APCOM & ISCM
11-14th December, 2013, Singapore

Simulation of Thermal behavior of a Two-speed Dual Clutch Transmssion

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Abstract

In order to study the thermal behaviour of a two-speed dual clutch transmission (DCT), mathematic calculation and simulation will be conducted. This paper presents a theoretical analysis of power losses and heat transfer in DCT. The power losses components include wet clutches, concentric shaft, power losses caused by gear meshing, gear windage, churning, and bearings. In order to demonstrate the effectiveness of the model, simulations are conducted based on the presented theoretical analysis and developed powertrain model using different vehicle test driving cycles. Thermal behaviour study can contribute to the optimization design of future transmissions and calculating its reliability.

Keywords: Thermal Capacity, heat dissipation, dual clutch transmission, DCT, Simulation.

Introduction

Pure EVs being currently widely used in the market are mainly equipped with single speed transmission, with tradeoffs between dynamic (such as climbing ability, top speed, and acceleration) and economic performance (driving 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 multi-speed transmissions for electric vehicles is likely to improve average motor efficiency and range capacity, or even can reduce the required motor size. There are a number of multi-speed transmissions available for pure EVs, as shown by the author (Rudolph, 2007), indicating that DCTs have higher fuel efficiency than other automatic drives, making them extremely suitable.

It is necessary to point out that the vehicle performance improvement by using multi-gear transmissions for an electric vehicle is not as much as for an engine driven vehicle, as a result of significantly different characteristics of these two systems. In an engine driven system, the available output speed range of engine is narrower than that of electric motor. The inertia of motor is smaller than that of engine as well, usually smaller than a quarter of engine. And the motor speed is also more controllable than that of engine. Therefore, the number of gear ratios for a pure EV is not the more the better, as it will increase the transmission manufacture cost and the overall vehicle mass without contributing significantly to the overall performance of the vehicle. Consequently, the detailed performance difference between one-speed and two-speed EV can refer from our research work by the author (Zhou, 2012)**.**

A general two-speed wet DCT is suggested to be equipped into a pure EV, as shown in Figure 1. It is made up of two clutches, C1 and C2. The two clutches have a common drum attached to the same input shaft from the motor, and the friction plates are independently connected to 1st or second gear. C1 connects the inner input shaft engaged with 2nd gear, and C2 connects the outer input shaft, engaged with 1st gear. And there are no synchronisers in this two-speed DCT. Thus, the transmission can be looked at as two half manual transmission, and, in this sense, shifting is realised through the simultaneous shifting between these two half transmissions. For this special layout, it is the reason why that the author (Goetz, 2005) points out that vehicle equipped with DCT can change speed smoothly with nearly no power.

