
Theoretical and experimental investigation of the thermal behaviour of a two-speed dual clutch transmission

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Abstract: Understanding the heating behaviour of automotive transmissions in an important step in prototype development, it plays a significant role in achieving appropriate tolerances at the design phase and strongly influences the system's durability in the long term. Furthermore, lubricant viscosity is strongly dependent on operating temperature, understanding this behaviour will provide insight into how transmission efficiency is affected during vehicle warm-up. In this paper, an experimental and simulation based study has been conducted to investigate the thermal behaviour of a two-speed dual clutch transmission (DCT) that has been developed for electric vehicle applications. This is achieved through the modelling of the power losses in the system that is then integrated with a compact thermal model of the transmission. Transient heating simulations for the transmission are conducted at constant speeds and compared with experimental studies, showing a strong correlation between results. Simulations are then utilised to investigate the load and speed dependencies of the transmission's thermal behaviour.

Keywords: thermal behaviour; heat dissipation; power losses; DCT; dual clutch transmission.

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Biographical notes: Xingxing Zhou received his PhD in 2014 from the University of Technology Sydney. His research interests include the analysis of transmission dynamics, transmission thermal behaviour and transmission shifting control.

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Nong Zhang received his BE in 1982 from Northeastern University, ME in 1984 from Shanghai Jiao Tong University, China, and PhD in 1989 from the University of Tokyo, Japan. He worked at several universities in China, Japan, USA and Australia, respectively before joining the Faculty of Engineering of the University in 1995. Since 2009, he has been Professor of Mechanical Engineering, at School of Electrical, Mechanical and Mechatronic Systems. For more than 20 years, he has been involved in research in areas of dynamics and control of automotive systems including powertrains with various types of transmissions, hybrid propulsion systems, vehicle dynamics, passive and active suspensions, and mechanical vibration including experimental modal analysis, rotor dynamics, cold rolling mill chatter and machine condition monitoring.

1 Introduction

In recent years, there has been considerable effort devoted to producing energy efficient vehicles and machines, not only for economic reasons, but also to contribute to meet the requirement for emissions reduction. Improvement of the fuel consumption of existing technologies and reduction of greenhouse gas emissions are some of the most important reasons for developing new vehicle technologies. Generally, in conventional combustion vehicles, one-third of the fuel energy is used to overcome friction in the powertrain system, including the engine, transmission, tyres and brakes. By making the best use of new technologies for friction or wear reduction in vehicles, friction losses can be cut down by 18% in the short term (5–10 years) and by 61% in the long term (15–25 years). Furthermore, the energy losses related to friction in a pure electric vehicle (EV) are

estimated to be only half those of a conventional combustion vehicle (Holmberg et al., 2012). As a consequence, the development of commercially viable hybrid electric vehicles (HEVs), fuel cell vehicles (FCVs) for using in the short to mid-term, and pure EVs in the long term represent the automotive industry's major initiatives towards solving these related issues (Chan et al., 2010).

Pure EVs currently being available on the market are equipped with fixed reduction transmissions (i.e., one reduction gear), trading between the vehicle performance characteristics of acceleration and grade climbing against vehicle top speed, whilst maximising driving range (i.e., driving efficiency). More research and development is now focused on the deployment of multi-speed transmissions in place of fixed reduction gears, as these are expected to improve the EV performance without degrading the overall driving range. As has been discussed in our previous work (Roser et al., 2013; Walker et al., 2013; Zhou et al., 2013a, 2013b), the use of these multi-speed transmissions in EVs can improve the average operating efficiency of the motor and therefore range, it is even possible to reduce the size of the electric motor. Examples of the application of two-speed transmissions include the deployment in the front wheel drive of the BMW i8 (Merwerth, 2014), applications to novel four-wheel drive powertrains (Chen et al., 2013) and even applications in electric buses (Yang et al., 2014).

Computer aided engineering has enabled the rapid reduction in design and development times across all forms of engineering, reducing the requirements for prototyping and testing through the generation of reasonably accurate simulated data earlier in the design life. Lechner (Naunheimer et al., 2011) indicates that the thermal behaviour of automotive transmissions is not considered early in the design life due to its inherent complexity. However as indicated in Naunheimer et al. (2011) among, the thermal expansion of transmission cases, for example, can change gear axes geometry positions, gears clearances, and lubricant regime and film conditions, and consequently, dissipated heat during normal operation. Furthermore, it is of great importance to know the temperature of oil lubricated transmission systems, and the quantity of required oil cooling. The prediction of the thermal behaviour of a transmission is beneficial to the evaluation of cooling and lubrication conditions, and can aid in the analysis of transmission efficiency. Therefore, understanding the thermal behaviour of automotive transmissions is essential for a number of design considerations, notably cooling system design, operational behaviour and reliability. Consequently, significant benefits can be attained through the development of models to predict the thermal behaviour earlier in the design life.

Transmission thermal behaviour is directly affected by the heat generated from transmission power losses. Hence, it is important to study the sources of power losses in a transmission as part of the model development. The total power losses within a gear-train for any transmission are generally made up from several sources, including gear friction (Heingartner and Mba, 2003; Diab et al., 2006; Xu et al., 2007; Li et al., 2009), windage (Diab, 2004; Dawson, 2006; Eastwick and Johnson, 2008) and oil churning (Changenet et al., 2005; Changenet and Vex, 2007). Other important drag torque sources have to be considered as well e.g., bearings and seals (Harris, 1966; Hashimoto et al., 1984; Changenet et al., 2005), synchroniser and free-pinion losses (British Standards Institute, 2001; Walker et al., 2011). There are only a few reported works on the

entire transmission power losses (Iritani et al., 1999; Michlin and Myunster, 2002; Rosander et al., 2008; Seetharaman, 2008; Patel and Patel, 2012), which mostly focus on the manual transmission (MT) gearbox, these, therefore, do not include wet clutch losses in the transmission models.

Some works (Joule et al., 1988; Coe, 1989; Changenet et al., 1996; Phillips and Association, 1996; Handschuh and Kilmain, 2002) have performed thermal analysis of manual and automatic transmissions, but there is limited published work that combines gearbox components and wet clutch losses, an example being (Marklund et al., 2007) where limited slip differentials are studied. In particular, there is no published report on the study of thermal behaviour within a wet dual clutch transmission (DCT). 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 (Kato and Murasugi, 1993; Iqbal et al., 2011), as these will significantly influence the power losses and heat generated in the transmission, with direct implications for vehicle warm up and heat rejection needs for the transmission.

