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# IMPROVEMENT OF HEAT DISSIPATION RATE OF AN AUTOMOBILE BRAKE DRUM USING FINS INCORPORATION

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## Abstract

The concept of incorporation of fins in automobile brake drum came up as a measure to subdue or address the thermal problems associated with it, which ultimately leads to brake failure. In order not to compromise the original weight of brake drum,1/10th of the overall wall thickness of the brake drum was converted into fins on the outer surface of the brake drum for effective heat dissipation. Modeling and simulation analysis were carried out using Solidworks (2013) software, on both the existing and modified brake drum, followed by validation using theoretical finite element analysis. The minimum temperatures observed from the simulation analysis were 4935K and 4927K for the existing and the modified brake drum model respectively. While maximum von Mises stress were $22{,}378.9Nm^2$and $21{,}971.2Nm^2$ and the maximum displacements were $5142 \times 10^{-5}mm$ and $5102 \times 10^{-5}mm$ for the existing and the modified brake drum model respectively. This implied that the modified brake drum have improved strength and better heat dissipation rate than the existing model.

**Keywords:** Brake drum, fins, temperature, heat dissipation, model, SolidWorks.

## 1. Introduction

Brake failure or fade prevalent rate is higher in drum brake than disc brake and the reason could be adduced to the indirect exposure of drum brake equipment to air which results in slow process of convective heat transfer thereby making it to dissipate low quantity of heat (Sinha & Gahir, 2018). One of the ways of improving heat transfer of the brake drum is by incorporating fins in the outer surface (Alin-Marian et al., 2015), Travaglia and Lopes (2014), Rong et al. (1997), Puhn (1985). However, in critical search for effective ways in which heat dissipation in brake drum can be enhanced, Sinha & Gahir (2018) varied the design of drum brake to obtain three different models by adding different number of fins to the circumference of the models. They carried out CFD analysis using ANSYS fluent software to simulate heat transfer rate among others in the models. They concluded based on their results that heat transfer rate increased with increase in number of fins. Hsueh (2012) developed a brake drum cooling device based on thermoelectric cooling system. The cooling side of the system was made to contact with the brake system to reduce the temperature of the brake pads and the other side of the system was made to dissipate heat which was carried away by water and discharged to the atmosphere by water –cooled radiator. The result of performance evaluation of the device revealed that there was about 20% decrease in brake system working temperature and 30% increase in braking force. He remarked that an efficient braking force can be attained with the device, when brake is applied to reduce the speed of vehicle for a long period of time and aid the safety of the driver.

Puncioiu et al. (2015) used structural and thermal FEA models to simulate a whole braking process of wheeled armored personnel carriers and concluded based on the outcome of the simulation, that decrease in the surface temperature of the lining was facilitated by forced convection as a result of high air speed propelled by the rotation of the drum brake. They stated that if the contacting surfaces of the lining with the roller is made to be wide, the high rise in temperature experienced, when brake shoe is applied to the brake drum, will be reduced by cooling as a result of heat transfer. Shodhganga (2018) designed and optimized an aluminum metal matrix composite brake drum that have effective heat dissipation among other attributes and to be used in passenger vehicle, instead of the heavy cast iron brake drum. Based on their results, it was concluded that the temperature rise in the drum decreased to 208.5°C, when aluminum metal matrix composite brake drum was substituted for the common heavy cast iron brake drum. Raju et al. (2016) developed and used a finite element model to evaluate time dependant temperature distribution in brake drums produced from aluminum composite, cast iron

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and stainless steel 304. This was done with a view to selecting the best brakedrum for use in automobile. They concluded, based on their results, that aluminum composite drum brake should be selected for use due to its low temperature and thermal deformation instead of cast iron or stainless steel 304 drum brakes.

