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Thermal Modelling of a Disc Brake System at Different Vehicle Weights and Constant Speeds

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Abstract

In this study, three-dimensional thermal modelling has been made for a disc brake mechanism of an automobile by using the COMSOL Multiphysics programme. Frictional heat has been calculated by means of multibody dynamic module, and heat distribution has been calculated by means of heat transfer module. In such model, an automobile with an initial speed of 25 m/s and 35 m/s has been decelerated by -10 m/s^2 braking speed, and the braking state has been realised at time interval of 2-4 s. Thermal analysis has been made for two different vehicle weights (1200 kg-1500 kg) under the same braking scenario. The results obtained from the thermal analysis have shown similarity to various studies carried out within the literature. An increase of 300 kg in vehicle weight has resulted in a temperature increase by 3.33% during motion at 25 m/s vehicle speed, and by 6.03% during motion at 35 m/s vehicle speed. According to temporal temperature change, maximum pad temperature has been obtained at the 4th second; and in the case that the vehicle with a weight of 1500 kg and moving at 35 m/s speed has braked, maximum pad temperature has been obtained as 450 K.

Keywords: Brake system, COMSOL Multiphysics, Disc brake, Thermal analysis.

Research Article

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1. Introduction

Disc brake systems are important machinery equipment, which enable the vehicles to decelerate or stop during the cruise. Conditions of the contact between brake lining and disc surface are of great importance in such brake systems in terms of braking performance. In disc brake systems, when the brake pedal is pressed, the location of the hydraulic fluid included in the system changes, a friction force occurs for the brake discs by the help of piston and the vehicle is decelerated with the effect of such force. During braking, kinetic energy of the vehicle is transformed into heat energy and most of such heat is swallowed by the brake lining and the disc [1]. Transformation of mechanical energy into heat during braking results in temperature increase in the lining and the disc [2]. Braking distance of the vehicle is affected by the efficacy of the brake system and braking performance [3]. Primary factors affecting the braking performance can be listed as follows: vehicle weight, design of brake system, state of hydraulic and mechanic components of the brake, environmental conditions that may affect the brake system, road conditions, state of tires, adhesion coefficient between the wheel and road surface, and temperature of the brake system elements [4-8]. The high temperature occurring during braking causes sudden friction loss between disc and lining, early wearing of the brake lining, evaporation of the brake fluid, thermal deformations and brake vibrations [9-12]. With relation to thermal analysis of the brake linings, Wolff [13] has examined the modelling studies and stated that thermal analyses are made by means of one-dimensional, two-dimensional and three-dimensional models, the latter of which are more complicated. Zhu et al. [14] have examined three-dimensional temperature distribution of a brake lining during an emergent braking by means of finite elements method, approximate integral method, green functions method, Laplace transformation method and integral transformation method. Among such methods, the integral transformation method has provided more reliable results when compared to the others. Yevtushenko and Grzes [15]; Yevtushenko et al. [16]; Yevtushenko and Grzes [17]; Yevtushenko and Grzes [18] have examined the effect of different temporal friction power profiles on temperature distribution on brake disc and brake lining. Analysis of heat generation between disc and lining has been made by means of finite elements method by using COMSOL Multiphysics programme [19-20]. In this study, friction heat has been calculated by composing a finite element brake disc and lining model by means of COMSOL Multiphysics programme, and thermal analysis of brake disc-lining system has been made.

2. Disc Brake Model

In this study, 3D thermal analysis of the disc-brake system of an automobile has been made by means of COMSOL Multiphysics programme for 3 different weights during deceleration by 2 seconds with a deceleration speed of 10 m/s^2 while cruising at 25 m/s and 35 m/s speeds. Selected speeds are the ones commonly used by the automobiles on intercity roads. Firstly, 3D geometry of the disc-lining system has been formed. Diameter of the disc has been determined as 0.14 m, lining thickness has been determined as 0.05 m and disc thickness has been determined as 0.013 m (Figure 1). The model also contains the heat transmission in disc and lining by means of heat transfer (transient heat transfer equation) equation. Heat distribution from disc and lining surfaces towards the environment has occurred by both convection and radiation. In Figure 1, brake disc-pad geometry has been given. In Table 2, thermal properties of the materials used in modelling have been given [21].

