CLUTCH TRANSIENT HEAT TRANSFER SIMULATION FOR HILL START VEHICLE TEST CONDITION

Çakmak T.*, Kılıç M.**
*Author for correspondence
*Valeo Automotive Industry and Trade Co., Bursa, Turkey
**Uludağ University, Engineering Faculty, Department of Mechanical Engineering, Bursa, Turkey

E-mail: tolga.cakmak@valeo.com

ABSTRACT

The successive repetitive engagement operation causes temperature rise on the friction surfaces of the automotive dry clutches. This condition is simulated via hill start vehicle tests by clutch manufacturers during thermal performance validation of new transmission platforms or vehicle configurations. The aim of this study is to determine the temperature variation of the clutch pressure plate by the use of a 1D transient thermal simulation for the hill start vehicle test conditions. Simulation model includes pressure plate, flywheel and clutch housing elements. Clutch housing air temperature variation was simulated numerically for 120 successive repetitive cycles of hill start truck vehicle test. Parameters effecting pressure plate temperature change were analyzed. It is observed that increasing the mass of the clutch pressure plate results a decrease in the temperature variation in early stages of engagement cycles. However, increasing the value of the pressure plate convection coefficient significantly reduces the temperature variation during all stages of cycles. Finally, lightweight design proposals of dry clutch pressure plate for better energy dissipation are demonstrated.

INTRODUCTION

Clutches connect and disconnect two shafts, either when both are stationary or relatively rotating [1]. In vehicles, relative speeds of engine and gear box shaft may increase during vehicle launch, depending on the conditions like driving style, vehicle weight or road slope. Clutch design should withstand extreme conditions by providing good thermal performance. In order to predict the thermal performance of clutches in design phase, mathematical models and finite element analysis are solved for thermal boundary conditions. Following the availability of clutch and vehicle prototypes, hill start vehicle tests are performed for final validation.

Temperature rises during clutch engagement due to friction between disc facing and counter parts; pressure plate and flywheel. In engagement phase, clutch facing friction area is not totally in contact with metal parts. This phenomenon causes spot temperature rise. Khamlichi et al. [2] defined analytically spot and nominal temperature rise during engagement and concluded that the ratio of fibers to matrix in an organic facing is effective on temperature rise. Slipping duration and energy dissipation during engagement phase of clutch have been investigated in many studies. Duque et al. [3] used a hybrid math model of previous literature studies to calculate heat generation during vehicle launch. Same researchers studied different friction models during vehicle launch and investigated of these models effect on slippage duration and dissipated energy [4].

In the present study, Ø430 mm size clutch has been considered. There are available studies in the literature for the same size. In one of them, Coelho and Rabelo [5] investigated clutch pressure plate mass effect on temperature rise. Results show that both temperature rise and clutch wear were lowered when the pressure plate mass was increased. In another same size clutch study, Fan and Ji-ping [6] investigated thermal stresses of clutch pressure plate numerically. In their study, convection coefficient was considered as a function of Reynolds and Prandtl numbers.

NOMENCLATURE

A	[m^2]	Area
c	[J/kgK]	Specific heat
h	[W/m^{2}K]	Convective heat transfer coefficient
m	[kg]	Mass
E	[J]	Energy dissipation
PP	[-]	Pressure plate
T	[°C]	Temperature
T_c	[Nm]	Clutch torque
t	[s]	Time
w	[rad/s]	Speed
ρ	[kg/m^3]	Density

Subscripts

c	Clutch
r	Rotational
s	Slippage

Successive repetitive vehicle start-ups result temperature rise on frictional surfaces of clutch parts. In the literature, there are available numerical studies for 5 [7] and 10 [8] number of successive start-ups. 2D finite element analyses have been used in these studies. 3D FEA and CFD analysis for 20 start-ups had been performed by Kilic et al. [9]. It is known from the previous studies that 2D and 3D numerical analysis requires large computer storage as well as quite long computational times especially when the number of successive start-ups increases. Therefore, in the present study, a 1D numerical simulation method for the large number of successive start-ups is introduced to analyze the effect of several important parameters in a short calculation time. In the current study, the simulation analysis are performed for the number of successive start-up cycles up to 120.

FRICTIONAL PARTS OF DRY CLUTCH SYSTEM

Dry clutch system shown in Figure 1 is used in manual and automated manual transmissions. The time at which all the torque on the engine is transmitted to the gearbox input shaft is the effective engagement time. The time at which the gearbox input shaft is pulled into synchronism with the engine is the synchronizing time [10]. In manual transmissions, during this period on a sloping road, driver releases the clutch pedal and at the same time presses the accelerator pedal, this action leads to temperature rise due to friction between clutch disc facings and counter solid parts (pressure plate and flywheel).

