연구논문

Journal of Drive and Control, Vol.12 No.1 pp.1-8 Mar. 2015
ISSN(print) 2234-8328 ISSN(online) 2287-6146
http://dx.doi.org/10.7839/ksfc.2015.12.1.001

A Study on Thermal Analytical Model for a Dry Dual Clutch
건식 듀얼 클러치의 열해석 모델에 대한 연구

Hao Liu[1], J. C. Lee[1*], Y. J. Noh[1], J. H. Cho[2], H. R. Lee[2], J. E. Koh[2] and J. W. Kang[2]
류하오 · 이재천 · 노유정 · 조종환 · 이희락 · 고재욱 · 강지우

Received: 27 Nov. 2014, Revised: 12 Jan. 2015, Accepted: 9 Feb. 2015

Key Words : Dry dual transmissions (DCT), Dry dual clutch (DCC), Thermal model, Lumped parameter, Heat convection

Abstract: The stability of friction characteristics and thermal management for a dry type dual clutch transmission (DCT) are inferior to those of a wet clutch. Too high temperature resulting from frequent engagement of DCT speeds up degradation or serious wear of the pressure plate or burning of the clutch disk lining. Even though it is significantly important to estimate the temperature of a dry double clutch (DDC) in real-time, few meaningful study of the thermal model of DDC has been known yet. This study presented a thermal analytical model of lumped parameters for a DDC by analyzing its each component firstly. Then a series of experimental test was carried out on the test bench with a patented temperature telemetry system to validate the proposed thermal model. The thermal model, whose optimal parameter values were found by optimization algorithm, was also simulated on the experimental test conditions. The simulation results of DDC temperature show consistency with the experiment, which validates the proposed thermal model of DDC.

Nomenclature

A_d : Convection area of disk
A_{hin} : Convection area of inside housing
A_{hout} : Convection area of outside housing
A_p : Convection area of pressure plate
c_a : Specific heat of air at constant pressure
c_d : Specific heat of disk
c_h : Specific heat of clutch housing
c_p : Specific heat of pressure plate
f : Cost function
h_d : Convection coefficient of disk
h_{hin} : Convection coefficient of inside housing
h_{hout} : Convection coefficient of outside housing
h_p : Convection coefficient of pressure plate
m_a : Air mass inside clutch housing
m_d : Disk mass
m_h : Clutch housing mass
m_p : Pressure plate mass
$\dot{Q}_{cd}$: Convection heat of disk in unit time
$\dot{Q}_{cp}$: Convection heat of pressure plate in unit time
$\dot{Q}_f$: Friction heat in unit time
T_a : Air temperature inside clutch
T_d : Disk temperature

* Corresponding author email: ljcds@kmu.ac.kr
1 Department of Mechanical and Automotive, Engineering, Keimyung University, Daegu 704-701, South Korea
2 Technical Research Center, Valeo Pyeong Hwa Co., Daegu 704-190, South Korea

T_h : Clutch housing temperature

T_p : Pressure plate temperature

α_d : Thermal distribution coefficient of disk

α_p : Thermal distribution coefficient of pressure plate

λ_d : Thermal conductivity of disk

λ_p : Thermal conductivity of pressure plate

ρ_d : Density of disk

ρ_p : Density of pressure plate

ω_{in} : Angular velocity of input shaft from engine

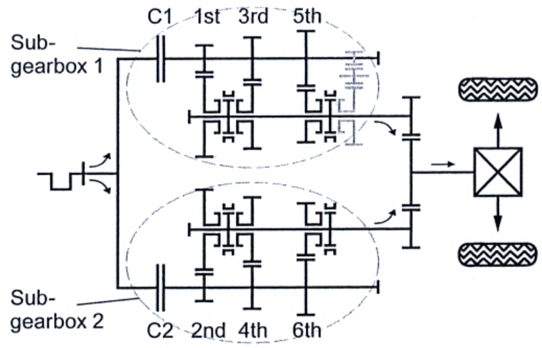
ω_{out1} : Angular speed of output shaft 1

