STEADY STATE THERMAL AND STRUCTURAL ANALYSIS OF DISC BRAKE USING FEM (CASE STUDY ON DISC BRAKE OF VOLVO B11R MODEL BUS)

A thesis submitted to Addis Ababa Institute of Technology, School of Graduate Studies, Addis Ababa University

In Partial Fulfillment of the Requirements for the Degree of

Masters of Science in Mechanical Engineering

(Mechanical Design)

By

Wondwosen Eshete

Advisor

Tollosa Deberie (Phd candidate)

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Addis Ababa, Ethiopia

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Approved by board of Examiners

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Chair	Signature
Tollosa Deberie (Phd candidate)	__________________
Advisor	Signature
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Internal Examiner	Signature
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External Examiner	Signature

Declaration

This thesis work is original and has not been presented for a degree in any other university and that all source of materials are duly acknowledged

Wondwosen Eshete ___________________

Student name signature

Tollosa Deberie (Phd candidate) ___________________

Advisor Signature

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Abstract

Braking system represents one of the most fundamental and critical safety components in modern vehicles. Brake absorbs kinetic energy of the rotating parts (Wheels) and the energy dissipated in the form of heat energy to the surrounding atmosphere. It decelerates or stops the vehicle. The thesis describes the finite element analysis techniques to predict the failure region on the pre maturity faller on disc brake to identify the critical locations of the components. The disc brake implemented on the front axle model with gray cast iron materials were studied in Disc brake which commonly used in auto motives. Despite all the stresses experience by the disc doesn't damage the disc due to high tensile strength but the disc may fail under fatigue loading. There is frictional heat generated over disc surfaces, which leads to high temperature. Due to number of brake actuation, stress and deformations have generated on the disc surface. Hence, it reduce the overall performance and efficiency of the brake system. The problem in Volvo B11R Bus of ODAA integrated plc, which is imported in May, 2019 is that their disc brake is involving pre-mature failure. i.e before service life of 500,000 km. This will initiate accident and leads to loss to the supplier due to complain from the customer. The objective of this paper was to model and steady state thermal and static structural analysis using finite element method (FEM). The method used to this research is to obtain necessarily data to study parametric values are retention time, force and torque. In order to make compare between actual prematurity failure and ANSYS result values decreasing torque and force while keeping retention time and mesh size at constant result in decreasing in both total deformation and Von miss stress. The maximum total deformation is 2.2424e-005m, the Von miss stress is 1.067e+008 Pa compressive ultimate strength 8.2e+008 pa, tensile ultimate strength 2.4e+008, compressive yield strength 1.e+009, tensile yield strength 1.e+008 and temperature result 420 ℃ . The ANSYS result shows that the crack that is happening in the bus's disc break is due to the increase in toque, force and temperature which result in higher deformation and stress. Thus we obtained the values of total deformation, Von-miss stress and thermal effects on disc brake, and interpreted them to explain the cause of the failure.

Keywords - Disc Brake, Solid work, ANSYS, FEA (finite element analysis).

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Nomenclature

c	Specific heat
ν	Poisson's ratio
α	Thermal expansion
E	Elastic modulus,
ε_t	Total strain amplitude
ε_p	Plastic strain
A_c	Surface of the pad
T_{wi}	The braking forces (normal force)
r_r	The rolling (or dynamic) radius
τ_{wi}	The neck area of torque
Q_i	Total heat energy
K.E	Kinetic Energy
ϱ_g	The heat generated
φ	Heat flux
T_i	Traction vector acting on the path
u_i	Displacement vector
ρ	Density
N	Number of cycle
A	$Effective\ area$
GVW	Growth vehicle weight
n/a	Not applicable
V_0	Initial speed of the vehicle
T_0	Initial temperature
ΔT	Temperature change from the reference temperature T_0
Q	Heat generation
V	Relative sliding velocity
μ	Coefficient of friction
p	Air pressure
Rt	Radius of tire
Rr	Radius of rotor

R_0	External radius of the disk
R_i	Internal radius of the disk
V_0	Initial speed of the vehicle
Q	Heat generation

Acknowledgements

Special thanks to God. In addition, my supervisor Mr. Tollosa Deberle (PhD Candidate) further more I would like to thank Dr. Samuel and Tesfaye Gonite. Advised and support who has been involved in most of what I have been doing during my time at steady state thermal and structural analysis of disc brake using FEM (case study on disc brake of Volvo B11R model bus) Without my advisers, the project would never have reached the results as it did.

To sum up, my thesis project has been widely scoped among different departments and areas, so there are many people to thank for making this project possible. At last, I would like to thank my friends and my family for all support and encouragement throughout the project and during the thesis work.

Chapter One

1. Introduction

1.1 Background

At braking, the kinetic (and/or potential) energy of a vehicle converted into thermal energy. Heavy road vehicles nowadays often use disc brakes. The discs and pads exposed to high thermomechanical loads, resulting in high stress and deformation of the discs such phenomena lead to performance and safety problems.

The materials used in heavy-duty buses brake discs and pads have been used grey cast iron, whereas pads are used, ceramic composites investigated for road vehicles and implemented on some high-performance bus. In recent years, the general development in computer-aided engineering and simulation methods have made possible analysis of more complex physical phenomena and detailed geometries [1].

This general trend also includes the challenging task of analyzing method of calculations and what stresses occur in the discs. The Finite Element Method, FEM done largely at Haldex in earlier works or experimenter on stress on a disc. Brake disc rotor is the rotating part of a disc brake assembly normally located on the Bus throughout the wheel axle but specifically investigated on the front wheel brake most important safety feature of an automotive.

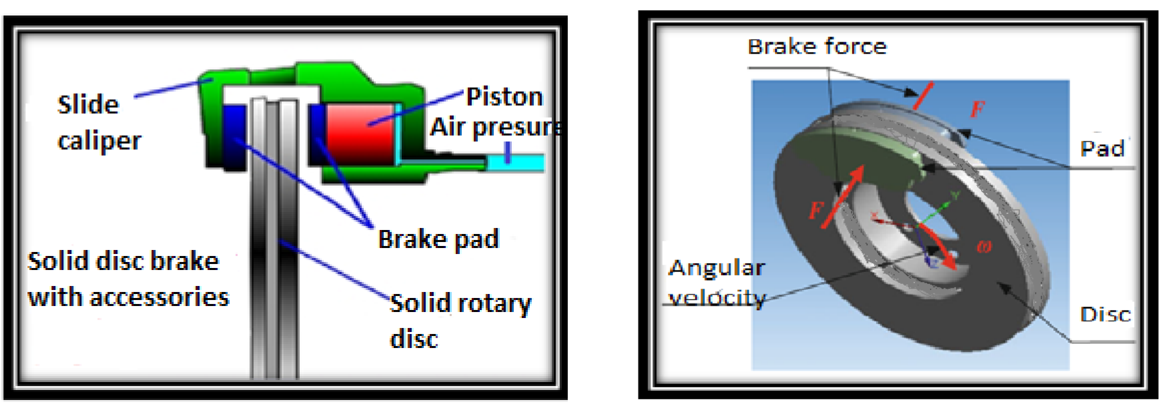


Figure 1.sliding solid disc-pad brake with accessories and assembly with forces applied to the disk [1].

Heavy-duty vehicles such as commercial vehicles built for a high utilization. The life of a commercial vehicle designed for a minimum of 15years (2,000,000km) and during this time, some of its components should be replace at planned intervals. A costly and potentially dangerous situation may arise when specific observations made for a component, demanding unintended replacement causing immobilization and downtime of the vehicle [1].

The brake system is most important safety control part of any vehicle. It used for stop or retard the motion of vehicle under emergency or normal condition [2].

During the braking phase, large amount of kinetic energy is transform in to heat energy through the friction at interface of disc and pad. Out of total heat energy, part of heat dissipated in disc and pad by conduction and remaining amount of heat dissipates to surrounding atmosphere by convection and radiation [3].

Due to the heat dissipation, temperature gradient generated through the thickness of disc and compressive stresses are generates on both side of disc. After released the brake an outer surface cooled and generates the tensile Stresses. Due to these tensile stresses, after certain number of brake actuation. Volvo B11R model bus are now become the most common passenger vehicles throughout the worldwide.

1.2 Statement of the problem

There were researches done on transient thermal and structural analysis of disc brake which are described in literatures. Most of the researches are concentrated on analytical (theoretical method) experimental or finite element method, in each study only one of the methods was applied, which is not satisfactory for accuracy of the analysis. But this thesis is done both analytical and by finite element which strengths the pre-mature failure result obtained. This research done on the problem in Volvo B11R Bus disc brake, which Involves pre-mature failure as shown in the figure 2. Among the 50 Volvo B11R buses imported, 10 of the buses exposed to the premature disc brake failure before service life of 500,000 km or within 2-3 months of service life. This high stress and deformation on the bus disk brake rotor results in:-

- weak brake efficiency and
- Increases the rate of accident. In addition, this
- leads to financial loss to the supplier and customer
- The business relation and trust between the customer and supplier will be badly affected

Experimental analysis is difficult to estimate the thermal structural analysis between disc and pad, where there is heat dissipation over time and high speed rotation of disc. Experimental determination of temperature, stress and deformation of a surface of contact concerning authentic objects in most cases causes significant technical difficult and is connected with pre maturity failure on Volvo bus disc brake. Instruments which pass the obstacle which is expensive of these instruments. Taking this in to consideration, the purpose of this paper is aimed at solving of the above challenges by showing the reason why early failure happened on the bus disk brake rotor by using mechanical engineering software's. Using solid work modeling & finite element /ANSYS software method, the results have obtained.

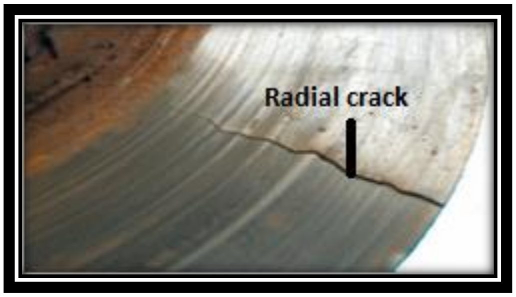


Figure 2 Actual photo of a premature failure of disc brake on Volvo B11R buss model

1.3 Objectives

1.3.1 General objective

The general objective of this paper was to make steady-state thermal and static structural analysis using finite element method (FEM)

1.3.2 Specific Objective

The specific objectives of this research works were-

- To obtain necessarily data for model the front disc brake
- To study the deformation and von misses stress distribution to apply a parametric values of retention time, force and torque on this modeling of solid disc brakes
- To compare the FEM result with the actual prematurity failure on Volvo B11R Buss disc brake.

1.4 Research Methodology

The methods employed to achieve the above objective are:

- Literature review: giving consideration to the researches done relate to this problem and creating a new way of solving the problem based on those researches.
- Necessarily data collection: taking the input data from the bus's workshop and Volvo disc brake supplier from Gutenberg by using my Valdo ID number.

- Software modeling and analysis: the modeling and numerical simulation is done using solid works and ANSYS software with respectively.

1.5 Significance of the thesis

The significance of the proposed thesis work is the following

- Reduce accidents.
- Protects the supplier from financial losses.
- Good satisfaction for the customer

1.6 Beneficiaries

- Automotive industries, Volvo dealers, vehicle owners, researchers, society and country as a whole would benefitted with this work.

1.7 Delimitation (Scope and limitation of the study)

- The scope of this work was necessarily data collect, modeling by solid work and finally analysis part was interpreted by finite element work bench which is steady-state thermal and static structural analysis using finite element method (FEM)
- The limitation of this work was the lack of the minimum allowed dimension of the neck area torque (force applied area on a pad) on Volvo B11R model disc brake in our country for the analysis that forces me to go to Dubai (Doha marketing service).
- Another limitation is that only the front brakes investigated in this study because they normally prioritized due to the brake torque distribution higher in front, that normally provides a lot of absorbed heat energy.

1.8 Organization of thesis

This thesis organized by five chapters;

In the first chapter, background and justification of this thesis work and the objectives be achieved are discussed.

In chapter two, a review of literature, which investigated by different researchers, were given.