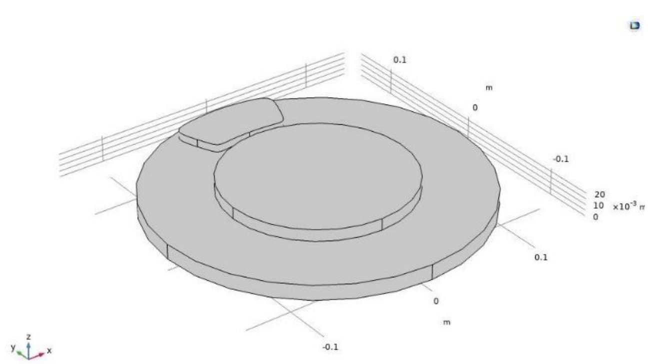

Fig. 1. Brake disc-pad geometry

Table 1. Input parameters of the model

Parameter	Symbol	Explanation	Value given to the model
Initial speed of the vehicle	v_o	$90\ (\frac{km}{h})$	$25\ m/s$
Diameter of the wheel	r_w	$0.3\ (m)$	$0.3\ m$
Friction coefficient	μ	0.7	0.7
Braking force	F_b	$200\ (N)$	$200\ N$
Angular velocity of the wheel	ω	v_0/r_w	$83.33\ 1/s$

Table 2. Thermal properties of the materials used in disc-pad system

Property	Explanation	Disc	Pad	Air
$\rho\ (kg/m^3)$	Density	7870	2000	1.170
$C_p (J/(kgK))$	Specific heat at constant pressure	449	935	1100
$k\ (W/(mK))$	Thermal conductivity	82	8.7	0.0026
ε	Surface emissivity	0.28	0.8	
$E\ (Pa)$	Young's module	200e9		
nu	Poisson's ratio	0.3		

The geometry is meshed with the free triangular element to transform the continuous structure into a finite number of elements. The described model contains approximately 15620 elements and 20466 degrees of freedom. The maximum and minimum element size of the created mesh is 0.0098 m and 4.2e-4 m [22]. Transient analysis was carried out.

Brake power is obtained by means of derivation of kinetic energy of the vehicle as per time. In equation no. (1), “m” refers to the vehicle weight/mass, and “v” refers to the velocity/speed.

$$P_b = -\frac{d}{dt}\left(\frac{mv^2}{2}\right) = -mv\ \frac{dv}{dt} \tag{1}$$

In Figure 2, speed scenario of the vehicle has been given. A vehicle cruising with a speed of 25 m/s has braked at the 2nd second of the motion and decreased its speed down to 5 m/s at the 4th second, and continued its motion at 5 m/s speed.

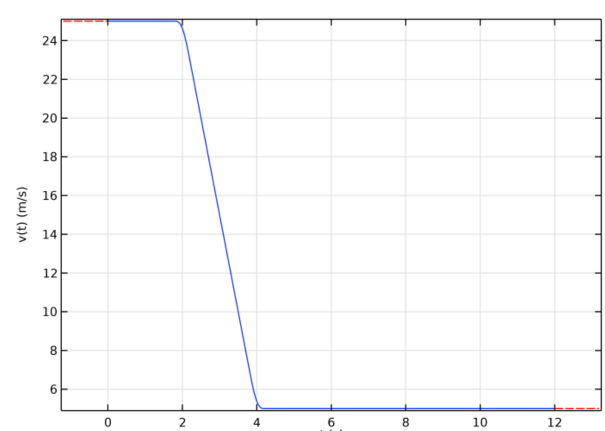

Fig. 2. Vehicle speed scenario identified for the thermal model

Contact pressure between the disc and pad surfaces:

$$p = \frac{P_b}{\mu\, v} \tag{2}$$

In equation no. (2), “μ” ($\mu = 0.3$) refers to friction coefficient, "v" refers to vehicle velocity, "P_b" refers to the heat energy occurring during braking. Velocity vector of the brake disc:

$$V_d = \frac{v}{R}(-y, x) \tag{3}$$

Amount of heat generated after braking and released to the environment:

