CLUTCH PRESSURE PLATE COMPACTNESS EFFECT ON THE CLUTCH SYSTEM HEAT DISSIPATION

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ABSTRACT

In the recent years, the role and importance of clutch size and weight in transmission system have been continuously increasing primarily due to its geometrical and functional constraints in the clutch housing and high fuel efficiency target. Increasing demands on vehicle performance within a limited clutch system envelope lead to an increase of the clutches thermal solicitations. In this paper finite element method was used to investigate the effect of heavy truck clutch pressure plate compactness on the clutch system heat dissipation. Three-dimensional computational fluid dynamics (CFD) model was used to study temperature variation of the clutch system representing repetitive successive hill starts condition. Structural analysis based on CFD output was also studied and both CFD and structural analyses results were compared. Clutch plate stress distribution was also studied with structural analyses for the clutch burst condition. 2 cases with different plate compactness were investigated and the results show that clutch plate compactness has a significant impact on the clutches thermal performance.

NOMENCLATURE

A	[m^2]	Area
c	[j/kgK]	Specific heat
F	[N]	Load
h	[W/m^{2}K]	Convective heat transfer coefficient
k	[W/mK]	Thermal conductivity
m	[kg]	Mass
P	[W]	Energy
Q	[W]	Heat Transfer
T	[K]	Temperature
T	[Nm]	Transmitted torque
w	[rpm]	Rotational speed
μ	[-]	Friction coefficient
ρ	[kg/m^3]	Density
σ	[W/m^{2}K^4]	Stefan-Boltzmann Constant
ε	[-]	Emissivity coefficient

Subscripts

h	Hot body
c	Cold surrounding
c	Convection
p	Constant pressure
r	Radiation

INTRODUCTION

Minimizing energy loss to rotating mass in manual transmission powertrain is one of the important target in order to approach the maximum theoretical thermal efficiency of internal combustion engines [1]. On the other hand mass decrement which is one of the effective parameter to reach this goal is limited for a clutch plate due to required heat absorption, induced high stresses during engagement conditions and inertia need for clutch system dampening function. The thermal simulation of clutch parts are of particular concern for engineers, because they much evaluate the extreme vehicle conditions to determine if the particular product can withstand high temperatures generated between the contacting surfaces. Coelho and Rabelo [2] made an experimental study for heavy duty clutch pressure plate to show the pressure plate increased mass positive effect on thermal absorption.

Zhang and Bao [3] showed that it is feasible to improve thermal performance while decreasing the clutch pressure plate mass and structural analysis of clutch plate has been studied with the assumption of convection coefficient determination based on an empirical definition. Minereau [4] determined experimentally an empirical definition for convection coefficient as a function of rotational speed and facing mean radius and showed that convective heat transfer is mainly effective type in the cooling of clutches. Abdullah and Schalttmann [5] studied transient thermal analysis of a dry friction clutch system during repeated engagements based on the uniform pressure and uniform wear theories. Temperature variation of pressure plate, facing and flywheel with time was presented for two-dimensional axisymmetric model.

Many researchers have studied clutches thermal phenomena with CFD. Wittig S. et. al [6] presented numerical results of the flow in a high speed rotating clutch with emphasis on geometrical variations. Wang et. al [7] applied CFD in order to determine convection coefficient of pressure plate and used thermal structure analysis to develop a new model to reduce the stress of the clutch plate. Instead of unified convection coefficient for all convective surfaces of pressure plate, its effecting factors air flow speed, pressure plate structural characteristics and surrounding temperature were taken into account for local surfaces. Anthony et. al [8,9] investigated the effect of clutch housing size on the flow and the thermal behaviour of single clutch and the impact of holes on the air flow path for double dry clutch thermal behaviour.

It is obvious that the main advantage of CFD application in clutch thermal phenomenon investigation compared to structural thermal analysis approach is obtaining heat transfer coefficient distribution and variation through solid parts surfaces.

This study examines improved clutch plate compactness, considering weight reduction without compromising thermal performance on the clutch system heat dissipation by taking into account the advantage of heat transfer coefficient variation and distribution based on geometry and fluid flow speed over surfaces computation with CFD. Geometrical change due to compactness improvement effect was also investigated through structural analyses.

