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BRAKE DISC TEMPERATURE PREDICTION IN FADE TEST USING COMPUTATIONAL FLUID DYNAMICS AERO-THERMAL SIMULATION

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Abstract

The Brake fade test is one of the critical validation activities during brake sizing in new project development. The size of the brake disc is designed according to the satisfaction of severe braking applications such as downhill, city traffic with heating and cooling cycle. During braking, the vehicle's kinetic energy is converted to thermal energy which is absorbed by both disc rotor and brake pad. For cast iron rotors coupled with organic/semi-metallic pads the coefficient of friction may be dropped to as low as 0.2 beyond 350°C, resulting in brake fade. This highlights the need for thermal performance evaluation of disc rotors. The rotor temperature also has a greater impact on the wear and tear of the brake pad material which directly affects the frequency of the service or replacement. The brake disc thermal performance is evaluated by repeated braking and cooling cycles which helps the brake disc achieve thermal equilibrium. Evaluating heat generation and dissipation characteristics in this phase would help us determine maximum temperature of rotor for all vehicle driving conditions. In this work, an attempt has been made to predict the temperature of the brake disc for light passenger vehicles with a standalone brake disc and vehicle level Computational Fluid Dynamics (CFD) simulation. Also, to further establish a good correlation both test track and vehicle level CFD results are compared with an analytical solution given by Rudolf Limpert. The CFD results shows a 97% correlation with vehicle level physical test.

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1. Introduction

Overheating of the brake system components is a safety issue that can cause serious problems such as decreased friction coefficients, i.e., brake fading. Moreover, such overheating can result in generating brake judder and / or brake squeal, increasing wear, thermal cracking and even brake fluid vaporization [1].

Good design of braking system is very important in many aspects such as Performance, Noise Vibration and Harshness (NVH), Safety of nearby components. Wang, Peiyu, et al [2] in their study carried out thermal degradation performance check, where they evaluated the disc temperature from repeated braking and cooling cycles. Chaudhary et al [3] have discussed about the importance of studying complex heat transfer of brake disc in successive braking conditions. Cai, Ran, et al [4] in their work studied about different types of coating that could combat corrosion issues in brake disc. They have decreased the number of stops in brake dynamometer from 15 to 6 to maintain the test temperature below 500°C, this indicates the repeated braking strongly influences in brake disc temperature. Chopade [5] et al in their work have studied the influence of the vane profile on ventilated disc cooling. The boundary condition and heat transfer mechanism used in the CFD analysis is similar to Chaudhary et al [3].

Conducting the downhill & city traffic [6] for all the vehicles will consume time and cost. Instead brake fade test can be done on test tracks with repeated braking to imitate the worst-case scenario.

On the other hand, the revolutionary advancements in computers strengthen the simulation prediction capability closer to the real-world applications. Where the Computational Fluid Dynamics simulations are extensively used in many case studies. Vdovin et. al [7] investigated the changes in heat flux during downhill and the experimental validation is closer to computational fluid dynamics simulation. The results reveal that the inner vane of the brake disc design contributes more on its cooling performance specifically convection mode of cooling. The transient thermal analysis on brake disc in heavy duty vehicles has been studied with the objective of weight reduction. However, the article does not reveal the effect in passenger vehicles [8]. Pranta et. al has investigated the structural and thermal behaviour of modified brake disc with constant thermal assessment which is less efficient compared to the time dependant transient thermal analysis [9]. The transient thermal simulation for brake disc has been carried out for a standalone configuration and correlated against the bench test. However, the impact of front and surrounding components is critical in brake disc heat transfer [10]. The air flux generated by the aerodynamics of bumper and wheelhouse that

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has to be taken into consideration for accurate capturing of brake disc heat transfer coefficient [11]. Although the cited article is more than 10 years old, it has been retained because it specifically considers the influence of surrounding components in the simulation. This aspect is often simplified or neglected in similar studies due to the high computational and processing cost involved, making the reference still relevant to the present work. Computational Fluid Dynamics has gained recognition as a versatile and powerful tool for predicting brake disc temperature, particularly due to its ability to accurately model all three modes of heat transfer conduction, convection, and radiation [12, 13]. The advancement of computational capabilities has played a pivotal role in enhancing the accuracy and efficiency of CFD simulations. Modern computational resources now enable the use of high-fidelity turbulence models, adaptive meshing techniques, and advanced thermal boundary conditions, allowing for a more realistic representation of the intricate heat transfer phenomena occurring in braking systems. Parallel processing and cloud-based computing further extend the scope of CFD by reducing computation times, enabling simulations of more complex scenarios and larger datasets than ever before.

By leveraging these advancements, this study aims to utilize CFD to predict brake disc temperatures and validate the results against empirical test data. This approach underscores the potential of CFD as a cost-effective and reliable alternative to traditional testing methods, while also highlighting its capability to provide deeper insights into the thermal behaviour of automotive braking systems under dynamic operating conditions.

The simulation method used in this study is similar to the existing articles [7, 13, 14], where the authors investigated the thermal performance of brake disc using numerical methods. In addition to the thermal performance researchers also focused on complex test profiles such as repeated braking [15] and 1D model for brake energy input prediction [16]. The main difference being the boundary conditions calculations are different such as temperature dependant material properties, heat flux and validation procedure in this work is repeated braking applications with bench and vehicle level test.

The following contents are organised as follows: the explanation of brake test procedure is discussed in section 2 and followed by detailed descriptions used in the Computation fluid dynamics in section 3. The bench and vehicle level simulations are compared in different aspects in section 4 followed by conclusion at the end.