ω_{out2} : Angular speed of output shaft 2

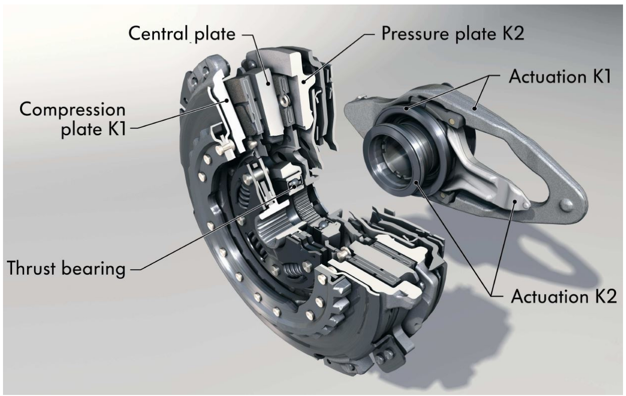
1. Introduction

Dual clutch transmissions (DCT) combine advantages of both manual transmissions (MT) and automatic transmissions (AT), which are easy mechanical structure with high efficiency and easy control convenience for drivers[1]. Since Volkswagen launched the first DCT with a dry dual clutch (DQ250) applied on some light class passenger vehicles in the beginning of 2008, a large number of drivers and automotive experts have experience with this technology[2]. Since then other automotive manufacturers have also been competitively developing DCT for their passenger vehicles[3,4].

The principle of DCT is based on the idea of two independent sub-gearboxes each connected to the engine via its own clutch as illustrated in Figure 1(a)[1]. One sub-gear contains the odd gears (1, 3, 5...) and the other does the even gears (2, 4, 6...). By dividing the gears through the dual clutch, the DCT becomes fully power shiftable. The DCT utilizes actuators to realize the function of gear shift between the odd gear set and the even gear set. The first clutch and the second clutch cooperate together to implement the smooth gear shift without power interruption. Figure 1(b) shows a commercial dual clutch of Volkswagen.



(a) DCT



(b) Dual clutch

Fig. 1 Schematic of DCT and view of dual clutch (Volkswagen DSG)

In general, friction clutches of DCT are classified into two types: dry friction clutches and wet friction clutches. Dry clutches are characterized by a high level of efficiency (no drag torque) as well as by a small moment of inertia, whereby the moment of inertia can increase considerably with additional demands regarding damping or automated crawling starts or multiple starts. The advantage of wet clutch is good controllability as well as a high power/weight ratio and large torque capacity. They are suited to vehicles with little installation space and high torque. In contrast to wet clutches, the stability of friction characteristics, thermal management and durability of dry clutches are generally inferior to wet multi-plate clutches and hence they have limited application to small and compact automobiles[5]. The optimal engagement of DCT requires proper effects of the friction works of DDC whose friction coefficient is a function of temperature[6].

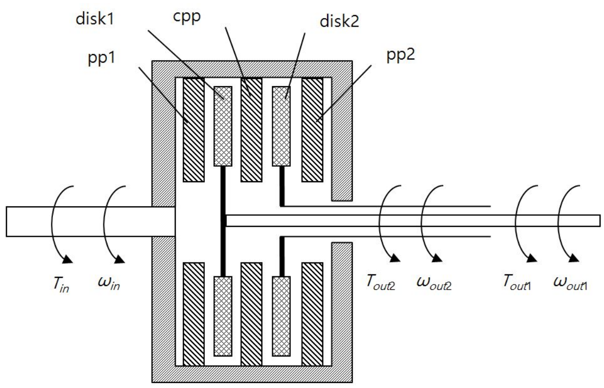
As the DDC works, the pressure plate and the disk frequently engage and disengage so that the temperature of the contact facing increases higher. Too high temperature would speed up degradation or wear of the pressure plate and burning of the disk. Hence, for the fail-safe of DCT, it is highly significant to obtain the thermal model of the DDC in order to estimate the temperature of the pressure plate and the disk in real-time.

Some researchers have already presented clutch thermal models used in real-time for other type of clutches and transmission[7-10]. However few meaningful study results for the thermal model of DDC were known yet. Chen, G., et al.[7,8] proposed a virtual clutch temperature sensor for Chrysler automatic transmission in which the thermal model includes the heat generation, clutch cooling by transmission oil flow, oil vaporization and heat conduction. Velardocchia, M., et al[9] presented a linear thermal model of diaphragm spring clutch for manual transmission. Kong, G. and Zaimin, Z.[10] proposed ideal clutch thermal model of single dry clutch for an automated mechanical transmission (AMT).