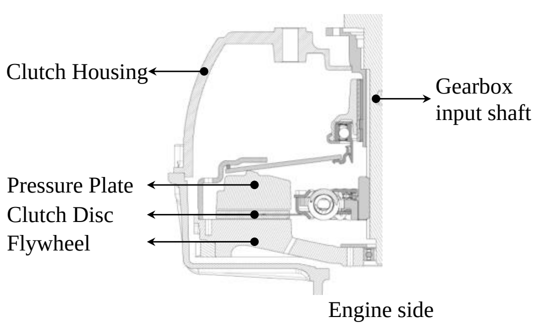

Figure 1 Dry clutch system

Energy dissipation E is a function of transmitted clutch torque T_c [Nm] and rotational speed difference w_r [rad/s] between engine and gearbox shaft;

$$E = \int_0^{t_s} T_c \omega_r(t)dt \tag{1}$$

In Figure 2, an engagement duration sample based on vehicle test measurements is shown.

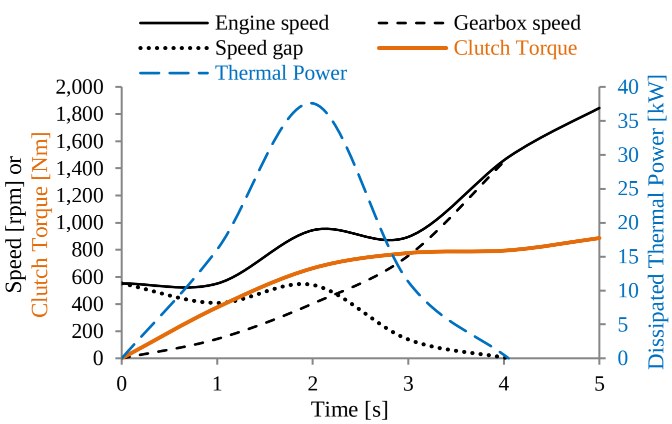

Figure 2 Speed, Clutch Torque and Dissipated Thermal Power variation during vehicle start-up

EXPERIMENTAL TEST

Test purpose is to evaluate on vehicle the characteristics of friction and the thermal and mechanical behavior of a clutch. For this purpose, vehicle move cycles on a slope road 10% were carried out. In order to have high energy dissipation following conditions are tuned; maximum vehicle load, the gearbox ratio used for starting the vehicle, engine rotation speed to start. Each cycle includes two steps; first step is slippage duration and second step is cooling duration at idle speed till 60 s. This cycle was repeated successively till 120 cycles at total 7200 s. Engine and gearbox shaft speeds and clutch housing air temperature were recorded by sensors for all cycles.

Slippage duration, energy dissipation and clutch housing air temperature variation were obtained from the measurements. Sensor instrumentation on the vehicle is shown in Figure 3 and vehicle configuration is shown in Table 1.

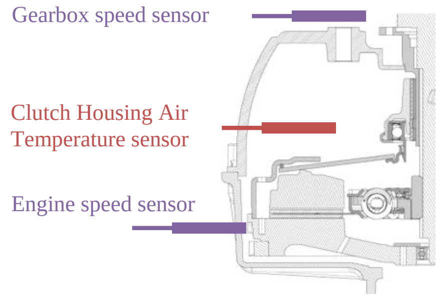

Figure 3 Sensor instrumentation for hill start vehicle test

Table 1 Hill Start Vehicle Test information

Configuration	Value
Engine Torque [*Nm*]	2500
Vehicle weight [*kg*]	40000
Wheels Radius [*mm*]	504
1st Gear Ratio	13.8
Differential Gear Ratio	3.08

Dissipated energy and slippage duration of each cycle of hill start vehicle test is shown in Figure 4.