2. Brake fade test

The thermal performance of the brake disc is calculated/predicted in the upstream stage of the new project development with various criteria. Brake fade test is one of the crucial activities where the frequent brake applications are applied and the rotor temperature is measured to understand its capacity to absorb and dissipate the heat energy in various

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situation such as downhill and city traffic conditions. This evaluation is important to understand whether the brake disc is getting enough air circulation to evacuate the heat into atmosphere. Temperature of the brake disc greatly influence the friction pad wear rate. Inefficient heat dissipation in brake disc could increase the surface temperature of brake-pad, accelerated wear may start 50°C to 100°C prior to brake fade temperature for organic/semi-metallic brake pads.

3. Governing Equations and Input Calculation

Thermal behaviour in brake disc is studied by solving the fluid flow and heat transfer governing equations in CFD. Due to the nature of repeated braking, the brake disc's temperature reaches significantly high levels. This simulation incorporates all three modes of heat transfer: conduction, convection, and radiation, to accurately represent the heat absorption and dissipation characteristics of the brake disc [17, 18]. Although the referenced article is more than 10 years old, it has been included because it accounts for all modes of thermal heat transfer in the simulation, an approach that remains fundamentally accurate and relevant to the present work.To precisely model conduction, we have utilized temperature-dependent material properties for the brake disc. The material properties of the brake disc change with temperature, significantly affecting its thermal behaviour. Figure 1 shows the temperature dependent material properties. The Convective heat transfer coefficient (HTC) is computed based on airflow generated by linear wind speed and rotor's angular speed. Given the high temperatures experienced by the brake disc, radiation becomes a significant contributor to heat dissipation. The thermal radiation emitted by the hot brake disc plays a crucial role in the overall heat transfer process. By integrating these three modes of heat transfer, the simulation provides a comprehensive thermal analysis of the brake disc under repeated braking conditions.

Siemens Star CCM+ [19] is used to simulate the heat generation and dissipation characteristics of brake disc. Conservation of Mass, Momentum and Energy are the three main governing equations solved.

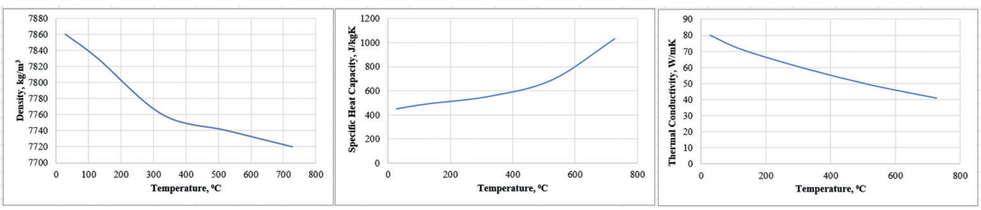

Fig. 1. Brake disc's Temperature dependent material property

Conservation of Mass (Continuity Equation): $\nabla \cdot (\rho u) = 0$

Conservation of Momentum (Navier-Stokes Equation): $\rho\left(\frac{Du}{Dt}\right) = -\nabla p + \mu\nabla 2u + f$

Conservation of Energy: $\rho C_p\left(\frac{DT}{Dt}\right) = \nabla \cdot (k\nabla T) + Q$

Where:

$\rho - Fluid\ density\ \left(\frac{kg}{m^3}\right)$ $f - body\ force\ per\ unit\ volume\ (N)$

$u - velocity\ vector\ (\frac{m}{s})$ $C_p - Specific\ heat\ capacity\ \left(\frac{J}{kgK}\right)$

$p - pressure\ (MPa)$ $T - Temperature\ (°C)$

$\mu - dynamic\ viscosity\ \left(\frac{Ns}{m^2}\right)$ $k - thermal\ conductivity\left(\frac{W}{mK}\right)$

$Q - Heat\ source\ (j)$

About 90% of the kinetic energy is absorbed by brake disc as thermal energy and rest by pad [20]. Heat generated during braking is applied as heat flux across rotor-pad interface. The heat flux is calculated from power generated during braking.

$$Power\ generation\ during\ braking\ (per\ rotor) = \left(\frac{Brake\ Torque\ (Nm) \cdot Angular\ Velocity\ (rad/s)}{2}\right)$$

$$Heat\ flux = \left(\frac{Power\ generated\ during\ braking\ (per\ disc\ rotor)}{Rotor, pad\ contact\ area}\right)$$

3.1. 3D Model considered for CFD Analysis:

The geometry preparation for Conjugate Heat Transfer (CHT) analysis of a passenger car's brake disc involves creating a detailed 3D model that includes the disc rotor, wheel, tire, bumper, fender, and wheel cap. Front portion of right side of the vehicle alone is considered to significantly reduce the cell count and focus the analysis on the localized thermal behaviour. The virtual wind tunnel surrounding the components is carefully designed to be sufficiently large, ensuring it does not interfere with the airflow around the domain, providing an accurate representation of the real-world environment. The boundary conditions are meticulously defined: slip conditions are applied to the right side and top walls to simulate an unobstructed flow, while the left wall is defined as a symmetry plane to represent the mirrored half of the vehicle. A no-slip moving wall condition is applied to the floor to simulate the relative motion between the car and the road surface accurately. The linear and angular velocity