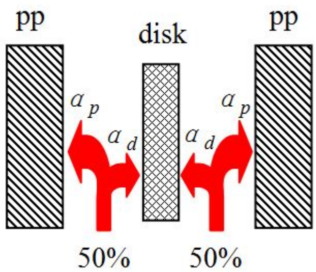
This study proposed the thermal model with lumped parameters for a DDC first, which considers the heat conduction and convection among the pressure plate, disk, clutch housing, and internal air. The thermal model with lumped parameters adopted in the research makes it possible to estimate the temperature in real-time, Since the computational load of proposed thermal model should not be too heavy to be implemented into a micro-controller for DDC.

2. Model Equations

Figure 2(a) shows the schematic of DDC. The pressure plate 1 (pp1), central pressure plate (cpp), and pressure plate 2 (pp2) all rotate synchronously with the engine output shaft. Two disks are positioned between these three pressure plates. There are two output shafts, and which shaft can transmit power depends on the operation. If the pp1 moves towards the right direction and presses the disk 1, then the input power can be transmitted to the first output shaft. Similarly, if the pp2 moves left and presses the disk 2, the power can be transmitted to the second output shaft. Clutch housing encloses the pressure plates and disks, making them as an entire DDC.



(a) DDC model



(b) Heat transfer path of DDC

Fig. 2 Schematic and heat transfer path of DDC

In order to simplify the problem in the allowable error range, the following assumptions are given:

- There is no temperature distributions for the pressure plate, disk, and housing, in other words lumped parameter method is used in the research.
- All radioactive heat transfer inside the clutch is neglected because the amount of radioactive heat is very small, comparing with that of heat conduction.
- The air inside the clutch housing has the same temperature distribution due to the pressure plate rotation with high speed.
- All slipping energy is converted to frictional

heat.

• When the clutch is engaged, the amount of heat flowing into the pressure plate is evenly absorbed by the left and right pressure plate. That is to say that the left and right pressure plate absorbs 50% of total heat, respectively, as illustrated in Figure 2(b). But the heat flow distribution to the pressure plate and disk depends on their material properties.

Based on above assumptions, the thermal model of lumped parameters for a DDC is established as follows. Frictional heat is generated because of a relative slipping speed of the input and output shaft, meanwhile the output torque is not zero. The following equation describes this process.

$$\dot{Q}_f = T_{out}(\omega_{in} - \omega_{out}) \tag{1}$$

Due to difference material properties of the pressure plate and disk, they are capable of absorbing different amount of heat. Reference [7] gave the ratio of heat absorbed by the pressure plate and disk. Thus thermal distribution coefficients of them are expressed by the following equations.

$$\alpha_d = \frac{\sqrt{\rho_d c_d \lambda_d}}{\sqrt{\rho_p c_p \lambda_p} + \sqrt{\rho_d c_d \lambda_d}} \tag{2}$$

$$\alpha_p = \frac{\sqrt{\rho_p c_p \lambda_p}}{\sqrt{\rho_p c_p \lambda_p} + \sqrt{\rho_d c_d \lambda_d}} \tag{3}$$

Frictional heat increases the temperatures of pressure plate and disk, while convective heat transfer makes them decrease. Thus, the thermal model of the pressure plate is described by

$$m_p c_p \frac{dT_p}{dt} = \frac{1}{2}\alpha_p \dot{Q}_f - \dot{Q}_{cp} \tag{4}$$

Where, convection heat transfer is denoted by

$$\dot{Q}_{cp} = h_p A_p (T_p - T_a) \tag{5}$$

Similarly, for the disk the following equations are to describe its thermal characteristics.

$$m_d c_d \frac{dT_d}{dt} = \alpha_d \dot{Q}_f - \dot{Q}_{cd} \tag{6}$$

$$\dot{Q}_{cd} = h_d A_d (T_d - T_a) \tag{7}$$

There exist internal and external heat convection for the clutch housing, as described in equation (8).

$$m_h c_h \frac{dT_h}{dt} = h_{hin} A_{hin} (T_a - T_h) - h_{hout} A_{hout} (T_h - T_s) \tag{8}$$

The variation of air temperature inside the clutch results from the heat convection of the pressure plate and disk, as well as internal convection of the housing as denoted by

$$m_a c_a \frac{dT_a}{dt} = \sum \dot{Q}_{cp} + \sum \dot{Q}_{cd} - h_{hin} A_{hin} (T_a - T_h) \tag{9}$$

As a result, equation (1) is applied for two clutches; equation (4) and equation (5) are used for pp1, cpp, and pp2; and equation (6) and equation (7) are used for disk1 and disk2, respectively.