Li (2018) developed an automatic brake drum cooling system based on semi conductor refrigeration. It was stated that the developed cooling system operates by blowing cooled air across the brake drum outer wall's fin thereby carrying away the heat and therefore reduces the entire brake drum temperature. Rao et al. (2016) developed rectangular and triangular finned brake drums by putting the fins on the outer part of the brake drum surface. When it was subjected to a test for effective heat dissipation, the result showed that there was an increase in the amount of dissipated heat. It was reported that when selection for use is based on the maximum amount of dissipated heat or heat flow, brake drum with the incorporation of rectangular fins at the outer surface should be selected for use. According Rao et al. (1993), brake drum is the heaviest part of brake assemblies and it has to be designed to be lighter so as to cope with increase in energy absorption and dissipate heat effectively. Undoubtedly every automobile brake drum in use have been designed and produced taking the remarks made by Rao et al. (1993) into consideration. In this research work, existing brake drum which has been designed and produced is being redesigned by converting 10% or 1/10$^{th}$ of the wall thickness to fins, without altering the original weight, with a view to improving heat dissipation rate of the brake drum.

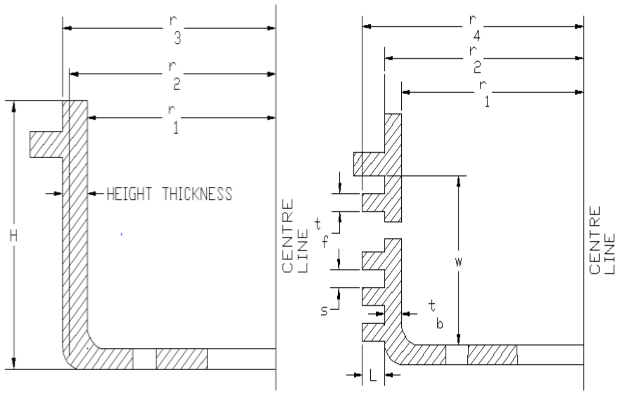
## 2. Materials and Method

Peugeot 406 D8 brake drum was used as a control experiment for this study. The dimensions of the brake drum were taken by measurement, using veneer caliper. One-tenth (10%) of the brake drum wall thickness was calculated and converted to the dimensions of the annular fins of the modified brake drum, shown in Table 1.

Table 1 Calculated Fins Parameters

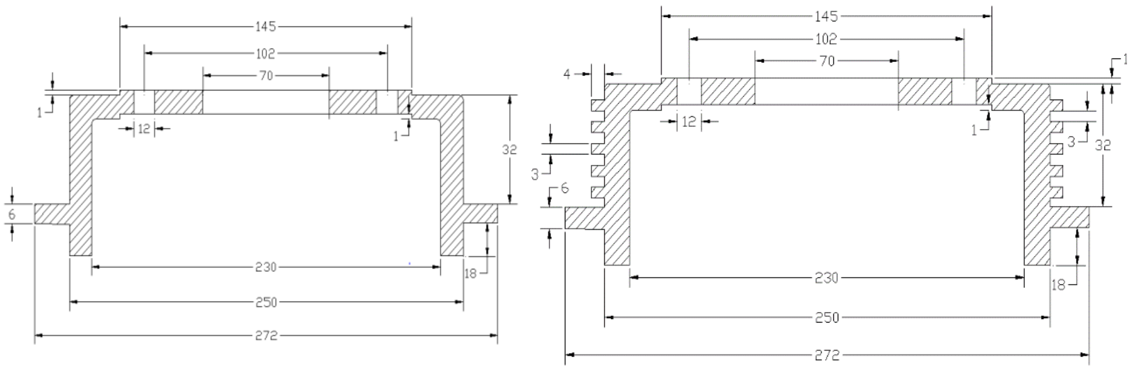
| S/N | Geometry/Parameter | Value | Unit |
|---|---|---|---|
| *1* | Fins Base Radius | 125 | mm |
| *2* | Fins Base Thickness | 9 | mm |
| *3* | Fins Thickness | 3 | mm |
| *4* | Fins Radius | 128 | mm |
| *5* | Fins Spacing | 3 | mm |
| *6* | Fins Length | 4 | mm |
| *7* | Number of Fins | 5 | - |

These dimensions as well as the ones obtained from existing brake drum were used as the references for modeling of the existing and the modified brake drum for simulation analysis. The sectional views of existing and modified brake drum are shown in figure 1. For the sake of validation purpose, both simulations by SolidWorks (2013) software and Finite Element Method (analytical approach) were carried out.