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in the simulation is defined based on vehicle decelerating and accelerating between 100 and 0 km/h. The rotational motion of air around the disc rotor is defined by moving reference frame (MRF) [13]. The velocity inlet and pressure outlet are set for the inlet and outlet boundaries, respectively, to mimic the airflow entering and exiting the domain. Unsteady Reynolds Averaged Navier-Stokes approach was used to model the turbulence. These settings create a realistic simulation environment, crucial for accurately predicting the thermal performance of the brake disc under various operating conditions. This detailed geometry preparation and boundary condition setup are essential steps in ensuring the CHT analysis captures the intricate interactions between the solid and fluid regions around the brake system. As a first step to keep the 3D model complexity and simulation time minimum, only brake disc is considered for the analysis inside the virtual wind tunnel in phase 1. All boundaries of virtual wind tunnel except inlet and outlet is defined with slip wall condition as shown in Figure 2. In phase 2 the surrounding parts which affects the airflow – front quarter symmetric bumper, wheel cover, wheel and tire is considered for the analysis of the brake disc shown in Figure 3. The importance of wheel cover in cooling performance is discussed by Ratamero et.al [21].

Phase 1: Only Brake disc is considered for the analysis to predict temperature.
The length, width, and height of the enclosure is considered 5 times the rotor diameter to ensure the pumping flow from the rotor vanes doesn't interact with the boundary conditions.

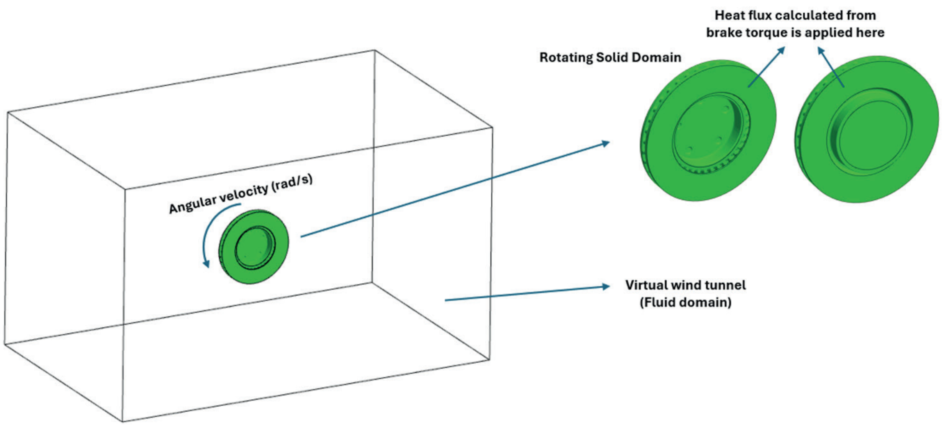

Fig. 2. CFD model of brake disc for phase 1

Phase 2: Brake disc along with wheel, tire and symmetry front bumper is considered to simulate real time vehicle condition.
Bongkeun Choi in their work have considered only the wheel, tire, knuckle, calliper and brake disc for evaluating thermal performance in brake disc and shown good corelation with measured results [22]. So, it is evident that only parts in proximity of disc rotor have major impact in its thermal performance. These articles [21, 22] focus on importance of thermal

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management of vehicle brake discs. Authors developed numerical procedure to replicate physical testing of brake discs under critical operating condition. In our present work we have developed numerical methods for evaluating thermal performance of brake discs that compares bench test setup and real time vehicle conditions with simplified surrounding parts. Hence, these articles are mainly cited for its close relevance.

Although these articles are more than ten years old, its inclusion is warranted because it introduces a unique methodology that remains distinctive within the field. The experimental design and analytical approach continue to serve as a benchmark, providing insights that newer studies still reference and build upon.

The vehicle model considered in phase 2 is limited to simplified bumper, wheel cover, wheel and tire along with brake disc is to reduce the complexity of the simulation without missing out the geometry parts that has significant impact on the disc rotor thermal behaviour. Also, the solid modelling in the simulation is limited to wheel and tire apart from disc rotor as they are in close proximity to the heat source (disc rotor). The main aim of modelling the simplified bumper is to mimic the real-world air flow condition to the CHT bodies (disc rotor, wheel and tire). Stationary parts like Knuckle, brake calliper and brake pad were not considered in this analysis to reduce the complexity of modelling stationary parts inside the MRF region of simulation

To prevent undesired interference from the tunnel boundary conditions, the virtual tunnel is 60 meters long, 40 meters wide, and 20 meters tall [7].

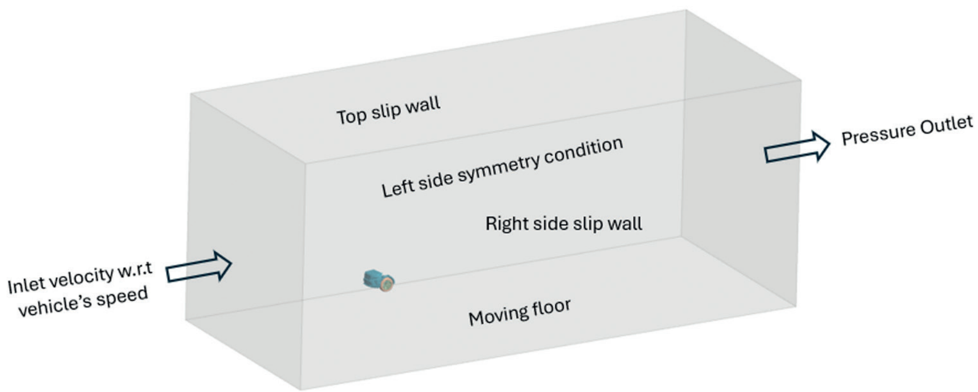

Fig. 3. CFD model of brake disc for phase 2

Polyhedral mesh in Star CCM+ is used to discretise the 3D model, prism layers are grown on walls to capture the boundary layers adjacent to wall region. The meshing ended with several million polyhedral cells to solve the governing equations. The meshed model is shown in Figure 4.