HEAT GENERATION DURING CLUTCH ENGAGEMENT

The purpose of this study is to simulate the behaviour of the clutch pressure plate in situations with high heat generation in single clutch transmissions. High heat generation, so the power dissipation within the clutch occurs during the engaging operation as a function of transmitted torque and rotational speed difference;

$$P(t) = T(t)\,(w_{engine(t)} - w_{gearbox(t)}) \tag{1}$$

The energy dissipated in heating within the clutch during launches or gear shifts only depends on vehicle parameters (engine torque, gear ratios, wheels radius, vehicle weight, road slope) or driver choices (engine speed at synchronism) but it doesn't depend on clutch parameters.

Friction coefficient between metal components (pressure plate and flywheel) and damper facings slightly decreases when the temperature reaches a critical level (350 – 400 °C) that leads to loss of torque transmission. Gas generation that lubricates the facings area at high temperature levels is the main reason of friction coefficient decrease. In this case clutch is not able anymore to transmit the engine torque. Torque transmission $T(t)$ [Nm] is directly proportional to clamp load F [N] and friction coefficient μ.

$$T(t) = f\,(F,\mu) \tag{2}$$

Generated heat flux during slippage phase is distributed between components pressure plate, flywheel and disc assembly depending on their thermal effusivity. The heat is transferred by conduction within the solid parts. Temperature of solid parts increases due to heat flux, depending on their specific heat c_p [J/kgK] and mass m [kg].

$$Q = m\, c_p\, \Delta T \tag{3}$$

On the other hand heat Q, is dissipated from solid parts to clutch house air by convection and radiation.

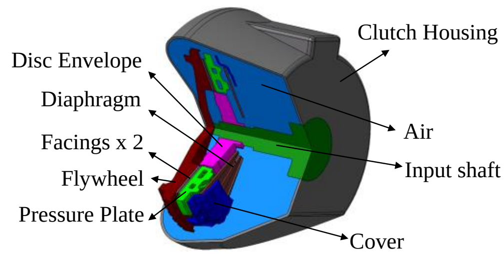
$$Q_c = h\, A\, \Delta T \tag{4}$$

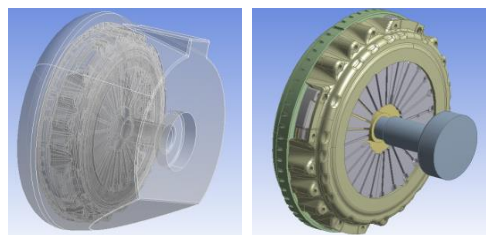
$$Q_r = \sigma\, \varepsilon\, A\,(T_h^4 - T_c^4) \tag{5}$$

Since the convective heat transfer is mainly effective type in the cooling of clutch components, if the increased convective surface area positive effect on heat dissipation (Eq.4) is equal or more than mass decrement negative effect on heat absorption (Eq.3), then it is feasible to have solid parts mass reduction without compromising thermal performance.

MATERIAL AND METHOD

In this approach firstly, thermal model of a clutch system used in this study was created by using CFD method. 3D CAD model of the clutch system components were obtained from real components CAD data and is shown in Figure 1. CAD model consists of ten different domains which were defined as 9 solid and a fluid domain in CFD model.





(a) Fluid domain (b) Solid domains

Figure 1 3D-CAD models of the fluid domain (a) and solid domains (b)

Today, there are many commercial software packages available for CFD analysis. With the improved computer technology and CFD techniques, analysis of complex engineering systems based on numerical calculations with sufficient accuracy and acceptable results is now possible for researchers. In this study, Ansys Fluent software is used for three-dimensional flow and heat transfer field analysis, this software has ability solving continuum, energy, and transport equations numerically with natural convection effects [10]. The first step of the CFD method was creating air domain from the CAD models of the solid parts. In this step, the disc assembly which includes many components inside was simplified according to its envelope. The next step was forming a mesh structure for transient computations to get precise results and reduce computing times. The section view of the mesh structure and mesh distribution of the surfaces of the solid parts are shown in Fig.2-3. The full assembly mesh model consist 9.10^6 tetrahedron elements. Solid mesh was conformal to fluid mesh structure.

