The effect of duct shape on the heat dissipation performance of ventilated clutch pressure plate

T Cakmak¹, G Sevilgen², M Kilic³, A K Pehlivan⁴

¹, ⁴ Valeo Automotive Industry and Trade Co., Bursa, Turkey
² Automotive Engineering Department, Uludag University, Bursa, Turkey
³ Mechanical Engineering Department, Uludag University, Bursa, Turkey

*Corresponding author: tolga.cakmak@valeo.com

Abstract. This study presents the effects of duct design on thermo-fluidic characteristics of ventilated clutch pressure plate via validated numerical model. Two different duct designs are considered. Moreover, variation of duct width and airfoil shaped duct designs are investigated. Compared to non-ventilated conventional clutch pressure plate, results show that implementing duct shape enhances the overall cooling performance by up to 154% at idle speed 550 rpm, 137% at 1200 rpm, and 142% at 2000 rpm. Once only the duct shapes are considered, it is found that the variation of inlet and outlet duct width has negligible effect on overall cooling performance. Further, radial airfoil shaped duct design shows the best cooling performance in overall among all duct shapes, and enhances the overall cooling performance by up to 11% at 550 rpm idle speed compared to non-airfoil radial duct design. In addition, implementation of ventilated design results weight reduction up to 25%. Heat transfer enhancement mechanisms clarified in the present study are helpful to motivate further innovative lightweight solutions in order to overcome the challenges related to the recent CO₂ regulation for Heavy Duty Vehicles.

1. Introduction

Transport sector is a very competitive market in which the main driving force is the weight reduction while the costs are maintained or reduced. Thus, only the lightweight solutions that are accomplished by a performance improvement are worthy to be developed. Trucks face several challenges: one is related to the recent CO₂ regulation for Heavy Duty Vehicles (HDV); and thus, innovative lightweight solutions are required. On the other hand, clutches suffer from the consecutive and repetitive clutch engagements within fully loaded HDVs on roads with slope or with heavy traffic result in premature failure of these parts by thermal fatigue and excessive temperature rise.

The heaviest part of the clutch system is the cast iron manufactured pressure plate. The material choice of the pressure plate is gray or ductile cast iron due to its good surface finish, machinability, thermal capacity, wear resistance and low cost. Convective heat transfer can be enhanced by increasing the convective heat transfer surface [1]. Clutch system ensures the comfort during gear change. During the engagement, generated heat flux is spread between opposing frictional parts; flywheel, pressure plate and disc facings (Figure 1).

Repetitive and consecutive engagements can result many clutch failures due to temperature rise on the friction surfaces [2]. Consequently, the clutch transmission characteristic is affected negatively by the temperature rise [3]. Many studies have been performed on thermal problems of dry frictional systems like clutches and brakes. Mouffak and Bouchetara determined the heat transfer coefficient by
CFD and showed ventilation grooves and the material positive effect on the clutch disc facings [4]. Nejat et al. studied on the vanes model of a brake disk in a steady state approach and find out the heat transfer coefficient of the vanes by CFD [5]. Their results indicated that increasing the flow momentum and limiting the flow separation region close to the leading edge of the vanes improve the overall heat transfer coefficient. In the previous work, it has been demonstrated that one of the most effective clutch design parameter to prevent dramatic temperature rise is convection surface improvement of the pressure plate [6]. Similarly, Adamowicz showed the positive effect of convective cooling in brake discs [7]. It has been also experimentally verified that the enhancement of cooling performance is feasible via implementation of ventilation channels into clutch pressure [8].

The aim of this research is to enhance the convective heat transfer of clutch pressure plate thanks to ventilation channels, and thus prevents dramatic temperature rise during repetitive engagements. In addition, a popular technique used to meet fuel efficiency standards is lightweighting automotive components. Innovative designs demonstrated in this study results lower fuel consumption by weight reduction.