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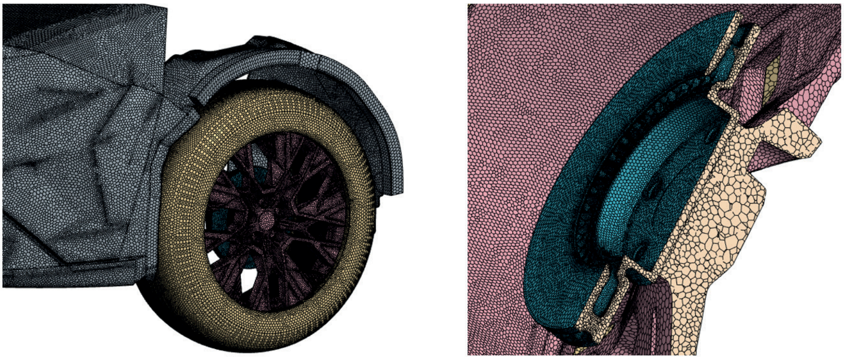

Fig. 4. Meshed model for CHT analysis

3.2. Inputs for CFD Analysis

Phase 1: Only Brake disc is considered for the analysis to predict temperature

The rotation of brake disc with respect to time is given as angular velocity representing corresponding braking and acceleration phase. No linear speed is given for this condition, The heat generated during braking phase is calculated from brake torque and applied on rotor-pad interface area. The heat flux application is removed during accelerating phase.

Phase 2: Brake disc along with wheel, tire and ½ front bumper is considered to simulate real time vehicle condition.

In phase 2 as we are attempting to replicate the real time vehicle scenario. Linear vehicle speed is simulated by defining linear velocity (transient) in fluid enclosure's front facing boundary. The heat generation and rotational speed of rotor, wheel and tire are same as in phase1 analysis. The boundary condition used in the CFD simulation is listed in Table 1.

Tab. 1. The boundary conditions for and phase 2

Boundary	Boundary type	Value
Inlet	Velocity inlet	Repeated 27.78 to 0, 0 to 27.78 m/s at 0.5*9.81 m/s^2
Outlet	Pressure outlet	Gauge pressure = 0 Pa
Floor	Moving wall	Repeated 27.78 to 0, 0 to 27.78 m/s at 0.5*9.81 m/s^2
Left	Symmetry	–
Top, Right	Wall	slip

The heat generated during braking is applied as heat flux across the cheeks of the brake disc. Heat flux is calculated from vehicle's brake torque and applied with respect to braking time in the simulation as mentioned in Figure 5. Heat flux application is removed during acceleration phase.

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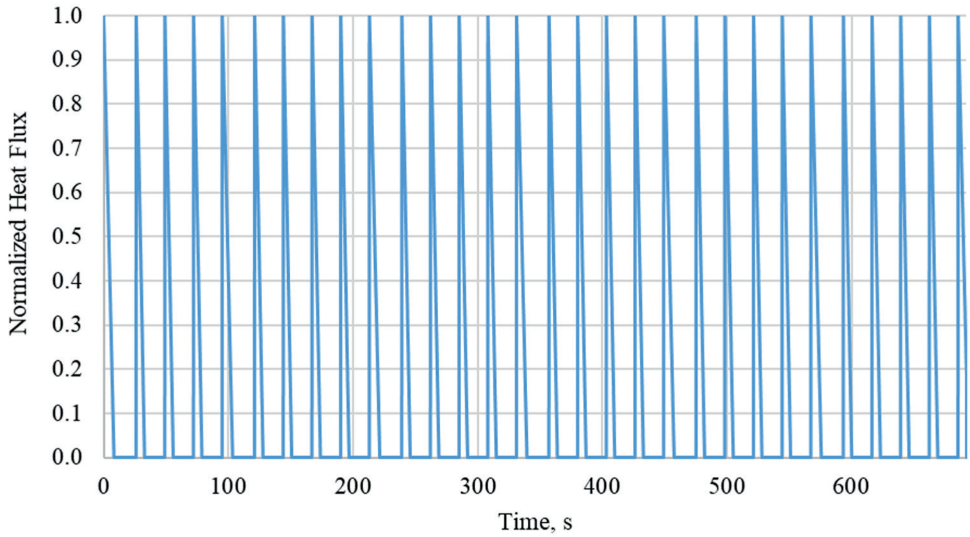

Fig. 5. Heat generation with respect to time

4. Results and Discussion

Phase 1: Only Brake disc is considered for the analysis to predict temperature

The comparison of the time vs temperature curves between CFD simulations and test results serves as a pivotal aspect of this study. The temperature of disc rotor is probed from the disc effective radius in the rotor-pad interface, As in Figure 6 the CFD analysis predicted a maximum temperature of 707°C, whereas the physical tests recorded a maximum temperature of 648°C. This difference of 59°C (8.5% deviation from physical test) between the CFD and test results is significant and warrants a detailed examination.