Chapter three is about analytical method in contact between solid disc and pads stress analysis. With finite element method (FEM) is used, to develop 3-D model of disc brake.

In addition, different loading and mesh size on gray cast iron disc brake and conditions discussed in detail.

In chapter, four results of the analysis summarized and discussions made based on the outputs of the FEM. In addition, comparison of analytical and numerical solutions made. Finally, chapter five gives conclusion and recommendation achieved from this thesis work and propose future work in this field of study.

Chapter Two

2. Literature Reviews

The application of brakes to retard the motion of vehicle could cause tribological effect. As a result frictional heat over the surface of disc could developed. At normal braking, slowly apply the brake to retards the motion of vehicle and generates small heat at interface of disc and pad. However, in emergency or sudden braking, large force is applied and generates large amount of heat energy at interface of disc- pad. This generated heat is dissipates by conduction, convection and radiation out of the total heat dissipated, most of the heat dissipation is by conduction and convection and very small amount of heat dissipation is by radiation.

At braking time, there is prevention to transfer the heat by convection at contact surface and hence heat dissipates by conduction through the disc thickness and generates the temperature gradient in disc. Thus, outer surface has the maximum temperature as compared to interior parts of disc, which helps to generates high stress and deformation. The condition of braking is very much severe and thus the steady state thermal and structural analysis attracts the attention of researchers. Taking this in to consideration, many researchers have been done about the brake disc.

Abd Rahim Abu-Bakar and Huajiang Ouyang [4] this paper studies the contact pressure distribution of a solid disc brake as a result of structural modifications. Before modifications are simulated, four different models of different degrees of complexity for contact analysis are investigated. It is shown that the contact pressure distributions obtained from these four models are quite different. This suggests that one should be careful in modeling disc brakes in order to obtain correct contact pressure distributions. This work could help design engineers to obtain a more uniform pressure distribution and subsequently satisfy customers' needs by making pad life longer.

M. Nouby, et.al [5] proposes an approach to investigate the influencing factors of the brake pad on the disc brake squeal by integrating finite element simulations with statistical regression techniques. Complex eigenvalue analysis (CEA) has been widely used to predict unstable frequencies in brake systems models. The finite element model is correlated with experimental modal test. The 'input-output' relationship between the brake squeal and the brake pad geometry is constructed for possible prediction of the squeal using various geometrical configurations of the disc brake. Influences of the various factors namely; Young's modulus of back plate, back plate thickness, chamfer, distance between two slots, slot width and angle of slot are investigated using design of experiments (DOE) technique. A mathematical prediction model has been developed based on the most influencing factors and the validation simulation experiments proved its adequacy.

P. Liu a, et.al [6] An attempt is made to investigate the effects of system parameters, such as the hydraulic pressure, the rotational velocity of the disc, the friction coefficient of the contact interactions between the pads and the disc, the stiffness of the disc, and the stiffness of the back plates of the pads, on the disc squeal. The simulation results show that significant pad bending vibration may be responsible for the disc brake squeal. The squeal can be reduced by decreasing the friction coefficient, creasing the stiffness of the disc, using damping material on the back plates of the pads, and modifying the shape of the brake pads.

Rajendra Pohane, and R. G. Choudhari [7] FEM model is prepared for contact analysis. A three dimensional finite element model of the brake pad and the disc is developed to calculate static structural analysis, and transient state analysis. The comparison is made between the solid and ventilated disc keeping the same material properties and constraints and using general purpose finite element analysis. This paper discusses how general purpose finite element analysis software can be used to analyze the equivalent (von-mises) stresses& the thermal stresses at disc to pad interface.

Mazidi,S. et.al [8] In this study, the heat conduction problems of the disc brake components (Pad and Rotor) are modeled mathematically and is solved numerically using finite difference method. In the discretization of time dependent equations the implicit method is taken into account. In the derivation of heat equations, parameters such as the duration of braking, vehical velocity, Geometries and the dimensions of the brake components, Materials of the disc brake rotor and the PAD and contact pressure distribution have been taken into account.

V.M.M.Thilak, [9] In this work, an attempt has been made to investigate the suitable hybrid composite material which is lighter than cast iron and has good Young’s modulus, Yield strength and density properties. Aluminum base metal matrix composite and High Strength Glass Fiber composites have a promising friction and wear behavior as a Disk brake rotor. The transient thermo elastic analysis of Disc brakes in repeated brake applications has been performed and the results were compared. The suitable material for the braking operation is S2 glass fiber and all the values obtained from the analysis are less than their allowable values.

From experimental setup, it has found out that coefficient of viscosity generally decreases with increasing sliding speed and applied load but increases with increasing disc temperature of 230 degree and then decreases with above disc temperature. Furthermore specific wear rate found to increase with increase in sliding speed and disc temperature. [10]

Thilak and Krishnaraj [11] conducted a transient thermal and structural analysis of a brake disk to evaluate its performance under severe braking conditions and there by assist in disk rotor, design and analysis. The usage of new materials was investigated which aims at improving the braking efficiency and providing greater stability to the vehicle. This study done using ANSYS 11 software to analyze the temperature distribution, variation of the stresses and deformation across the disk brake profile.

The new materials under study were Aluminum base metal matrix composite and High Strength Glass Fiber composites. These materials have a promising friction and wear behavior as a brake disk. The transient thermos-elastic analysis of disks in repeated brake applications performed and the results compared to that of cast iron disk. The friction heat generated between two sliding bodies causes thermos-elastic deformation, which alters the contact pressure distribution. This coupled thermo-mechanical process referred to as frictionally excited thermos-elastic instability or TEI (Total economic impact) [12].

The friction heat generated between two sliding bodies causes thermos-elastic deformation, which alters the contact pressure distribution. This coupled thermo-mechanical process cause frictionally excited thermos-elastic instability leading to the development of none uniform contact pressure. And local high temperature with important gradients called ‘hot spots’ ,the formation of such localized hot spots is accompanied by high local stresses that can lead to material degradation and eventual failure, also, the hot spots can be a source of undesirable frictional vibrations, known in the automotive disk brake community as ‘hot roughness’ or ‘hot judder’ [13].

Talati and Jalalifar [14] presented a paper on Analysis of heat conduction in a disk brake system. They used transient heat equations with heat generation that is dependent to time and space. In the derivation of the heat equations, parameters such as the duration of braking, vehicle velocity, geometries and the dimensions of the brake components, materials of the disk brake rotor and the pad and contact pressure distribution have taken into account. The problem solved analytically using Green’s function approach. They concluded that the heat generated due to friction between the disk and the pad should ideally dissipated to the environment to avoid decreasing the friction coefficient between the disk and the pad and to avoid the temperature rise of various brake components and brake fluid vaporization due to excessive heating.

A unidirectional temperature gradient is simulated [15] as part of a thermo mechanical fatigue experiment aimed at reproducing the temperature gradient in the brake disc` friction area. Here, cyclic temperature loading applied to a fully constrained hourglass shape specimen. The constrained thermal at increased temperature induces mechanical straining of the material. The controlled temperature from 110 °C to 600 °C at the center of the specimen and held at its maximum for 150 s before controlled cooling to 110 °C in 180 s. The increase in temperature performed in 80s.from the figure that shows defect occurring at the friction surface of brake disc among different types.

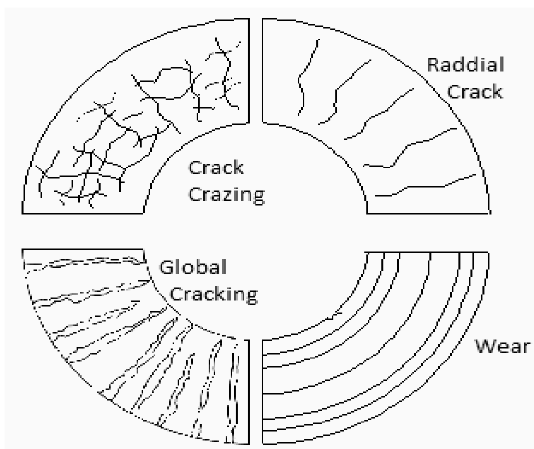


Figure 3. Types of defects occurring at the friction surfaces of brake discs (left) and permissible wear (right).

Chengal Reddy et al [16] describes the design, simulation and optimization of solid and ventilated brake disc using Pro-E modeling software, ANSYS-FLUENT.to evaluate the deformations, von-mises stresses was evaluated in the brake disc with the pressure applied on the pads attached to the disc.

To determine the temperature distribution on the disc brake from Equation this equation $\frac{\partial}{\partial x}\left(k_c \frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k_c \frac{\partial T}{\partial y}\right) + \frac{\partial}{\partial z}\left(k_c \frac{\partial T}{\partial z}\right) + \psi = \rho_c \frac{\partial T}{\partial t}$ physical conditions existing at the boundaries must be defined such as surface heat fluxes and convective heat transfer coefficients at free surfaces. By expressing this differential equation as a functional that is defined as some integral over the control volume and part of its boundary, the basic finite element method for heat transfer analysis can be derived using the variation approach the material properties required for the temperature analysis in brake discs include the density, specific heat capacity and thermal conductivity. Table 1 below shows the temperature dependent properties for the grade 150 cast iron used in the Rover disc design [8]. A constant density of 7050 kgm" was assumed.

Table 1. Material data for grade 150 grey cast iron [17]

Temperature (°C)	Conductivity (W m m 1 °C'1)	Specific heat (J kg'1 “C*1)
20	0.0533	103
100	0.0525	247
200	0.0515	427
300	0.0505	607
400	0.0495	697

In the scope of the present work both numerically by solid work, finite element method and analytical analysis have done and compared. None of the above approaches was available found fully compared with other result. In some cases, the geometry boundary conditions were inappropriate; in others, the published information was insufficient to allow direct implementation. As a result, the researchers decided to develop a new solution to meet our requirements.

Chapter Three

3. Analytical Analysis Methods and Conditions

3.1 Analytical Analysis Conditions

For comparison of steady state thermal and structural analysis between analytical and finite element, the Dimensions and specifications of Volvo B11R model bus has selected. The detail of this specification is in appendix I. Dimensions and specifications of this car has taken from Volvo supplier in Sweden. B11R model bus manufactured by Volvo Sweden Company, and Volvo Company one of a sole Deller Company in Ethiopia imports Assembled Vehicles and spare parts. The demand of this bus is increasing throughout the world because of comfortability, durability of engine; gearbox and differentials even if there should have good fuel economic quality, nearly expensive comparing to other buses. Horsepower of this bus is 430 hp (horsepower) this power leads to higher contacts between disc and pads (the minimum allowed dimension of neck area). Due to this effect, I would like to make steady state thermal and structural analysis on Volvo B11R bus model on disc brake.

