*.* *J. Mech. Des.* 2007; 129: 58-68.

7. Dawson PH. Windage loss in large high speed gears*.* *Proc. Inst. Mech. Eng. Part J: J. Engineering Tribology* 1984; 198A(1): 51-59.

8. Eastwick CN and Johnson G. Gear windage: a review*.* *J.Mech. Des.* 2008; 130.

9. Diab Y, Ville F, Houjoh H, Sainsot P, et al. Experimental and numerical investigations on the air pumping phenomenon in high speed spur and helical gears*.* *Proc. Inst. Mech. Eng., Part C: J. Mechanical Engineering Sciences* 2004; 219(G8): 785-790

10. Changenet C and Velex P. A model for the prediction of churning losses in geared transmissions—preliminary results*.* *J. Mech. Des.* 2006; 129(1): 128-133.

11. Changenet C, Oviedo-Marlot X and Velex P. Power loss predictions in geared transmissions using thermal networks-application to a six speed manual gearbox*.* *Trans. Am. Soc. Mech. Eng.* 2006; 128: 618–625

12. Harris TA, *Rolling Bearing Analysis*. 1st ed. 1966.

13. Hashimoto H, Wada S and Murayama Y. The performance of a Turbulent Lubricated Sliding Bearing Subject to Centrifugal Effect*.* *Trans. Jpn, Soc. Mech. Eng.,Ser, C* 1984; 49(446): 1753-1761.

14. Walker PD, Zhang N, Tamba R, and Fitzgerald S. Simulations of drag torque affecting synchronisers in a dual clutch transmissions*.* *Japan J. Indust. Appl. Math.* 2011; 28: 119-140.

15. British Standards Institute BS ISO/TR 14179-1:2001, *Gears-thermal capacity-Part 1: rating gear drives with thermal equilibrium at 95◦C sump temperature. Part 2:Thermal load-carrying capacity*. 2001.

16. Kato Y, Murasugi T and Hirano H. Fuel Economy Improvement through Tribological Analysis of the Wet Clutches and Brakes of an Automatic Transmission*.* *Society of automotive Engineers of Japan* 1993; 16(12): 57-60.

17. Iqbal S, Janssens T, Desmet W, and Al-Bender F. Transmitted power and energy flow behaviour of degrading wet friction clutches*.* *International Journal of Applied Research in Mechanical Engineering* 2012; 2(1): 94-100.

18. Kitabyashi H, Li C and Hiraki H. Analysis of the Various Factors Affecting Drag Torque in Multiplate Plate Wet Clutches*.* *JSAE Paper No.2003-01-1973.* 2003.

19. Yuan Y, Attibele P and Dong Y. CFD Simulation of the Flows Within Disengaged Wet Clutches of an Automatic Transmission*.* *SAE Technical Paper 2003-01-0320* 2003.

20. Aphale CR, Cho J, Schultz WW, Ceccio SL, et al. Modeling and Parametric Study of Torque in Open Clutch Plates*.* *Journal of Tribology* 2005; 128(2): 422-430.

21. Yuan S, Peng Z and Jing C. Experimental Research and Mathematical Model of Drag torque in Single-plate Wet clutch*.* *Chinese Journal of Mechanical Engineering* 2010; 23: 1-8.

22. Lechner G and Naunheimer H, *Automotive Transmissions—Fundamentals, Selection, Design and Application*. 1st ed. 1999, Berlin: Springer.

23. Patel PD and Patel JM. An Experimental Investigation of Power Losses in Manual Transmission Gear Box*.* *International Journal of Applied Research in Mechanical Engineering* 2012; 2(1): 1-5.

24. Iritani M, Aoki H, Suzuki K, and Morita Y. Prediction Technique for the Lubricating Oil Temperature in Manual Transaxle*.* *SAE Technical Paper 1999-01-0747* 1999.

25. Seetharaman S, Kahraman A, Bednarek G, and Rosander P. A Model to Predict Mechanical Power Losses of Manual Transmissions*.* *Automobiltechnische Zeitschrift* 2008(4): 346-357.

26. Per R, Georg B, Satya S, and Ahmet K. Development of an efficiency model for manual transmissions*.* *ATZ worldwide* 2008; 110(4): 36-43.

27. Michlin Y and Myunster V. Determination of power losses in gear transmissions with rolling and sliding friction incorporated*.* *Mechanism and Machine Theory* 2002; 37(2): 167-174.

28. Goetz M, *Integrated Powertrain Control for Twin Clutch Transmissions*. University of Leeds. 2005.

29. Yang L, Li H and Ma B, *Prediction Model of No-load Power loss for a DSG Transmission*, in *The 5th TM Symposium China, ICE, HEV and EV transmission*: Suzhou, China. 2013, p. 44-52.

30. Dudley DW and Townsend DP, *Dudley's gear handbook*. 1991: McGraw-Hill.

31. Schlichting H and Gersten K, *Boundary-Layer Theory*. 2000: MacGraw-Hill.

32. Li H, Jing Q and Ma B. Modeling and Parametric Study on Drag Torque of Wet Clutch,*.* *Proceedings of the FISITA 2012 World Automotive Congress, Lecture Notes in Electrical Engineering* 2013; 193: 21-30.