(a) (b)

Where;
$r_1$ = Brake Face Radius/ Inner Radius (mm)
$r_2$ = Fins Base Radius (mm)
$r_3$ = Shield recess radius/ Outer radius (mm)
$r_4$ = Outer Radius of the Fins (mm)
$t_b$ = Fin Base Thickness (mm)
s = Fin Spacing (mm)
$t_f$ = Fin Thickness (mm



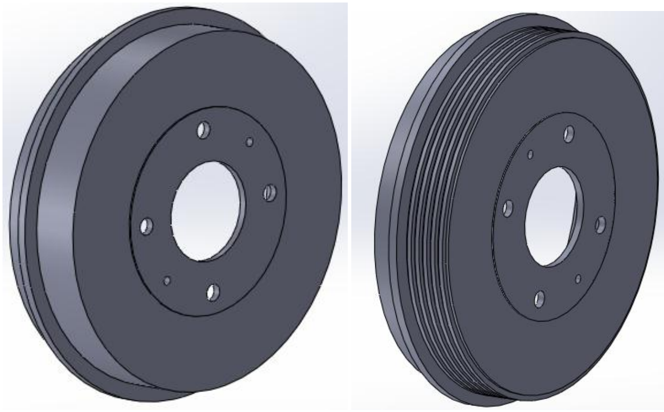

(c) (d)

(All dimensions in mm), Scale 1:1

Figure 1 Brake Drum Sectional view. (a) and (c) Existing brake drum (b) and (d) Modified brake drum.

## 2. 1 Brake Drum Modeling and Simulation

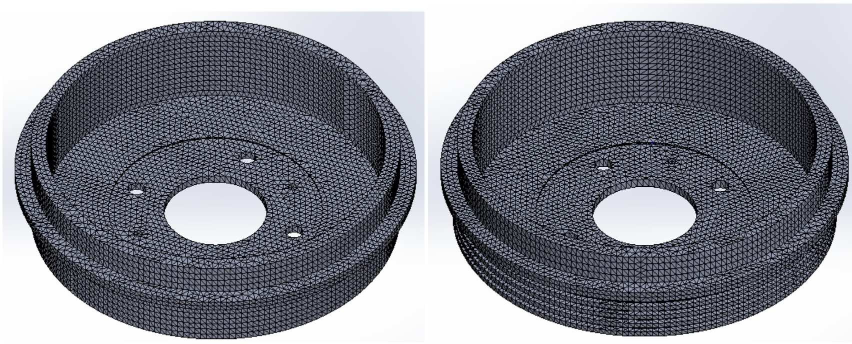
The brake drum models were developed by using SolidWorks (2013) software with the help of data measured and calculated fins parameters. The developed brake drum models are shown in figure 2.



(a) (b)

Figure 2. (a) Existing Brake Drum Model (b) Modified Brake Drum Model

It was shown in Andrzeji (2010), that at the ambient temperature of $20^0$C; the average brake drum temperature was $379^0$C. The exterior and the interior temperatures of the brake drum models were assigned to be 20℃ and 379℃ respectively in order to investigate the two models at the same condition. The models were subjected to pressure of 1500N/m$^2$. A fine mapped mesh of triangular element with 0.004m element size was used for the brake drum models. After setting these boundary conditions, the simulation was carried out. The meshed brake drum models are shown in figure 3.



(a) (b)

Figure 3. Meshed Models (a) Existing Model (b) Modified Model

### 2.2 Validation of Simulation Results

With the use of the finite element method and assuming a linear variation of temperature, the resulting stiffness matrix is given by Roland et al. (2004) as;

$$[K] = \frac{2\pi k}{l}\frac{(r_i + r_j)}{2}\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} + 2\pi r_o h \begin{Bmatrix} 0 & 0 \\ 0 & 1 \end{Bmatrix} \qquad 1$$

$$Q^e = hT_\infty 2\pi r_o \begin{bmatrix} 0 \\ 1 \end{bmatrix} \qquad 2$$

It was also noted in David (2004), that the Finite Element Equation with conduction and convection are expressed as