Figure 4. Volvo B11R model bus [18]

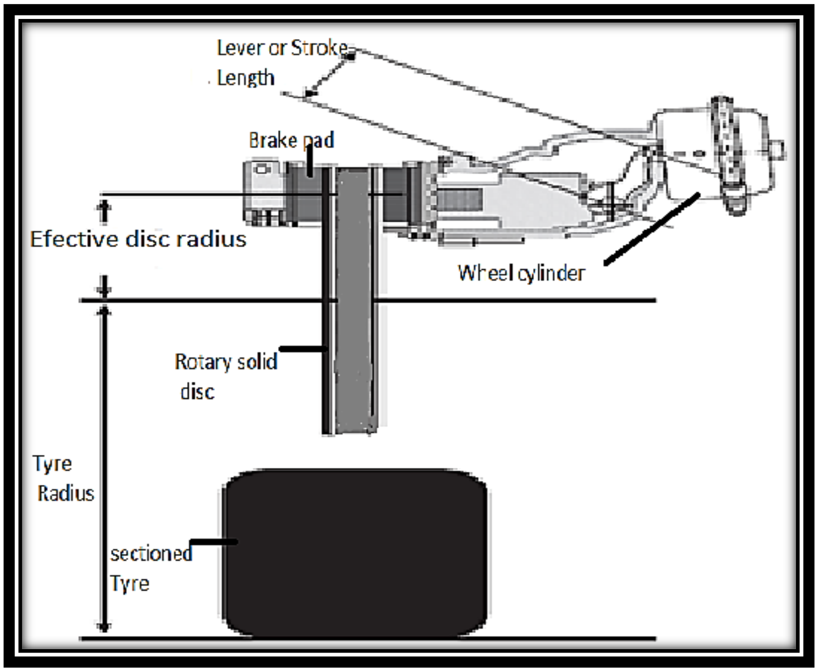


Figure 5. Photo of Brake actuators with sectional tyre on Volvo workshop

Dimensions of disc and pads used for development of 3-D drawing. Radius and thickness, as well as other complex structures used as an input for analysis. These dimensions have taken by measuring on Volvo busses front wheel disc brake-pads and some of the necessarily data's was taking from Volvo manual software, which is impact version four or URO 6 these are:-

Table 2 Geometrical dimensions and application parameters of Volvo B11R bus solid disc braking

Description	**Measured values**
Outer Disc brake diameter, Ø	434mm
Sweeping surface of the brake disc per pad, cm	948 cm^2
Brake disc thickness, mm	45mm
Weight of the bus GVW(kg)	24,750
Maximum permitted speed of the bus	120 mph
Contact conductance is φ	0.1= [W/NK]
Convection coefficient	50 [W/m2K]
Deceleration	15m/s^2
Tire size	295/80R22.5''
Over all height of the bus(mm)	3,236
Radius of inner brake pad on the brake disc, mm	130mm
Thickness of new brake pad, mm	20mm
Inner disc brake diameter,Ø	206mm
Elastic modules E (Gpa)	138
Poisson's ratio,*v*	0.28
Initial velocity m/s	16.84
Tire radius R_{tire} mm	800
Density [kg/m3]	7,250

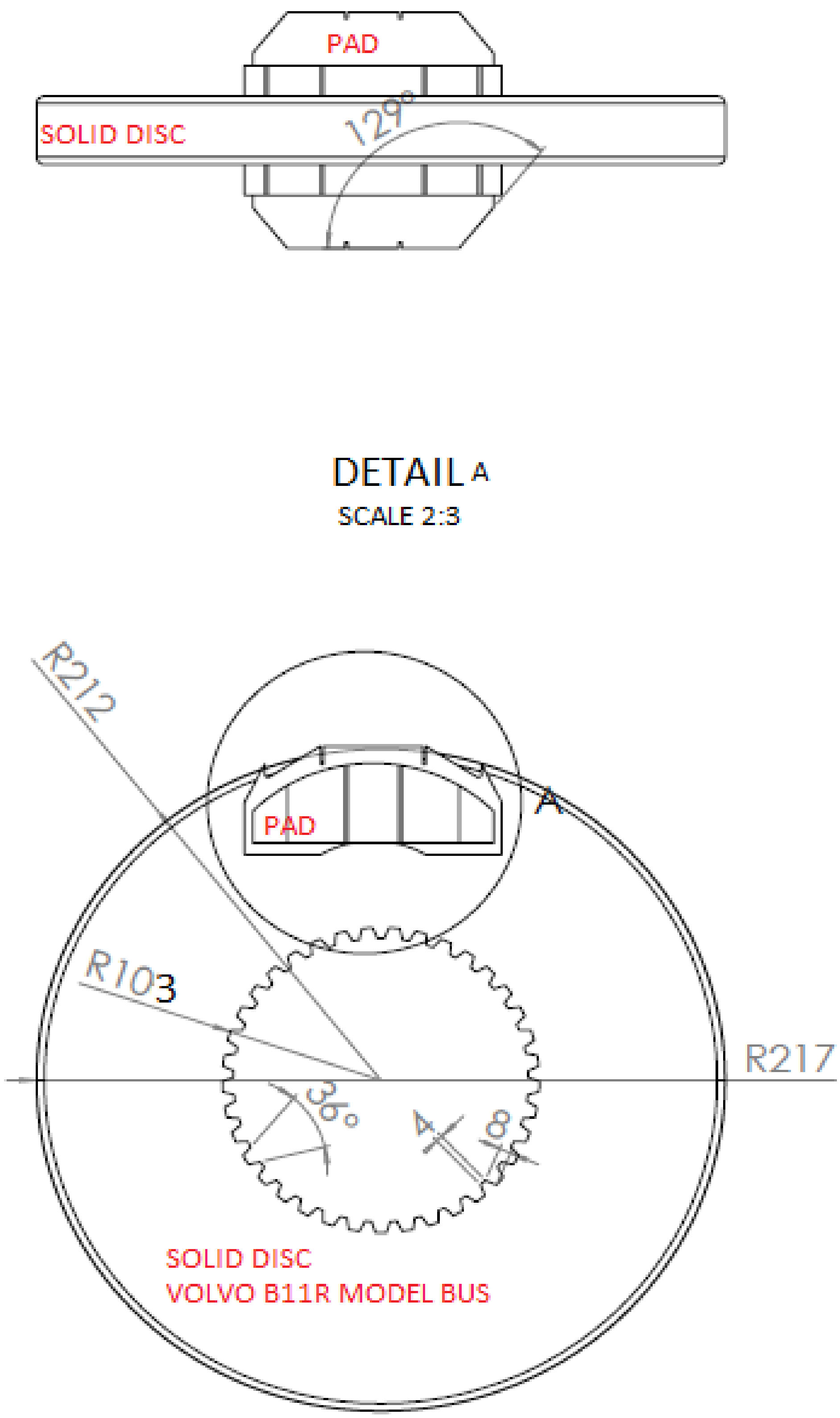


Figure 6. Modeling of actual Volvo rotary disc using solid work version 2015

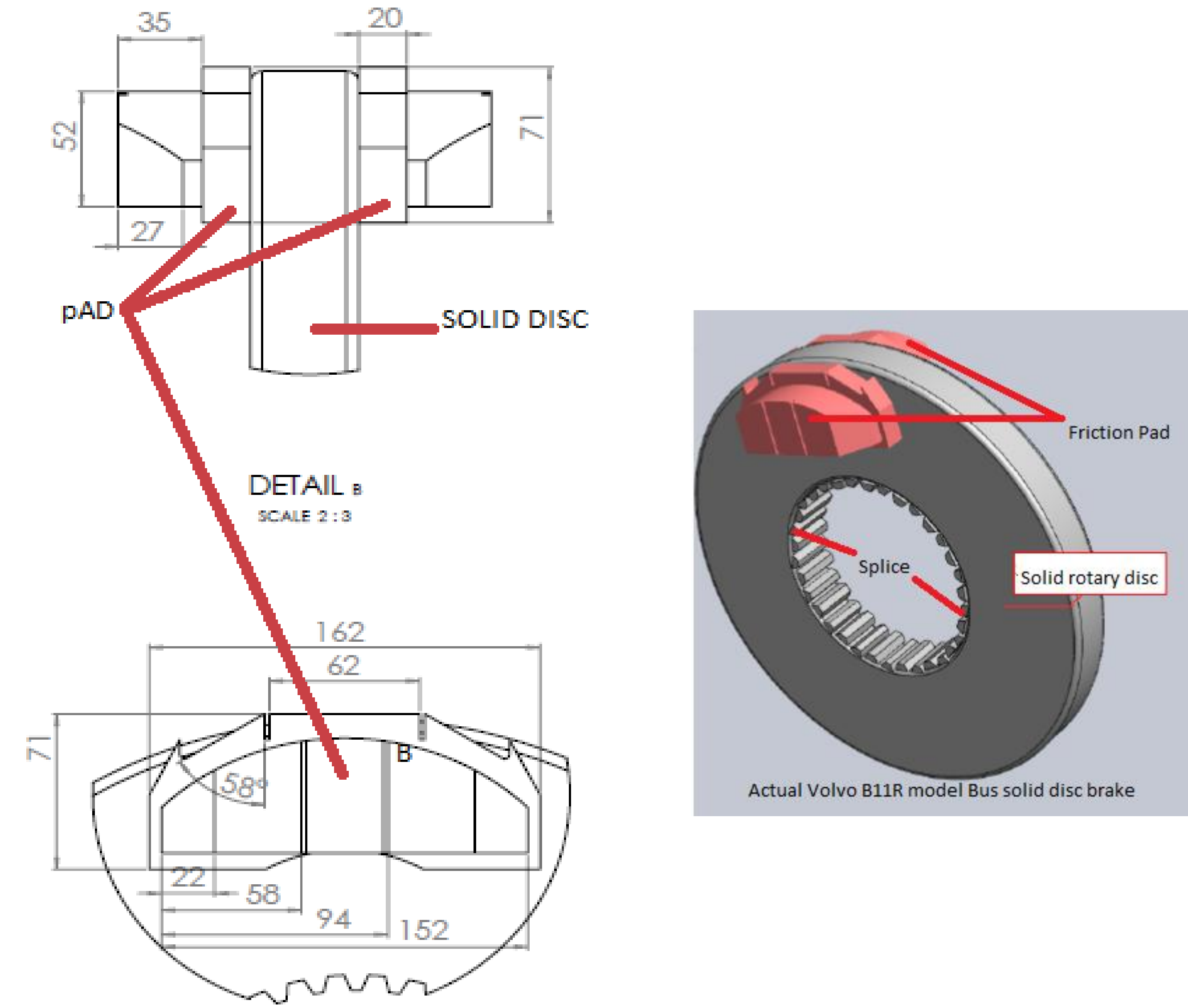


Figure 7.Modeling of actual Volvo disc brake-pad by solid work version 2015,

The values displayed in table 2 are found on figure 6 and figure 7. Cylindrical cordite system used to describe the dimensions. Different surfaces of disc or pad shown in figure 7 designated by the following symbols.

3.2.2 Method of Calculate the Maximum Brake Torque at Each Wheel

Brakes are usually specified in terms of the maximum braking torque that they are designed to deliver in use and this is related to the braking force at each axle of the vehicle using the following equations:-

- The maximum value of the torque was given by:-

$$\tau_{wi} = T_{wi} r_r = T_i r_r / 2 \ldots\ldots (1)$$

Where r_r is the rolling (or dynamic) radius of the tyre on the wheel and the maximum values of the braking forces (normal force) isT_{wi}.

1. The normal force is calculated by

$$T_{wi} = \frac{Frictional\ force}{cofficent\ of\ friction} = \frac{F_f}{\mu} \ldots\ldots (2)$$

2. The total Kinetic Energy (K.E) was calculated by

$$\text{K.} E_{Total} = K.E_{Transverse} + k.E_{Rotation} = \frac{1}{2}\text{m}v^2 + \frac{1}{2}I\omega^2 \ldots\ldots (3)$$

Where: - I=$\frac{1}{2}mR^2 and\ V_t = R\omega$

The total kinetic energy = the heat generated(Q_g)

3. Number of rotation before coming to stop can be calculated by

$$\text{K.}E_0 = heat\ lost = force\ of\ friction(F_f) \times numberof\ rotation(r) = 2\pi r \times number\ of\ rotation \ldots\ldots (4)$$

Therefore, number of rotation= $\frac{K.E_0}{2\pi r}$……………………………………………………….. (5)

4. The Area of the rubbing faces A = 2* Π * (0.4342 – 0.2062) * 0.25= 0.229 m^2……....(6)
5. Heat Flux = Heat Generated / Time / twice the projected …………………………….. (7)

The analysis has done by taking the Brake Efficiency of 30% and hence the distribution of braking torque between the front and rear axle is 70:30. Due to this, we can determine heat flux both front and rear wheel brakes

6. Thus Heat Flux can be calculated by

$$\varphi = \frac{Q}{A} \ldots\ldots (8)$$

$$where: -A\ is\ effective\ area = oter\ diameter - inner\ diameter\ of\ solid\ disc$$

N.B The wall thickness did not involve in the disc brake calculations; the valve of the heat flux does not change wall thickness.

