Analysis of air brake used in medium duty truck*

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ABSTRACT

Keywords:
Pressure,
Displacement,
Deceleration,
Thermal Analysis.

This project work involving modeling and analysis of brake drum of comet vehicle of Ashok Leyland. The work was done using mechanical desktop and Cosmos. The complete brake assembly with about 20 parts was modelled and assembly was created. The assembly is shown below shows the brake shoes, S Cam, cam rod and pneumatic cylinder and slack adjuster. Kinematic analysis was done on the model elements to see interference and limitation of movement. The thrust provided by the shoes on the drum face was computed from the pneumatic pressure under full braking condition and the same was given as a uniform pressure on the brake shoes and the displacement of the drum shell was studied. The maximum deformation of the drum shell was found to be 1.3 X 1o^{-5} mm.

The stress induced on the drum due to the force applied by the brake shoe was analysed. The stress plat shows the distribution of Y directional stress in the drum when the shoe applies maximum thrust.

Next thermal analysis was carried out on the drum under single application and multiple application of brake with full force. The vehicle inertia was taken when travelling at 60kmph and the brake absorbed the complete energy proportionately by this drum. This energy was applied as heat on the inner face of the drum for the width the shoe is in contact. The decelerating time was computed using empirical formula given by Rudolph Lumpert was arrived at as 10 sec. for deceleration from 60 – 0kmph. The maximum heat flux is applied at time t = 0 and at time t = 10 heat flux applied is zero as there is no braking action while speed is zero. The variation was assumed as linear. As a second alternative the vehicle was accelerated for 40 seconds and braked for 10 sec. And this acceleration and braking cycle was repeated 10 times and corresponding temperature variation was studied on a node on the outer face of the brake drum where convective cooling is taking place during accelerating phase.

The curve above shows the temperature variation with reference to time for a period of 370 sec. As can be seen the temperature has not stabilised but increasing. But the cooling curve doing acceleration phase as the temperature difference between ambient and the node is higher.

1. Introduction

The device for decreasing the speed of a body or for stopping its motion is known as brakes. Most of the brakes act on rotating mechanical elements and absorb kinetic energy mechanically, hydro dynamically or electrically.

The first truck brakes were brake shoes operating directly on the wheels. From this simple beginning has evolved, one of the most complex braking systems found on any type of vehicle. The first air brakes were introduced in 1918. Seven years later four wheel brakes were introduced on truck, and the internally expanding type was introduced by 1930. In the late 1930s the vacuum booster, or hydraulic brake, was introduced. A pressure gauge and a low pressure warming device, either audible or visual, indicate air pressure. Air is stored in the reservoirs and supplied to the brake valves, a foot valve supplies air to all brake chambers on the vehicle, including those being towed. Another brake valve is hand controlled and applies the brakes on the towed vehicle only. Both the foot

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pedal and hand valve supply air to the same service line, which extends back to the towed vehicles. The second, or emergency, line carries full air pressure when the vehicle is in operation. If this line is broken, the emergency brakes are applied on all towed vehicles from air reservoirs located on the towed vehicles. After reaching the brake chamber from the brake valves, the air acts on a diaphragm connected to a push rod, which in turn actuates a cam that moves the shoe against the rotating brake drum. A heat exchanger is a piece of equipment built for efficient heat transfer from one medium to another. The media may be separated by a solid wall, so that they never mix, or they may be in direct contact. They are widely used in space heating, refrigeration, air conditioning, power plants, chemical plants, petrochemical plants, petroleum refineries, natural gas processing, and sewage treatment. The classic example of a heat exchanger is found in an internal combustion engine in which a circulating fluid known as engine coolant flows through radiator coils and air flows past the coils, which cools the coolant and heats the incoming air. There are three primary classifications of heat exchangers according to their flow arrangement. In parallel-flow heat exchangers, the two fluids enter the exchanger at the same end, and travel in parallel to one another to the other side. In counter-flow heat exchangers the fluids enter In the 1970s a wedge replaced the cam. Recently wedge has been replaced by S Cam.

2. Objectives

The ultimate objective of this structural and the thermal analysis of brake drum are to evaluate its performance under normal vehicle operating conditions.