One primary factor contributing to this variance could be the idealized conditions inherent in the CFD simulation. In the simulation, since only the disc rotor was considered for analysis, the linear wind speed was not defined. Defining the linear air speed while simulating standalone rotor would result in direct impact of air over rotor and would resemble rotor soaking in air stream which would completely deviate real world scenario. This resulted in cooling by convection being solely from the rotation of the rotor, rather than from both linear wind speed and rotation as it happens in practical conditions. The absence of linear wind speed in the simulation underestimates the cooling effect, thereby resulting in a higher temperature compared to the test where the rotor is subjected to more comprehensive convective cooling.

Despite the differences, the overall trend observed in both the CFD and test results exhibits a consistent pattern of temperature rise and subsequent cooling during the braking cycles.

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This alignment in trends reinforces the validity of using CFD as a predictive tool for understanding the thermal behaviour of disc rotors under dynamic braking conditions. The higher temperature prediction in the CFD analysis suggests a need for calibration of the model parameters to better match the test data. The study is further extended to phase 2 with the aim of achieving a good time vs temperature corelation with test data.

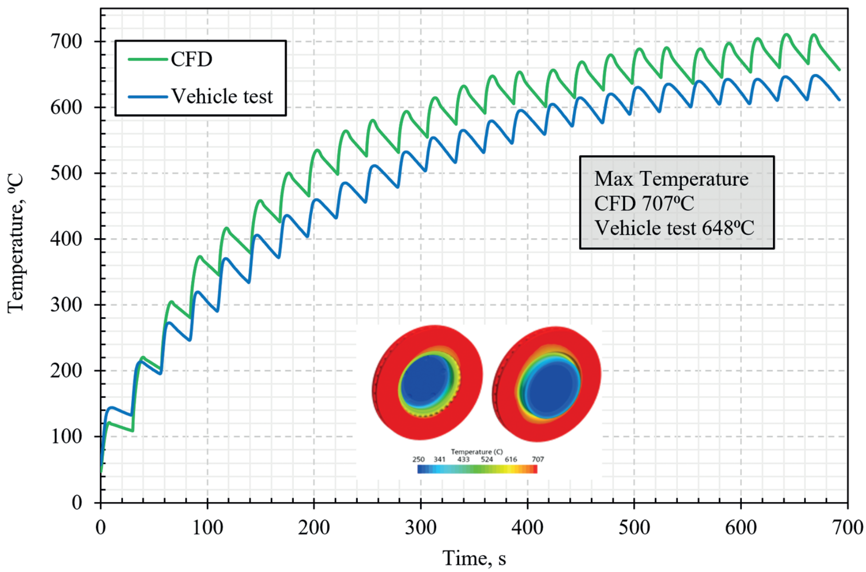

Fig. 6. Temperature profile during the fade test for phase 1

Phase 2: Brake disc along with wheel, tire and ½ front bumper is considered to simulate real time vehicle condition

In phase 2 of the study, the CFD analysis was enhanced by incorporating a simplified front bumper, wheel, and tire along with the disc rotor. Additionally, both linear wind speed and rotational speed were defined for the rotor, aligning the simulation conditions more closely with the real-world scenario. As a result, shown in Figure 7, the maximum temperature predicted by the CFD analysis decreased to 632°C, compared to 707°C in phase 1. This improved prediction is much closer to the test results, which recorded a maximum temperature of 648°C. The inclusion of linear wind speed in phase 2 played a crucial role in accurately representing the convective cooling effect experienced by the rotor in practical conditions, thus reducing the temperature difference between the CFD simulation and the tests. This refinement underscores the importance of realistic boundary conditions in CFD modelling to enhance the accuracy and reliability of the predictions.

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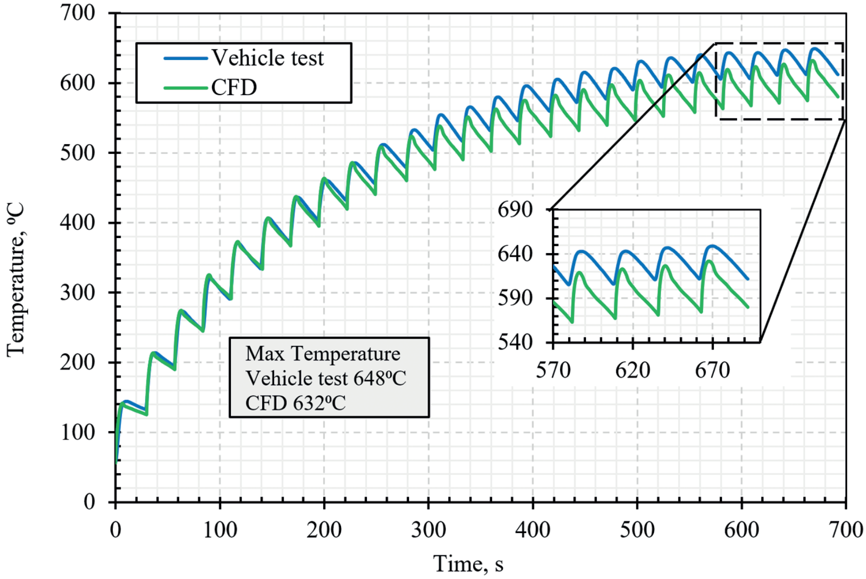

Fig. 7. Temperature profile during the fade test for phase 2

Figure 8 illustrate the temperature distribution on disc rotor, wheel and tire at the end of simulation cycle. Based on experts opinion the brake fade cycle is limited to 25 cycles within which most of passenger cars brake disc achieve thermal saturation. It can be seen clearly that the rotor-pad interface temperature distribution is more uniform with less temperature gradients at the end of 692 s.