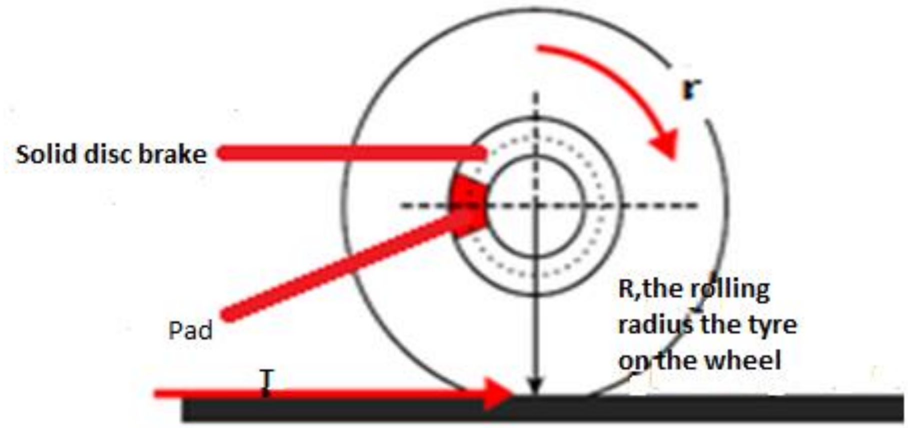


Figure 8. Disc Brake on Road Wheel Showing the Braking Force at the Tyre/Road Interface

Fitted to heavy commercial vehicles during the last 15 years, but drum brakes are still popular because of their robustness, low cost of ownership and ease of maintenance. Disc and drum brakes for commercial vehicles usually have standardized dimensions and the discs are usually the large amount of heat energy that needs to be dissipated because of the influence rotter size includes:-

- Torque and power rating
- Single-stop temperature rise calculations
- Fade and Alpine descent temperature prediction

Typical values of mean power density for a front wheel drive car during a 0.5 g deceleration to rest at GVW from 90% of vehicle maximum speed are: 8.5$MW/m^2$for front disc brakes; do you to these we can follow the next step.

3.3. Finite Element Analysis Methods

3.3.1 Conditions of simplified model of solid disc brake Volvo B11R model bus

The model of the disc is a circular clear from splines, the model cover of the hall entire disc. This means the degrees in the circumferential direction and the disc thickness. A circular part of the disc could possibly use the one, which has used in the more detailed model including the spline to cover the entire disc geometry. The simplified models shown in figure 6 & 7. The disc in the model has an outer diameter of 434 mm, an inner diameter of 206 mm and a thickness of 45 mm.

At the outer diameter of the disc model stress and deformation is placed (the solid disc is modelled full of a circle), this problem is found on Volvo B11R bus disc surfaces. To get a refined mesh around the surface, this way of working makes it possible to model any stress concentration in the full circular section shaped model; the stress exists in the model. The model placed at the same coordinates as the inner surface in the edge of the disc. This is bad condition when, placed in the middle of the disc the problem is loaded under plane strain conditions. The increased stress concentration is due to the transverse contraction strain, which is occurring in the z-axes direction, this strain makes the stress concentration near more emphasized.

Due to problems with convergence in the solution (probably because of a software error in ANSYS), only the stress-strain curve for 100^0C is used in the full circular model.

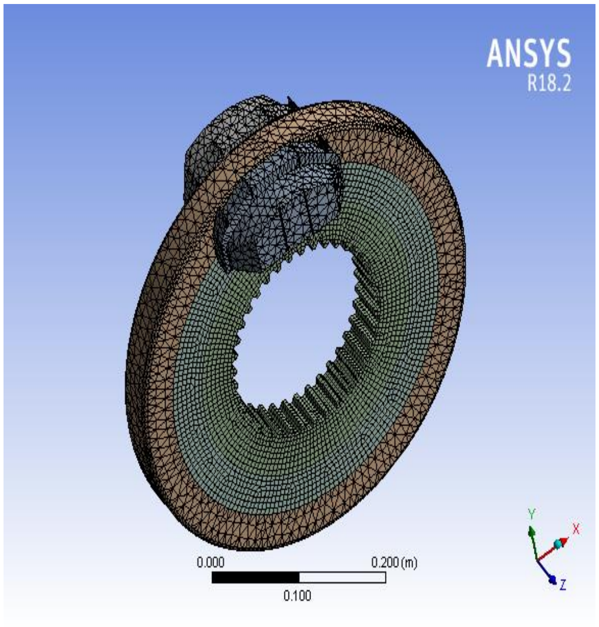
3.4 Modeling of the contract of disc to brake-pad

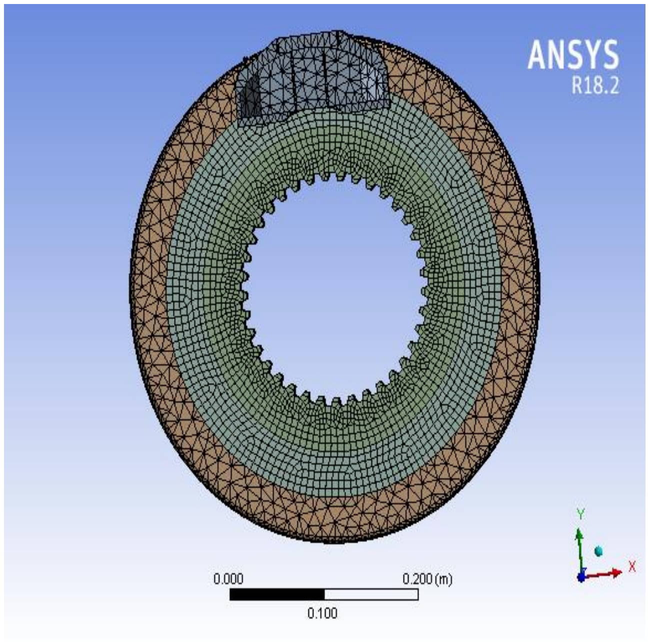
The following assumptions made when modeling the brake rotor in the finite element calculations:

- The brake pressure uniformly distributed over the contact area of the disc and pads.
- The friction coefficient remains constant during braking.
- The materials of the disk and pad are homogeneous and their properties are invariable with temperature.

3.4.1 Meshing the Model of Volvo B11R disc brake with pad

The finite element model of the rotor carried out such that the resulting elements came to 45,273 and 35,627 with a total of 146,514 and 114,641 nodes. The meshing sizes are six and seven respectively of the disc-pad as modeled in ANSYS R 18.2 is presented in Figure 10.as follow:-





a, Meshing size six b, Meshing size seven

Figure 9. Meshing of the disc and pad gives total number of nodes and elements are respectively number of nodes are 145,514, 114,641 and total number of elements are 45,273 and 35,627

3.4.2 Results of meshed models and comparison

A FEM test intended to evaluate the influence of the mesh on the accuracy of the numerical simulation. Three cases of meshing were tried (fine, medium, course) whose characteristics are shown in Tables 3&4 and Figures 8.

Table 3 Result of the different case of meshing (meshing size is six)

Mesh type (span angle center)	Number of Nodes	Number of Elements	Step End Time (s)	Analysis type	MAPDL Elap-sed time(s)
Finest	146,514	45,273	40	3-D	237
Medium	146,514	45,273	40	3-D	214
Course	145,960	44,855	40	3-D	244

Table 4 Comparison between finest and fine meshes

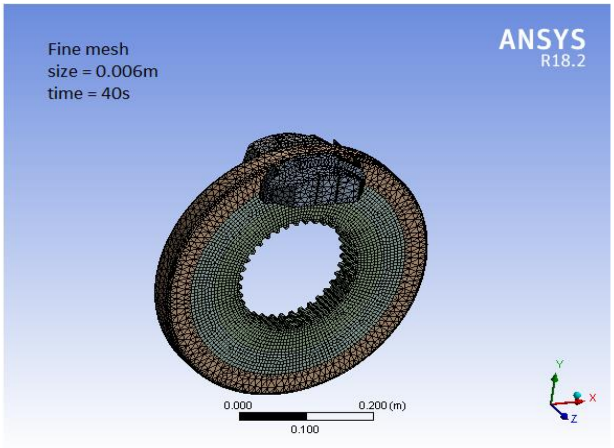
	Finest mesh size = 0.006m		Fine mesh size = 0.007m	
	nodes	elements	nodes	elements
	146,514	45,273	114,641	35,627
	Min	Max	Min	Max
Total deformation	0	2.2424e-005	0	2.2409e-005
Von miss stress in (pa)	38804	1.0846e+008	54944 Pa	1.0732e+008 Pa
MAPDL Elapsed time	237 second		186 second	

3.5 Tensile, compression and shear stress in the disk on Volvo B11R model bus

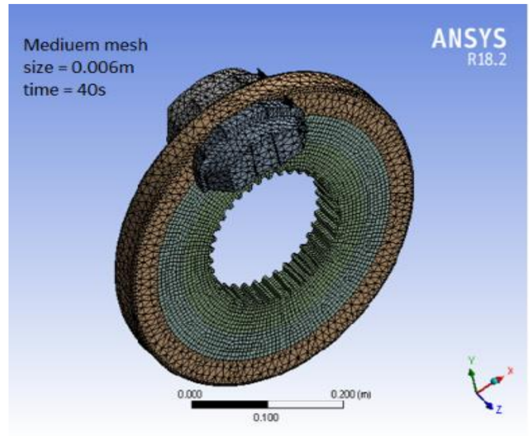
Tensile of compression stresses and shear stresses in the disc shown in Figures 13-16. During the rotation of the disc, there is a concentration of stresses at the fixing teeth (splice) and the connection area of the tracks to the bowl. Stress propagated on to the friction track versus time. The maximum value of the compressive stress is in the order of 1.0656e+007 Pa. This loading tensile stresses of 1.1757e+008 Pa. Shear stresses vary from (-2.7572e+007) Pa to (2.7826e+007) Pa. This loading format has an influence on the total deformations of the disc, which could take the shape.

3.6 Case of a disk without rotation

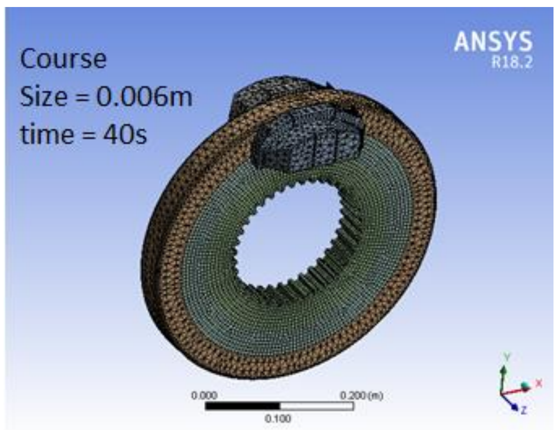
Assuming the disk is at rest, it noted that according to Figure 9, the Von Mises stress concentrations are located only in the bowl, but it does not spread on to the friction tracks, contrary to the case of disc with rotation. The total deformation varies from 0 m to 2.2424e-005 m



a, fine mesh



b, medium mesh



c, course mesh

Figure 10. Different meshing of the disc a, fine mesh it has 146,514 nodes and 45273 elements

3.6 Method of applying Loads

The first step in the calculations (Lagrange method) is to apply the heat load; this has done in a steady state thermal analysis where the second one is structural analysis shown below with the entire heat load applied outside the mean radius of the disc

3.6.1 Modeling of loading and boundary condition

For thermal analysis, the temperature distribution depends upon the heat flux entering the disc through both sides of the disc and heat transfer coefficient. For analysis, the initial and boundary conditions introduced in the transient thermal nodule of ANSYS WORKBENCH. The conditions for numerical analysis are as follows:-

Table 5 Conditions of thermal numerical analysis

Number Of Steps	180
Current Step Number	One
Step End Time	30s and 40s
Auto Time Stepping	Off
Define By	Time
Time Step	0.25s
Time Integration	on
Initial temperature of disc	22℃
Material of disk	Gray cast iron

For structural analysis, temperature and corresponding stress in disc brake vary under free way driving conditions. For analysis, the initial and boundary condition introduced in structural module of ANSYS workbench. Initial and boundary conditions area as follow:-

Table 6 Conditions of structural numerical analysis

Initial temperature of a disc	100℃
Pressure applied on both surface	1Mpa
Rotational speed of solid disc (ω)	45 rad/s

The loadings and boundary conditions of the finite element model were such that the following conditions were imposed (figure 10)

- The disc rotation was assumed constant at ω=45 rad/s according to Volvo impact software (just like specification manuals) specification.[uro 3]
- The cylindrical support used where axial and radial direction is fixed; and tangential direction free.