The solid modeling of brake drums, brake assembly using mechanical desktop package and analysing the brake drum model using finite element analysis package Cosmos for study of structural and thermal behaviour.

Brake drum, static stress analysis and the nodes along the hub bore are fixed in all the translational and rotational directions. The present calculations are limited to study of temperature variations along the cross section of the drum. The results of the temperature study indicate the quality and performance of the brake drum.

3. Specifications

I. Diameter of the air brake chamber (d) = 11 cm
II. Brake chamber pressure = 8.3 kg/cm^2
III. Outer radius of the brake drum (R) = 21.5 cm
IV. Inner radius of the brake drum (r) = 19.7 cm
V. Distance between the centre of the brake drum to the centre of the pivot (b) = 17.75 cm
VI. Distance between the centre of the S cam to the centre fo the pivot (h) = 35.5 cm
VII. Coffcient of friction of lining (μL) = 0.4
VIII. Arc of angle (α) = 131.8^0 (2.3 rad)
IX. Angle between beginning of the lining and straight line connecting the centre and the pivot point (α1) = 58^0
X. $\alpha 2= \alpha 1 + \alpha 0 = 189^0$
XI. $\alpha 3 = \alpha 2 + \alpha 1 = 247^0$
XII. Lining thickness = 1.27 cm
XIII. S Cam radius = 1.27 cm

Materials

Brake drum = Cast Iron
Lining = Molded asbestos based – HF7
Properties of brake drum

Density	=	7228 kg/m^3
Specific heat	=	419 Nm/kgK
Thermal conductivity	=	174465 Nm/hmK
Thermal diffusivity	=	0.0576 m^2/h
Young's modulus	=	2.1 X 10^{11}N/m^2
Poisson's ratio	=	0.29

4. Theoretical Calculation

Brake torque analysis

The drum rotation and the force applied on the shoe are in the same direction is called leading shoe.

The drum rotation and the force applied on the shoe are in the opposite direction is called trailing shoe.

Basic Torque analysis for drum brake (Ref. Fig, 1)

Moment balance around the shoe pivot yields

$-Fa.h – Fd.c + Fd.b / \mu L$ -------------1

Where b – Distance between the center of the pivot to the drum diameter

c – Horizontal distance between the center of the pivot to the inner radius of the drum

h – Vertical distance between the center of the pivot to the force applied

μL – friction coefficient of shoe

Brake factor for air S cam brake (Ref, fig. 2)

$BF1 = \mu Lh/r/(a/r)[\alpha^0 - \sin\alpha^0 \cos\alpha 3/4 \sin(\alpha^0/2) \sin(\alpha 3/2)] - \mu L[1+ (a/r) \cos(\alpha^0/2) \cos\alpha 3/2]$ --------2

$BF2 = \mu Lh/r/(a/r)[\alpha^0 - \sin\alpha^0 \cos\alpha 3/4 \sin(\alpha^0/2) \sin(\alpha 3/2)] + \mu L[1+ (a/r) \cos(\alpha^0/2) \cos\alpha 3/2]$ ------3

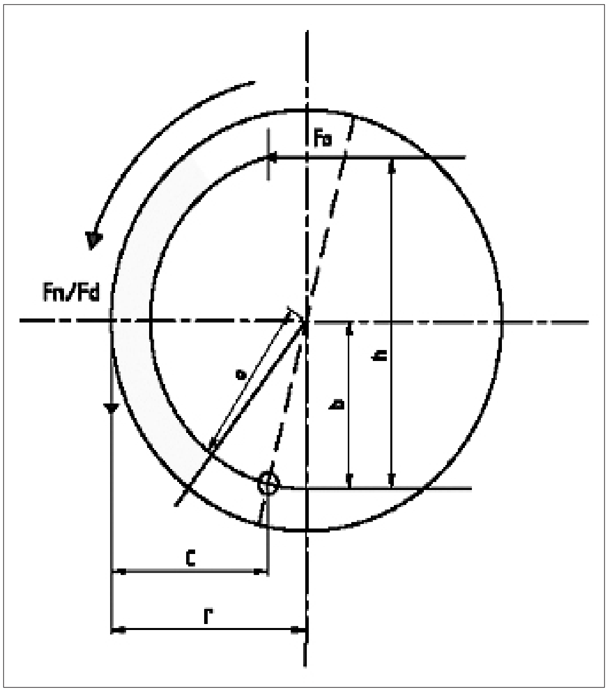

Fig. 1. Brake torque.