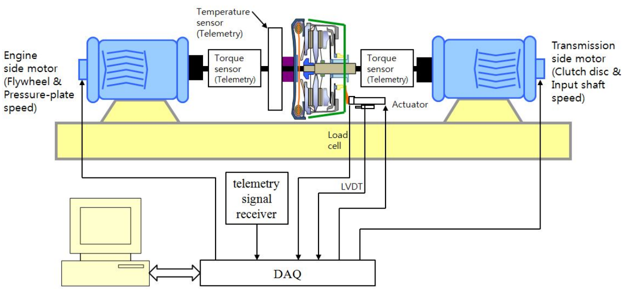
3. Model Validation

3.1 Test Apparatus and Operation Mode

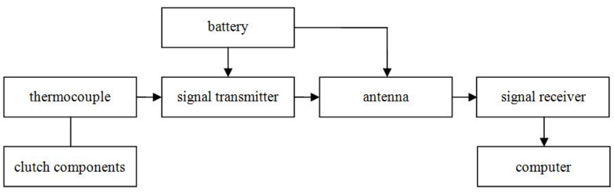
In order to validate the proposed thermal model, a series of thermal test of DDC was conducted using the product of Valeo Pyeong Hwa Co. The layout of the test bench is illustrated in Figure 3(a). The rotary speeds and output torques of two electric motors, as well as engagement and disengagement of DDC, are controlled by a computer. As a result, one electric motor drives a DDC as an engine torque input while the other imposes a torque load on it. The engine input torque, load torque, and the temperatures of pp1, cpp, and pp2 are coincidentally measured and transmitted to the computer by using telemetry device. Meanwhile the displacements and operating forces of the clutch actuators are also obtained.

The layout of the temperature telemetry system, which was patented by Valeo Pyeong Hwa Co.[11], is illustrated in Figure 3(b). And Figure 3(c) shows the view of thermocouples and telemetry in

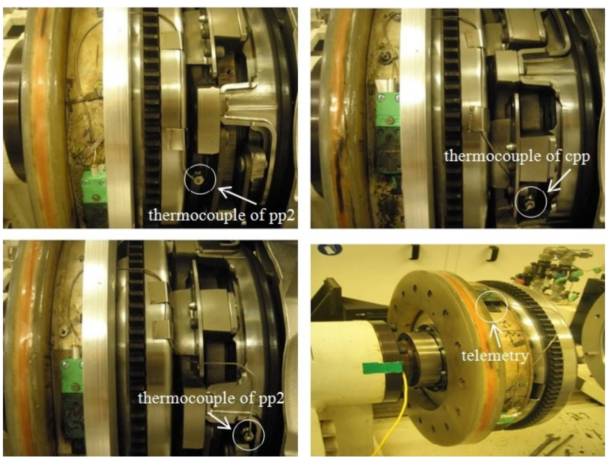
DDC test sample. Thermocouples are attached on the pp1, cpp, and pp2, respectively, and the temperature signals are emitted by transmitter whose power is provided by a battery. The thermocouples, transmitter and battery all are installed on the DDC, so they rotate with it together. The key point of thermal model validation test is how to reliably measure temperature by telemetry system, which includes following three objectives. The first is to develop the robust telemetry system for hot operating environment. By adopting new telemetry, the operating temperature condition can be increased. For example, the operating temperature of the clutch housing is increased from 60℃ to 120℃. The second is to utilize battery parallel technology to develop long life-span battery. So the device operating time is extended from 4 hours to 8 hours. The last significantly important objective is to perform telemetry system functional validation. The thermocouple signal transmitter is placed in the temperature chamber of 110℃, and the temperature reading from the signal receiver is compared with that of the temperature chamber indicator to ensure reliable communication between the transmitter and receiver. It should be mentioned that, due to the installation of the signal transmitter and battery on the DDC, the balancing weight is manufactured and installed on the DDC so as to eliminate the unbalance centrifugal force.



(a) Layout of test bench



(b) Layout of temperature telemetry system



(c) View of thermocouple and telemetry

Fig. 3 Schematic of DDC test system

According to the properties of DDC and normal driving conditions, the following 4 different operation modes were implemented. Cooling process was also accomplished after each operation.