  The disc is attached to the wheel hub by splines and clamping plate, which keep the disc fixed in the three dimensional space.

The boundary conditions applied to the pads defined based on the movements allowed by the brake housing mounting that is the side motion. Indeed, the mounting allowed the brake housing and the pads to follow the movement of the disc when the two structures are in contract. The brake housing also holds the pads in the Z direction.

The mechanical loading represented by the plates, which in turn pressed by the brake air pistons. The pads press the disc, which generated friction on the rotating disc. The clamping force of the plates comes from the cylindrical air pistons. Thus the conditions imposed on the pads were;

- The pad is clamped at its edges in the orthogonal plane on the contract surface, thereby permitting a rigid body movement in the normal direction to the contract surface, such that found in Volvo B11R model bus brake assembly.
- Fixed support of the outer pad. The mechanical loading represented by the pads pressing on to the rotating disc, generating friction. The pad clamping force comes from the cylindrical air brake piston, pressing the metal back of the pads, which was the only load acting on the pads, which is a value of 24.5KN force acting on it.
- Air pressure P of 1MPa applied to the inner pad

For all the calculations that involve friction between surfaces, the coefficient of friction comes into play, whose value depends on many factors, including pressure, sliding speed, temperature, humidity and surface roughness between the two surfaces. Thus, the real value varies during braking. For simplicity, it assumed constant and a value of $\mu = 0.3$ used in all the calculations in this study.

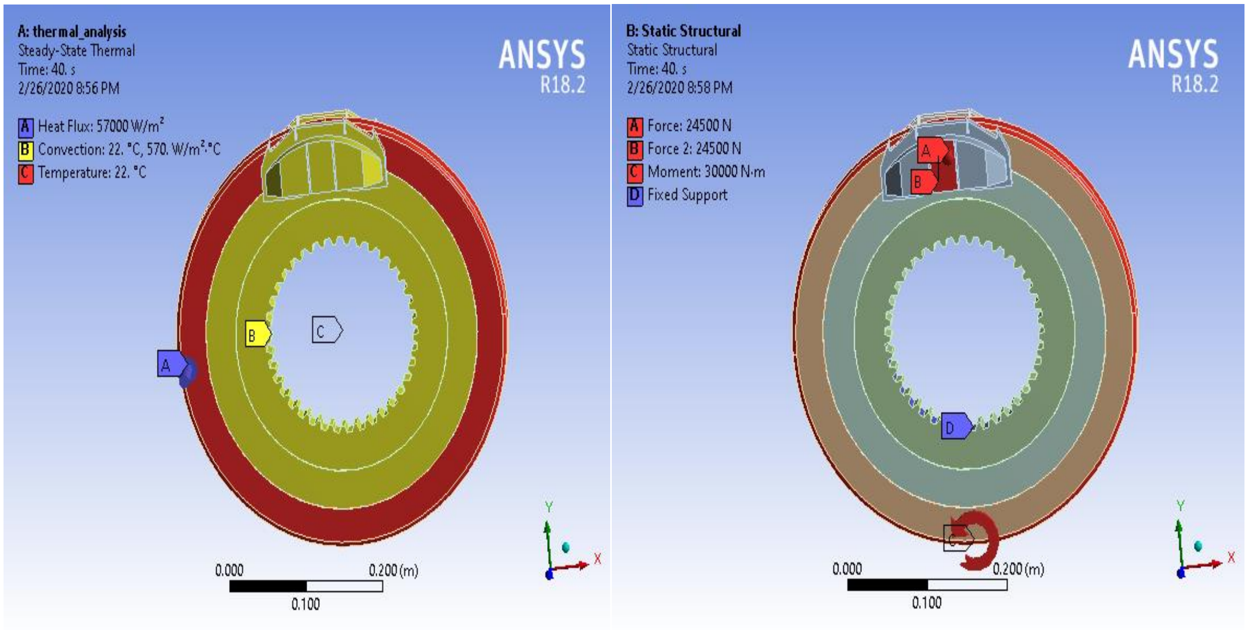


Figure 11. Boundary conditions and loading imposed on the disc-pad interface steady state thermal and static structural

3.7 ANSYS simulation of the problem

In this section, results of finite element approach for Steady-State Thermal and Static Structural analysis the result of the analysis summarized as follow:-

The finite element code ANSYS R18.2 (3D) used to simulate the behavior of the contact friction mechanism of the two bodies (disc and pad) during a braking stop. This code has the frictional contract management algorithms of the disc was about 138 times higher than that of the pads. The simulation presents in this study, considered the frictional pad to be deformable pad on a rigid disk.

In this study, it is assumed that 70% of the braking forces supported by the front brakes. According to Volvo, impact software .Using the Volvo B11R model bus data as given in table 3.and equations (41) - (44), braking force on the disc, rotational speed, brake pressure on the pad and torque can be calculated, respectively.

$$F_{disc} = \frac{(30\%)\frac{1}{2}mv_0^2}{2\frac{R_{roter}}{r_{tire}}\left(v_0 t_{stop} - \frac{1}{2}\left\{\frac{v_0}{t_{stop}}\right\} t_{stop}^2\right)} = 24.5\text{KN} \ldots\ldots\ldots\ldots\ldots\ldots (9)$$

The rotational speed of the disc calculated as follows:

$$\omega = \frac{v_0}{R_{tire}} = 45\frac{rad}{s}\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(10)$$

The external pressure between the disc and the pads calculated by the force applied to the disc

For a flat track, the air pressure is

$$\mathrm{P} = \frac{F_{disc}}{A_c\mu} = 1[Mpa]\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(11)$$

Where A_c is the surface of the pad in contact with the disc and μ is the coefficient of friction

Finally, the minimum allowed dimension of the neck area of the torque calculated by:

$$\tau_{wi} = T_{wi}r_r = 30\ \mathrm{KNm}\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(12)$$

Where r_r is the rolling (or dynamic) radius of the tyre on the wheel and the maximum values of the braking forces (normal force) isT_{wi}.

Due to these, the application of the contact pressure on the brake pad input as frictional contact data, and the disc rotational speed kept constant during the entire simulation. The material chosen for the disk was gray cast iron GG 25 high carbon content steel. The brake pad has considered to be made of an isotropic elastic material. The overall mechanical characteristics of the two parts summarized in Table 7. Parts design features provided by the ANSYS package; whose data is given in Table 8.

The friction coefficient is 0.3 in the contact zone. Friction is the product of the inter-surface shear stress and the contact area and it is a very complicated phenomenon arising at the contact interface. The coefficient of friction is actually a function of many parameters such as pressure, sliding speed, temperature and humidity. ANSYS is capable of using various methods to solve for the coefficient of friction, such as the Lagrange multipliers the augmenter Lagrangian method, or the penalty method. The latter selected for this work.

Table 7 Mechanical characteristics of disc and pads

	Disc	**pad**
Young's Modulus Pa	1.1e+011	3.93e+011
Poisson's Ratio,ν	0.28	0.28
Density	7200 kg m^-3	4900 kg m^-3
Coefficient of friction,μ	0.3	0.3
Angular velocity of the disc	45 rad/s	0(fixed)

Table 8 Design characteristics of disc and pads

parameters	Disc	pad
Volume (m^3)	6.1188e-003	4.0315e-004
Surface area (m^2)	0.945	0.080
Mass (kg)	38.25	3.951
Exterior Faces only	59	10
Nodes	145,514	6,764
Elements	45,273	3,784
Moment of Inertia Ip1	7.3299e-002 kg·m^2	1.0119e-003 kg·m^2
Moment of Inertia Ip2	7.333e-002 kg·m^2	3.6863e-003 kg·m^2
Moment of Inertia Ip3	0.14325 kg·m^2	3.7569e-003 kg·m^2
Mesh Metric	None	None

Chapter four

4. Results and Discussion

4.1 Mechanical calculation results and discussion (stress, deformation, von miss and thermal analysis)

ANSYS computed code also allows the determination and visualization of the structural deformations due to sliding contact between the disc and the pads. The results of the contact stress calculations described in table 15 and table 16 this section relate to some of the parameters with result values show on shows about the deformation and von miss stress on Volvo B11R bus solid disc brake in addition to this an overall results on Steady-State thermal and structural analysis as follow:-

The project relates to the heat generated during braking; prediction of the thermal increase, the consequences from increased temperature of components and increased cycling of temperatures and how the thermal energy be handled from a design perspective. The internal temperature of the brake disc can be quickly increased, even for relatively small amounts of brake usage. To show the thermal aspect the following example can be considered: A braking that lasts 40 s duration at only 20 % of the maximum brake pressure which is 1:5 ratio of brake pedal travel distance, the braking procedure is repeated 5 times with 1 mints cooling time between the braking applications; the initial temperature was 100 °C which is looking as a typical working temperature. After the 1st brake application the brake disc temperature is 169.8 °C, after the 2nd 231.7 °C, 3rd 206.3 °C, and 4th 351 °C after the 5th 420 °C. As stated above, these are internal disc temperatures, however, it is not the core of the brake disc which is prone to fatigue cracks, it is the surface of the disc. The temperature of thought surface will be significantly higher for the braking procedure stated above. Due to the severe working conditions during operation, macro-cracks might develop on the disc surface in the radial direction. Therefore it is necessarily to study a disc for the existence of stresses and deformations and estimate its fatigue life. In the second part thermal stresses have been computed for the disc. The stress analysis results show that during hard braking, high compressive stresses are generated in the circumferential direction on the disc surface which causes plastic deformation for the stress analysis, a frictional heat analysis was performed first. High temperatures in the middle of the disc surface. For this simulation of ANSYS, a brake force of 24.5 [KN] is applied for 40 [s] on the back surface of the support plate. The angular velocity of the disc is 45 [rad/s] and held constant throughout the simulation. The friction coefficient is μ =

0.3, the contact conductance is φ = 0.1 [W/NM] and the convection coefficient is set to 50 [W/m2K]. The temperature history of this frictional heat analysis is imported to commercial software, FEM, and a stress analysis is performed. To perform stress analysis for repeated braking, it is assumed that braking conditions are same for all the brake cycles so they generate similar temperature. From the results it can also be seen that the thermal strain has higher range than the mechanical strain, so it influences the total strain more. This can be explained by the temperature dependent thermal expansion coefficient. Such stress-strain cycles might generate radial cracks after a few braking operations resulting in low cycle fatigue of a disc brake.

Table 9. Ansys result values of steady state thermal with a retention time of 40sec.as follow in the table.

Temperature result in (℃)	Thermal Conductivity (W m^-1 C^-1)	Specific Heat (J kg^-1 C^-1)	Bulk Modulus (Pa)	Shear Modulus (Pa)	Young's Modulus (Pa)
169.8	59.73	800	2.9773e+011	1.5352e+011	3.93e+011
231.7	58.85	810	3.0002e+011	2.0010e+011	3.93e+011
260.3	57.01	845	3.2000e+011	2.4451e+011	3.93e+011
351	40.12	899	3.9213e+011	3.8875e+011	3.93e+011
420	35.22	899	4.0002e+011	4.2351e+011	3.93e+011

Table 10. Ansys result values

	Gray Cast Iron
Compressive Ultimate Strength Pa	8.2e+008
Compressive Yield Strength Pa	1.e+009
Tensile Yield Strength Pa	1.e+008
Tensile Ultimate Strength Pa	2.4e+008

This Ansys result value greater than specification values of gray cast iron because of this the material will come to fail before service life of the buss disc brake which is known as pre maturity faller.

To show the thermal aspect we use infrared thermometer operational manual measures the surface temperature of an object the following results was a braking procedure repeated five times within 60 seconds rest each brake when the bus moves fully loaded.