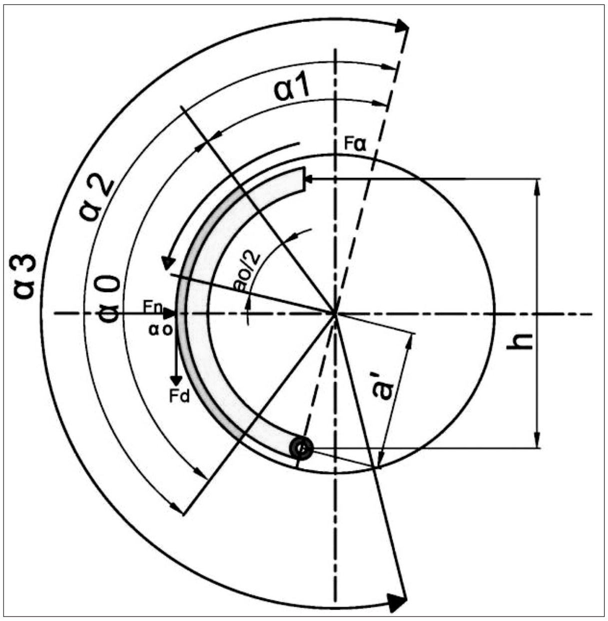

Fig. 2. Brake factor.

$BF = 4 (BF1)(BF2) / (BF1 + BF2)$-------4

Brake Torque

$TB = (P1 - P0)\ Ac\ BF\ \eta m \rho k A k T r$ -----------5

Kt – brake temperature reduction factor

P1 – brake line pressure, N/ cm^2

P0 – push out pressure, N/ cm^2

R – Effective drum radius

ηm – Mechanical Efficiency

STATIC AXLE LOADS (Ref, fig. 3)

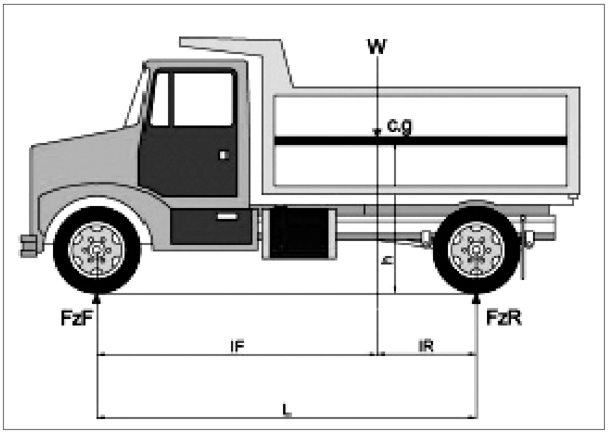

Fig. 3. Static axle load.

FORCES ACTING ON A DECELERATING VEHICLE (Ref, fig. 4)

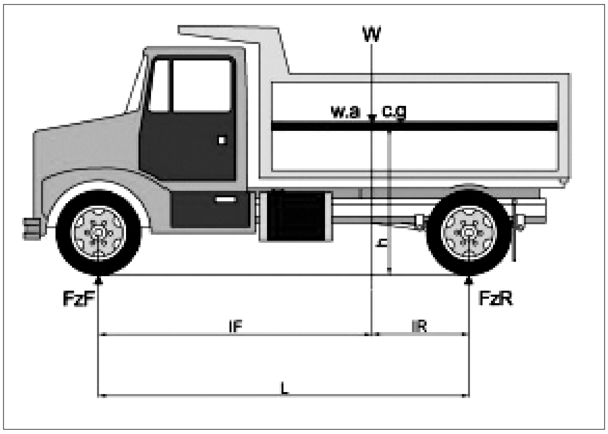

Fig. 4. Decelerating vehicle.