The purpose of this paper is to develop a compact thermal model of a DCT that integrates the power losses developed and relates these to the heating behaviour of the transmission. Such models have specific applications early in the design lifecycle as they allow prediction of the thermal behaviour, with direct impact on transmission durability and reliability. Where designing to accommodate thermal effects will directly impact on tolerance requirements for critical components (i.e., bearing or shaft alignment), which will significantly influence the system durability and reliability. Furthermore (Chiara and Canova, 2013) identifies the reduction in lubricant viscosity through improved thermal management as an important avenue for improving transmission efficiency. Specific applications of this model could also include simulating the cold start behaviour of simulated vehicles in driving cycle based studies for evaluation of fuel economy, etc.

Given the analysis of drag torque of DCT (Changenet et al., 1996) and simulation existing result thermal behaviour (Coe, 1989), a comprehensive study of the power losses and resulting thermal behaviour of a wet DCT is evaluated in this paper, then, validated in the testing bench. Given the layout of the DCT under investigation, that contributions to power losses in the transmission include wet clutches, concentric shaft viscous shear, as well as gear and bearing power losses (i.e., gear meshing, gear windage, churning, bearing losses and oil seal losses). Utilising the developed drag torque model as the heat source a thermal model of the system is developed and simulations with experimental verification are performed to demonstrate the effectiveness of the model. This paper is organised as follows. In Section 2, after briefly discussing the different source of power losses in the DCT, the thermal behaviour model will be introduced and modelled. Then, the experimental aspects comprising of the description of the University of Technology Sydney (UTS) test rig, and the power loss and thermal rise test procedures are presented and discussed in Section 3. In Section 4, results from three aspects will be presented. These include the experimental test results, simulation results from implemented drag torque model and the thermal behaviour model and comparison of said models. Finally, major conclusions are drawn from the study, these are summarised in Section 5.

2 Theoretical analysis of power losses and heat dissipation in a two-speed dual clutch transmission

The system under consideration is modified from a 6-speed DCT (DQ250) into a two-speed DCT as shown in Figure 1. The two-speed DCT housing is made from an aluminium alloy. The DCT used in this testing comprises of dual wet clutches with a single drum providing input power from the electric motor. The concentrically aligned output shafts are connected to the inner clutch (C1) and the outer clutch (C2). Each output shaft is connected to either the 1st or the 2nd gear. C1, shown in green, connects the outer input shaft with the 1st gear, and C2, shown in red, connects the inner input shaft with the 2nd gear. In order to make the transmission control system simpler and reduce manufacturing costs, there are no synchronisers in this DCT. Shifting is achieved through the simultaneous opening and closing of both clutches to achieve power on gear shifts. For this special layout, a vehicle equipped with a DCT not only changes speed smoothly with nearly no power hole, as identified by Goetz et al. (2004), but it also improves the EV efficiency.

There are several major differences between the EV powertrain and that of a conventional vehicle. These make the powertrain simpler and influence the rate of heat generation in the transmission. These are:

- The EV power train is launched from a stationary state using motor torque only, therefore no heat is generated in the friction clutch, this significantly reduces the thermal load. A conventional DCT will use friction clutches to launch the vehicle thereby generating considerable friction energy that must be dissipated, frequently using small heat exchangers.
- No synchronisers are used in the powertrain to isolate the open wet clutch from the gear-train, consequently, wet clutch drag will always contribute to powertrain losses.
- The wider operating region of an electric motor requires fewer gear pairs in the transmission, in this case only two gears are used in the DCT. Therefore, there are significantly fewer gear changes in the transmission (Walker et al., 2013), reducing the rate of heat generation in comparison to a conventional vehicle.

2.1 Power losses analysis

The sources of power loss in a two-speed DCT (Figure 2), 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 loss source, the following formulae have been implemented in the simulation code.

$$P_L = P_{Con} + \sum P_B + \sum P_G + P_{Ch} + P_{Cl}, \quad (1)$$

where P_L is total power losses in DCT, P_{Con} is power losses caused by concentric shaft drag torque, P_B is power losses caused by bearings drag torque, P_G is power losses caused by gear meshing drag torque, P_{Ch} is power losses caused by gear churning, P_{Cl} is power losses caused by wet clutch plates drag torque. Figure 2 shows the energy losses flow from the electric machine to the environment through heat dissipation.

Figure 1 Schematic of a two-speed DCT powertrain system (see online version for colours)

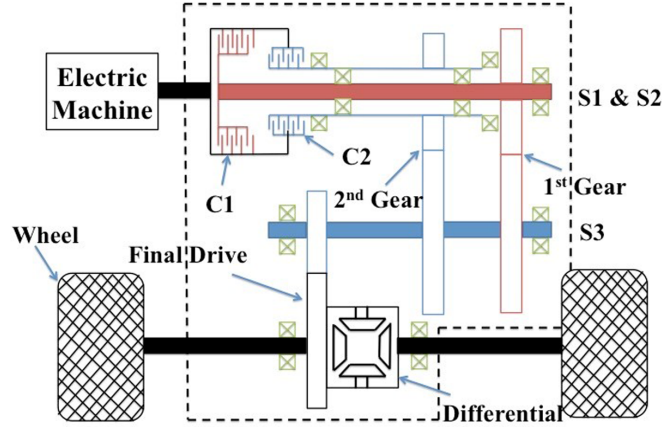
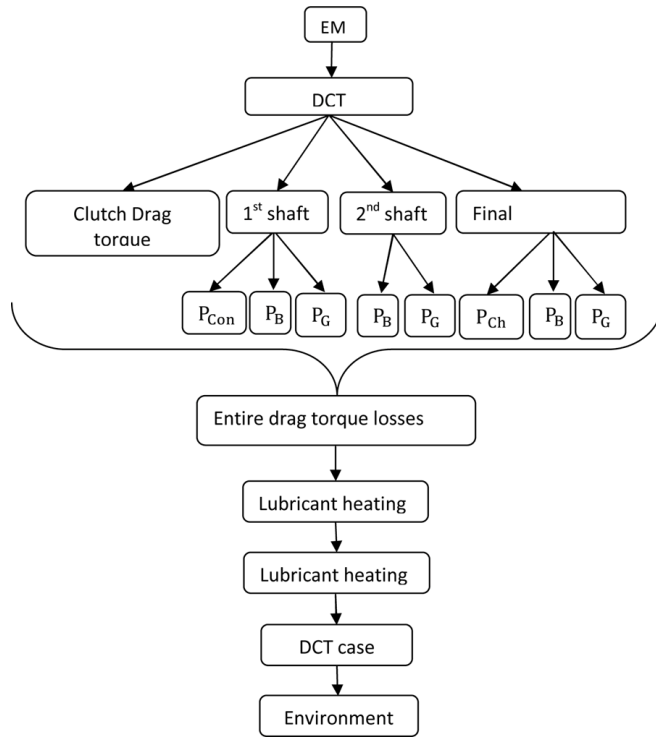