The heat transfer coefficient of the clutch pressure plate is estimated by means of a steady state conjugate heat transfer 3D CFD analysis. Different duct types are analyzed and compared with that of conventional non-ventilated version. Implementation of specific duct design into the casting pressure plate provides weight reduction up to 25%, and heat transfer enhancement up to 44% at idle speed.

![Dry clutch system](image)

**Figure 1.** Dry clutch system.

2. **Material and Method**

In manual transmissions, the time at which the gearbox input shaft is pulled into synchronism with the engine is the clutch engagement duration \( t_s \) (s). In real life, this period corresponds to releasing of the clutch pedal by the driver. This action leads to temperature rise due to friction between clutch disc facings and counter solid parts (pressure plate and flywheel). With the assumption of all frictional work conversion into heat, energy dissipation \( E \) (kJ) is a function of transmitted clutch torque \( \tau_c \) (Nm) and rotational speed difference between engine and gearbox \( \omega_r \) (rads \(^{-1}\));

\[
E = \int_0^{t_s} \tau_c \omega_r(t) \, dt
\]  

Generated heat flux during slippage phase is spread between components depending on their thermal effusivity. The conduction heat fluxes \( q_{x,y,z} \) in \( x,y \) and \( z \) directions increases the temperature \( T \) of solid parts that varies with the coordinates as well as the time \( t \), depending on their specific heat \( c_p \) (Jkg\(^{-1}\)K\(^{-1}\)) and specific mass \( \rho \) (kgm\(^{-3}\)) [4];
\[- \frac{\partial q_x}{\partial x} - \frac{\partial q_y}{\partial y} - \frac{\partial q_z}{\partial z} = \rho c_p \frac{\partial T}{\partial t} \]  

(2)

The conduction heat flux can be written in the form of temperature using Fourier's law. Assuming constant and uniform thermal properties, the conduction heat flux relations are;

\[ q_x = -k_x \frac{\partial T}{\partial x}, \quad q_y = -k_y \frac{\partial T}{\partial y}, \quad q_z = -k_z \frac{\partial T}{\partial z} \]  

(3)

\( k_x, k_y \) and \( k_z \) are thermal conductivity in \( x, y \) and \( z \) directions, respectively.

For the case of a clutch pressure plate, the boundary conditions are usually the conduction and energy dissipation from solid parts to surrounding clutch housing air environment mainly by the convection. The boundary condition is given as follow;

\[ T_c = T(x, y, z, t) \]  

(4)

\[ -q_s = h(T_c - T_\infty) \]  

(5)

where \( T_c \) is the convective surface temperature, \( T_\infty \) is the surrounding clutch housing air temperature, \( q_s \) is the convective heat flux (Wm\(^{-2}\)), \( h \) is the coefficient of heat convection at the solid-fluid interface (Wm\(^{-2}\)K\(^{-1}\)).

In this study, Ansys Fluent software is used for three-dimensional flow and heat transfer field analysis. It has ability to solve continuum, energy and transport equations numerically with natural convection effects. The numerical model consists of all heat transfer modes and the governing equations including continuity, momentum, and energy equations for steady-state conditions of fluid flow with constant properties can be written by using Equations 6-11 where, \( u, v \) and \( w \) are the velocity (ms\(^{-1}\)) components, \( \alpha \) is the thermal diffusivity (m\(^{2}\)s\(^{-1}\)), \( \nu \) is the kinematic viscosity (m\(^{2}\)s\(^{-1}\)), \( T \) is the temperature (°C), \( \rho \) is the density (kgm\(^{-3}\)) of fluid in the computational domain. More descriptions and detailed information can be found in the reference [9].

\[ \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \]  

(6)

\[ u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + u \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \]  

(7)

\[ u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial y} + u \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) \]  

(8)

\[ u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} = -\frac{1}{\rho} \frac{\partial p}{\partial z} + u \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) \]  