$$W_{prod} = \int_0^{t_0} Q_{prod}dt \tag{4}$$

$$W_{diss} = \int_0^{t_0} Q_{diss}dt \tag{5}$$

3. Results and discussion

In Figure 3, temperature distribution between brake disc and pad at 4th and 6th seconds of the braking has been shown for vehicle weights of 1200 kg and 1500 kg. The increase in vehicle weight has caused the temperature on brake disc and pad surfaces to increase. At the diameter point on which the pad has been located, temperature points have been obtained unambivalently. Moreover, temperature distribution is uniform. The results obtained show similarity to the results of the analysis carried out by Belhocine et al. by means of ANSYS programme [22].

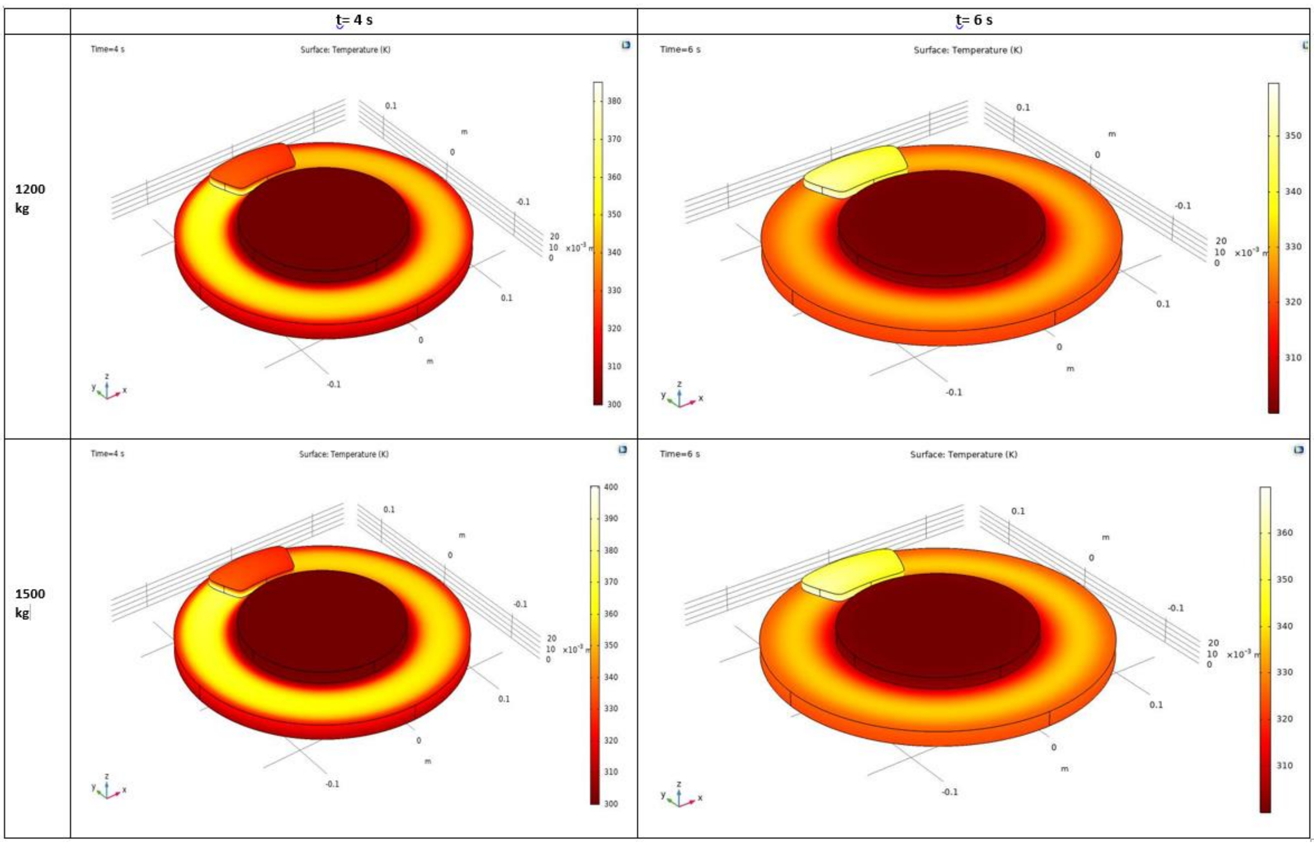

Fig. 3. Temperature distribution occurring on brake disc and brake lining at the 4th and 6th seconds for different vehicle weights (25 m/s)