RIGHT SIDE VIEW OF BRAKE ASSEMBLY (Ref, fig. 5)

MESHED MODEL

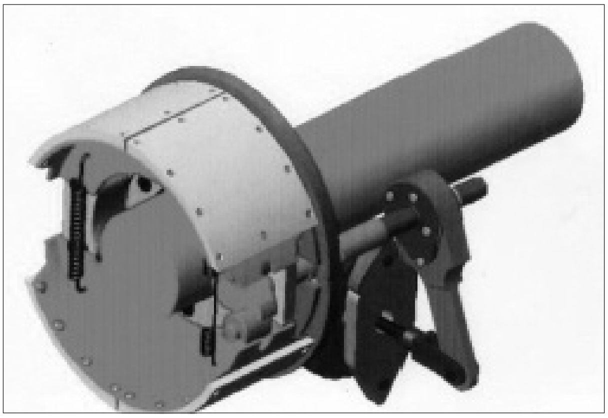

Fig. 5. Brake drum assembly

Element Group- 4 noded tetrahedral solid element
Number of nodes – 23782
Number of elements - 12354

Simulation of brake drum using finite element Analysed. The model considers only the drum. The drum is tapered externally in order to withstand the stresses. The loads are applied radically along the contact of drum and lining. The drum consists, provisions for the axle. The hole provided in the geometry is made constraint while performing static stress analysis.

In case of heat transfer analysis, the section of the drum is considered and analyzed for various conditions.

Meshed Model of Brake Drum (Ref, fig. 6)

Von Mises stress is used as a criterion in determining the onset of failure in ductile materials. The failure criterion states that Von Mises stress should be less than the yield stress of the material. This distribution is considered to compare with the yield stress and the results are obtained.

The figure shows the von Mises stress distribution for the radial load case. The maximum Von mises stress is obtained at the node number 1608 with the magnitude of 214.25 N / mm^2.

The stress distribution is more between hole and hub.

Yield stress for the drum material = 419 N / mm^2
Von Mises stress from FEA result = 214.25 N / mm^2

As the Von Mises stress, which was obtained from FEA, is lying below the yield stress of the material/

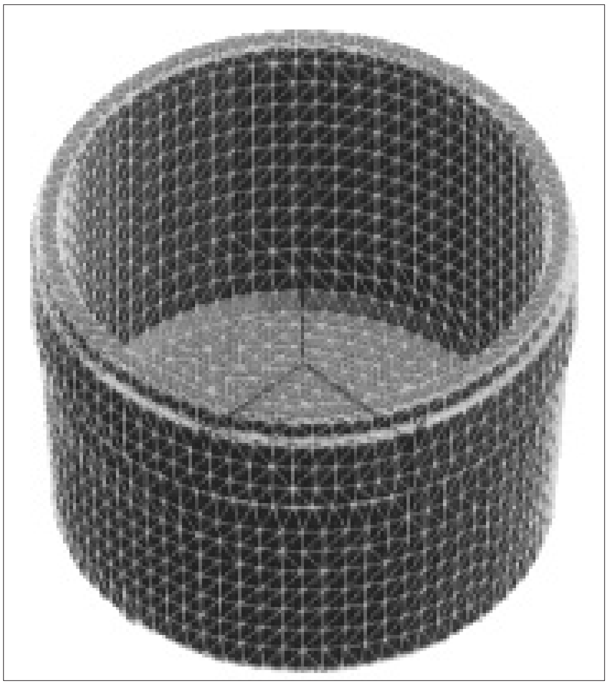

Fig. 6. Von - mises stress plot in brake drum.