(9)

\[ u \frac{\partial \theta}{\partial x} + v \frac{\partial \theta}{\partial y} + w \frac{\partial \theta}{\partial z} = \alpha \left( \frac{\partial^2 \theta}{\partial x^2} + \frac{\partial^2 \theta}{\partial y^2} + \frac{\partial^2 \theta}{\partial z^2} \right) \]  

(10)

\[ \left( \frac{\partial^2 \theta}{\partial x^2} + \frac{\partial^2 \theta}{\partial y^2} + \frac{\partial^2 \theta}{\partial z^2} \right) + \frac{q}{k} = 0 \]  

(11)

2.1. Numerical Analysis

Heat transfer coefficient of pressure plate is computed by means of 3D conjugate CFD analysis. Temperature variation of the ventilated version is analyzed and compared with that of conventional non-ventilated version for the three operating range rotational speeds. Visuals of analyzed pressure plates for heavy duty truck applications are shown in Figure 2. Considering similar boundary conditions of clutches and brakes, literature of brake technology is also reviewed. The findings in this review reveal that the curved vane design is the best choice in terms of cooling performance among many other types [10]. Taking into account these findings, a similar radial duct design (model 2 in Figure 2), and additionally a straight duct design (model 4 in Figure 2) (used in brake discs for many
decades) are considered in the clutch pressure plate. Furthermore, in order to analyze the effect of airfoil shape and variation of duct width, two more models (model 3 in Figure 2 and 3, and model 5 in Figure 2 and 4) are considered.

![Figure 2. Clutch Pressure Plates Top Section View](image1)

Conventional non-ventilated (1), Radial duct (2), Radial duct airfoil (3), Straight duct (4), Straight duct width varied (5).

![Figure 3. Clutch Pressure Plates Section View - Radial duct (2), Radial duct airfoil (3).](image2)

![Figure 4. Clutch Pressure Plates Top Section View - Straight (4), Straight duct width varied (5).](image3)

Geometric properties of the pressure plates are shown in Table 1. Compared to straight duct shape, radial duct shape has more compactness value. Compactness ($m^{-1}$) states the ratio of the convection surface ($m^2$) to the volume ($m^3$). Radial duct shape (2) presents the most weight reduction up to 25% compared to conventional non-ventilated version (1).

In the previous numerical models, the simulations were analyzed in two ways; first, clutch system 3D full model including flywheel, clutch housing and disc assembly [1], latter, single pressure plate full model [11]. In this work, quarter model of the clutch pressure plate is considered (Figure 5). This approach enables faster computation compared to previous models. In the numerical analysis, pressure plate is defined as a solid and surrounding air is modeled as rotating fluid.
Table 1. Geometric properties of the pressure plates.

| Geo (Fig.2) | Mass (kg) | Convection surface (m²) | Compactness (m⁻¹) |
|-------------|-----------|-------------------------|------------------|
| (1)         | 25.15     | 0.216                   | 65.25            |
| (2)         | 18.88     | 0.353                   | 135.77           |
| (3)         | 19.52     | 0.349                   | 129.74           |
| (4)         | 19.54     | 0.338                   | 125.18           |
| (5)         | 20.14     | 0.328                   | 118.41           |

Figure 5. Clutch Numerical Model and Boundary Conditions.

Rotational periodic pattern is defined to model the real clutch pressure plate geometry. Difference of translational and rotational periodic pattern is shown in Figure 6. Rotational periodic pattern (Figure 6b) results same geometry as the real one (Figure 6a).

Figure 6. Real Pressure plate Top view (a), Rotational periodic (b), and Translational periodic (c).