4 th second is the duration in which the braking ends. After such duration, a rapid cooling trend has been detected on brake disc-pad surfaces of both vehicle weights up to the 6th second. As a result of the thermal analysis; for the vehicle weight of 1200 kg, a temperature of 360 K has been obtained between brake disc and pad at the 4th second of the motion, and a temperature of 330 K has been obtained at the 6th second; for the vehicle weight of 1500 kg, a temperature of 380 K has been obtained between brake disc and pad at the 4th second of the motion, and a temperature of 340 K has been obtained at the 6th second.

In Figure 4, change of heat generated and released during braking has been given. Braking has occurred at 2-4 second interval of the vehicle motion. Rate of increase of the heat generated and released at 2-3 s interval of braking is higher when compared to 3-4 s interval. Such situation has occurred due to the change in thermal resistance of disc-lining surfaces. As braking is completed after 4th second, the amount of heat generated has remained constant. Although the amount of heat has continued to increase, whole of the heat generated during braking could not be released to the environment. The amount of heat generated during braking and released to the environment has increased as the vehicle weight has increased. The temporal changes of heat generated and released at vehicle speeds of 1200 kg and 1500 kg show similarity to each other.

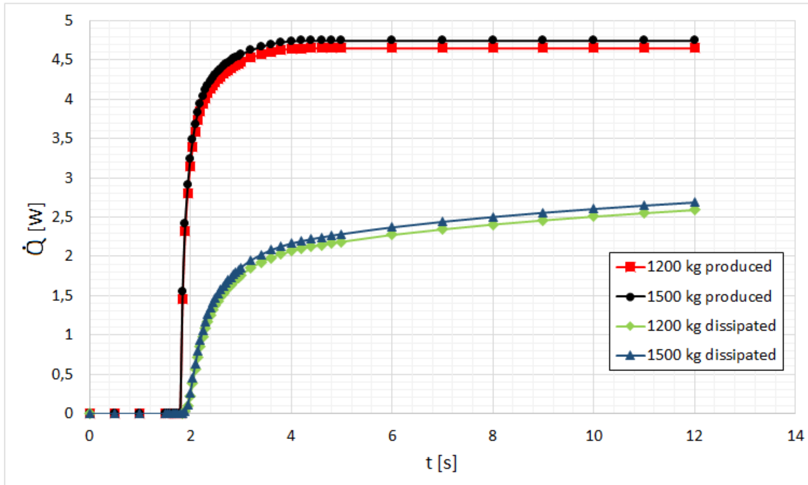

Fig. 4. Temporal changes of heat generated for brake systems of different vehicle weight stand released (25 m/s)

In Figure 5, surface temperatures of vehicles with two different weights at the end of braking (4th second) have been shown for speeds of 25-35 m/s. The increases in vehicle speed and weight have caused brake disc-pad temperature to increase at the end of the braking. As a result of the thermal analysis, brake disc-pad surface temperatures are uniform. According to the results of the analysis, for the vehicle weight 1200 kg; a brake disc surface temperature of 360 K has been obtained at vehicle speed of 25 m/s and a brake disc surface temperature of 398 K has been obtained at vehicle speed of 35 m/s. Moreover; for the vehicle weight 1500 kg; a brake disc surface temperature of 373 K has been obtained at vehicle speed of 25 m/s and a brake disc surface temperature of 422 K has been obtained at vehicle speed of 35 m/s.