From that we infer the stresses are lying well within the limits. (Ref, fig. 7 & 8)

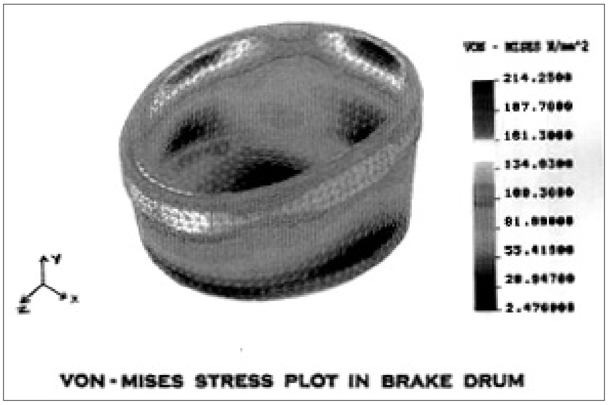

Fig. 7. Stress plot in brake drum

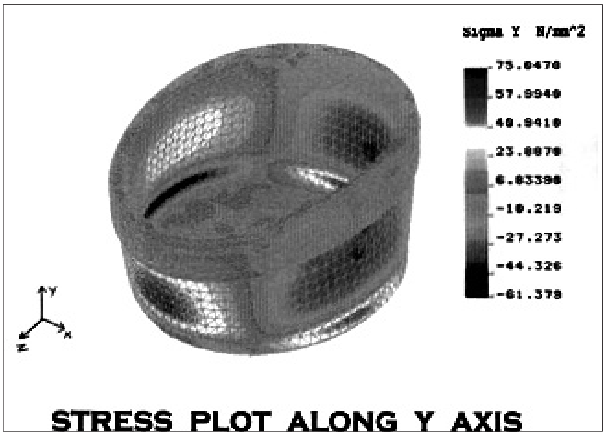

Fig. 8. Stress plot along y axis.

Radial Load Case

The radial load is applied along the area of contact of lining and the drum. i.e., by considering probable case of failure.

The load applied along the positive Z axis and negative Z axis are different, this is due to the radius of the S Cam. (Ref. Fig. 9)

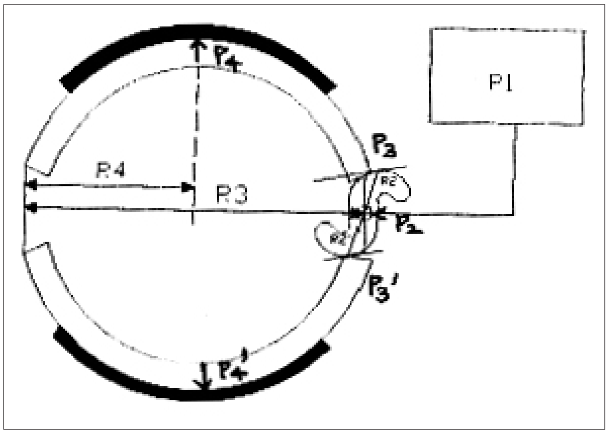

Fig. 9. Radial Load in Brake drum.

R1 – Effective length of slack adjuster
R2 – radius of S Cam (along the leading shoe)
R2' – radius of S Cam (along the trailing shoe)
Area of the diaphragm = 95cm^2
Brake chamber pressure (P1) = 8.3kg/cm^3
P2 – force applied on S cam
P3 – force applied on the leading shoe
P3' – force applied on the trailing shoe
P4 – force applied on the drum by the leading shoe
P4'- force applied on the drum by the trailing shoe
P2 = 95 x 8.3 = 789 kg
At positive Z direction
P3 = (P1R1) / R2 = (789 X 14) / 2.64 = 4184 kg
P4 = (P2R3) / R4 = (4184 X 27.6) / 13.8 = 8368 kg
At negative Z direction
P3' = (P1R1) / R2' = (789 X 14) / 2.8 = 3945 kg
P4' = (P2'R3) / R4 = (3945 X 27.6) / 13.8 = 7890 kg

Displacement Along Z Axis

The displacement will be maximum along the Z direction. The rod color at the bottom shows the maximum displacement. The maximum and minimum displacement along the Z direction is given below.