- Mode1: vehicle launch (M1): when the vehicle starts off, only the 1st clutch is engaged.
- Mode2: vehicle reverse (M2): when the vehicle reverses, only the 2nd clutch is engaged.
- Mode3: gear shifting (M3): gear shift is operated, where slip of each clutch occurs in very short time (2 sec).
- Mode4: creep (M4): two clutches are alternatively engaged; meanwhile slip occurs in long time (30 sec).

Table 1 An example of test mode: vehicle launch

mode	rpm	rpm	clutch torque	slip time	interval
M1-1	1000	500	C1 24Nm /C2 OPEN	1 sec	5 sec
M1-2			C1 54Nm /C2 OPEN		
M1-3			C1 72Nm /C2 OPEN		
M1-4			C1 98Nm /C2 OPEN		
M1-5			C1 120Nm /C2 OPEN		
M1-6	2000	1000	C1 24Nm /C2 OPEN	2 sec	
M1-7			C1 54Nm /C2 OPEN		
M1-8			C1 72Nm /C2 OPEN		
M1-9			C1 98Nm /C2 OPEN		
M1-10			C1 120Nm /C2 OPEN		
M1-11	3000	1500	C1 24Nm /C2 OPEN	3 sec	
M1-12			C1 54Nm /C2 OPEN		
M1-13			C1 72Nm /C2 OPEN		
M1-14			C1 98Nm /C2 OPEN		
M1-15			C1 120Nm /C2 OPEN		

Note: repeat M1-(1~15) until pressure plate temperature MAX 350℃or 1hour

In each operation mode, the angular speeds of engine and transmission were set as several fixed values, and the transferred torque of clutch 1 or clutch 2 were also adjusted from small to large values. The torque was acted in the slip time and there was an interval between two slip periods. All operation was able to be set in the computer shown in Figure 3(a). Table 1 gives the detail test values of the operation mode of vehicle launch and Figure4 illustrates a test process from M1-5 to M1-6.

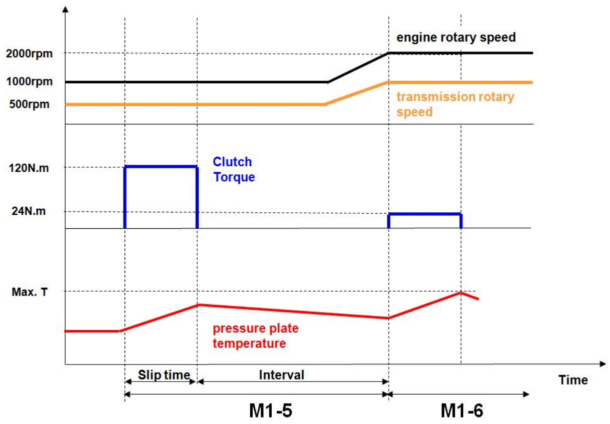


Fig. 4 A test process from M1-5 to M1-6

3.2 Numerical Estimation of Temperature

Based on proposed thermal model of DDC in the section 2, a Matlab/Simulink simulation model, shown in Figure 5, was constructed to estimate the temperature of DDC. The data of experiment, such as the rotary speeds of the input/output shafts and the transmitted torque of the output shaft, were imported into the Simulink model.

Most important problem in simulation was to determine uncertain parameters, such as heat convection coefficients of the disk, pressure plate, and housing. The values of them in the simulation were able to be optimized using the experimental results. That is to say, the optimization process was performed through co-simulation of Simulink and PIAnO[12], applying the optimization algorithm of Progressive Quadratic Response Surface Method (PQRSM)[13]. The cost function of optimization, as expressed in equation (10) was the sum of minimization of maximal absolute temperature differences of the pp1, cpp, and pp2 between experiments and simulations.

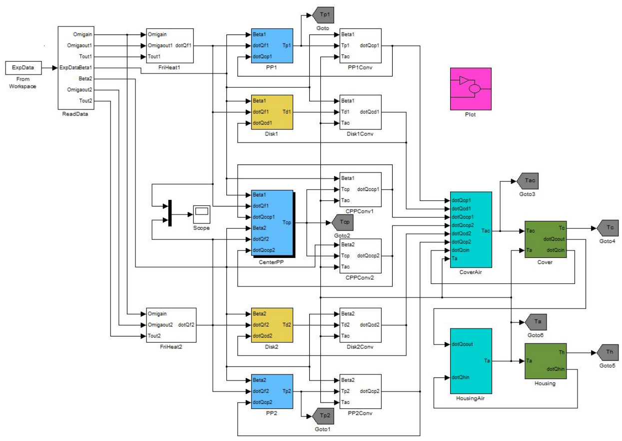


Fig. 5 Matlab/Simulink thermal model of DDC

$$\begin{aligned} f = & \min(\max(|T_{cp_{sim,i}} - T_{cp_{exp,i}}|)) \\ & + \min(\max(|T_{p1_{sim,i}} - T_{p1_{exp,i}}|)) \\ & + \min(\max(|T_{p2_{sim,i}} - T_{p2_{exp,i}}|)) \end{aligned} \tag{10}$$

where subscript *sim* stands for simulation temperature, and *exp* for experimental temperature, and i means the i^{th} sampling moment.