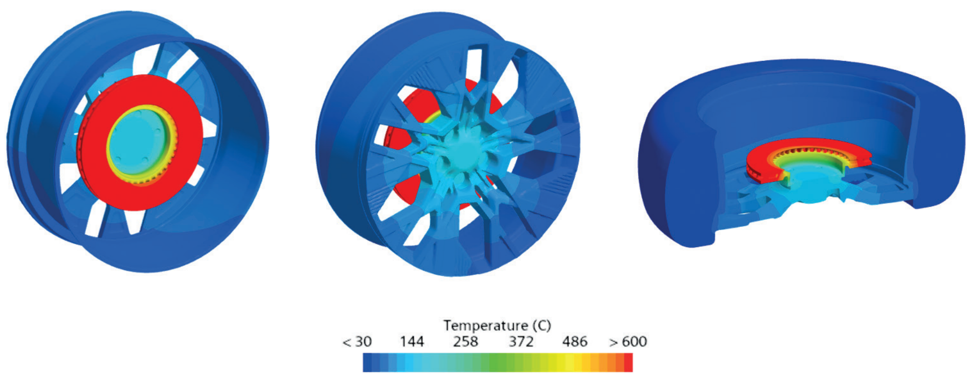

Fig. 8. Temperature distribution over brake disc, wheel and tire at the end of cycle (692 s)

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Also, from Figure 7 we can see that the maximum temperature rise, and temperature drop after cooling cycles (last 4 cycles) remain almost same after 600 s, indicating the disc rotor has attained thermal equilibrium after 600 s. So, any heat addition and cooling in similar pattern will have very little impact on rotor's temperature rise and drop.

The heat transfer coefficient (HTC) for disc rotor is shown in the Figure 9. In the context of a ventilated brake disc subjected to repeated braking and cooling cycles, the scalar image of HTC provides crucial insights into the cooling performance. The elevated HTC values observed near the vane roots (close to the rotor's inner diameter) can be attributed to the constriction of airflow at the entry region. This constriction restricts smooth air penetration, and the centrifugal forces generated by the rotor's rotational motion drive airflow towards the outer diameter, intensifying heat dissipation in the vane roots. At the displayed speed of 100 km/h, these HTC distributions highlight the influence of rotor geometry and airflow dynamics. Given the transient nature of the simulation, the airflow pattern, and consequently the HTC distribution, exhibits chaotic behaviour at any fixed time step. This suggests that localized high HTC zones are sensitive to temporal variations in airflow, underscoring the complex interplay between rotor motion and airflow patterns. The average heat transfer coefficient for inner vanes of the disc rotor is equal to 71.8 W/m^{2}K. The disc rotor HTC will be more useful when comparing the vane shapes that dissipates heat quickly.

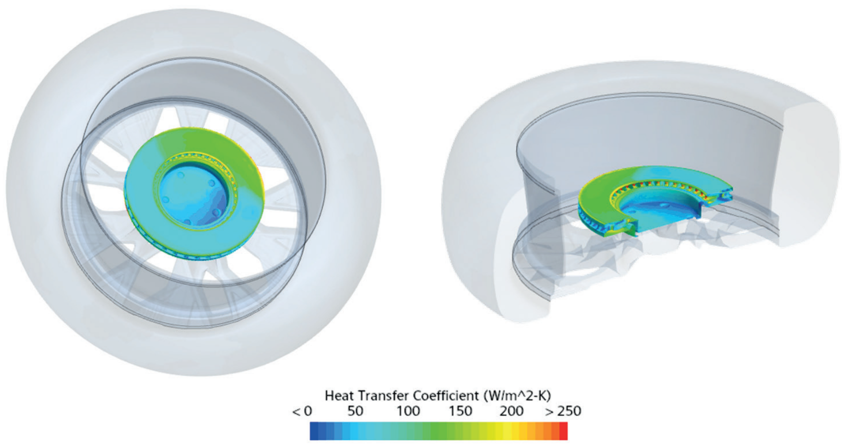

Fig. 9. Heat Transfer Coefficient for Disc rotor, at speed 100 km/h

Figure 10 (a) represents the velocity streamlines over the bumper, wheel cover and CHT bodies. The velocity streamlines help in understanding the air flow around the disc rotor. Figure 10 (b) shows air flow velocity distribution in a cut section parallel to floor. The highlighted zone in Figure 10 (b) shows localized high velocity zones. This is due to the proximity of wheel cover and tire, the air circulation created by tire rotation in the narrower gap between wheel cover

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and tire creates a localized high velocity zone. Though wheel covers block the linear wind, it can generate a localized air flow from tire rotation that may assist brake disc cooling.

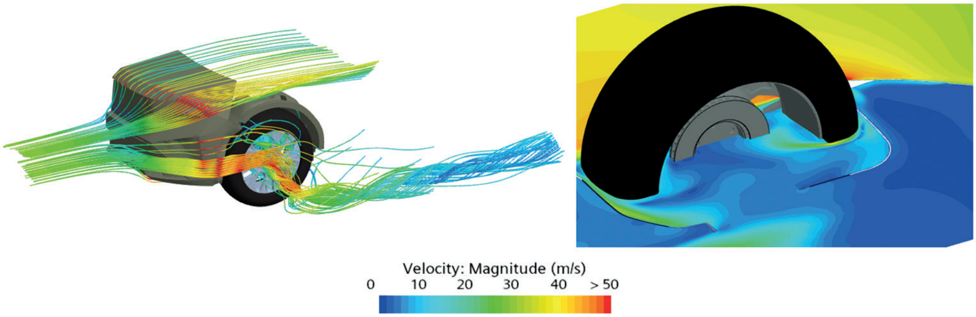

Fig. 10. a) Velocity streamlines, b) Sectional velocity magnitude at 100 km/h

4.1. CFD Results vs Analytical Solution:

Rudolt Limpert has derived an analytical expression to calculate brake disc's temperature for repeated braking and cooling cycles. The expression is well valid for lumped systems, where the temperature is uniform throughout the brake disc making it a function of only time and not space. To use this expression, heat transfer coefficient and thermal properties of brake disc are assumed as constants and single value is taken [23]. Although the referenced book is more than 10 years old, it has been retained because it is a standard reference with strong technical contributions in the brake field.