The mesh structure for the computations is shown in Figure 7. The mesh model consist total 3.5x10⁶ tetrahedron elements (Table 2). In Figure 7 inflation details and single pressure plate mesh structure is highlighted.
Table 2. Number of Elements and nodes of the domains.

| Domain  | No of Elements | No of Nodes |
|---------|----------------|-------------|
| Fluid   | 1.6x10⁶        | 8.5x10⁵     |
| Solid   | 1.9x10⁶        | 8.5x10⁵     |
| Total   | 3.5x10⁶        | 1.7x10⁶     |

Inflation (in the fluid Boundary Layer) is defined for better resolution of heat transfer from pressure plate to air [12]. Mesh face sizing is defined as 1 mm at contact faces. Due to complex geometry and cyclic periodic pattern application, tetrahedron mesh structure is used, and solid mesh is conformal to fluid mesh structure.

3. Results and Discussion

3.1. Validation of the Numerical Model

Numerical model is verified with a bench test measurement. Bench test aims to determine the global coefficient of heat convection at the interface between the pressure plate and surrounding air in the clutch housing. In the bench test experimental work, clutch system is exposed to friction till the pressure plate mass temperature reaches up to 300 °C, and then it was cooled down to the ambient temperature. During the cooling phase, temperature drop of pressure plate and clutch housing air are measured by type K thermocouples. Computation duration is maximum 3 hours. On the other hand, experimental bench test duration including set-up is up to 10 hours and more costly. In Figure 8, the comparison of the output of numerical method followed in this study and the output of experimental work is shown. The comparison is presented only for the non-ventilated conventional pressure plate, since the real part is only available for this model. The difference is in acceptable level. Even the computation is a steady state condition; it gives similar results in comparison with transient experimental work. In the transient experimentation, global convection coefficient is obtained from the real temperature variation of the pressure plate and clutch housing air [8].
3.2. The effect of duct design on heat transfer

In Figure 9 comparison of convection coefficient and its multiplication with convective surface is shown. Implementing ventilation into clutch housing does not only improve the convective surface, but also enhances the convection coefficient. Radial airfoil shaped duct design shows the best cooling performance in overall among all duct shapes. Compared to conventional non-ventilated version, it enhances the overall cooling performance; \( h_x s \) (WK\(^{-1}\)) by up to 154% at 550 rpm idle speed, 137% at 1200 rpm, and 142% at 2000 rpm.

Temperature distribution at 2000 rpm of Geo 1 (Conventional non-ventilated) and Geo 3 (radial duct airfoil) is shown in Figure 10. Temperature of Geo 3 reduces to 70 °C at pressure plate surfaces in contact with fluid air, and its mass temperature is lower than Geo 1 conventional non-ventilated.
4. Conclusion

To determine the effect of duct shape on the cooling performance of the clutch casting pressure plate, the numerical 3D conjugate CFD simulation is presented. The approach is cost effective and time saving, and eliminates the need for prototyping to estimate the cooling performance. This study is limited to thermo-fluidic performance but can be extended for the mechanical performance. Conclusions drawn from this study are summarized as follows.

- Convection coefficient of the conventional clutch pressure plate increases linearly with the rotational speed ($R^2 = 0.9986$).
- Applying ventilation to conventional clutch pressure plate improves convection coefficient at minimum 29% within the operating range of the vehicle.
- Applying ventilation into conventional clutch pressure plate improves the convection coefficient at maximum, by 44% at idle speed, by 36% at 1200 rpm and by 37% at 2000 rpm.
- Variation of inlet and outlet duct width has negligible effect on overall cooling performance.
- Radial airfoil shaped duct design shows the best cooling performance in overall among all duct shapes, and enhances the overall cooling performance ($h_{xs}$; convective heat transfer coefficient x convection surface) by up to 154% at 550 rpm idle speed compared to conventional non-ventilated version.
- Airfoil shaped radial duct design improves the the convection coefficient by 11% at idle speed compared to non-airfoil radial version.
- These findings suggest that there is an opportunity to reduce the weight of the heaviest part of the clutch system by up to 25%, while also improving its cooling performance. The results of this research also support the target of greenhouse gas emission reductions in heavy duty vehicle applications.

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