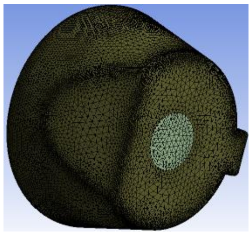


Figure 2 A section view of mesh structure of all domains

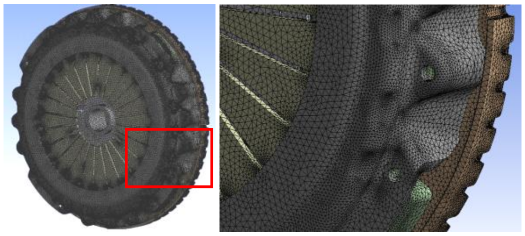


Figure 3 Mesh structure on the surfaces of the solid parts

The third step of the CFD method was defining appropriate boundary conditions for possible numerical solution. Defined material properties were based on experimental studies. In our CFD model four different materials were defined as described in Table 1. Thermal properties of these materials vary with temperature during the numerical simulation. Cover, disc-envelope, input shaft and diaphragm part's material was defined as steel, two facing part's material was defined as facing composite, clutch house, flywheel and pressure plate part's material was defined as cast iron.

Table 1 Material Properties definition of Clutch Parts

Component	Material	Properties *f(T)*
Cover, disc-envelope, input shaft and diaphragm	Steel	ρ [kg/m^3] c_p [J/kgK] k [W/mK]
Facings	Facing composite	
Flywheel, Pressure Plate, Clutch Housing	Cast iron	

Fluid was defined as air which is listed in Ansys Fluent default material library. The initial temperature of the pressure plate and flywheel surfaces contact with the other solid parts were considered as a value of 90°C. And all other parts include air zone was started as a value of 80°C. The k-ω SST model was chosen for turbulence modelling and continuum, momentum and energy equations solved simultaneously during computations. Single Rotating Reference Frame (SRF) method was used for modelling the rotating parts and the clutch housing was the only fixed part. All other parts were assumed as rotating parts. For fluid domain, frame motion was used which has a variable rotational speed (rpm). The values of rotational speed and the duration applied to these parts are based on previous experimental vehicle tests. For solid domain, all parts without clutch housing were defined as stationary wall relative to adjacent cell zone with no slip condition and the rotational speed was considered as 0 rpm. Relative speed between pressure plate and facing was ignored.

Power dissipation given in Eq.1 so the corresponding heat flux was applied to the contact surfaces between casting parts (pressure plate, flywheel) and facings (shown in Figure 4 and labelled as 1-2).

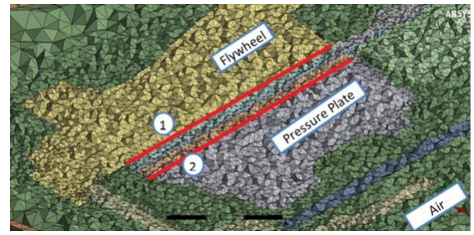


Figure 4 Contact surfaces of pressure plate and flywheel parts with facing parts.

In this study one cycle repetitive engagement and cooling phase that lasts within 60 seconds was employed. Slipping duration was calculated for 12% slope of road and maximum vehicle weight. Calculated heat flux is applied half and half between pressure plate and flywheel contacting surfaces with facings. A convective heat transfer boundary condition was applied to the exterior surfaces of the clutch house contact with the under hood ambient air and the exterior surfaces of the clutch components contact with the clutch house interior air. The convergence is assumed when the normalized residuals of flow equations are less than 10^{-4} and the energy equations are less than 10^{-6}.

Since the CFD computations of whole clutch system take long time duration, simple but similar to CFD analysis approach; single component pressure plate was also investigated through structural transient thermal analysis for repetitive successive hill start condition for 20 cycles with Ansys Workbench. Mesh structure in structural analysis is shown in Figure 5.

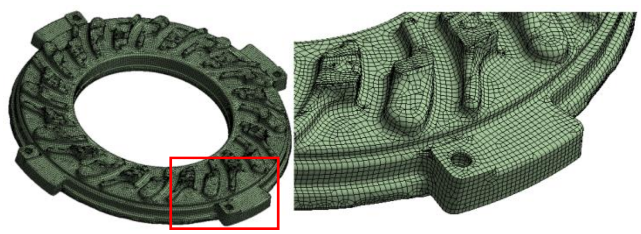
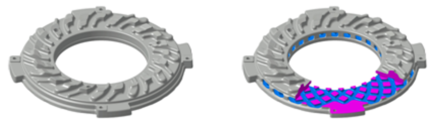


Figure 5 Mesh structure of the pressure plate

Convection coefficient was applied through exterior surfaces of pressure plate was obtained from CFD simulation. Heat flux was applied to facing contact surface of pressure plate. Burst strength limit of pressure plates were also investigated for two cases.