Figure 2 Energies flow chart for the two-speed DCT



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

$$(T_m - T_{con} - T_{B(1,2)})r_{1st} = T_{1st_output_outer} \quad (2)$$

$$\left[T_{1st_output_outer} - T_{GM_{1st_pair}} - T_{B(6,7)} - (T_{GM_{2nd_pair}} + T_{Cl_{C2}} + T_{B(3,4,5)})r_{2nd} \right] = \frac{T_{2nd_output}}{r_{3rd}} \quad (3)$$

$$T_{2nd_output} - T_{GM_{3rd_pair}} - T_{Ch} - T_{B(8,9)} = T_{final_output} \quad (4)$$

where T_m is motor output torque; T_{con} is drag torque caused by concentric shafts viscous shear resistance; T_B is drag torque caused by bearings; and r is gear ratio, $T_{1st_output_outer}$ represents the output torque of the outer concentric shaft. T_{GM} is drag torque caused by gear pairs meshing, T_{Cl} is the drag torque caused by wet clutch packs. T_{ch} is the drag torque caused by churning. And the individual power losses can be calculated in rotational speed multiplied by drag torque:

$$P = T\omega \quad (5)$$

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

$$P_{Con} = T_{con}\omega_{motor} \quad (6)$$

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

$$\sum P_B = \left(T_{B(1,2)}\omega_{motor} + T_{B(3,4,5)}\omega_{motor} \frac{r_{2nd}}{r_{1st}} + T_{B(6,7)} \frac{\omega_{motor}}{r_{1st}} + T_{B(8,9)} \frac{\omega_{motor}}{r_{1st} \times r_{3rd}} \right) \quad (7)$$

$$\sum P_G = \left(T_{GM_{1st_pair}} \frac{\omega_{motor}}{r_{1st}} + T_{GM_{2nd_pair}} \omega_{motor} \frac{r_{2nd}}{r_{1st}} + T_{GM_{3rd_pair}} \frac{\omega_{motor}}{r_{1st} \times r_{3rd}} \right) \quad (8)$$

The power losses resulting from gear windage and churning is:

$$P_{Ch} = \frac{T_{Ch}\omega_{motor}}{r_{1st} \times r_{3rd}} \quad (9)$$

The overall power loss caused by the clutch package is:

$$P_{Cl} = \frac{T_{Cl_{c2}}\omega_{motor}r_{2nd}}{r_{1st}} \quad (10)$$

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

$$(T_m - T_{con} - T_{B(3,4,5)})r_{1st} = T_{1st_output_inner} \quad (11)$$

$$\left[T_{1st_output_inner} - T_{GM_{2nd_pair}} - T_{B(6,7)} - (T_{GM_{1st_pair}} + T_{Cl_{c1}} + T_{B(1,2)})r_{1st} \right] = \frac{T_{2nd_output}}{r_{3rd}} \quad (12)$$

$$T_{2nd_output} - T_{GM_{3rd_pair}} - T_{Ch} - T_{B(8,9)} = T_{final_output} \quad (13)$$

$$P_{Con} = T_{con}\omega_{motor} \quad (14)$$

$$\sum P_B = \left(T_{B(1,2)} \frac{\omega_{motor}r_{1st}}{r_{2nd}} + T_{B(3,4,5)}\omega_{motor} + T_{B(6,7)} \frac{\omega_{motor}}{r_{2nd}} + T_{B(8,9)} \frac{\omega_{motor}}{r_{2nd} \times r_{3rd}} \right) \quad (15)$$

$$\sum P_G = \left(T_{GM_{1st_pair}} \omega_{motor} \frac{r_{1st}}{r_{2nd}} + T_{GM_{2nd_pair}} \frac{\omega_{motor}}{r_{2nd}} + T_{GM_{3rd_pair}} \frac{\omega_{motor}}{r_{2nd} \times r_{3rd}} \right) \quad (16)$$

$$P_{Ch} = \frac{T_{Ch} \omega_{motor}}{r_{2nd} \times r_{3rd}} \quad (17)$$

$$P_{Cl} = \frac{T_{Clc2} \omega_{motor} r_{1st}}{r_{2nd}}. \quad (18)$$

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

$$\frac{\omega_{final} T_{final_output}}{\omega_{motor} T_{motor}} \times 100\% = E_{DCT}. \quad (19)$$

Further detail equations regarding calculation of components' drag torques and power losses can be referred from our previous work (Zhou et al., 2014), where both numerical and experimental investigation are conducted to study the entire two-speed DCT's drag torque. It was demonstrated that the proposed model performs well in the prediction of drag torque for the DCT, with an acceptable correlation between simulated and experimental results. Particularly, it was shown that the entire drag torque for the transmission is dominated by the viscous shear in the wet clutch pack and gear churning losses. Accordingly, the model can be applied to calculate the power losses the transmission.

2.2 Heat dissipation

The BS/ISO (British Standards Institute, 2001) model is adopted in this paper for the simulation of heat exchange through the transmission (British Standards Institute, 2001). The Remainder of this section was initially presented in our previous work (Zhou et al., 2013a, 2013b).

The quantity of heat, Q_{Ca} , dissipated through the DCT case by convection can be calculated by:

$$Q_{Ca} = \int k A_{ca} (T_{oil} - T_{en}) dt, \quad (20)$$

where, k represents the heat transmission coefficient, which includes the internal heat transfer between oil and case, and the heat conduction through the case wall and the external heat transfer to the environment, usually surrounding air. T_{oil} and T_{en} mean the oil temperature and environment temperature, respectively with the unit of Kelvin.

$$\frac{1}{k} = \frac{1}{\alpha_{oil}} \frac{A_{ca}}{A_{oil}} + \frac{\delta_{wall}}{\lambda_{wall}} \frac{A_{ca}}{A_{oil}} + \frac{1}{\alpha_{ca}}. \quad (21)$$

The heat dissipation via the DCT case is determined by the larger value air-side, i.e., external side, thermal resistance at the case surface. The front two terms in the above equation can then be ignored. For high air velocities and thus good external heat transfer, it will be necessary to also consider the oil-side heat transfer. According to (British Standards Institute, 2001) an oil-side heat transfer coefficient of $\alpha_{oil} = 200 \text{ W/m}^2\text{K}$ is assumed. But it requires investigation and revision for different oil types. The heat conduction through the transmission case should only be considered in special cases,

such as in the case of double-walled cases, cases with insulation and non-metallic cases. And the appropriate coefficient of thermal conduction, λ_{wall} , has to be expressed in the case material in question.

The air-side heat transfer, α_{ca} , includes a convection part, α_{con} , and a radiation part, α_{rad} , which performs as

$$\alpha_{ca} = \alpha_{\text{con}} + \alpha_{\text{rad}} \quad (22)$$

$$\alpha_{\text{rad}} = 0.23 \times 10^{-6} \varepsilon \left(\frac{T_{\text{wall}} + T_{\text{en}}}{2} \right)^3, \quad (23)$$

where the emission ratio, ε , is obtained as 0.05.