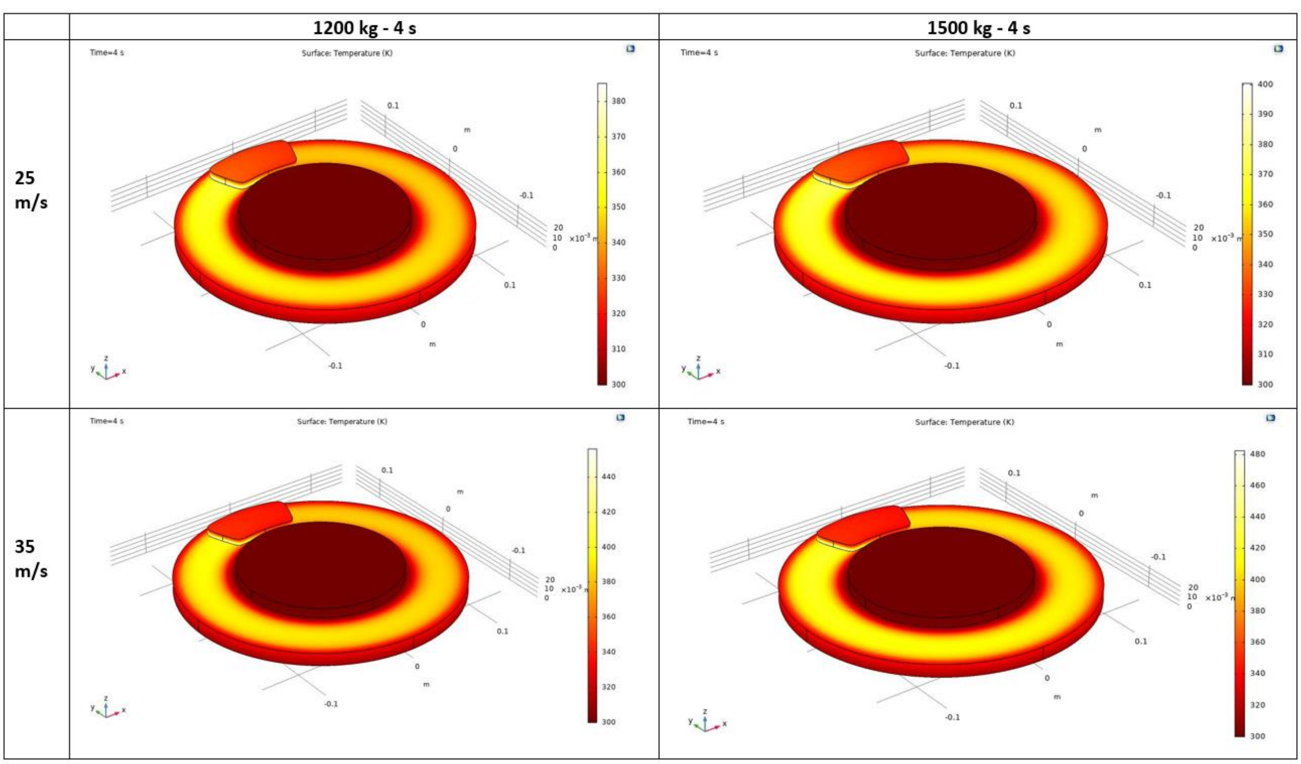

Fig. 5. Distribution of temperature occurring on brake disc and brake lining at the end of braking for different vehicle speeds

In Figure 6, temperature distribution on brake disc and pad at the 6th second of vehicle motion has been shown. As braking is completed at the 4th second, temperatures of brake disc and pad have decreased until the 6th second. According to the results of the analysis, for the vehicle weight 1200 kg; a brake disc surface temperature of 329 K has been obtained at vehicle speed of 25 m/s and a brake disc surface temperature of 348 K has been obtained at vehicle speed of 35 m/s. Moreover; for the vehicle weight 1500 kg; a brake disc surface temperature of 337 K has been obtained at vehicle speed of 25 m/s and a brake disc surface temperature of 360 K has been obtained at vehicle speed of 35 m/s. At the same time interval, a state of cooling by 50 K has occurred on brake disc at vehicle speed of 35 m/s, and a state of cooling by 62 K has occurred on brake disc for the vehicle weight 1500 kg.