In this study CFD numerical study of a clutch system and corresponding structural analysis for single pressure plate were described for repetitive hill start vehicle test. The validation of computed convection coefficient will be considered in further studies by bench tests. Investigated 2 pressure plate cases with different plate compactness are shown in Table 2. Case 2 compactness [convection surface, m^2 / volume, m^3] value is doubled of case 1.

Table 2 Comparison of pressure plate Case 1 and Case 2



	Case 1	*Case 2*
Material	GJL 200	
Volume [m^3]	0,00347	0,00285
Mass [kg]	25,13	20,63
Convection surface [m^2]	0,216	0,3665
Compactness [m^2/m^3]	62,24	128,6

RESULTS AND DISCUSSION

The initial temperature values of the contact surfaces of the pressure plate and flywheel surfaces were considered 90°C and for clutch house air 80°C. Heat flux application duration was 0,595 s, then the system was cooled at 550 rpm till 60 s. This cycle was performed once in CFD simulation that was resulted within 60 s at total simulation of clutch system and 20 times in structural analysis of single component pressure plate that was resulted within 1200 s at total simulation. Applied convection coefficients in structural analysis are the ones that were obtained from one cycle CFD results

Transient Thermal CFD Simulation Results

The maximum predicted temperature value was suddenly increasing to 185,8 °C for Case 1 and 188,5 °C for Case 2 when the heat flux was applied to the contact surfaces. After heat flux application duration, the cooling phase was applied till 60 s in which the heat flux was remained constant as a value of 0 [W/m^{2}K]. During this cooling phase, rotational speed was decreased from maximum engine torque 1300 rpm to idle speed 550 rpm in a certain time based on knowledge from previous vehicle test experiments. This cycle was performed once in CFD computation. Maximum temperature variation of two cases of CFD results is shown in Figure 6 and in Figure 7.

During the cooling period, the temperature values of pressure plates decreased slowly compared to the heating period. Within cooling duration, maximum temperature values decreased slowly, compared to first 15 seconds period. The temperature distribution of two cases at 60 seconds is shown in Figure 8.

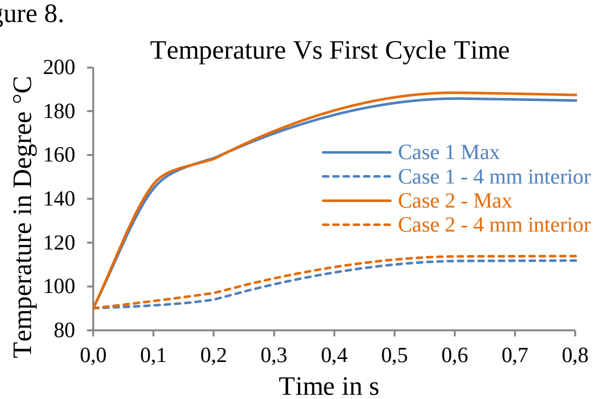


Figure 6 Maximum Temperature variation of pressure plates Case 1 and Case 2 in the first cycle heating phase

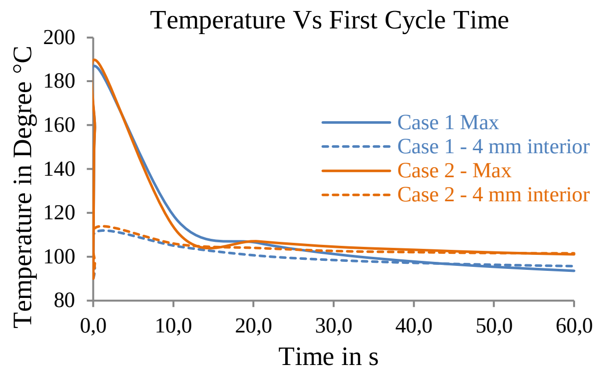
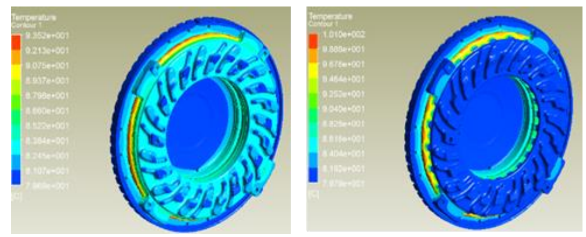