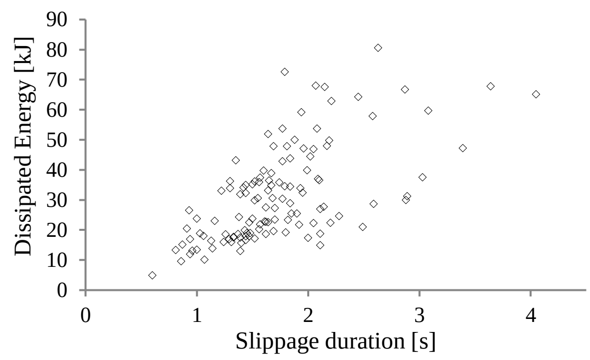

Figure 4 Distribution of each engagement cycle during hill start vehicle test

During the hill start vehicle test, limits of sensor instrumentation do not allow to measure temperature on frictional solid parts. To estimate temperature rise on pressure plate and its effecting parameters, numerical simulation has been used.

NUMERICAL SIMULATION

Numerical simulation aims at describing a system thermal model developed under Amesim software which had been used and validated by several researches such as Bataus and Vasiliu [11] and Sorihan [12] for powertrain modeling. The main objective of this model is to calculate the mass temperatures reached by the pressure plate and the flywheel and the air temperature inside the clutch housing during the hill start vehicle test. Energy dissipation and slippage duration applied in the numerical simulation are those obtained in the experimental tests.

The current transient thermal 1D model includes 3 elements representing pressure plate, flywheel and clutch housing. Schematic view of 1D simulation sketch is shown in Figure 5.

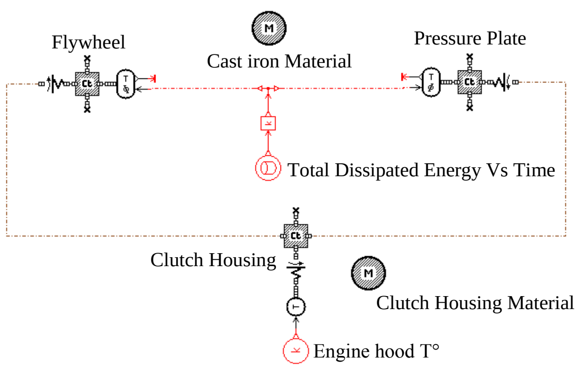

Figure 5 Schematic view of 1D simulation (Amesim)

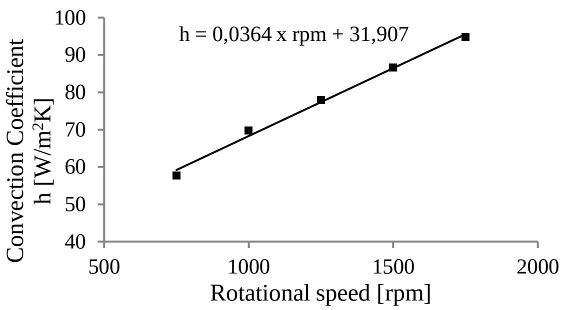

Figure 6 Convection Coefficient for Ø430 mm size Dry Clutch Pressure Plate

The pressure plate (PP) was set at 90 ° C, assuming that the initial temperature was about 10 °C higher than the clutch housing temperature. Inputs are No of cycles; 120, cycle duration; 60 s, average dissipated energy; 31.55 kJ, average slippage duration; 1.732 s, PP convection coefficient, measured experimentally, has been defined as a function of rotational speed as shown in Figure 6, PP mass is 22 kg and specific heat is 550 J/kgK, engine average and idle speed are 700 and 550 rpm, respectively. Outputs are clutch housing air and pressure plate mass temperature variation. Simulation duration is less than 1 minute.

RESULTS AND DISCUSSION

Temperature rise can lead to many problems; poor torque transmission due to friction coefficient drop [13], facing fade [14], poor gear shift quality [15] and pressure plate cracks [16]. Experimental measurements of clutch housing air temperature during 120 repetitive engagements and 1D numerical computation results are shown in Figure 7. Temperature of the clutch housing air reaches 96°C at the end of 120 repetitive successive experimental vehicle start-ups. This is just 13 °C above of initial condition. Reached temperature is quite below of acceptable levels and experimental measurements proof that clutch design shows very good thermal and mechanical performance for the vehicle configuration shown in Table 1. Numerical computation shows parallel characteristics to experimental results. Hill start vehicle test computation is accepted as “Base” data that the following improving scenarios outputs were compared with it.