Node number: 1065
Maximum displacement = 1.3641e^-5
Node Number: 1080
Maximum displacement = -9.6864e^-6

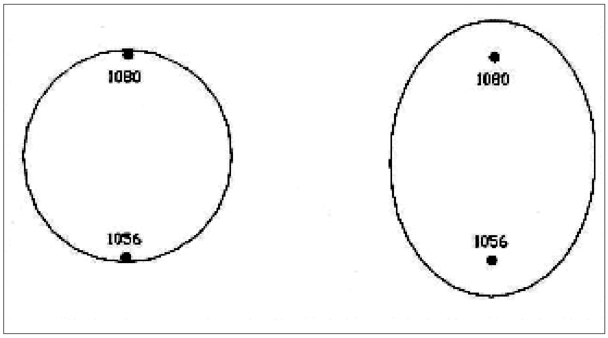

Fig. 10. Displacement plot.

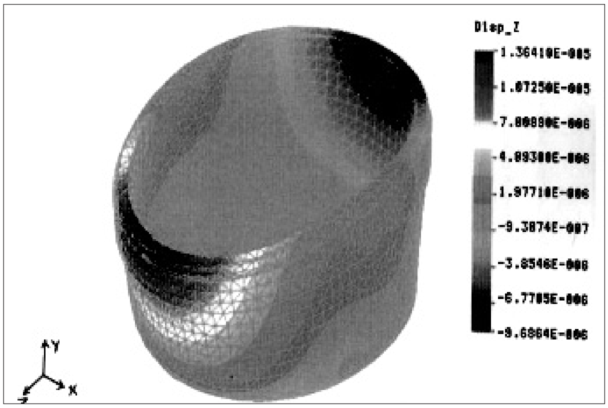

Fig. 11. Displacement along Z Axis.

Displacement plot (Ref. Fig. 10)

Displacement Along Z Direction (Ref. Fig. 11)

TRANSIENT TEMPERATURE ANALYSIS

Element Group – 3 – 6 noded triangular element

Total number of elements – 390

Total number of nodes – 277

The heat flux is given at the inner surface of the brake drum along the width of the lining, which is on the brake shoe. Convective heat transfer coefficient is given by the outer surface. The brake is applied for 10 seconds and then acceleration of the vechicle to reach 0 – 60 kmph is assumed as 30 seconds.

In this manner, brake is applied ten times repeatedly. The time curves for heat flux and convention coefficient are plotted.

Total time for ten brake application is found to be 370 seconds. The brake fade has not been taken into consideration because it will increase braking time in later cycles. The transient temperature analysis is done for full time period of 370 seconds.

The temperature distribution is plotted all along the cross section of the drum after 10 cycles. The time versus temperature graph is obtained for nodes at inner and outer surface. (Ref. Fig. 12)

TIME CURVE- HEAT FLUX (Ref. Fig. 13)

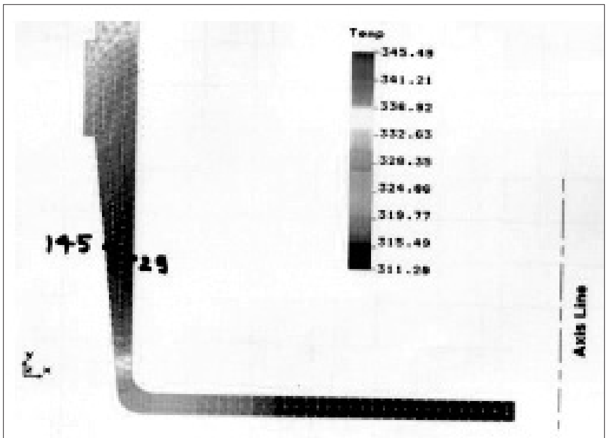

Fig. 12. Transient temperature analysis.

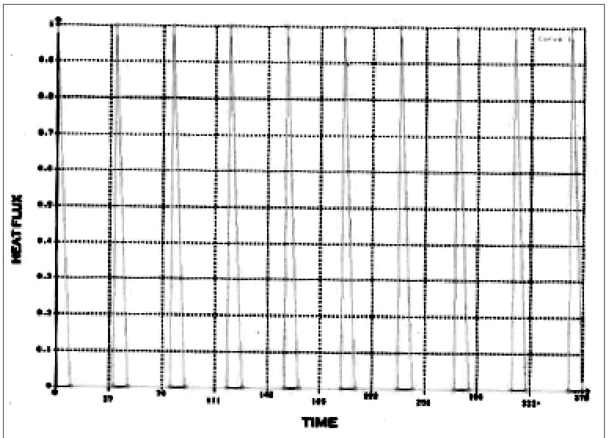

Fig. 13. Temperature curve – heat flux.