Figure 7 Maximum Temperature variation of pressure plates Case 1 and Case 2 for the first cycle



Case 1 *Case 2*

Figure 8 Temperature distribution of the pressure plate surfaces at t=60s of the simulation time

Transient Thermal Structural Simulation Results

CFD analysis of complete clutch system takes too long time. If convection coefficient is computed by CFD or measured experimentally then similar but simple approach could be applied to single component pressure plate in structural analysis in order to reduce computation time. In this study, convection coefficients used as input in structural analysis were taken from CFD results. Ansys Workbench software was used for three-dimensional transient thermal analysis of pressure plate.

Maximum Temperature variation of facing contact surface and 4 mm interior probe is shown in Figure 9 and Figure 10 respectively for 20 repeated cycles. It is observed that the clutch pressure plate compactness has significant influence on the heat dissipation and on the maximum temperature reached during repeated successive hill start condition. It is seen from figures that the maximum temperature for Case 2 is higher within first cycles where mass decrement effect seen as a reason. On the other hand temperature difference is reduced each cycle the maximum temperature for Case 2 is getting lower where the cooling performance is getting more effective than mass decrement after 18th cycle. The temperature distribution of two cases at 1200 s is shown in Figure 12. As seen from Table 3, improved compactness of pressure plate influenced positively the heat dissipation and resulted 2 °C less maximum temperature after 20 cycles even for approximately 20% mass decrement. This temperature difference can be increased with the convection coefficient increment which varies with rotational speed.

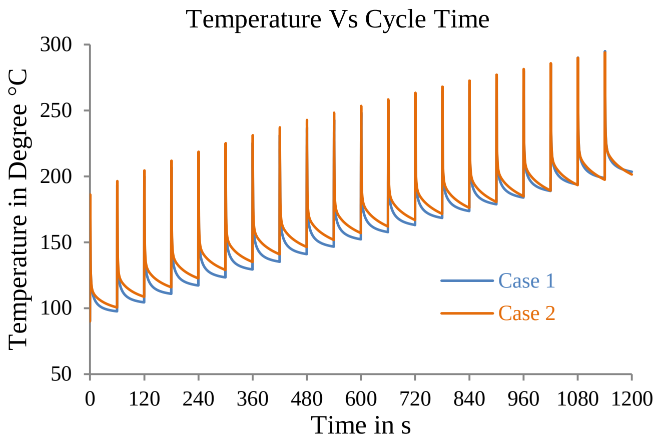


Figure 9 Maximum Temperature variation of pressure plate facing contact surfaces for 20 cycles

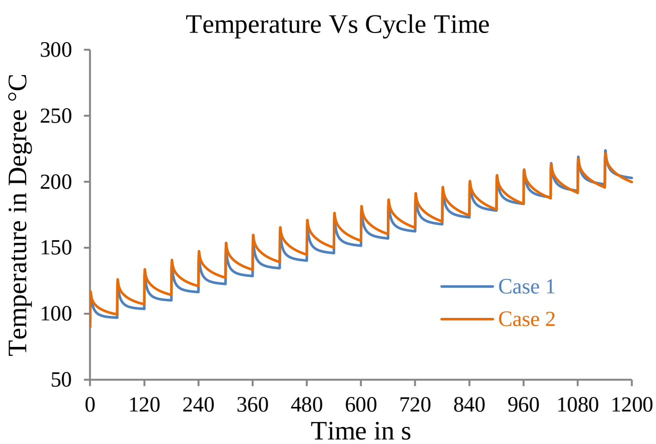


Figure 10 Temperature variation of pressure plate 4 mm interior probe for 20 cycles

Location of 4 mm interior from facing surface probe of pressure plate is shown in Figure 11.