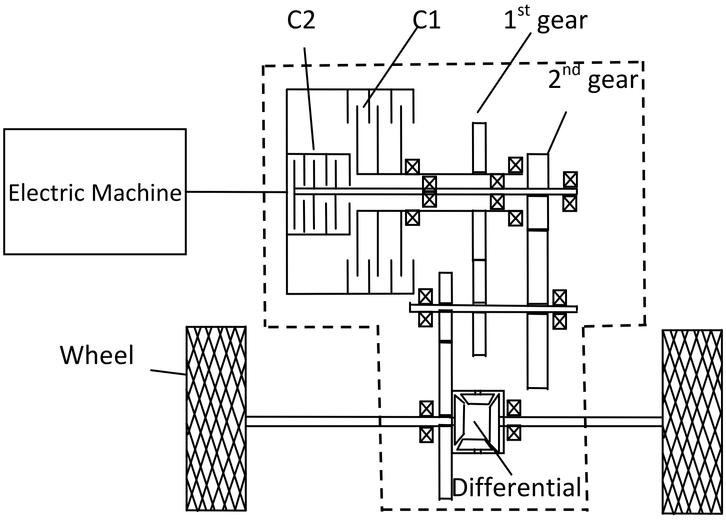

Figure 1. EV powertrain system equipped with two-speed wet DCT schematic

Computer calculations and simulations are now part of design new types of transmissions process, as prototype tests have become more and more expensive and time consuming as shown in the work by the author (Play, 1978). Thermal behaviour is not often considered in the preliminary design step as pointed by the author (Lechner, 1999). In fact, a thermal expansion of transmission cases, for example, can change gear axes geometry positions, gears clearances, lubricant types and film conditions, and consequently, dissipated heat during work. Furthermore, it is of great importance to know the temperature of oil lubricated transmission systems, and the quantity of demand cooling oil. The prediction of thermal behaviour of a transmission might be beneficial to evaluate cooling and lubricating conditions. Therefore, it appears necessary to develop numerical models to predict the transmissions thermal behaviours.

Some works (Changenet et al., 1996, Coe, 1989; Joule et al., 1988; Phillips, 1996) have been done considering the whole transmission system for thermal analysis. However, these approaches are mainly focus on manual or automatic transmission. There are limited, if any, publicised works and reports on dual clutch transmission thermal behaviours.

This paper performs a research of dual clutch transmission power losses and heat dissipation. Due to the particular structure of DCT, the power losses components include wet clutches, concentric shaft, power losses caused by gear meshing, gear windage, churning, and bearings. In order to demonstrate the effectiveness of the model, simulations are conducted based on the presented theoretical analysis and developed heat dissipation model using different assumed temperature and power losses. Thermal capacity study can contribute to the design of future transmissions and calculating its reliability.

1 Theoretical Analysis of Heat Dissipation of Dual Clutch Transmission

Given the rigorous development required for standards, the BS/ISO (2001) model is adopted to analysis the transmission heat dissipation.

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

$$Q_{Ca} = kA_{ca}(T_{oil} - T_{en}) \tag{1}$$

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 unit of Kelvin.

$$\frac{1}{k} = \frac{1}{\alpha_{oil}}\frac{A_{ca}}{A_{oil}} + \frac{\delta_{wall}}{\lambda_{wall}}\frac{A_{ca}}{A_{oil}} + \frac{1}{\alpha_{ca}} \tag{2}$$

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 neglected. For high air velocities and thus good external heat transfer, it will probably be necessary to also consider of the oil-side heat transfer. As a reference value, oil-side heat transfer coefficient, $\alpha_{oil} = 200\,\mathrm{W/m^2K}$, can be assumed. But it requires for 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 sound insulation and non-metallic cases. And the appropriate coefficient of thermal conduction, λ_{wall}, has to be expressed for the case material in question.

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

$$\alpha_{ca} = \alpha_{con} + \alpha_{rad} \tag{3}$$

$$\alpha_{rad} = 0.23 * 10^{-6}\varepsilon\left(\frac{T_{wall}}{T_{en}}\right)^3 \tag{4}$$

where the emission ratio, ε, is assumed as 0.15.

The convection part can be divided into two parts, free and forced convection. According to the investigations by the author (Funck, 1985), the following can be presented:

$$\alpha_{con} = \alpha_{free}(1 - \frac{A_{air}}{A_{ca}}) + \alpha_{forced}\frac{A_{air}}{A_{ca}}\eta^* \tag{5}$$

Where

$$\eta^* = \frac{T_{wall} - T_{air}}{T_{wall} - T_{en}} \tag{6}$$

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

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

$$\alpha_{free} = 18h_{ca}^{-0.1}\left(\frac{T_{wall} - T_{en}}{T_{en}}\right)^{0.3} \tag{7}$$

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

$$\alpha_{forced} = \frac{0.0086(R_e{}^*)^{0.64}}{l_x} \tag{8}$$

Where

$$R_e{}^* = \sqrt{R_e{}^2 + \frac{G_r}{2.5}} \tag{9}$$

$$R_e = \frac{v_{air} l_x}{V_{air}} \tag{10}$$

$$G_r = \frac{g h_{ca}^3 (T_{wall} - T_{en})}{T_{en} v_{air}^2} \tag{11}$$

2 Simulation Results and Discussion

The simulations present the relationship between DCT power losses and predicated temperature.