$$([K_T^e] + [K_C^e])[T] = [Q^e] + [q^e] \qquad 3a$$

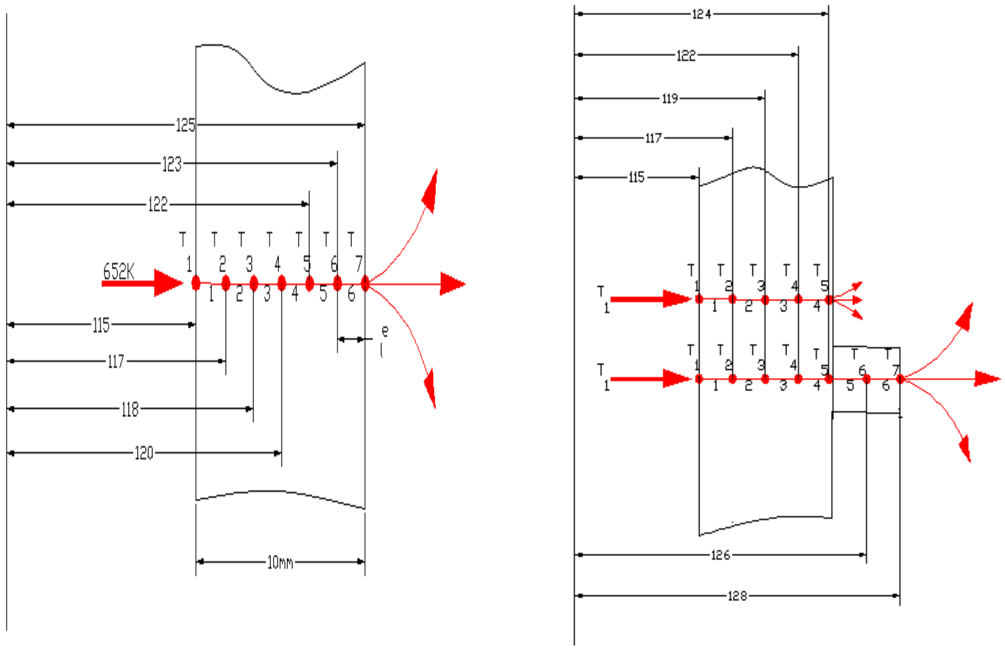
$$\left(\frac{2\pi k}{l}\frac{(r_i + r_j)}{2}\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix} + 2\pi r_o h \begin{Bmatrix} 0 & 0 \\ 0 & 1 \end{Bmatrix}\right)\begin{bmatrix} T_i \\ T_j \end{bmatrix} = \begin{bmatrix} Q_i^e & q_i^e \\ Q_j^e & q_j^e \end{bmatrix} \qquad 3b$$

Where, e = Element. i, j = Nodes. $l$ = Length.(m), $Q^e$ = Thermal load (W/m$^3$)

**q** = Vector of nodal heat flow (W/m$^3$), $T_\infty$= The ambient temperature (K)

## 2.3 Determination of Temperature Distribution at the Main Wall of the Brake Drum Mode

To calculate the thermal load vector and element conduction matrix for each element; equation 2 and 3b were used. The matrices were assembled and simplified by applying Gauss Elimination Method. The wall was divided into seven nodal points (7, 8, 9, 10, 11, 12, 13) andequation (2) and (3b) were used. The matrices were assembled and simplified by applying Gauss Elimination Method. The nodal temperature distributions are shown in figure 4.



(a) (b)

$T_1$, $T_2$, $T_3$, $T_4$,$T_5$, $T_6$ = Nodal Temperature, 1, 2, 3, 4, 5 = Elements

Figure 4. Nodal Temperature Distributions. (a) Existing Model (b) Modified Model

## 2.4 Determination of the Displacement at the Walls of the Brake Drum

For the purpose of this paper, only the displacement analysis was validated using theoretical calculation. Since the brake drum is fixed at the closed end and free to deflect at the opened end; therefore the brake drum can be treated as a cantilever beam with an internal force applied to the inside inner surface of the brake drum as shown in Figure 5. This was done in order to simplify the complexity of the models.