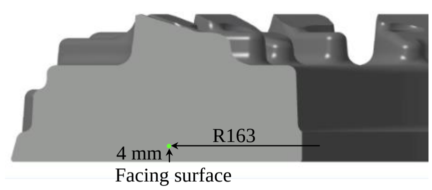


Figure 11 Location of 4 mm interior probe of pressure plate

Figure 13 shows the comparison for the computation results of CFD and structural analysis for the first cycle. As it can be seen from the Figure 13, even the pick values of reached temperature levels were closer of both CFD and structural analysis results, the cooling phase showed different characteristics. This difference can be expressed in terms of change in the heat dissipation in CFD and structural analysis, like heat conduction was also taken into account in CFD analysis. Table 4 shows the comparison of number of elements and computation duration of CFD and structural FEA.

Table 3 Maximum Temperature **c**omparison of pressure plate case 1 and case 2

Time	*Case 1*		*Case 2*	
	FS*	4mm	FS*	4mm
1200 s	203,55 °C	202,90 °C	201,46 °C	199,85 °C
1140.595 s	294,90 °C	223,80 °C	293,89 °C	221,70 °C

*FS; Pressure Plate Facing Contact Surface

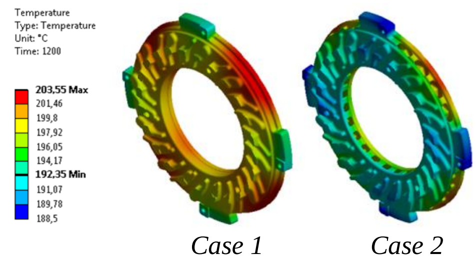


Figure 12 Temperature distribution of the pressure plate surfaces at t=1200s of the simulation time

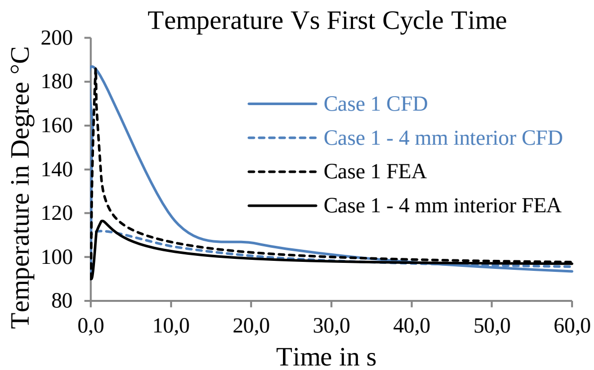


Figure 13 Maximum Temperature variation comparison of CFD and structural analyses

Table 4 Comparison of CFD and Structural Simulation

	CFD		**Structural FEA**	
	Case 1	*Case 2*	*Case 1*	*Case 2*
No of Elements	9.10^6	10.10^6	$2,83.10^5$	$3,27.10^5$
Simulation time	60s		1200s	
Simulation Duration	72h	80h	7,4h	26h

As seen from Table 4, CFD computation duration is quite long compared to structural FEA analysis even for 20 times reduced simulation time. In further studies 1D mathematical model will be studied in order to eliminate computation time.

Convection coefficient effect was also investigated to see the effect on the heat dissipation since it can vary with the rotational speed. As seen from Figure 14 Case 2 shows better cooling performance in earlier stages of cycles when the rotational speed so the convection coefficient is increased for both cases.

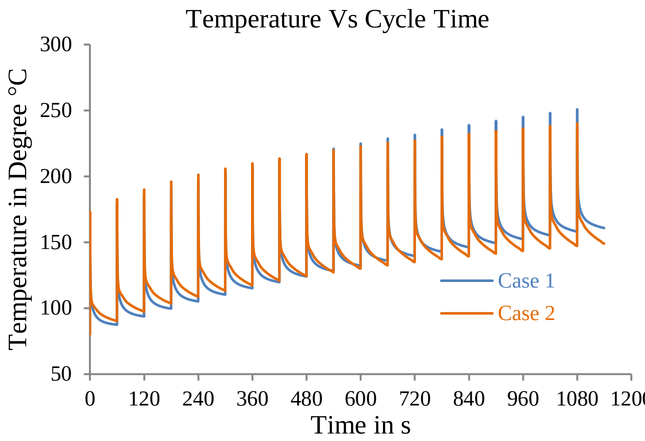


Figure 14 Temperature variation of pressure plate facing contact surfaces for 20 cycles with increased Convection Coeff.