The convection part can be divided into two parts, free and forced convection. According to the investigations by Funk (1985) and employed in British Standards Institute (2001), the following expressions can be presented that accommodates exposed case surface area against that exposed to forced cooling:

$$\alpha_{\text{con}} = \alpha_{\text{free}} \left(1 - \frac{A_{\text{air}}}{A_{ca}} \right) + \alpha_{\text{forced}} \frac{A_{\text{air}}}{A_{ca}} \eta^*, \quad (24)$$

where

$$\eta^* = \frac{T_{\text{wall}} - T_{\text{air}}}{T_{\text{wall}} - T_{\text{en}}}. \quad (25)$$

As this DCT case without thermal finning, the free and forced convection can be followed as:

For free convection ($V_{\text{air}} \leq 1.5$ m/s):

$$\alpha_{\text{free}} = 18 h_{ca}^{-0.1} \left(\frac{T_{\text{wall}} - T_{\text{en}}}{T_{\text{en}}} \right)^{0.3}. \quad (26)$$

For forced convection ($V_{\text{air}} > 1.5$ m/s):

$$\alpha_{\text{forced}} = \frac{0.0086 (R_e^*)^{0.64}}{l_x}, \quad (27)$$

where

$$R_e^* = \sqrt{R_e^2 + \frac{G_r}{2.5}} \quad (28)$$

$$R_e = \frac{v_{\text{air}} l_x}{V_{\text{air}}} \quad (29)$$

$$G_r = \frac{g h_{ca}^3 (T_{\text{wall}} - T_{\text{en}})}{T_{\text{en}} v_{\text{air}}^2}. \quad (30)$$

2.3 DCT temperature change model

In this section a compact model of the lubricant and transmission case developed utilising simplified lumped inertias. This is chosen over a more detailed geometric heat transfer model, particularly for the transmission case, as the complex geometry of the case with a large number of flanges and bosses does not lend well to simplified geometric representation. This is similar to the modelling strategy provided in Cao et al. (2004), where a thermal model is used as part of a wider investigation of friction estimation in clutches. This method allows for a computationally efficiency analysis of the temperature change, but can and will exclude the benefits of a more complex analysis. Particularly, the identification of any localised heating phenomena that may be identified through other more complex heat transfer methods. For example, Konstantinidis et al. (1997) studies the transient behaviour of exhaust systems using dynamic thermal models with and without the use of flanges and demonstrates a minor but noticeable difference in simulated results.

In the DCT, all of the power losses P_L caused by friction and viscous shear will be changed into heat. From equation (1), the overall quantity of DCT heat Q_{all} can be obtained as:

$$Q_{all} = \int P_L dt. \quad (31)$$

The generated heat is partially absorbed by oil Q_{oil} and DCT structure Q_{DCT} , and partially dissipated by the transmission case Q_{case} ,

$$Q_{all} = Q_{Ca} + Q_{oil} + Q_{DCT}. \quad (32)$$

The heat absorbed by the lubricant oil and the DCT can be obtained using specific heat equation, respectively.

$$Q_{oil} = c_{oil} m_{oil} \Delta T \quad (33)$$

$$Q_{DCT} = c_{DCT} m_{DCT} \Delta T \quad (34)$$

$$Q_{all} = Q_{Ca} + c_{oil} m_{oil} \Delta T + c_{DCT} m_{DCT} \Delta T, \quad (35)$$

where c_{oil} and c_{DCT} are the specific heat of the oil and average transmission respectively, and the m_{oil} and m_{DCT} are the mass of oil and DCT. ΔT is the increased temperature caused by absorbing heat.

Combining equations (31)–(34), the increased temperature can be achieved; therefore the rise of the temperature can be expressed as:

$$T_{oil} = T_0 + \Delta T. \quad (36)$$

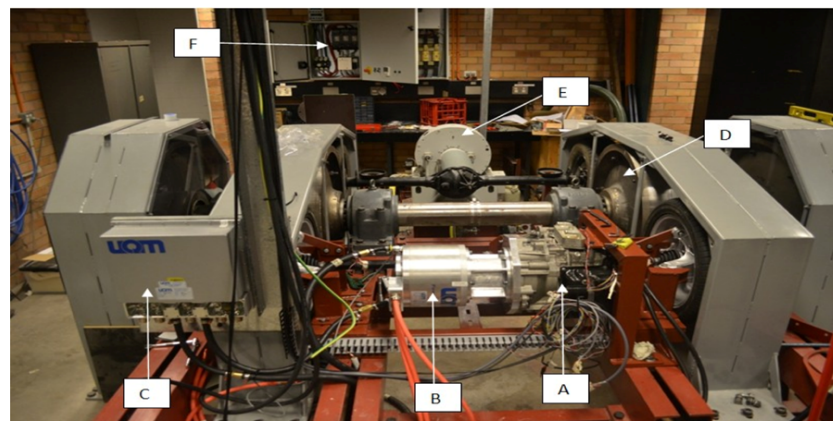
3 Experimental apparatus

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

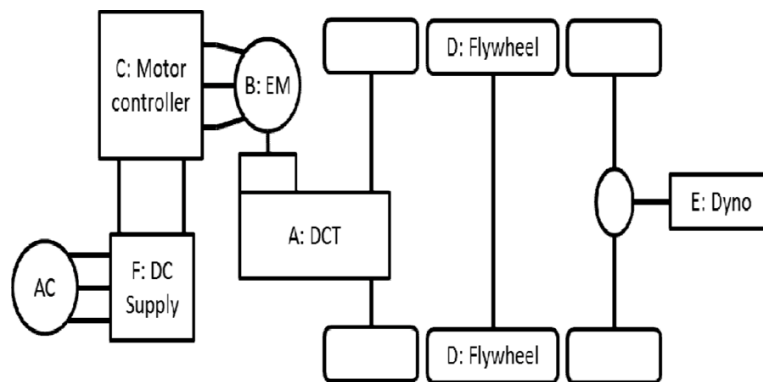
3.1 Test facility and hardware

The test facilities used for this study are shown from Figures 3–5 with several changes to the original UTS powertrain system test rig. The new test rig is based on an electric vehicle power train system, with a 330 V DC drive permanent magnet motor which is powered from AC high voltage electric power. The motor drives the wheels through the modified two-speed DCT, and the transmission parameters are shown in Table 1. A group of flywheels and tyres are used to simulate the vehicle inertia. The dynamometer, HPA engine stand 203, as shown in Figure 3, is used to apply load on the flywheels. This part of the test rig consists of two wheels which make contact with flywheels, and another final drive which is connected to the shaft before the dynamometer. The DCT has a separate supply pump. The lubricate fluid will be instead with DCTF rather ATF is shown in Table 2. And the vehicle control unit (VCU) and transmission control unit (TCU) are developed using the dSPACE Micro-Auto Box, as shown in Figure 5.