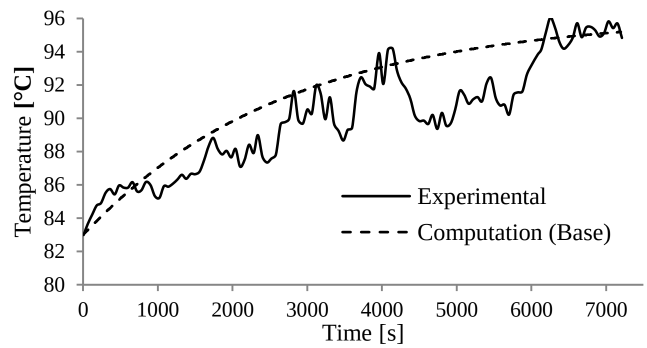

Figure 7 Clutch housing air temperature variation during hill start vehicle test

Pressure plate mass temperature rise during hill start vehicle test and 4 different improving scenarios were demonstrated in Figure 8. Scenario descriptions are shown in Table 2. According to numerical computation of hill start vehicle test, pressure plate mass temperature (Computation Base in Figure 8) reaches maximum 122 °C at the end of 120 cycles.

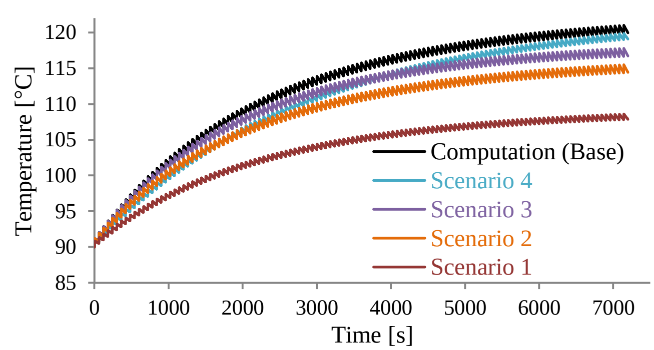

Figure 8 Pressure Plate mass temperature variation during hill start vehicle test condition

Table 2 Scenario descriptions

Scenario	Description
Scenario 1	Base - 30% Dissipated Energy, kJ
Scenario 2	Base + 30% Pressure Plate conv. coeff., W/m^2K
Scenario 3	Base + 30% Clutch housing conv. coeff., W/m^2K
Scenario 4	Base + 30% Pressure Plate mass, kg

Base ; Hill Start Vehicle Test Computation

To prevent dramatic temperature rise of pressure plate, 30% change of 4 different effecting parameters was analyzed. According to the simulation results, most effecting parameter is energy dissipation reduction; Scenario 1, which depends on many factors; engine torque, gear ratios, wheels radius, vehicle weight, road slope and driving style, while it is independent of clutch parameters. Compared to conventional manual transmissions, automated or e-clutch adapted manual transmission systems are providing better thermal performance in terms of dissipated energy, since the slippage duration management is under control with these systems. On the other hand, most effective clutch design based parameter seems convection coefficient of pressure plate; Scenario 2. Clutch housing convection coefficient increase; Scenario 3 shows similar characteristic to Scenario 2 but not as effective as its. Pressure plate mass increase; Scenario 4 is more efficient at early cycles than Scenario 2 and 3. After approximately 60 cycles its effect results less than the others.

Simulation results give an idea for further improved design of clutch pressure plate. Increase of convection coefficient, convection surface and mass of the pressure plate will avoid its dramatic temperature rise. On the other hand, powertrain mass increase is playing negative role in terms of CO_2 emissions [17] and cost. Therefore, lightweight design without compromising thermal and mechanical performance would be a better option. For this purpose, lightweight design proposals of pressure plate including increased convection surface for better energy dissipation are demonstrated in Figure 9.

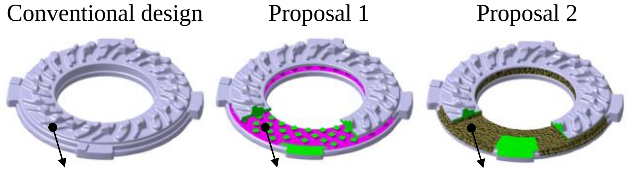

Figure 9 Lightweight design proposals of dry clutch pressure plate for better energy dissipation

In Figure 9, compared to conventional design, Proposal 1 is 18% lightweight and has 70% more convection area, thanks to cooling channels, Proposal 2 is 36% lightweight and has more than 100% convection area, thanks to open porosity metal foam implementation. Pressure plate mass optimization can be investigated with 3D FEA. Proposal visuals have partial section view in Figure 9. A further improvement option could be the upgrade of the casting material chemical content or graphite flake size with higher thermal diffusivity [18].

CONCLUSION

Dissipated energy and slippage duration of a hill start vehicle test is investigated in this research experimentally. Experimental measurements show that dissipated energy increases with slippage duration increase. Simulation of experimental study is demonstrated.