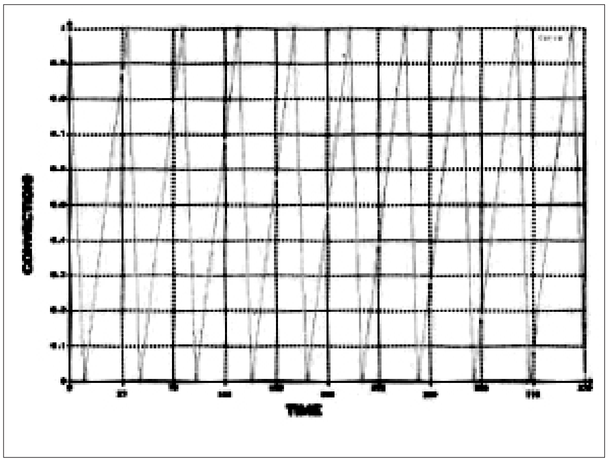

Fig. 14. Time curve - convection.

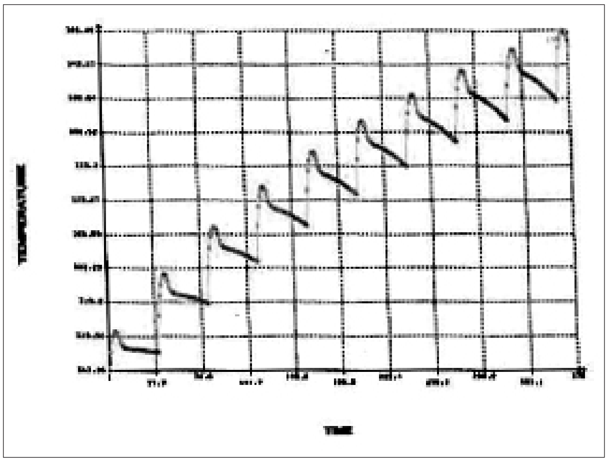

Fig. 15. Temperature variation at node 29.

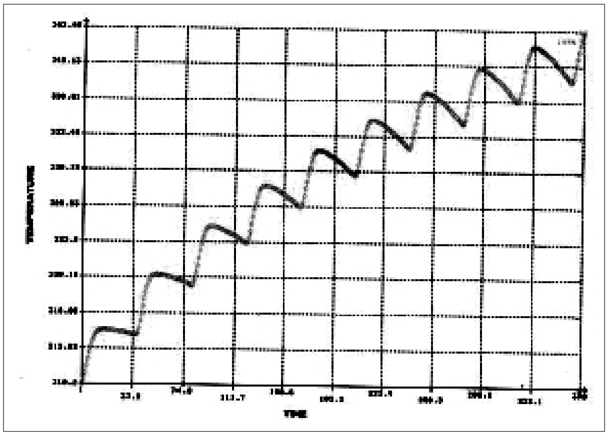

Fig. 16. Temperature variation at node 145.

TIME CURVE – CONVECTION (Ref. Fig. 14)
TEMPERATURE VARIATION AT NODE 29- INNER SURFACE (Ref. Fig. 15)
TEMPERATURE VARIATION AT NODE 145 – OUTER SURFACE (Ref. Fig. 16)

5. Conclusion

The modeling of brake assembly and analysis of brake drum has been carried out satisfactorily.

Static Analysis:

The entire brake drum is considered for the static analysis. The load is applied along the area of contact of the brake lining which is on the brake shoe. The load on the leading shoe will be more than the trailing shoe. The stress is found to be maximum on the leading shoe side.

Thermal Analysis:

Thermal analysis is done by taking the cross section of the drum and applying heat flux at the inner surface along the width of the brake lining. The graph has been obtained to show the variation in temperature for ten brake applications. It is found that there is a gradual temperature decreases due to convection while accelerating.

The drum could be modeled in Modeled in COSMOS/ M package and parametric mesh can be generated and analyzed for better results.

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