Figure 3 Modified test rig of two-speed DCT powertrain: (a) laboratory layout and (b) schematic (see online version for colours)



(a)



(b)

(A) DCT; (B) electric motor; (C) motor controller; (D) groups of flywheels;
(E) dynamometer and (F) High voltage power supply

Figure 4 Dual clutch transmission, original one (a), (b), and modified one (c) (see online version for colours)

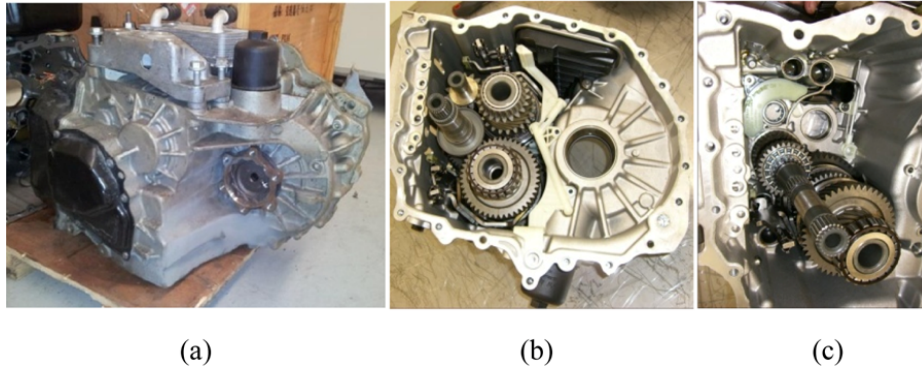


Figure 5 Schematic UTS EV powertrain system test rig and test procedure (see online version for colours)

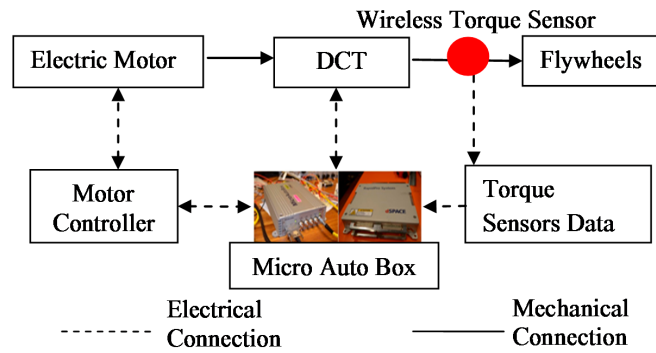


Table 1 DCT clutch plates geometric parameters

Clutch 1 plates		Clutch 2 plates		Clutch clearance	Plates number per clutch
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 Dual clutch transmission fluid (DCTF) properties and test conditions

Flow rate (m ³ /s)	Oil density (kg/m ³)	Oil viscosity (Ns/m ²)	Temperature (°C)
0.00005	853.4	0.0290	40
0.00005	835	0.0167	60
0.00005	820	0.0082	80
0.00005	810	0.0045	100

3.2 Instrumentation and data acquisition

The test instrumentation used in this study only requires measurement of the input and output speed and torque of the DCT respectively, and the DCT inside oil case

temperature. Two wireless torque sensors shown in Figure 5 are installed in the front half drive shafts, which are used to collect the output torque data of the DCT. The torque sensors are calibrated before testing and the DCT input torque is equal to the output torque of the electric motor, which can be collected by the motor feedback torque. This data was collected at 1000 samples per second throughout the test via the dSPACE control desk recorder, and then processed in a computer.

3.3 Test operation

In order to also reach the two-speed DCT maximum speed design requirements, during the temperature durability test, the vehicle speed should be able to reach over 70 km/h when running in 1st gear. And for the second speed, the vehicle speed should be more than 120 km/h. The final temperature results should be obtained from stable status, that is, the temperature will not continue to increase while the vehicle continues to run.

The procedures for the test can be described as:

- Check and monitor the primary temperature of the lubricative, environment.
- Apply load and set the expected top-speed. All the data are recorded during the process.
- After the temperature is stable, keep it run for another half hour.

4 Results and analysis

After implementing the developed drag torque and thermal behaviour model via simulation, and conducting the experimental test, results from the test and simulations will be presented and analysed in this section.

In order to investigate the effectiveness of the proposed thermal behaviour model, experimental tests are made on the UTS electric vehicle powertrain system test rig. For these tests, the motor speed is ramped up to the target speed using a constant throttle, and the throttle setting is then adjusted to maintain constant motor speed as the lubricant heats up and drag torque losses decrease.

Figure 6(a) and (b) present the experimental and simulation test results from the vehicle running in 1st gear and 2nd gear, with the overall gear ratio in 8.45 and 5.36, respectively. When the speed of the vehicle is over 70 km/h or 120 km/h, the corresponding DCT input speed from the output speed of electric motor is higher than 5000 rpm and 5500 rpm, respectively. In order to maintain the constant vehicle speed during running with the expected maximum speed, input torque of DCT is set as 23 Nm and 40 Nm, respectively, by operating the electric motor controller. From Figure 6(a) and (b), it can be seen that the simulation results agree reasonably well with corresponding experimental test results. The temperature of DCT rises with the increasing speed, first dramatically increasing, but gradually increasing as the lubricant temperature reaches steady state conditions. After two hours, both of the DCT's temperatures achieve steady state and stop increasing. The final stable temperature for first gear is 71.8°C, and 2nd gear in 104.5°C. And the DCT input speeds are 5276 rpm and 5690 rpm, respectively, and corresponding vehicle speeds are 73.5 km/h and 125 km/h respectively. From these

two groups of test, the stable temperature with certain input torque can be obtained, as well as the power losses, which will be shown in Figure 7.