Table 11. Actual ODAA bus temperature measurement values when the brake disc undergoes repeated with in 1mint every 10sec for five times. (1st,2nd ,3rd and mean iteration result values of temperature test using infrared thermometer)

Brake applied with in 10km intervals, retention time 40sec and brake pedal stroke 20% of maximum.	First iteration Temperature result in (°C)	Second iteration Temperature result in (°C)	Third iteration Temperature result in (°C)	Mean values of Temperature result in (°C)	Vehicle speed in (Km/Hr)
1st brake applied	170	170	169.5	169	100
2nd brake applied	230	225	240	231.7	100
3rd brake applied	260	261	260	260.3	100
4th brake applied	350	348	335	351	100
5th brake applied	420	419	421	420	100

Table 12. Actual ODAA bus temperature measurement values when the brake disc undergoes repeated with in 1mint every 10sec for five times.(2nd iteration temperature test using infrared thermometer)

Brake applied with in 10km intervals, retention time 40sec and brake pedal stroke 20% of maximum.	Temperature result in (°C)	Vehicle speed in (Km/Hr)
1st brake applied @ 12 sec.	170	100
2nd brake applied @ 24 sec.	225	100
3rd brake applied @ 36 sec.	261	100
4th brake applied @ 48 sec.	348	100
5th brake applied @ 60 sec.	419	100

Table 13. Actual ODAA bus temperature measurement values when the brake disc undergoes repeated with in 1mint every 12sec for five times.(3rd iteration temperature test using infrared thermometer)

Brake applied with in 10km intervals, retention time 40sec and brake pedal stroke 20% of maximum.	Temperature result in (℃)	Vehicle speed in (Km/Hr)
1st brake applied @ 12 sec.	169.5	100
2nd brake applied @ 24 sec.	240	100
3rd brake applied @ 36 sec.	260	100
4th brake applied @ 48 sec.	355	100
5th brake applied @ 60 sec.	421	100

Table 14. Actual ODAA bus average temperature measurement values when the brake disc undergoes repeated with in 1mint every 12sec for five times.

Brake applied with in 10km intervals, retention time 40sec and brake pedal stroke 20% of maximum.	Temperature result in (℃)	Vehicle speed in (Km/Hr)
1st brake applied @ 12 sec.	169.8	100
2nd brake applied @ 24 sec.	231.7	100
3rd brake applied @ 36 sec.	260.3	100
4th brake applied @ 48 sec.	351	100
5th brake applied @ 60 sec.	420	100

Generally it is known that gray cast iron maintain their mechanical properties up to approximately 400℃,Yield strength 204 Mpa and Ultimate strength 306Mpa According to study of Jae-Hoon Kim [19].

On account of this thesis, the way and methods to optimization of time verses temperature graph of a disk brake is to be studied and determination of the highest quality and the longest lifetime of the component, it is recommended to have the following values.

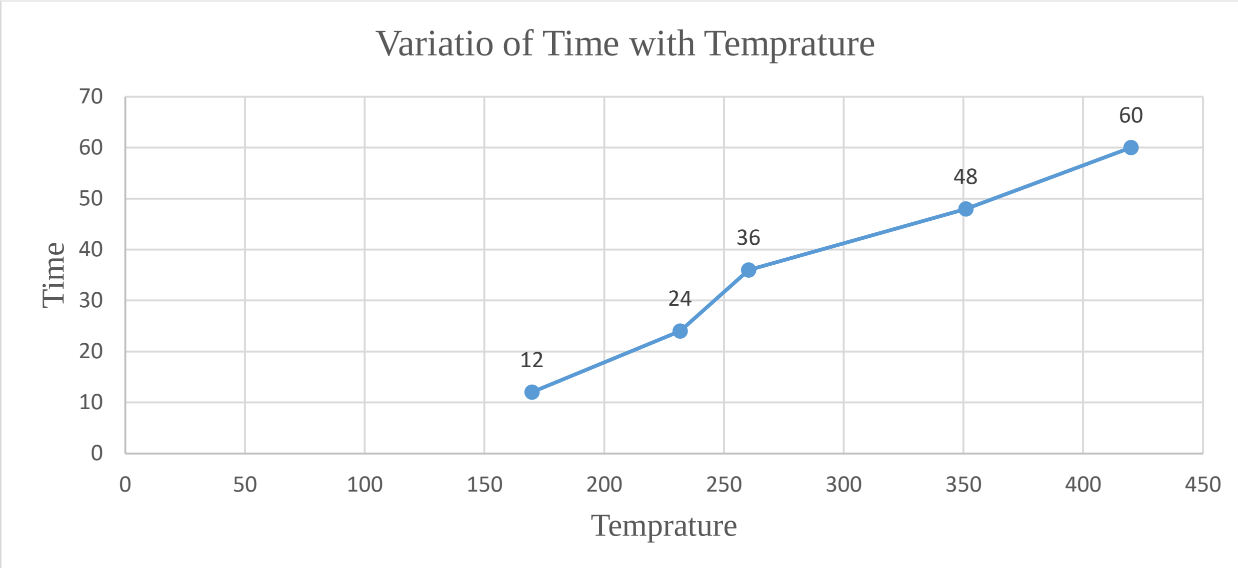


Figure 12. Optimization result on rotary Volvo bus disc brake time verses temperature.

Table 15 Result values of Steady-State Thermal and structural analysis

S. No	Parameter	Maximum Value	Minimum Value
1	Equivalent Elastic Strain	5.5735e-004 m/m	9.5006e-006 m/m
2	Directional Deformation	3.6653e-006 m	-3.6522e-006 m
3	Equivalent (Von Mises) Stress	1.0846e+008(Pa)	38804(Pa)
4	Constraints of friction	0.28	0
5	Sliding distance (m)	1.5857e-003 m	0
6	Specific Heat Constant Pressure	800 J kg^-1 C^-1	447 J kg^-1 C^-1
7	Maximum Principal Stress	1.1757e+008(pa)	-1.0656e+007(pa)
8	Thermal error between span angle center six and seven fine mesh	10.267 −10.267=0 (zero)	$3.5576e^{-5} - 3.5576e^{-5} = 0$ (zero)
9	Total Deformation (m)	2.2424e-005	0
10	$\sigma_{1,max}$	$1.2045e^{8}$	$-1.0919e^{7}$

11	$\sigma_{2,middel}$	$1.9744e^7$	$-2.2708e^7$
12	$\sigma_{3,minmum}$	$1.1275e^7$	$-3.1483e^7$
13	σ_{XY}	$2.3664e^7pa$	$-2.3664e^7pa$
14	σ_{YZ}	$3.5158e^7pa$	$-4.146e^7pa$
15	σ_{XZ}	$1.3552e^7pa$	$-107734e^7pa$
16	Maximum shear stress (pa)	5.3399e+007	-5.5347e+007
17	Temperature	By environment = 40.169((‘C))	By environment = 20.206((‘C))

Table 15. Different type of force, torque and retention time we get a value of Total deformation and Von miss stress

Mesh method	No. of nodes	No. of elements	Total deformation(m)		Von miss stress in (pa)		MAPDL Elapsed time(s)
at Torque=30KNM and Force= 24.5KN with retention time T=40s and mesh size is six			Max	Min	max	min	
fine	146,514	45,273	2.2424e-005	0	1.0846e+008	38804	237
medium	146,514	45,273	2.2424e-005	0	1.0846e+008	38804	214
course	145,960	44,855	2.2426e-005	0	1.0825e+008	53117	244
at Torque=25KNM and Force= 20KN with retention time T=40s and mesh size is six			Max	Min	max	min	MAPDL Elapsed time(s)
fine	146514	45273	1.9039e-005	0	1.0694e+008	30771	556
medium	146514	45273	1.9039e-005	0	1.0694e+008	30771	412

course	145960	44855	1.9039e-005	0	1.0673e+008	34023	253
at Torque=30KNM and Force= 24.5KN with retention time T=30s and mesh size is six			Max	Min	max	min	MAPDL Elapsed time(s)
fine	146514	45273	2.2424e-005	0	1.0846e+008	38804	327
medium	146120	45051	2.2424e-005	0	1.1479e+008	44819	324
course	145531	44598	2.2427e-005	0	1.1324e+008	39566	496
at Torque=25KNM and Force= 20KN with retention time T=30s and mesh size is six			Max	Min	max	min	MAPDL Elapsed time(s)
fine	146514	45273	1.9039e-005	0	1.0694e+008	30771	436
medium	146120	45051	1.9036e-005	0	1.1342e+008	33311	801
course	145960	44855	1.9039e-005	0	1.0673e+008	34300	328

The result values shows that decreasing torque and force while keeping retention time and mesh size at constant will result in decreasing in both total deformation and Von miss stress. Even though the variation of retention time and mesh size do not affected by total deformation and von miss stress having this result values interpreted the following ANSYS result figures as shown below.

The ANSYIS result in the figure with different colors represents the deformation and stress distribution in the disk brake. The colors represent the following:-

- Red color- the maximum total deformation and stress, which also means high chance of crack and wear happening.
- Orange color- high stress and deformation area but with a small decrease from the red region.
- Yellow color- show a decrease in both stress and deformation than the above.
- Green and light blue colors - represent a relatively low stress and deformation area which decrease the rate of cracks and wear.

- Blue color- the minimum stress and deformation occurrence area

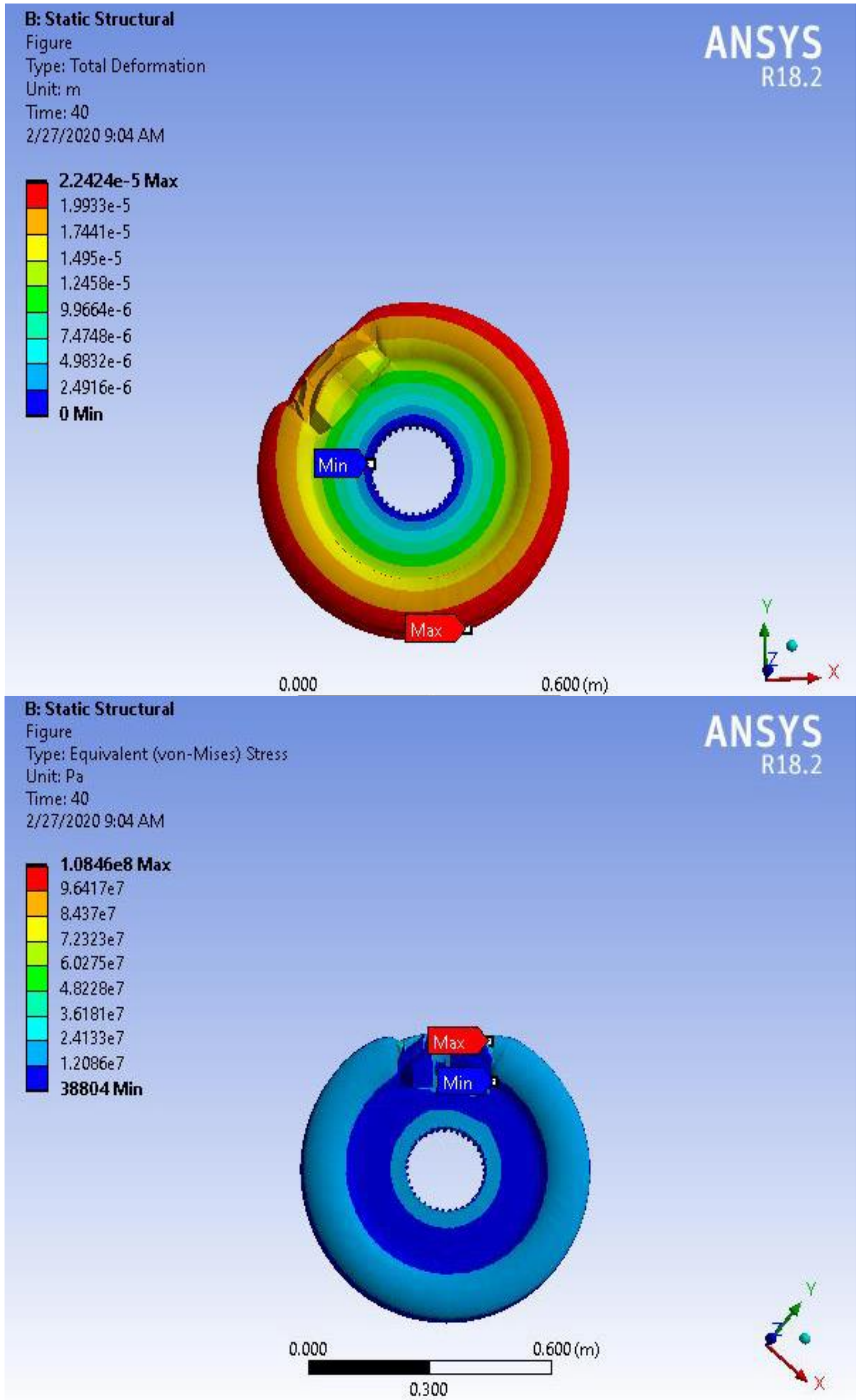


Figure 13. At Torque=30KNM and Force= 24.5KN with retention time T=40s and mesh size is six fine total deformation and von miss stress