Simulations were carried out to determine the impact of four different scenarios on the pressure plate temperature rise during hill start vehicle test. Numerical computation results indicate that the most effective scenario is reduction of dissipated energy. When the clutch design based parameters are analyzed, it is observed that, dry clutch pressure plate convection coefficient is more effective than its mass change. Based on these outputs, lightweight design proposals of dry clutch pressure plate for better thermal performance have been demonstrated.

ACKNOWLEDGEMENTS

The authors gratefully acknowledge the support of Technology Research Council of Turkey (TUBITAK) [grant number 112D082]; ongoing with collaboration between Uludağ University and Valeo Company in Turkey.

REFERENCES

[1] Peter R.N. Childs, Mechanical Design Engineering Handbook, Clutches and Brakes, Chapter 13, *Elsevier*, 2014.

[2] Khamlichi A., Bezzazi M., and Parron Vera M.A., Optimizing the thermal properties of clutch facings, *Journal of Materials Processing Technology*, 142 (2003) 634–642.

[3] Duque E.L., and Barreto M.A., Fleury A.T., Math model to simulate clutch energy during vehicle launch, *18th SAE Brasil International Congress and Exhibition*, 2009.

[4] Duque E.L., Barreto M.A., and Fleury A.T., Use of different friction models on the automotive clutch energy simulation during vehicle launch, *ABCM Symposium Series in Mechatronics, Vol. 5, Section VIII - Sensors & Actuators*, (2012) 1376–1389.

[5] Coelho R., and Rabelo T., Clutch 430 Heavy Duty, *XX SAE Brasil International Congress and Exhibition*, São Paulo, Brasil, 2011.

[6] Fan Z., and Ji-ping B.T., Finite Element Analysis and Improvement on the Thermal Stress of the Truck Clutch Pressure Plate, *Forestry Machinery & Woodworking Equipment*, 39 (2011) 23–26.

[7] Pisaturo M., and Senatore M.A., Simulation of engagement control in automotive dry-clutch and temperature field analysis through finite element model, *Applied Thermal Engineering*, (2016) 958–966.

[8] Abdullah, O., and Schlattmann, J., Finite Element Analysis of Temperature Field in Automotive Dry Friction Clutch, *Tribology in Industry*,34 (2012) 206–216.

[9] Kilic M., Cakmak T., and Sevilgen G., Clutch Pressure Plate Compactness Effect on the Clutch System Heat Dissipation, *12th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics*, Spain, (2016) 438–444.

[10] Fenton, J., Handbook of Automotive Design Analysis, A study into the analysis of vehicle transmission, *Butterworth Co. Publishers*, (1976) 62–63.

[11] Bataus M.V., and Vasiliu N., Modeling of a dual clutch transmission for real-time simulation, *U.P.B. Sci. Bull., Series D*, 74 (2) (2012) 251–264.

[12] Sorihan A.H.M., Simulation Based Comfort Evaluation for Vehicles with Automated Transmissions, *MSc Thesis, University of Applied Sciences Konstanz*, 2012.

[13] Zhao Z., He L., Yang Y., Wu C., Li X., and Hedrick J. K., Estimation of torque transmitted by clutch during shifting process for dry dual clutch transmission, *Mechanical Systems and Signal Processing*, 75 (2016) 413–433.

[14] Akhtar M.M.J., Abdullah O.I., and Schlattmann J., Transient thermoelastic analysis of dry clutch system, *FME Transactions*, 43 (2015) 241–248.

[15] Vasca F., Iannelli L, Senatore A., and Reale G., Torque Transmissibility Assessment for Automotive Dry Clutch Engagement, *ASME Transactions on Mechatronics*, 2011.

[16] Abdullah, O., and Schlattmann, J., and Lytkin M., Effect of Surface Roughness on the Thermoelastic Behaviour of Friction Clutches, *FME Transactions*, 43 (2015) 241–248.

[17] National Highway Traffic Safety Administration, Greenhouse Gas Emissions and Fuel Efficiency Standards for Medium- and Heavy-Duty Engines and Vehicles, United States Environmental Protection Agency, EPA-420-R-16-900, (2016) 2–44.

[18] Hecht, R.L., Dinwiddie, R.B., and Wang. H., The effect of graphite flake morphology on the thermal diffusivity of gray cast irons used for automotive brake discs, *Journal of Materials Science*, 34 (1999) 4775–4781.