Figure 6 Test and simulation results for thermal rise: (a) 1st gear and (b) 2nd gear (see online version for colours)

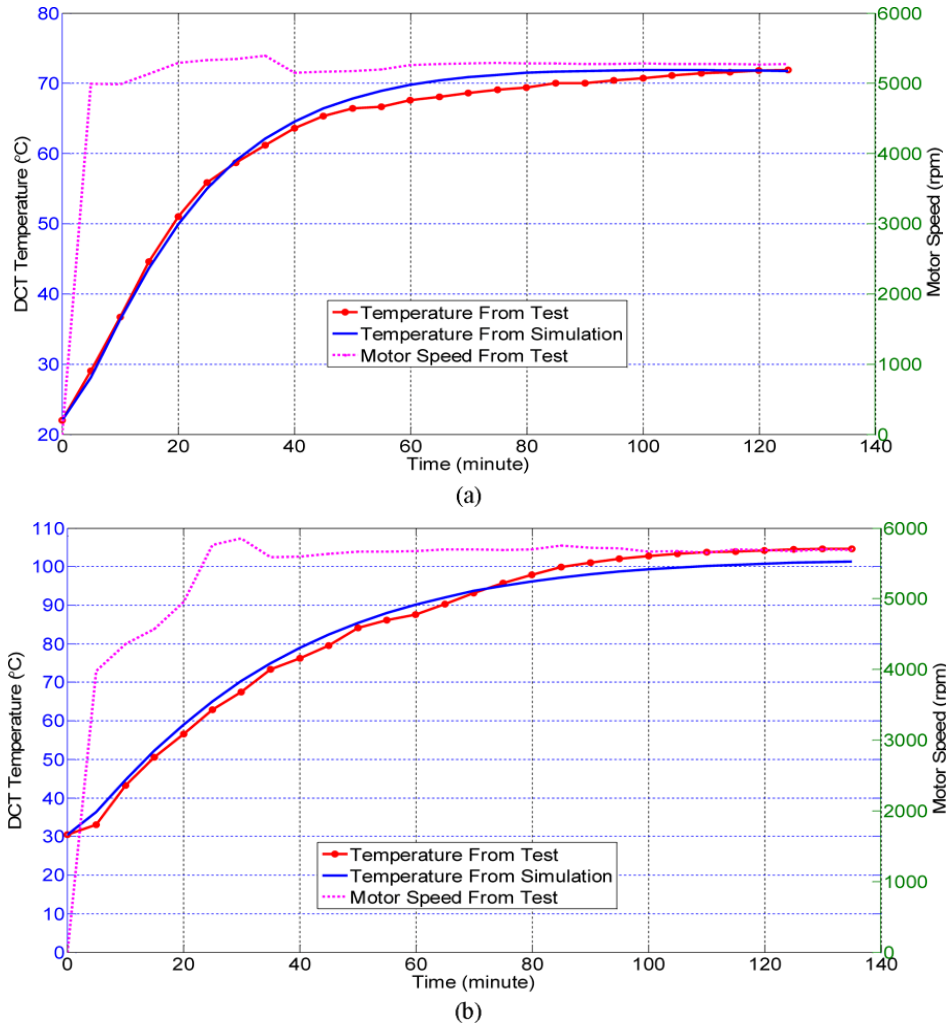
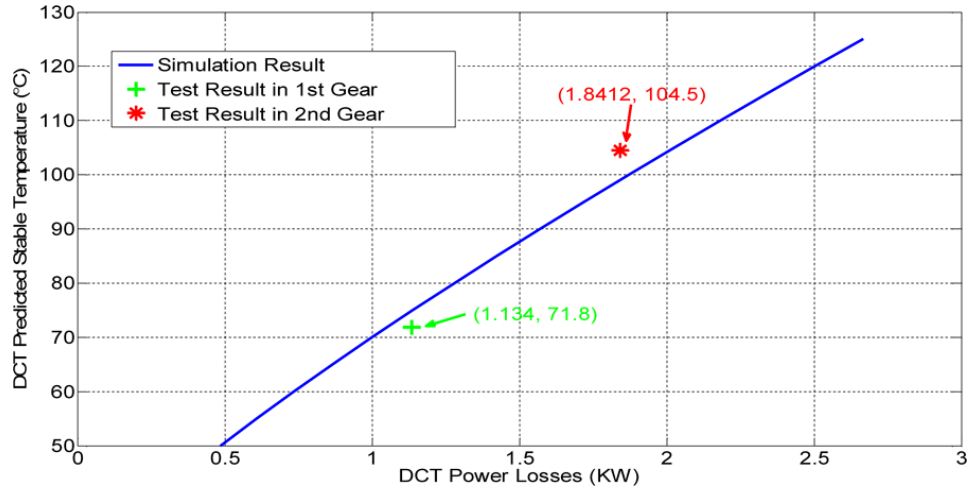


Figure 7 shows the simulation results by implementing the DCT thermal behaviour model. It presents the relationship of corresponding power losses leading to the DCT temperature in steady state rise from 50°C to 125°C. It indicates that the DCT steady state temperature increases linearly with the increased power losses. Additionally, experimental test results of DCT in two gears are also shown in Figure 7. The errors, less than 5°C, between the thermal prediction simulation results and the experimental test results, are within the predicted accuracy, around 10°C, shown in the British Standard (British Standards Institute, 2001). Therefore, the difference between the test results and thermal behaviour model is acceptable and the effectiveness of the thermal behaviour model is demonstrated by the experimental test.

Figure 7 Relationship between power losses and predicted stable temperature for DCT, and test results (see online version for colours)



Simulation results from implementing drag torque model are shown in Figure 8. The blue dash line shows the relationship of DCT overall power losses increasing with input speed rising when the vehicle runs in 1st gear, and the red solid line presents the results when the vehicle runs in the 2nd gear. The power losses obtained from drag torque simulation results and experimental test are nearly equal, so the estimated temperature and power losses by the thermal model able to accurately predict steady state temperature achieved. Moreover, when the input speed is low, less than 2500 rpm, the differences of power losses between two gears are minor. While in high speed, over 4000 rpm, the differences are becoming large with increasing speed. And at high speed, the DCT power losses in 2nd gear is larger than that in 1st gear, which demonstrates why the final steady temperature in 2nd gear is higher than that of in 1st gear shown in Figure 6. The discontinuity for both gears between 2000 RPM and 3000 RPM is a direct result from the lubricant failing to completely immerse wet clutches at higher speed. This is further discussed in Zhou et al. (2014).

Through the means of combining Figures 7 and 8, different stable temperatures can be estimated at different motor speeds, such as 2000 rpm, 3000 rpm, etc. Simulation results for DCT thermal rise characteristics are shown in Figure 9 with different input speed as well. The environment temperature of both gears is set at 25°C. It can be seen that the higher the input speed, the acceleration of thermal rise is higher. And the final stable temperature is influenced by the speed.

Thermal rise behaviour at different environment temperatures from 0°C to 40°C is simulated, with simulation results shown in Figure 10. The solid lines are for 1st gear, and dotted lines are for 2nd gear. From Figure 10, it can be summarised that the final stable temperature of DCT is directly influenced by the environment temperature. This is an important consideration for the use of such transmissions in different operating environments. In particular, there will be increased drag load on the transmission in cooler.