$$\frac{t}{r_1} << 1 \quad or \quad \frac{r_1}{t} > 1, \qquad 4$$

$$\sigma_c = \frac{pd_1}{2t}, \quad \sigma_a = \frac{pd_1}{4t,}, \qquad \sigma_r = p \qquad 5$$

Where $t$ = Thickness of the cylinder (m); $d_1$ = Inner diameter of the cylinder (m)

$\sigma_c$ = Circumferential stress (N/m$^2$); $\sigma_a$ = Axial stress (N/m$^2$); $\sigma_r$ = Radial stress (N/m$^2$)

From thinned wall cylinder criterion,

$$F = Pd_1 \qquad 6$$

Where F is the force acting on the brake drum

From beam formula (ANSI/AF&PA,2005); the deflection/displacement of a cantilever beam with a point load 'P' at a distance 'a' from the opened end and 'b' from the fixed end can be obtained by;

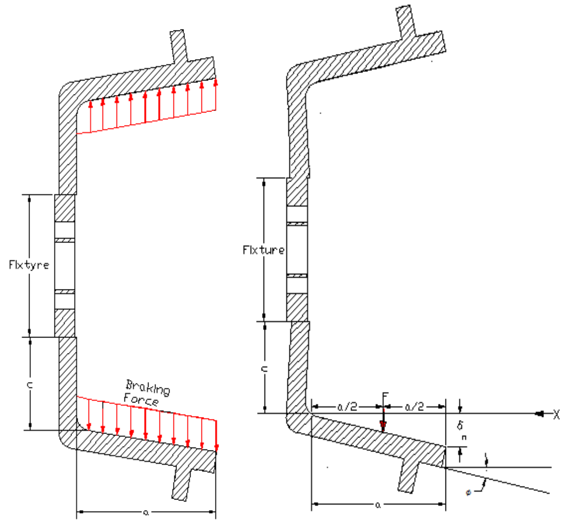



Figure 5. Brake Drum Displacement Analysis

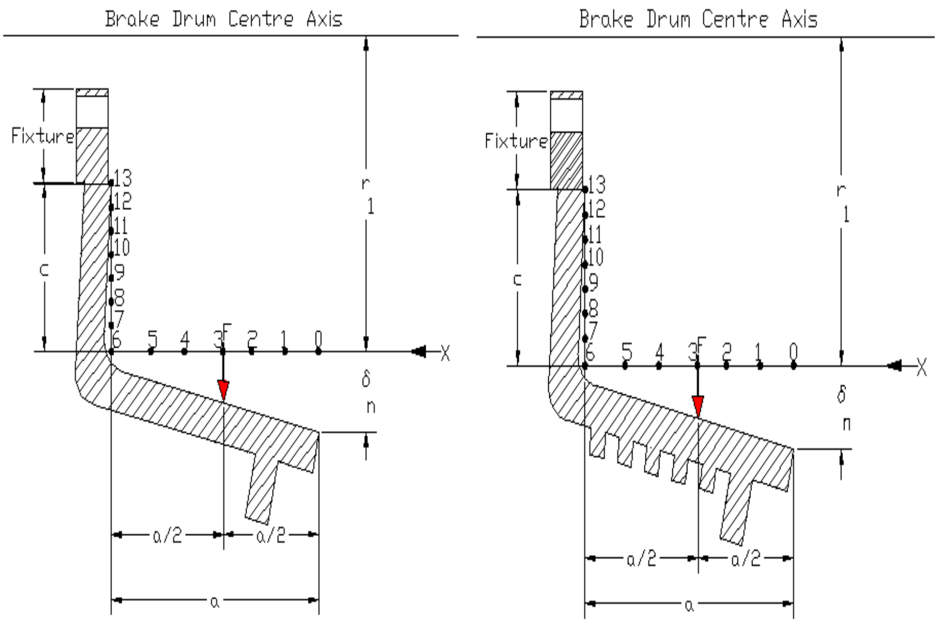
$$\delta_n = \frac{pd_1b^2}{6EI}(3l - 3x - b) \qquad (x < a) \qquad 7$$

$$\delta_n = \frac{pd_1(L - x)^2}{6EI}(3b - l - x) \qquad (x > a) \qquad 8$$

Where $\delta_n$ = Nodal deflection/displacement at distance x from the opened end (m)