To compare time-temperature curve from the CFD study with the Limpert analytical solution, we need to check that the Biot number from the brake disc is not greater than 0.1, so that the brake disc can be considered as an aggregate system and the Limpert analytical expression can be used to calculate the temperature.

$$Biot\ Number\ (Bi) = \left(\frac{h * L_c}{k}\right)$$

Where,

h – *heat transfer coefficient of brake disc (W/m²K)*

L_c – *Characteristic length of brake disc (m) [Volume of rotor/Surface area of the rotor]*

k – *Thermal conductivity of brake disc (W/mK)*

From CFD results, average heat transfer coefficient of rotor at maximum speed is taken (71.8 W/m²K) and thermal conductivity is taken as 55 W/mK. The Biot number of brake disc is

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calculated to be 0.00542 which is lesser than 0.1. Hence Limpert's analytical expression can be used to calculate the temperature of brake disc for repeated braking and cooling cycles.

The Limpert's analytical solution for temperature calculation for repeated braking and cooling cycles is given by,

$$T(t) = \frac{\left(1 - e^{\left(\frac{-n_a * h * A * t_c}{m * C_R}\right)}\right) * \Delta T}{1 - e^{\left(\frac{-h * A * t_c}{m * C_R}\right)}} + T_{amb} \qquad \Delta T = \frac{q_o * t_s}{m * C_R}$$

n_a – *Number of brake applications*
h – *heat transfer coefficient of brake disc* (W/m^2K)
A – *Rotor surface area* (m^2)
t_c – *Cooling time cycle* (s)
m – *Mass of the rotor* (kg)
C_R – *Specific heat capacity* (J/kgK)
T_{amb} – *Ambient temperature* (°C)
ΔT – *change in temperature* (°C)
q_o – *Braking power absorbed by the rotor* (W)
t_s – *Braking time to stop* (s)

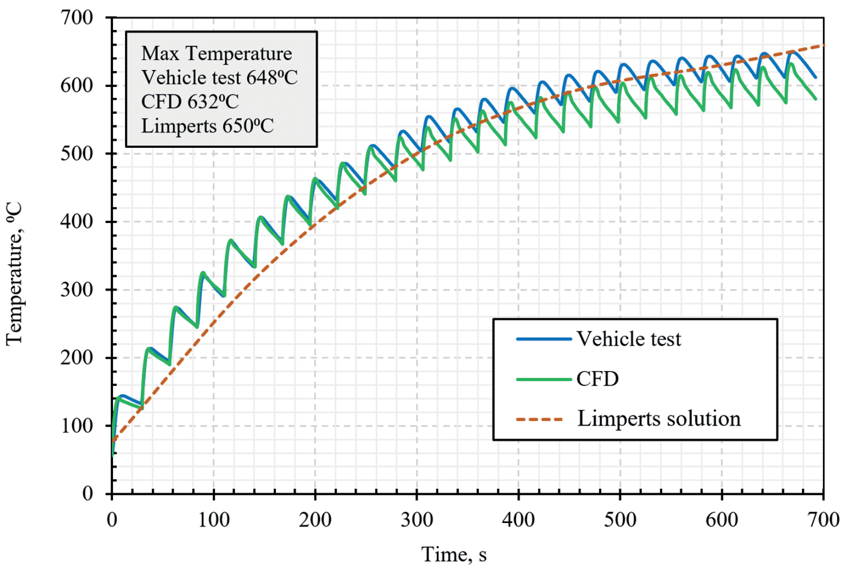

Fig. 11. Comparison of brake disc temperature using Limpert's analytical equation

The analytical solution for calculating temperature of brake disc in a repeated braking and cooling cycles given by Limpert shows very good correlation with test and CFD results, shown

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in Figure 11. The heat transfers coefficient used in getting analytical solution is taken from CFD results. Limpert has given an expression for calculating heat transfer coefficient for ventilated brake disc however the complexity arises when calculating the Reynold's number, which requires air flow velocity existing in vanes and not the speed of vehicles [23]. So, it is convenient to use the heat transfer coefficient from CFD results.

4.2. Grid Independency Study:

A CFD grid independency study was conducted to ensure the accuracy and reliability of the CHT analysis for a brake disc. Three different base sizes were employed for the rotor mesh: 3.5, 3 and 2.5 mm. The primary objective was to evaluate how variations in grid resolution affect the temperature distribution results. After running simulations for three mesh sizes, it was observed that the temperature results exhibited minimal deviation between the three scenarios. The temperature curve for all three mesh sizes is shown in Figure 12 This indicates that the solution had reached grid independence, implying that further refinement in the mesh would not significantly alter the results. Such consistency reinforces the credibility of the simulations, confirming that the chosen mesh densities are adequate for capturing the thermal behaviour of the brake disc accurately. This step is crucial in CFD analysis to ensure computational efficiency without compromising on result accuracy. Consequently, the study validated that a base size of 3 mm is sufficient for detailed thermal analysis, saving computational resources while maintaining precise results. Though the mesh size of 2.5 mm is closer to test results the mesh size of 3 mm is considered to balance between accuracy and number of cells.