Figure 8 Simulation results from drag torque model, 1st gear with input torque 23 Nm, and 2nd gear with input torque 40 N.m (see online version for colours)

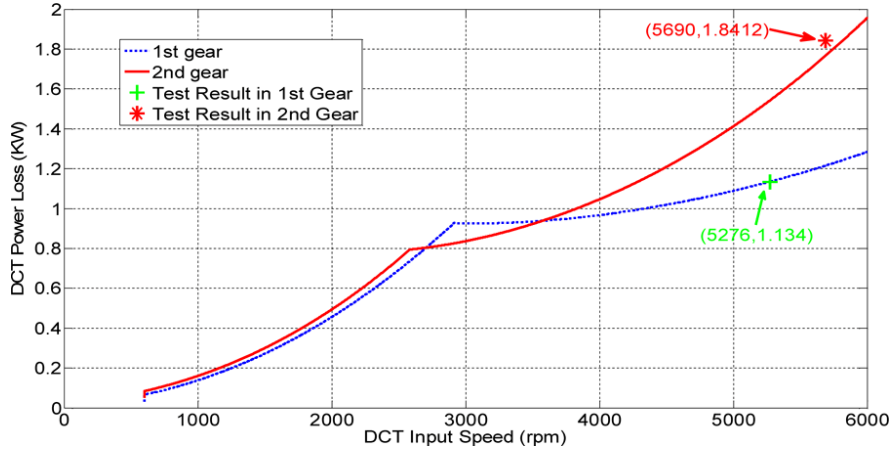


Figure 9 DCT thermal rise at different input speed: (a) 1st gear and (b) 2nd gear (see online version for colours)

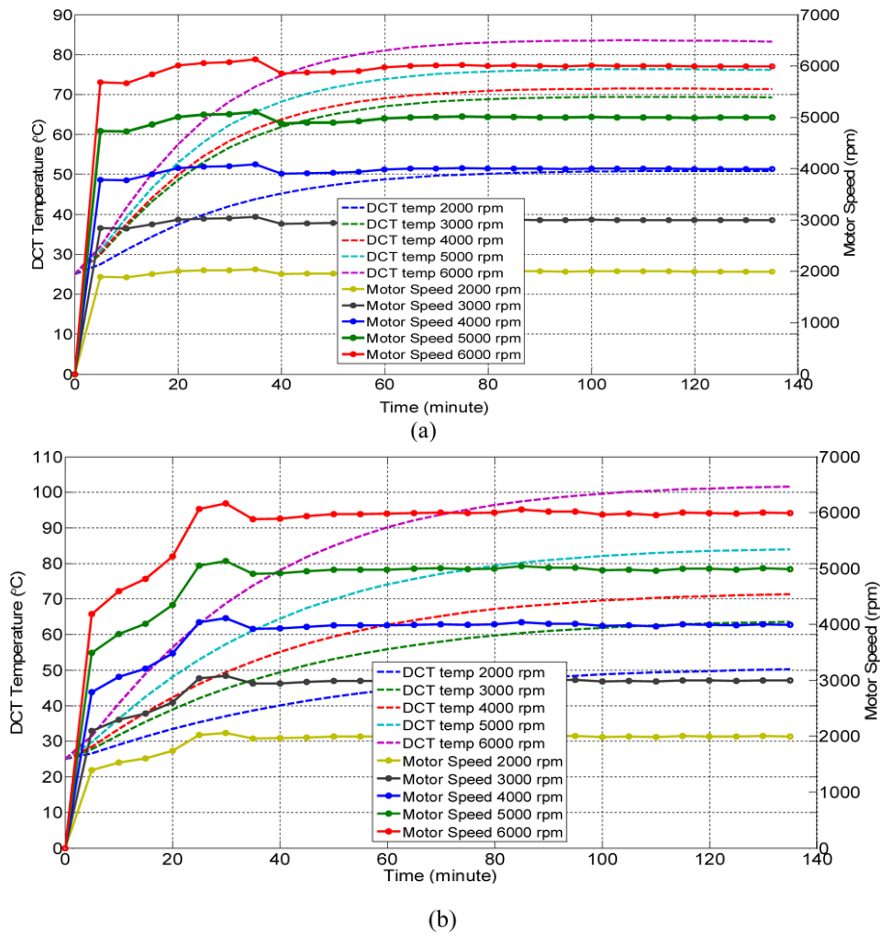


Figure 10 Simulations on DCT thermal rise behaviours at different environment temperatures (see online version for colours)

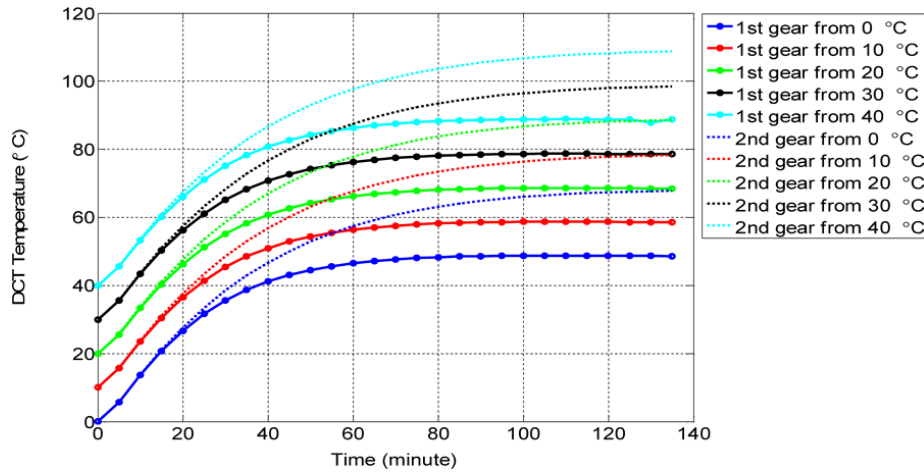
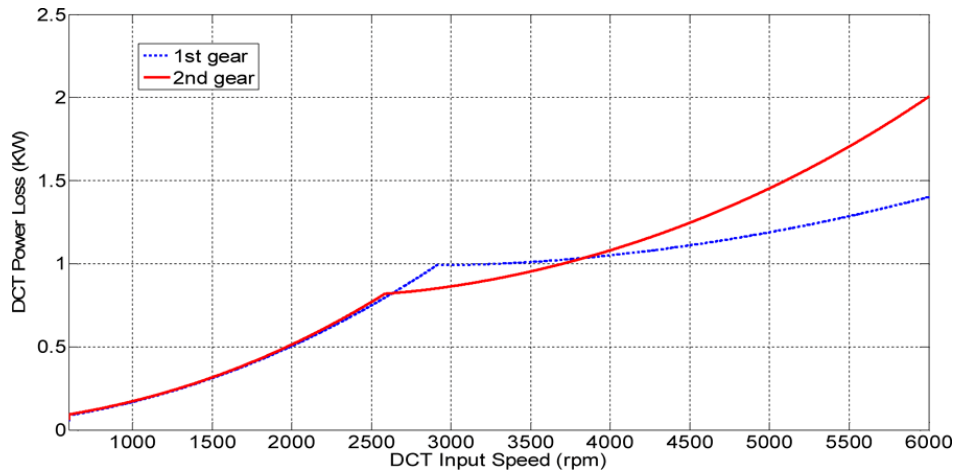


Figure 11 shows the simulation results on power losses from implemented drag torque model with constant input torques. Here, constant input torque 50 Nm for the DCT is set for both first and second gear. From Figure 10, it can be concluded that in low speed, i.e., less than 2600 rpm, the power losses in two gears are similar, while above 4000 rpm these losses are different. Furthermore, when the DCT input speed changes from 600 rpm to 6000 rpm, the DCT power loss increases by over 1.2 kW in each case.