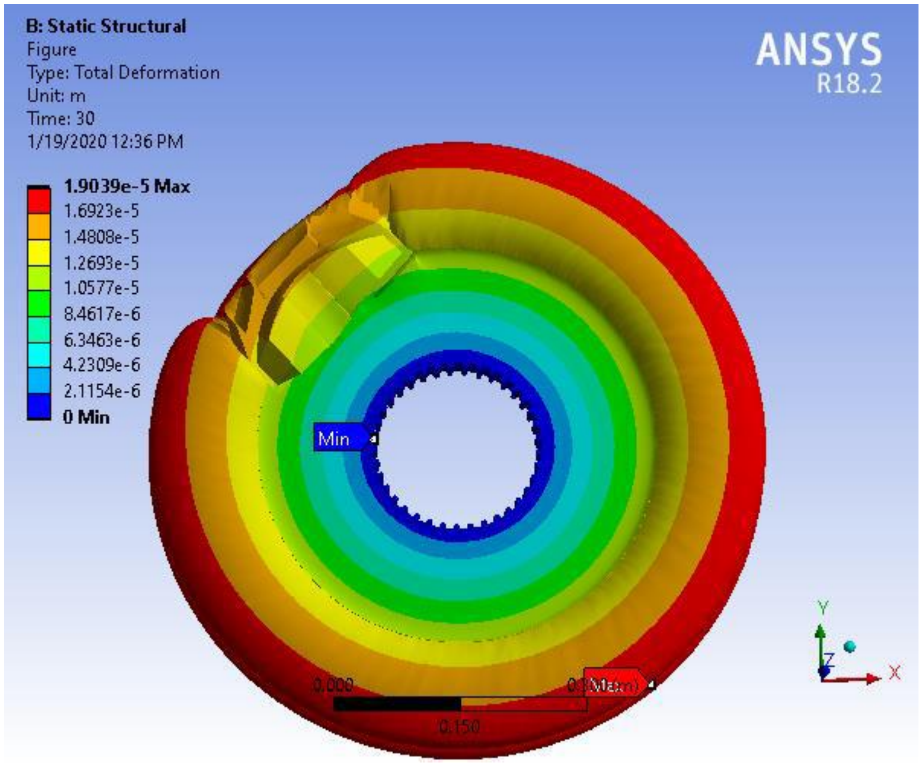
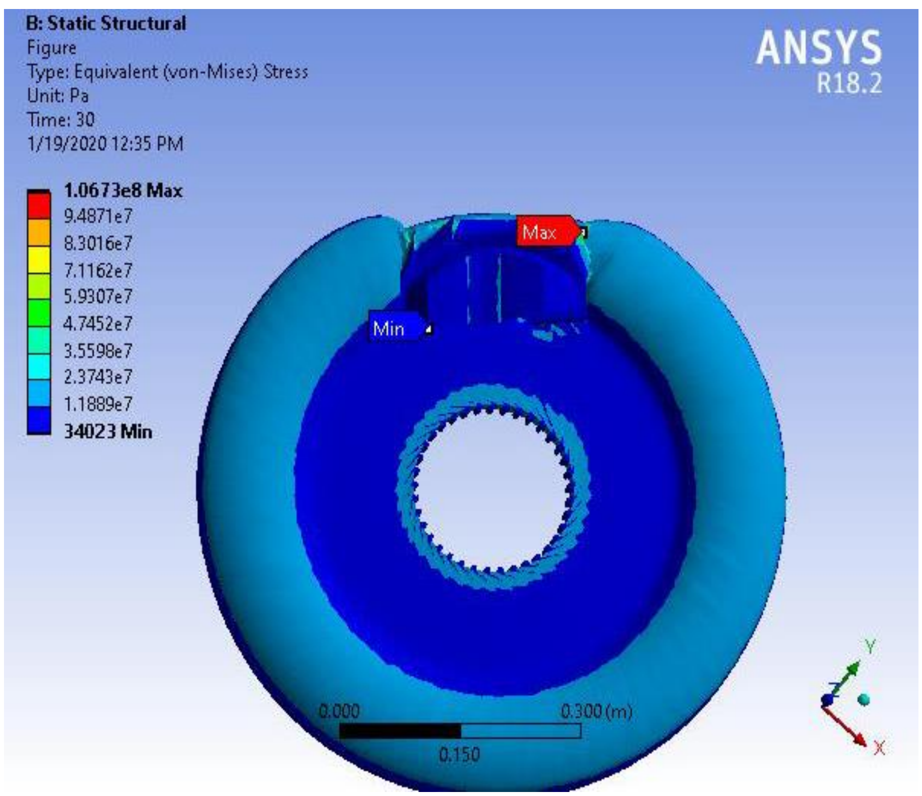


Figure 14. At Torque=25KNM and Force= 20KN with retention time T=40s and mesh size is six fine total deformation and von miss stress



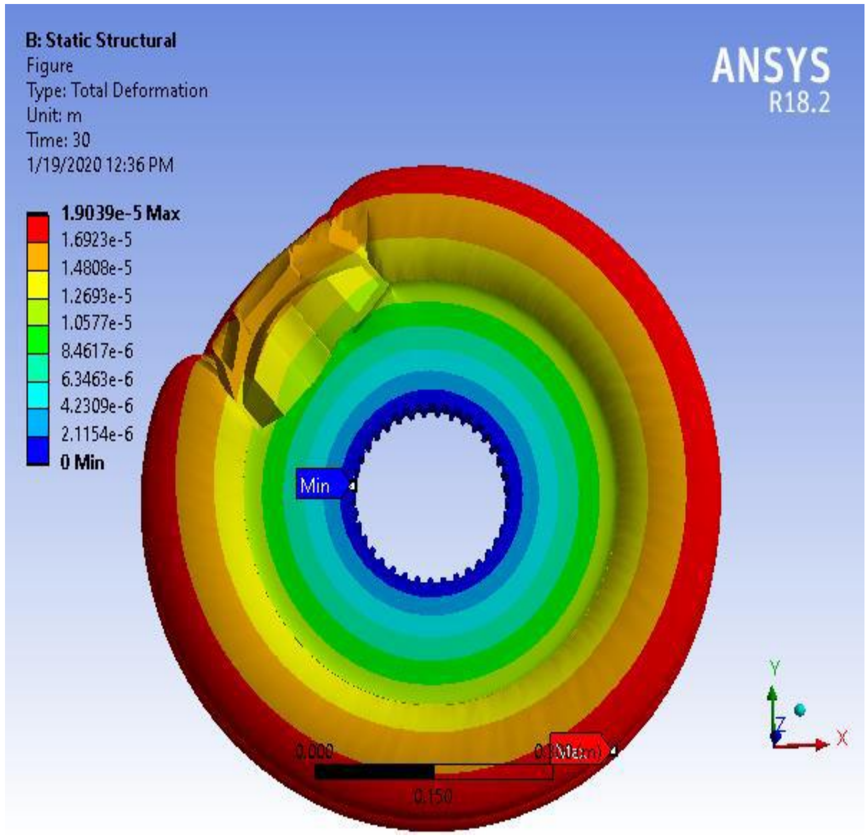
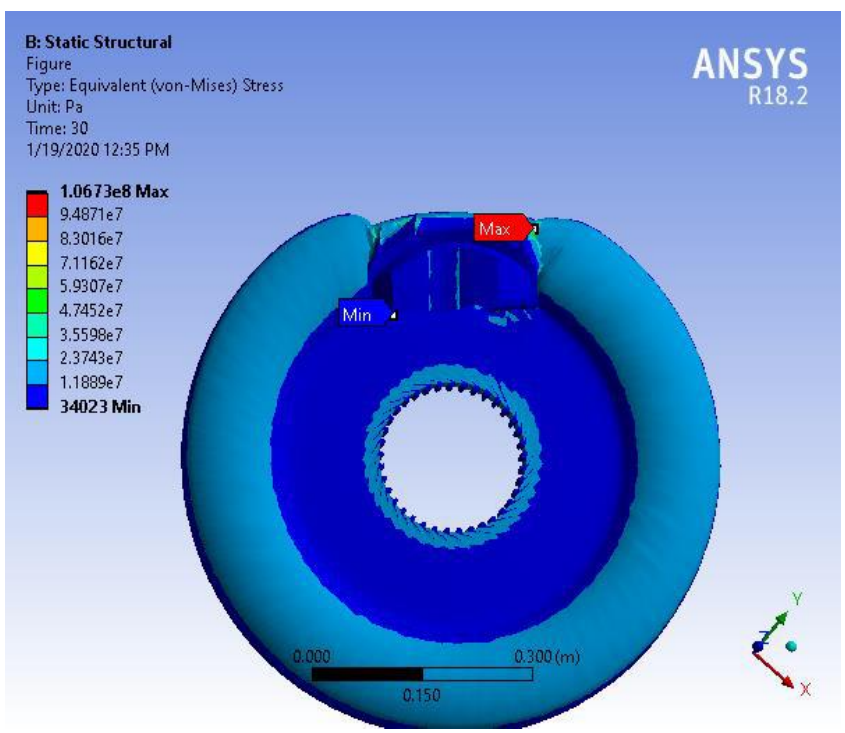


Figure 15. At Torque=25KNM and Force= 20KN with retention time T=40s and mesh size is six fine total deformation and von miss