Through optimization process, the optimal heat convection coefficients are obtained and listed in Table 2.

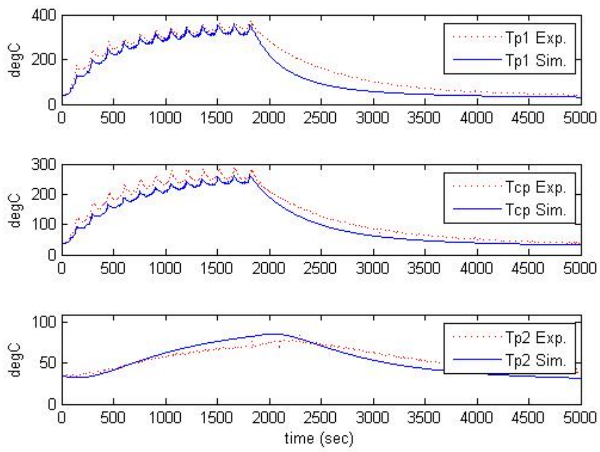
Table 2 Optimized heat convection coefficients and cost function

		Initial	Optimal
Design Variables	h_p	50	72.7
	h_d	50	158.4
	h_{him}	50	110
	h_{hout}	50	20
Objective Function	f	193.78	114.73

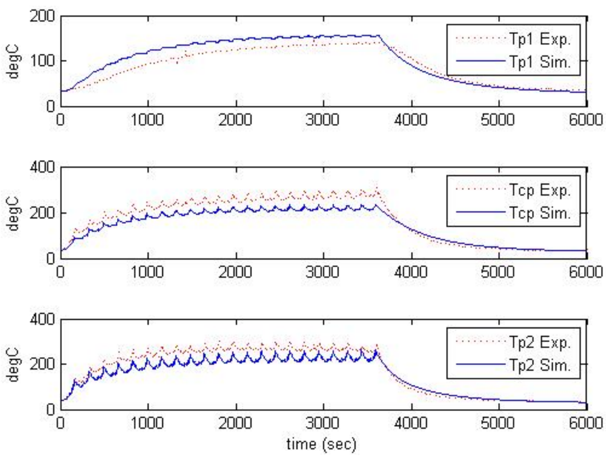
3.3 Results and Discussion

Figure 6(a) illustrates temperature comparison of simulation and experiment for operation Model (refer to Table 1), so only the first clutch was operated while the second one was disengaged during the whole test. The entire test period includes the heating (0~1,850 second) and cooling process (1,850~5,000 second) as shown in the figure. Figure 6(a) shows that simulation results present similar tendency with measured experimental data. In the heating process, the temperatures of the first and central pressure plates in simulation are very close to those in experiment, while in the cooling process, the temperatures in simulation decrease faster, which can be explained that the convection coefficients used in simulation are a little bit different from its real values. Obviously, it is very difficult to determine the accurate value of the convection coefficient, which is also the key point in modeling of the DDC.

Figure 6(b) shows temperature comparison of simulation and experiment when only the second clutch was operated by using operation Mode 2. It can be seen from the figure that there exists difference of simulation and experiment in the heating process, while there is not too much difference in the cooling process.



(a) For first clutch



(b) For second clutch

Fig. 6 Temperature comparison of simulation and experiment

The temperature difference between experimental test and simulation are given in Table 3. This difference is certainly caused because of the lumped parameter model which does not consider temperature distributions on the pressure plates and the disk. Inaccurate simulation values of uncertain parameters and the thermal expansion of clutch volume [14] are other causes resulting in the difference between experimental test and simulation. Further research to reduce temperature difference is still important to precision of the proposed thermal model. The effort is to figure out the representative temperature of the pressure plates and the disk at the location where temperature sensors were installed in test equipment.