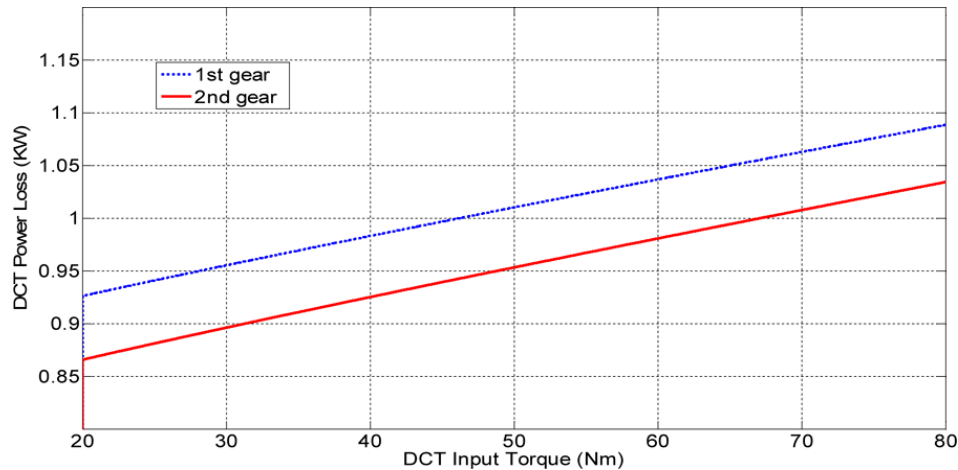
Figure 11 Relationship between input speed and power losses (1st and 2nd gear, 50 Nm) (see online version for colours)



With constant input speed of 3500 rpm and different input torque, the simulation results on DCT power loss are shown in Figure 12. It shows that the power loss in first gear is higher than in second gear by approximate 0.07 kW. Both lines are parallel with each other indicating that the power losses increase linearly with the input torque rising.

Besides, when the DCT input torque changes from 20 Nm to 80 Nm, the DCT power loss changes no more than 0.15 kW in either line.

Figure 12 Relationship between input torque and power losses (1st and 2nd gear, 3500 rpm)
(see online version for colours)



Via comparing Figures 11 and 12, it can be summarised that the impact of speed on DCT power loss is larger than that of torque, i.e., load. As all of the power losses will be converted into heat, and dissipated, it can also be concluded that the impact of speed on the thermal behaviour is higher than that by the load.

5 Conclusions

The purpose of this paper is to develop a mathematical model of the thermal behaviour of a prototype DCT. This is achieved through the integration of mathematical models of gear, bearing and clutch losses for the transmission. These losses are defined as the sources of energy generation to heat the lubricating oil. A thermal model of the lubricant and transmission case was also developed to evaluate transmission heat rejection, thus achieving a balanced heating/cooling model of the transmission. Experimental investigation of the transmission thermal behaviour is conducted to validate the effectiveness of the combined drag torque and thermal prediction models, with accurate steady state results achieved. The development of such models can enable the rapid analysis of thermal effects in the DCT case to help improve reliability in initial design and accelerate product development at reduced costs.

Results from simulation and testing of transmission drag torque and thermal behaviour resulted in the attainment of the following conclusions:

- This work reveals that the level of load applied to the transmission has a minor effect in comparison to the speed dependent operation.
- With only a limited number of gear changes required by the transmission, heat rejection through the casing only is sufficient.

- The impact of gear churning and windage losses become more and more significant with the increasing speed.
- Wet clutch drag loss is the most significant part of the DCT power losses. When designing the two-speed DCT, it is necessary to pay special attention to the thermal reliability to the clutch pack and the hydraulic actuators to avoid over-heating.
- The final stable temperature of DCT is significantly influenced by the environment temperature.

Future research includes a more extensive comparison of power losses under a wider range of speeds and load, expanding its application to transient heating models for driving cycle simulations, and applying this thermal behaviour model into powertrain system optimisation. The limitation of this study is that it only considers the steady state status of the thermal behaviours without considering the transient performance of the power losses and the thermal behaviour during shifting.

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Nomenclature

A_{ca}	Surface area of transmission case (m ²)	0.86
A_{oil}	Oil-side surface area of transmission case (m ²)	0.85
c_{oil}	Specific heat of oil (J/kg, K)	2090
c_{DCT}	The average specific heat of transmission (J/kg, Kelvin)	880
h_{ca}	Height of gearbox housing (m)	0.34
G_r	Grashoff number	–
K	Heat transmission coefficient (W/m ² Kelvin)	–
m_{oil}	Mass of oil (kg)	5.1
P_B	Power losses caused by bearings drag torque (kW)	–
P_{con}	Power losses caused by concentric shaft drag torque (kW)	–
P_{Ch}	Power losses caused by gear churning (kW)	–
P_{Cl}	Power losses caused by wet clutch plates drag torque (kW)	–
P_G	Power losses caused by gear meshing drag torque (kW)	–
P_L	Total power losses in DCT (kW)	–
Q_{ca}	Quantity of heat dissipated by the transmission case (J)	–
Q_{all}	Overall quantity of DCT heat (J)	–
Q_{oil}	Quantity of heat absorbed by oil (J)	–
Q_{DCT}	Quantity of heat absorbed by DCT structure (J)	–
Re	Reynold's number	–
T_{oil}	Oil temperature (Kelvin)	–
T_{en}	Environment temperature (Kelvin)	298
T_0	The primary temperature (Kelvin)	–
T_{wall}	Temperature of housing wall (Kelvin)	–
α_{oil}	Oil-side heat transfer coefficient (W/m ² Kelvin)	50
α_{ca}	Air-side heat conduction (W/m ² K)	200
α_{con}	Convection part(W/m ² K)	–
α_{rad}	Radiation part (W/m ² K)	–
α_{free}	Heat transfer coefficient due to free convection (W/m ² K)	–
α_{forced}	Heat transfer coefficient due to forced convection (W/m ² K)	–
λ_{wall}	Coefficient of thermal conduction of housing (W/m, K)	160
δ_{wall}	Mean housing wall thickness (m)	0.012
ε	Emission ratio	0.05
η^*	Temperature ratio	–
V_{air}	Impingement velocity (m/s)	1
ΔT	Increased temperature caused by absorbing heat (Kelvin)	–