$F$ = Force (N), $b$ = Distance between the fixed end and F (m)
$x$ = Distance from opened end toward fixe end (m), $l$ = Distance from fixed to free end (m)
$E$ = Modulus of elasticity of brake drum material (N/m$^2$)
$I$ = Moment if inertia the brake drum subjected to deflection/displacement (kg/m$^2$)

To determine the displacement at the wall of the brake drum; the walls of the brake drum were divided into thirteen elements with thirteen nodal points as shown in the figure 6.




(a) (b)

Figure 6. Displacement Analysis (a) Existing Model (b) Modified Model

Equation (7) was used to calculate the nodal displacement from node 1 to node 3. Recall,

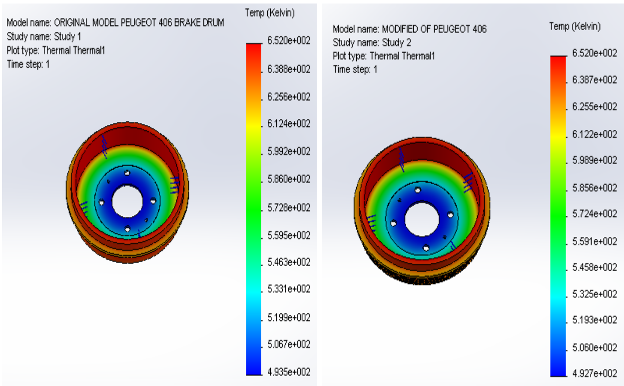
$b = c + \frac{a}{2};\ l = c + 2a$

$x =$ Linear distance from node 1 to node 13

Since $(x > a)$ from node 4 to 13, Therefore equation (8) was used to calculate the nodal displacement from node 4 to 13.

## 3. Results and Discussion

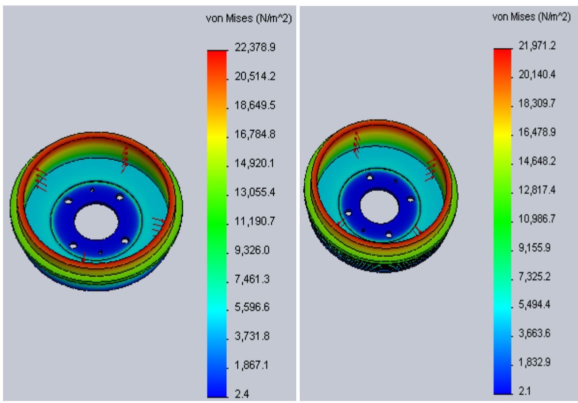
The simulation results shown in figure 7 depict the thermal (temperature) distribution of the two models simulated under the same initial conditions.




(a) (b)

Figure 7. Thermal Analysis Results (a) Existing Model (b) Modified Model

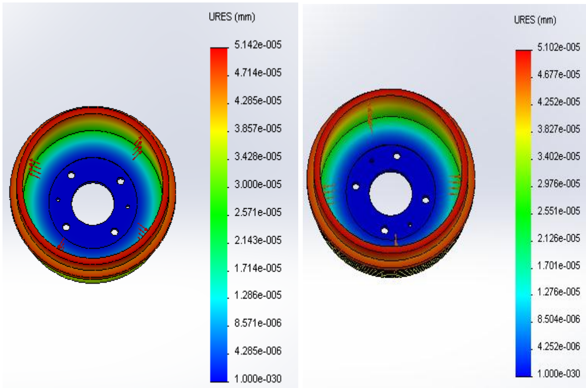
The simulation results shown in figure 8 depict the stress distribution of the two models simulated under the same initial conditions.