Table 3 Temperature error and relative error

Clutch No.	T_{p1} (℃)		T_{cp}(℃)		T_{p2}(℃)	
	Temp. error	Max. temp. in test	Temp. error	Max. temp. in test	Temp. error	Max. temp. in test
1st	74.1	377	56.2	288.5	12.5	78
relative error	19.7%		19.5%		16%	
2nd	31	139	98	309	88	308.5
relative error	22.3%		31.7%		28.5%	

4. Conclusion

A thermal analytical model of lumped parameters for DDC is established in the paper, in which heat conduction and convection among the three pressure plates, two disks, housing, and air inside the clutch are taken into account together. However it is difficult to determine the values of uncertain parameter of heat transfer coefficients for simulation. The key parameters of the proposed thermal model are optimized by co-simulation of Simulink and PIAnO on the basis of experimental data. The main reason of temperature difference between the simulation and experiment is inferred as a result of the lumped parameter model which does not consider temperature distribution on the pressure plates and the disk. However proposed lumped parameter model can be easily implanted into the embedded system for real-time temperature estimation. The further research to improve the real-time estimation accuracy of the temperature of DDC is still required very much.

Acknowledgement

This research was supported by the Ministry of Trade, Industry & Energy (MOTIE) and Korea Institute for Advancement of Technology (KIAT) through the Project of Source Technology Development for Industrial Fusion and also through the Center for Automotive Mechatronics Parts (CAMP) at Keimyung University.

References

1) H. Naunheimer, B. Bertsche, J. Ryborz, and W. N. Novak, Automotive Transmissions: Fundamentals, Selection, Design and Application, Springer, New York, pp.163-1822011.
2) K. Kimmig, K. Henneberger, M. Ehrlich, G. Rathke and J. Martin, “Success with efficiency and comfort – The dry double clutch has become established on the automatic transmission market”,The 9th Schaeffler Symposium, Detroit, Michigan, pp. 154-163, 2010.
3) C. Breitfeld, S. Rinderknecht, F. Munk, D. Schmidt-Troje, C. Gueter, J. Neuner and J. Eder, “The New BMW Dual Clutch Transmission”, ATZ, Vol. 110, No. 4, 2008.
4) P. Aversa and E. DeVincent, “Evolution and outlook: powershift DCT250”, Getrag Co., DTF 2010-09-16, 2010 (http://www.getlag.com/media/0000001557.pdf).
5) J. C. Wheals, J. McMicking, S. Shepherd and B. Bonnet, “Proven High Efficiency Actuation and Clutch Technologies for eAMT™ and eDCT™”, SAE International J, 2009-01-0513, 2009.
6) M. X. Wu, J. W. Zhang and C. S. Ni, “Research on optimal control for dry dual-clutch engagement during launch”, Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, Vol. 224 No.6, pp.749-763, 2010.
7) G. Chen, K. Baldwin and E. Czarnecki, “Real time virtual temperature sensor for transmission clutches”, SAE International J. Engines, Vol. 4, No.1, pp.1523-1535, 2011.
8) G. Chen, H. Dourraand D. Shah, “Rotating Clutch Temperature Model Development Using Rapid Prototype Controllers”, SAE International J. Passenger Cars 2012-01-0625, 2012.
9) M. Velardocchia, F. Amisano and R. Flora, “A linear thermal model for an automotive clutch”, SAE Technical Paper2000-01-0834, 2000.
10) G. Kong and Z. Zaimin, “AMT starting process analysis and clutch thermal balance estimation model”,International Conference on Electric Information and Control Engineering (ICEICE), Wuhan, China, pp. 2945-2949, 2011.
11) J. H. Cho and J. W. Kang, The device and method to measure the temperature of clutch component (in Korean), Korean Patent 10-2013-0061394, 2013.
12) PIAnO (Process Integration, Automation and Optimization) User's Manual (Ver. 3.3), PIPTECH Inc. 2011.
13) K. W. Park and S. J. Moon, “Optimal design of heat exchangers using the progressive quadratic response surface model”, International Journal of Heat and Mass Transfer, Vol. 48, No. 11, pp.2126-2139, 2005.
14) H. Matija, H. Zvonko, N. Kranjcevic and J. Deur, “Experimental characterization and modeling of dry dual clutch thermal expansion effects”, SAE International Journal of Passenger Cars - Mechanical System, Vol. 6, No. 2, pp.775-785, 2013.