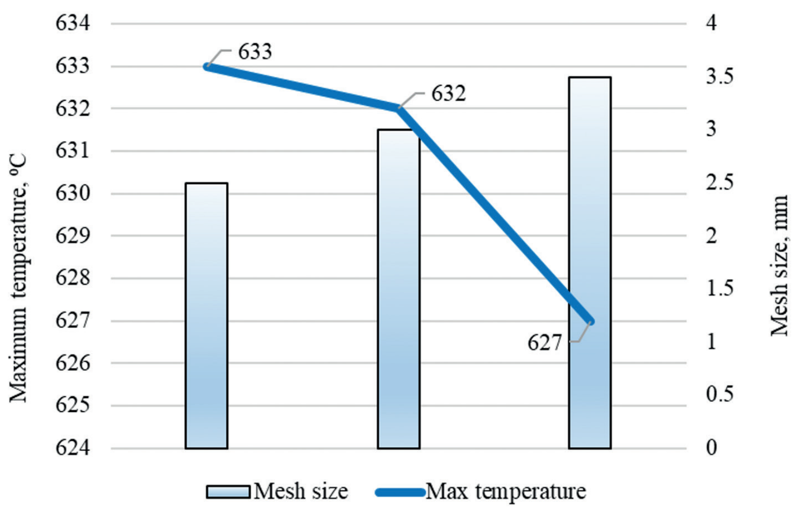

Fig. 12. Grid independence study

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4.3. Time Comparison:

The comparative study of aero-thermal simulation for calculating brake disc temperatures revealed significant differences in time consumption between two distinct phases. In phase 1, the focus was solely on the disc rotor, which involved model clean-up, meshing, and CPU solving, culminating in a total time expenditure of 16 hours. This phase allowed for a streamlined analysis of heat distribution and aerodynamic effects on the rotor itself. Conversely, phase 2 introduced additional complexities by incorporating the front half bumper, wheel, and tire along with the disc rotor. This comprehensive model aimed to capture more realistic interactions and thermal effects. The inclusion of these additional components markedly increased the computational effort and complexity, leading to a substantial total time requirement of 47 hours. This time encompassed the same stages of model clean-up, meshing, and CPU solving, but the increased model complexity demanded significantly more processing power and time. The comparison highlights how adding realistic vehicle elements to the simulation can vastly extend the overall duration of the process, underlining the importance of balancing model fidelity with practical computation constraints. Simulation time comparison between phase 1 and phase 2 is shown in Figure 13.

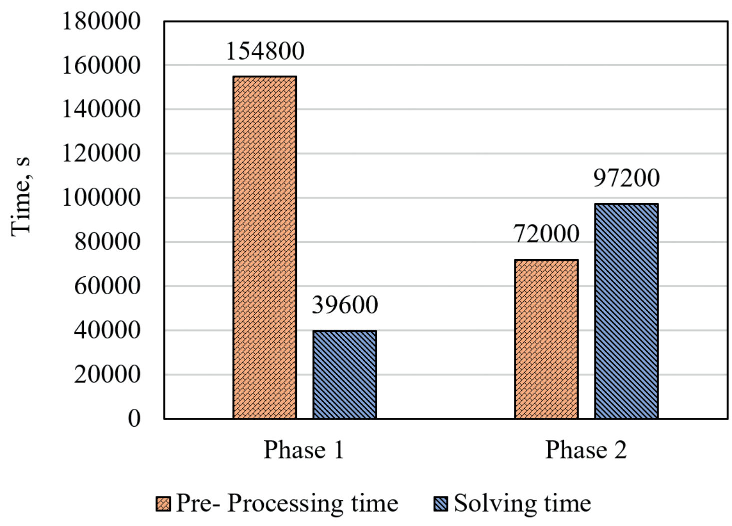

Fig. 13. Time comparison for phase 1 and phase 2 CFD models

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5. Conclusion

A detailed study was carried out to evaluate passenger cars disc rotor's thermal behaviour. To study rotor's thermal performance, it is always a best practise to study its maximum temperature rise after the rotor has reached thermal equilibrium. A repeated braking and cooling cycles would generally help achieve thermal saturation of rotors. However, the braking force, acceleration/deceleration 'g' values, 'from and to speed', shall be designed on the vehicle and regulation requirements.

Both numerical (CFD) and analytical method was discussed in this work to replicate the real-world vehicle level testing for rotor's thermal performance. In phase 1 CFD analysis there was a significant deviation in maximum temperature rise against the vehicle test results, which could be due to approximation in physical parts consideration and absence of linear wind speed in simulation. Phase 2 CFD simulation was aimed to reflect most of the real-world scenario, as a result we got good maximum temperature rise corelation (~ 97%) and time vs temperature curve trend while comparing with the vehicle test. Also, the sectional velocity post-processing gives a good examination on air flow distribution around the CHT bodies, it can be used to optimise the surrounding parts design for better air flow to brake disc.

The analytical calculation using Limpert's expression for calculating temperature of rotor also showed a good agreement with the vehicle test results. So, a combined Numerical–Analytical calculation can be adopted for evaluating thermal performance of passenger car's brake disc (in vehicle development phase). This would eliminate the time consuming, tedious vehicle level tests.

6. Reference

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