(a) (b)

Figure 8. Stress Analysis Results (a) Existing Model (b) Modified Model

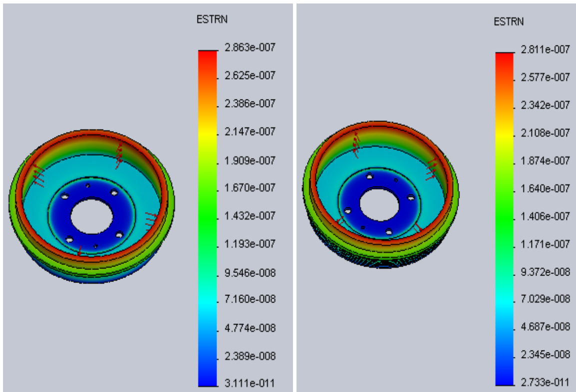
The results of simulation shown in figure 9 depict the displacement distribution of the two models simulated under the same initial conditions.




(a) (b)

Figure 9. Displacement Analysis Results (a) Existing Model (b) Modified Model

The results of simulation shown in figure 10 depict the strain distribution of the two models simulated under the same initial conditions.

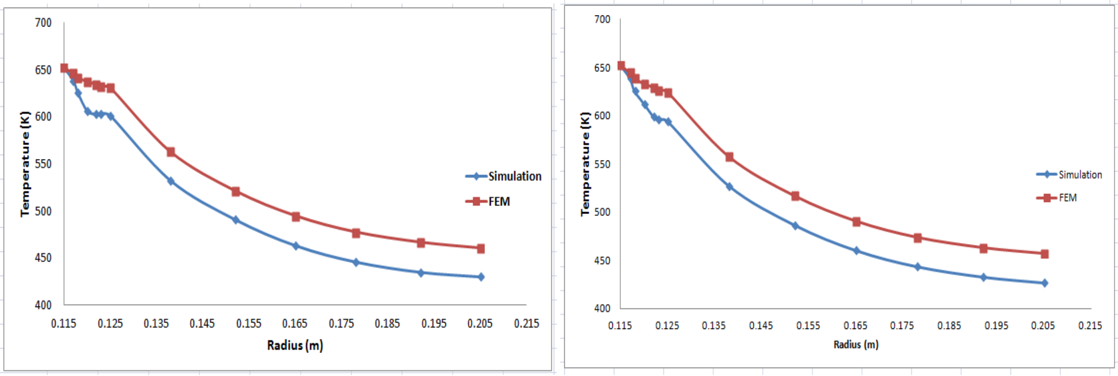



(a) (b)

Figure 10. Strain Analysis Results (a) Existing Model (b) Modified Model

The minimum von Mises stress and displacement are located at the adjacent side of the inner surface of the two brake drum models. This is due to the absence of braking forces acting directly on these surfaces. While the maximum von Mises stress of the two models are located on the walls of the brake drums. This is due to the action of the brake shoes force. The modified brake drum model showed a lower value of stress, strain and displacement than the original model. This has indicated that the fins have added more circumferential strength on the brake drum. This circumferential strength has increased the circumferential resistance of the brake drum to the action of the brake shoe force. This has also made the brake drum to be more rigid. This helps to resist the hoop or circumferential stress acting at the inner wall of the brake drum.

The brake drum temperature distribution or variation with radius of the two models is shown in figure 11. The initial point of intersection in the graphs (Figure 11) showed that the models were investigated at the same initial temperature while the lowest temperature signified the temperature at the outer surface of the brake drum.