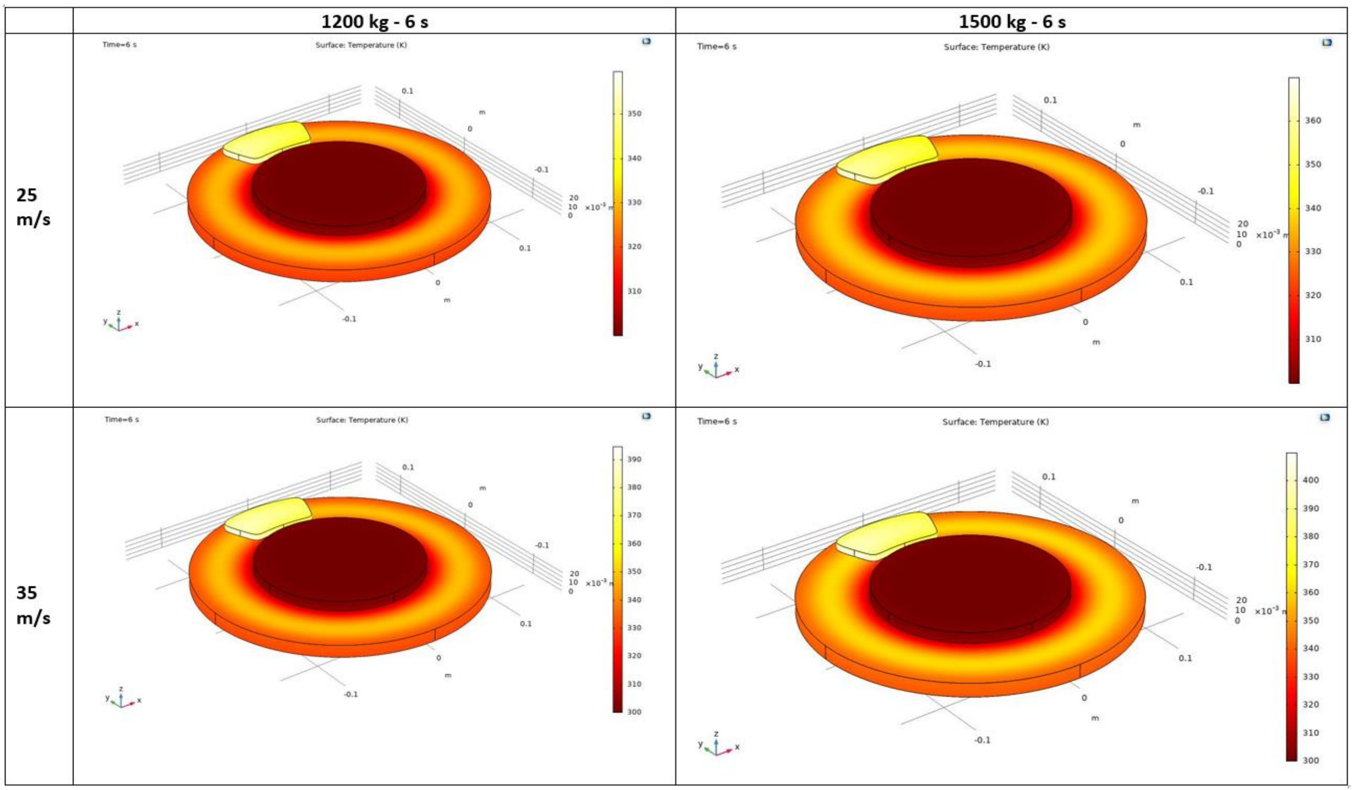

Fig. 6. Temperature distribution on brake disc and pad at the 6th second for different vehicle speeds

In Figure 7, temporal temperature change has been given for different vehicle speeds. The temperature distribution has been prepared according to two different vehicle weights. In the graphic, “x” axis shows the distance from the Wheel centre towards the circumference, “y” axis shows the time, and “z” axis shows the temperature. When temporal temperature distribution is observed, it is seen that the temperatures have increased rapidly from the beginning of braking until the completion. It is observed that there has been a rapid cooling on pad surface after the completion of braking (4th second). According to the graphics, when a vehicle with a weight of 1200 kg makes braking while cruising at 25 m/s speed, a maximum pad temperature of 380 K has occurred; and a maximum pad temperature of 420 K has occurred while cruising at 35 m/s speed. When a vehicle with a weight of 1500 kg makes braking while cruising at 25 m/s speed, a maximum pad temperature of 400 K has occurred; and a maximum pad temperature of 450 K has occurred while cruising at 35 m/s speed.