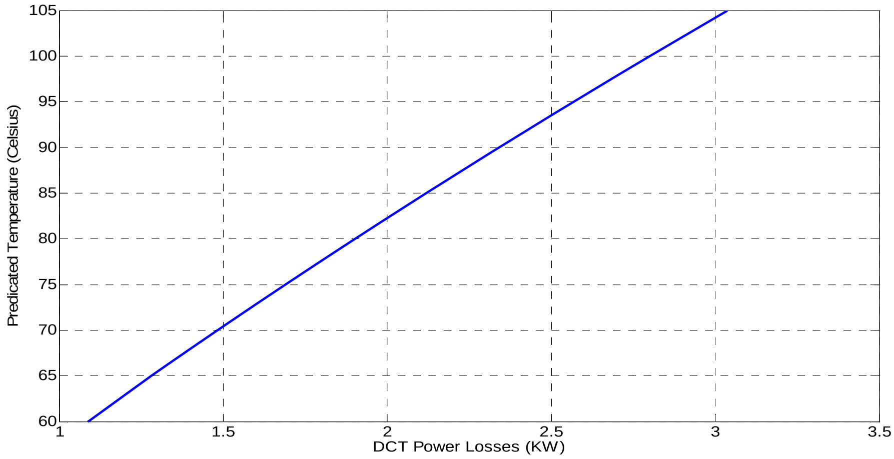

Figure 2. Relationship between DCT power losses and predicated temperature

Figure 3. Relationship between input power and power losses

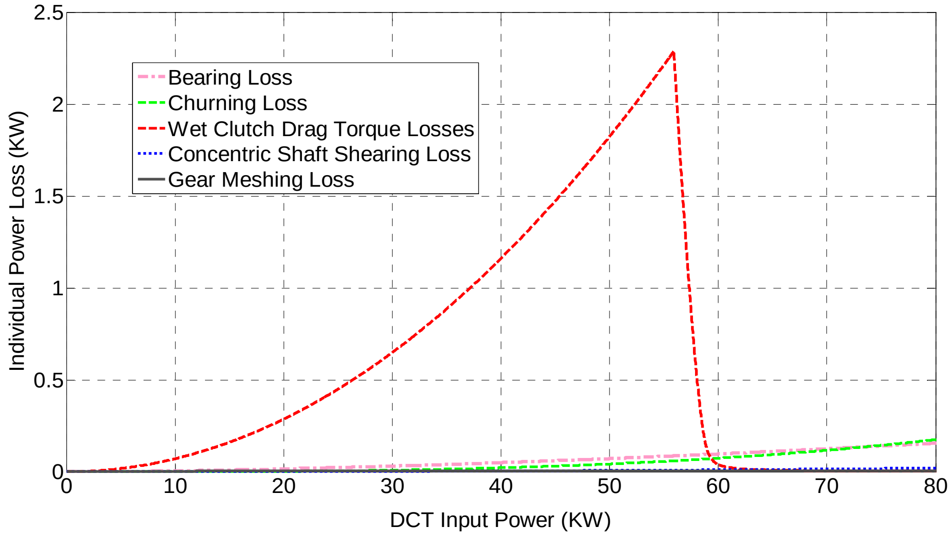

Figure 4. Individual DCT power losses verse input power

Figure 2 shows that the predicated temperature linearly increases with the DCT power losses. Figure 3 indicates that the maximum power loss for DCT is approximate 2.4KW. Consider Figure 3 and Figure 4 together, the power loss decreased sharply is caused by the wet clutch. It is because that the effective wet clutch radius will decrease which is caused by increased centrifugal force during high speed. Figure 4 also shows that wet clutch drag torque loss domain the main power losses in the two-speed DCT. From Figure 2 and Figure 4, as the power losses is domain caused by wet clutch drag torque, it is not only necessary to consider the DCT thermal capacity but carefully to calculate the wet clutch pack thermal capacity as well.

Conclusions

Thermal behaviour study can contribute to the design of future transmission prototypes and calculating its reliability avoiding unnecessary failures. It could help accelerating products development speed and save funds. Future research includes comparing power losses under different gears and operations, and using experiment to investigation the proposed model.

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