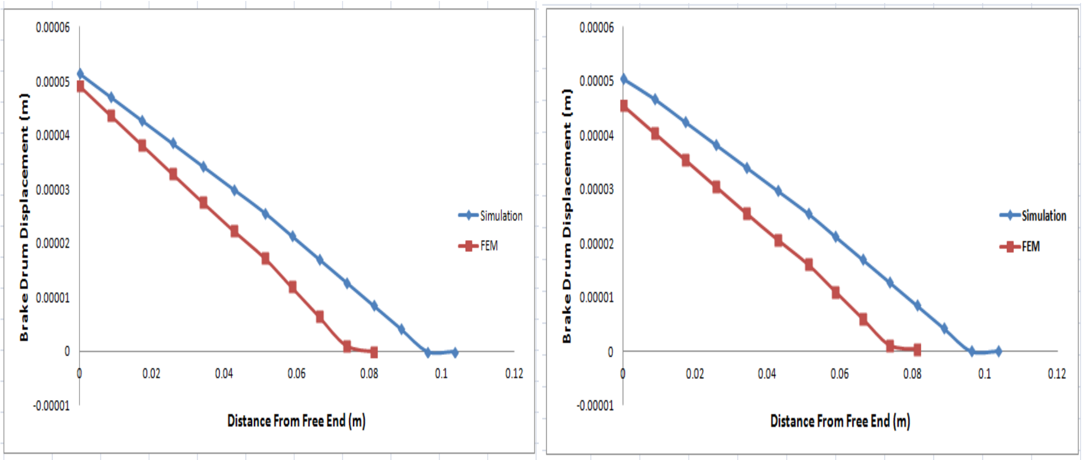



(a) (b)

Figure 11. Brake Drum Temperature Distributions (a) Existing Model (b) Modified Model

The temperature of both model declined from their initial (maximum) temperature to their minimum temperature as also justified by their computational (simulation) results and the theoretical calculations. But the existing model showed a lower rate of decrease in temperature than the modified model. This showed that the rate of heat dissipation from the modified model was higher than that of the existing model. Therefore some amounts of heat energy tend to remain inside the inner wall of the existing brake drum model as the result of the low heat transfer. This retained heat is the major cause of the thermal problems of the brake drum system earlier stated.

The displacement graphs (figure 12) showed the prediction of the displacement of the brake drum walls as result of the pressure acting on the inner wall of the brake drums.




(a) (b)

Figure 12. Brake Drum Displacement (a) Existing Model (b) Modified Brake

The maximum points on these graphs indicated the maximum displacement at the opened end of the brake drum, while lowest displacement signified the lowest displacement at the fixed end of the brake drum models. The modified brake drum model showed minimum displacement than the existing or original model. This is due to circumferential or annular arrangement of the fins on the outer surface of the brake drum. This indicated that the fins have added more strength circumferentially on the braking surface of the brake drum. This has also increased the circumferential resistance of the brake drum to the action of the brake shoe force. This made the brake drum to be more rigid and resistant to the action of the force acting on this surface.

The minimum temperature and maximum displacement differences between the modified and existing brake drum models were determined to be 0.8K and 4.0 x $10^{-7}$ mm respectively. The minimum temperature difference between the modified and existing brake drum of 0.8K obtained in this research work is higher than the 0.5k obtained from Bako et al (2015) and less than 0.9K obtained from Rao et al(2016) respective research work by Bako(2018). This showed that the modified brake drum dissipate more heat than that of Bako et al (2015) and less heat than that of Rao et. al (2016). Moreso, the maximum displacement difference between the modified and existing brake drum of 4.0 x $10^{-7}$ mm also obtained in this research work is less than 6.8 x $10^{-7}$ mm obtained from Bako et. al (2015) and 6.7x $10^{-6}$ mm obtained from Rao et al (2016) respective research work by Bako (2018). This implied that the modified brake drum is more rigid than that of Bako et. al (2015) and Rao et al (2016).

## 4. Conclusion

This work has provided a method for improving the strength and heat dissipation of an automobile brake drum without changing in its original weight and compromising the properties and requirement of the brake drum. The results obtained from the computational analysis and the theoretical calculations showed that the modified brake drum model have an improved heat dissipation and lower displacement.

The lower displacement and temperature shown by the modified brake drum model indicated that the model was stronger and rigid with better heat dissipation than existing brake drum model. The lower temperature of the modified model showed that more heat was transferred and dissipated from the inner to the outer surface of the brake drum while the high temperature of the existing model indicated that some heat were retained inside brake drum and has slower rate of heat dissipation.

This retained heat energy in the existing brake drum model is the major cause of thermal problems of the brake drum. The use of fins on the modified brake drum brake drum enormously increased the heat dissipation and the structural strength without a change in weight of brake drum of the existing model.

It can finally be concluded that a method of improving the heat transfer dissipation of an automobile brake drum has been developed without a change in weight and without compromising the existing properties and requirements of the automobile brake drum. This method of brake drum modification can assist Automotive Engineers to design a more effective and improved brake drums models. For the future, this work can further be extended by physical validation of the models.

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