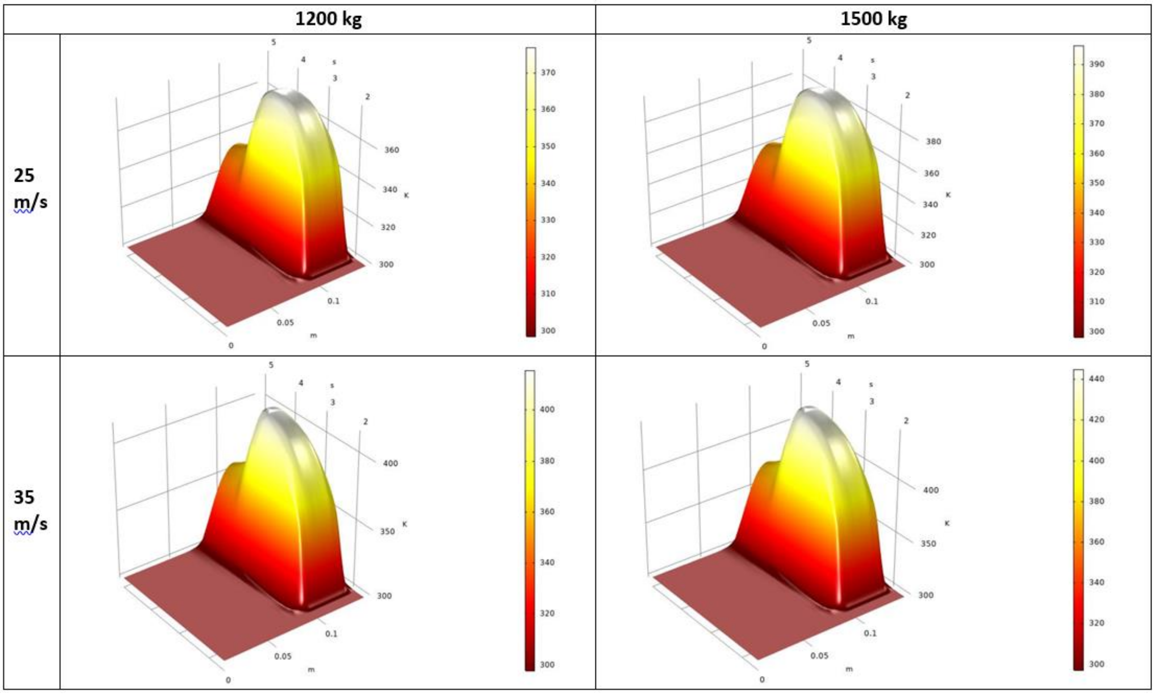

Fig. 7. Temporal temperature distribution for different vehicle speeds

4. Conclusion

In this study, thermal analysis of a disc brake system has been made by using COMSOL Multiphysics programme. The increases in vehicle weight and speed have caused the temperatures to increase during braking. Temperatures between brake disc and pad have been obtained as uniform. The results obtained Show similarity to the results of certain researches that study on the same subject. Temperatures generated on disc surface during braking and released to the environment have increased rapidly. While the amount of heat generated after the 4th second of the vehicle motion has not increased, the amount of heat released from disc surface to the environment has continued to increase. The increase in vehicle weight during cruising with 25 m/s speed at the 4th second when the braking is completed has caused the disc surface temperature to increase by 3.33%. The increase in vehicle speed (35 m/s) under the same conditions has caused the brake disc temperature to increase by 6.03%. When the modeling results obtained were evaluated, a maximum temperature of 420 K was obtained in a car with a mass of 1800 kg in a model with the same disc-pad dimensions and braking acceleration [23]. Considering that the vehicle masses used in the model are 1200 kg and 1500 kg, it can be said that the results show similarities with the literature.

Conflict of Interest Statement

The author declares that there is no conflict of interest.

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