Clutch plate compactness improvement geometry change was also investigated through burst condition in order to verify that the proposed design can withstand the specified rotational speed. Stress variation with rotational speed is shown in Figure 15. Even Case 2 shows worse burst mechanical performance both results are within the limits.

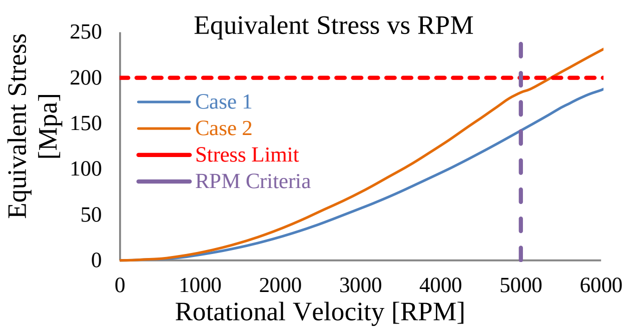


Figure 15 Stress variation Vs Rotational speed

Experimental Results Overview and Discussion

Temperature measurement of rotating solid parts in an enclosed envelope on the vehicle is a big challenge that requires complex, expensive and difficult telemetric instrumentation. Indirect measurement techniques, such as thermo-sensitive labels or colors, were used in the past to determine the reached maximum temperature. However temperature variation with time measurement of solid parts on the vehicle is required in this study to verify the numerical model. Clutch house air temperature measurement is also another option to analyze both numerically and experimentally, that can lead to solid components temperature levels prediction. This comparison will be considered in further studies when CFD computations finalized or 1D simulation model is built for 20 cycles in which the clutch house air temperature simulation will also be available.

In the literature there are many numerical but only one experimental study [Ref 2] available presenting temperature variation measurement interpolation based on test data of pressure plate on the vehicle for repetitive engagement conditions. Clutch size and the pressure plate mass presented are similar to our study case 1. In Figure 16 comparison of temperature variation for the first 20 seconds is shown. Even the reached maximum temperature levels are similar difference of the time where the temperature is maximum and difference of cooling performance is based on the vehicle configuration and driving style, so the applied heat flux amount and duration of heat flux differ for the literature experimental study [2] and our numerical study. This comparison is meaningful in terms of reached maximum temperature levels validation for the first cycle.

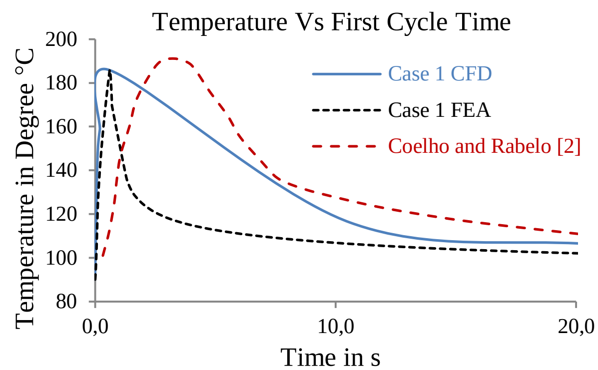


Figure 16 Maximum Temperature variation comparison for the first 20 seconds

CONCLUSION

In this study, a numerical simulation of the thermal behaviour of a complete clutch system with CFD and single pressure plate with structural finite element analysis was presented. Both analyses were investigated in transient state. Comparison of clutch compactness effect on heat dissipation during successive repeated hill start conditions of a heavy duty vehicle application was demonstrated. In addition, it is also described how CFD simulation of a clutch system output can be used as input to other parallel structural simulations. Described methodology can be used for further design improvements. Results showed that clutch components like pressure plate are heated in a short time for certain conditions while cooling of the component takes longer duration. It is observed that the clutch pressure plate compactness has significant influence on the heat dissipation and on the maximum temperature reached during repeated successive hill start condition. Compactness improvement effect on design was also investigated for the burst condition and stress variation with rotational speed was demonstrated.

1D simulation mathematical model build of clutch system and the experimental bench test for the convection coefficient measurement will be considered in further studies. Vehicle tests correlation is also essential to validate the numerical model.

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