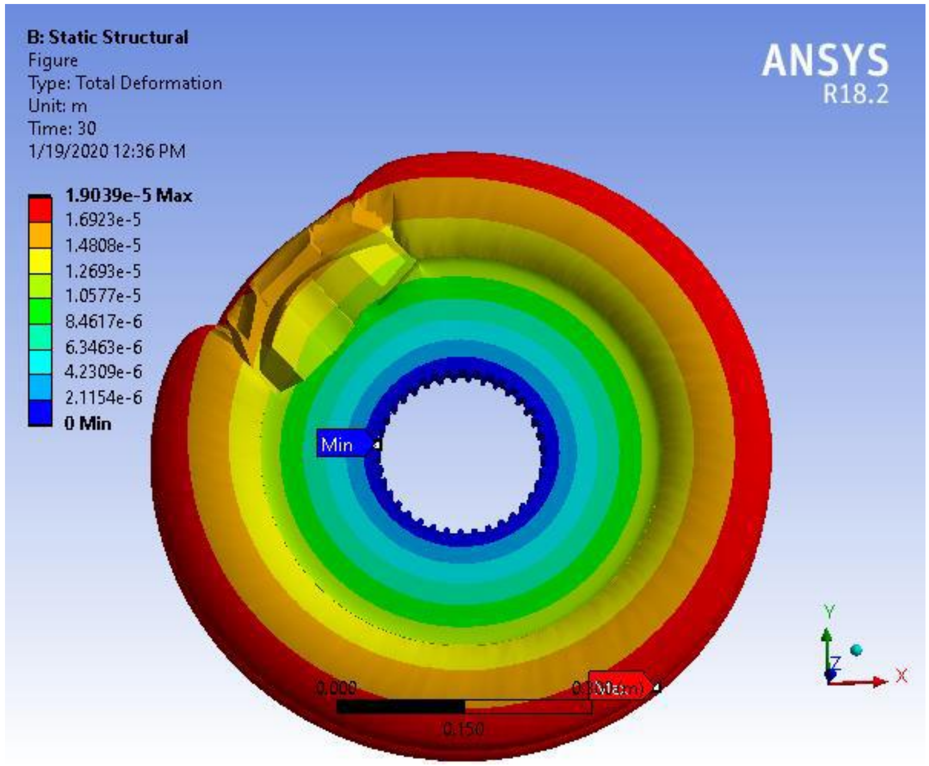
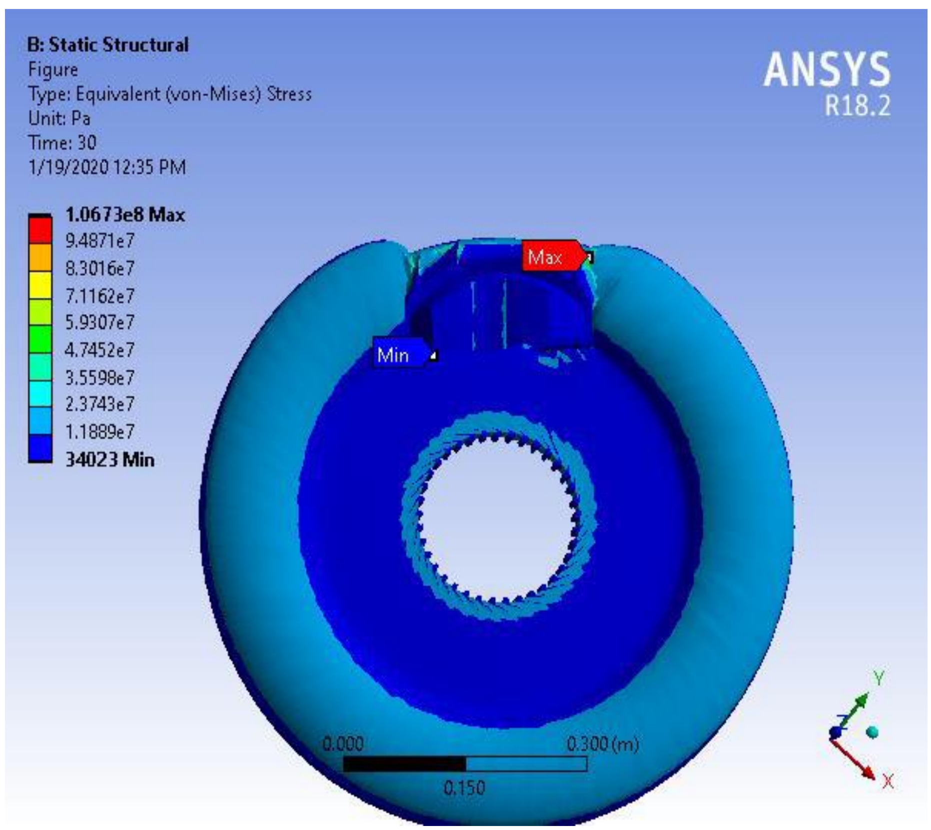


Figure 16. At Torque=25KNM and Force= 20KN with retention time T=30s and mesh size is six fine total deformation and von miss stress



4.2 The results can be concluded with this three findings (retention time, force and torque)

Decreasing torque and force while keeping retention time and mesh size at constant will result in decreasing in both total deformation and Von miss stress.

Example: - The maximum total deformation at Torque=30KNM and Force= 24.5KN with retention time T=40s and mesh size is six for fine materials is 2.2424e-005m. For the same material the maximum total deformation at Torque=25KNM and Force= 20KN with retention time T=40s and mesh size is six decreases to 1.9039e-005m.The Von miss stress also decrease from 1.0846e+00Pa to 1.0694e+008Pa.

- Keeping the torque, the force and the mesh size constant while varying the retention time the values in total deformation and Von miss stress remains constant.

Example 1: - The maximum total deformation at Torque=30KNM and Force= 24.5KN with retention time T=40s and mesh size is six for fine materials is 2.2424e-005m for the same material the maximum total deformation at Torque=30KNM and Force= 24.5KN with retention time T=30s and mesh size of six is 2.2424e-005m.

The Von miss stress also shows the same result.

- While keeping mesh size the same and decreasing all the other parameters the value of total deformation and von miss stress decreases.

Example 2: - The maximum total deformation at Torque=30KNM and Force= 24.5KN with retention time T=40s and mesh size is six for fine materials is 2.2424e-005m for the same material the maximum total deformation at Torque=25KNM and Force= 20KN with retention time T=30s and mesh size of six is 1.9039e-005m.

The Von miss stress also decrease from 1.0846e+00Pa to 1.0694e+008Pa.

Chapter Five

5. Conclusions and recommendation

5.1 Conclusions

The results have concluded with these three findings; (retention time, force, torque and thermal analysis) regarding the analytical results, we can say the maximum stress at the contact between rotary disc and pads and value of deformation, von miss stress and temperature are 2.2424e-005m, 1.067e+008 and 420℃ respectively.

Due to this and other values, they are satisfactorily in agreement with those commonly found in the literature investigations. It would be interesting to solve the problem in thermo mechanical disc brakes to validate the numerical results, for example, on ANSYS workbenches, in order to demonstrate a good agreement between the model and reality. Regarding the outlook, there are recommendations for the expansion of future work related to disc brake that would done for further understand the effects of thermo mechanical contact between the disc and pads, the recommendations are as follows:

5.1 Recommendation

- The parts with high stress concentration usually found in the disc bowl, the friction area between disc and pads, which could cause mechanical failures such as radial cracks, wear, and possibly breakage (fracture).
- The most favorable loading is the dual pressure bracket. Finer mesh increases we get the accuracy of the solution.
- The permitted temperature of gray cast iron disc brake is not more than 400℃.
- The choice of pad material would produce a different friction coefficient.
- Increasing the disc rotation speed decreased the equivalent Von Miss stress, disk shear stress, and principal stresses and the total deformation of the pads.
- The use of gray cast iron brake discs positively affects the mechanical behavior of the brake pad.
- There is no significant change in disc pad deformation with respect to the constant friction coefficient.

5.2 Future Work

In this thesis work steady state thermal and structural analysis of disc brake using solid work and ANSYS (case study on disc brake of Volvo B11R model bus) is studied, other influencing factors are not studied. So that the following additional works suggested.

- Study with considering tribological and vibratory study on the contact disc and pad of Volvo B11R model bus.
- Study using different materials to compare the effect of vane number, shape and van angle on Volvo B11R model bus disc brake.
- Study on hot spots analysis because of high local temperature, on solid disc brake on Volvo B11R model bus. Desirable performance hindrances such as brake fade or vibrations and judder.
- Vibration analysis caused by stress components Wear and noise analysis of disc brake caused by thermal heat indicate surface stress.
- Study the finest mesh using super computer in order to get a more accurate value of thermal error and large number of nodal and elements.
- Study the crack growth on Volvo B11R model Bus solid disc brake using contact fatigue mechanism.
- Test different material types

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Appendix I:

Specification of Volvo B11R model bus ant over all result of steady State thermal and structural analysis respectively as follow

Over all dimensions of Volvo B11R model bus, axle configuration 4 × 2	
Over all chassis length (mm)	10,188-12,688
Bogie (mm)	n/a (not applicable)
Frame height at rear structure (mm)	1,618
Transport wheel base (mm)	3,500
Track, front (mm)	2,068
Track, rear (mm)	1,836
Overall width front wheels or housing	2,500 mm
Overall width rear wheels or housing	2,480
Weight	
Permitted front axle load (kg)	7,500
Permitted drive axle load (kg)	12,000
Bogie axle load (kg)	n/a (not applicable)
Permitted tag axle load (kg)	n/a (not applicable)
Permitted GVW (Kg)	19,500
Engine	
Emission standard	Euro 6
Engine system	EGR, unit injector
Displacement (dm^3)	7.7
Cylinders/arrangement	6/inline
Output (hp)	430
Torque ISO 1985(Nm)at Engine speed (1,200-1,600)	1,400
Fuel tanks	

Diesel (L)	220,300,310,360,410
Transmission and axle	
Gear box	Volvo I-shift AT2412F
Front axle	Volvo RFS-L
Rear axle	Volvo RS1228C
Tag axle	n/a
Differential lock	optional
Suspension and steering system	
Air bellows, front	2
Air bellows, drive	4
Kneeling	optional
Maximum wheel angle	53^0 with 275/70R22.5 tyres
Power steering	Electric driven-hydraulic stering
Steering wheel position	LHS/RHS

Table 16 Overall results of Steady-State Thermal analysis

	Temperature	*Total Heat Flux*	*Directional Heat Flux*	*Directional Heat Flux 2*	*Directional Heat Flux 3*	*Thermal Error on span angle 6,fine*
Minimum	20.204 °C	34010 W/m²	-1.438e+005 W/m²	-1.164e+005 W/m²	-67039 W/m²	7.1293e-006
Maximum	40.166 °C	1.451e+005 W/m²	1.3045e+005 W/m²	1.3275e+005 W/m²	68563 W/m²	7.826

Table 17 Overall results of Static Structural analysis

	Total Deformation	Directional Deformation X Axis	Directional Deformation 2 Y Axis	Directional Deformation 3 Z Axis	Equivalent Elastic Strain	Maximum Principal Elastic Strain	Middle Principal Elastic Strain	Minimum Principal Elastic Strain	Elastic Strain Intensity	Shear Elastic Strain	Shear Elastic Strain 2
		Directional Deformation									
Minimum	0. m	-2.0792e-005 m	-2.2389e-005 m	-3.6522e-006 m	9.5006e-006 m/m	-2.1201e-005 m/m	-1.3157e-004 m/m	-3.7591e-004 m/m	1.4469e-007 m/m	-3.22e-004 m/m	-3.6053e-004 m/m
Maximum	2.2424e-005 m	2.2399e-005 m	2.2232e-005 m	3.6653e-006 m	5.5735e-004 m/m	4.2956e-004 m/m	5.2515e-005 m/m	-3.6284e-008 m/m	8.0547e-004 m/m	2.9856e-004 m/m	3.4784e-004 m/m

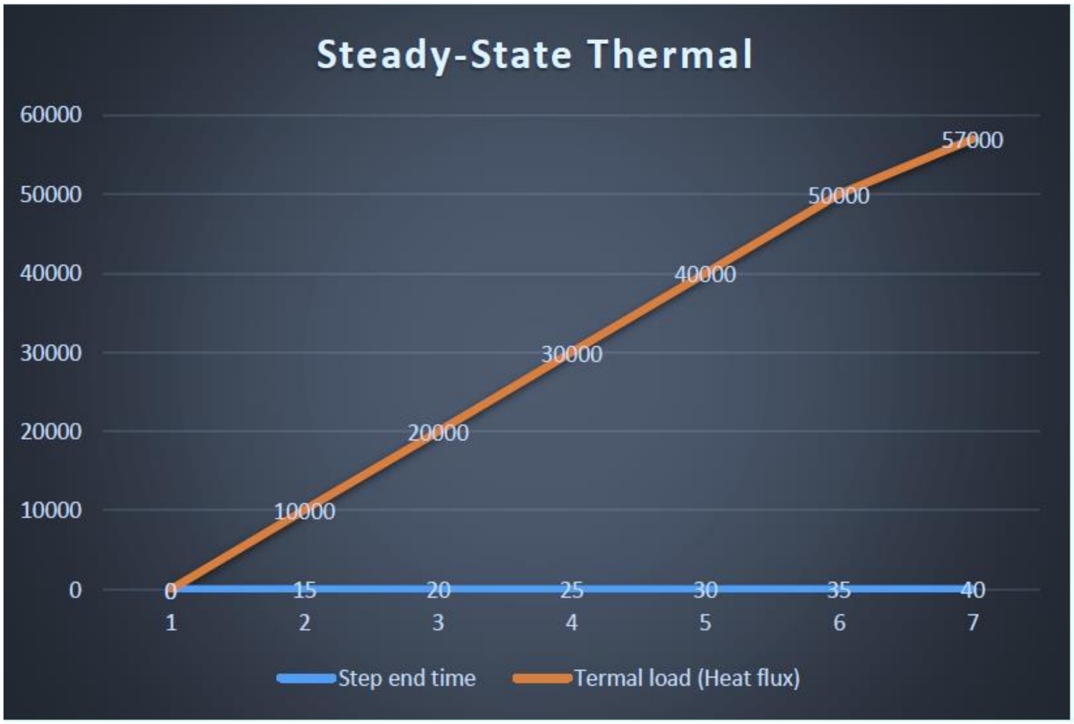

Figure 17 Steady state thermal vs time

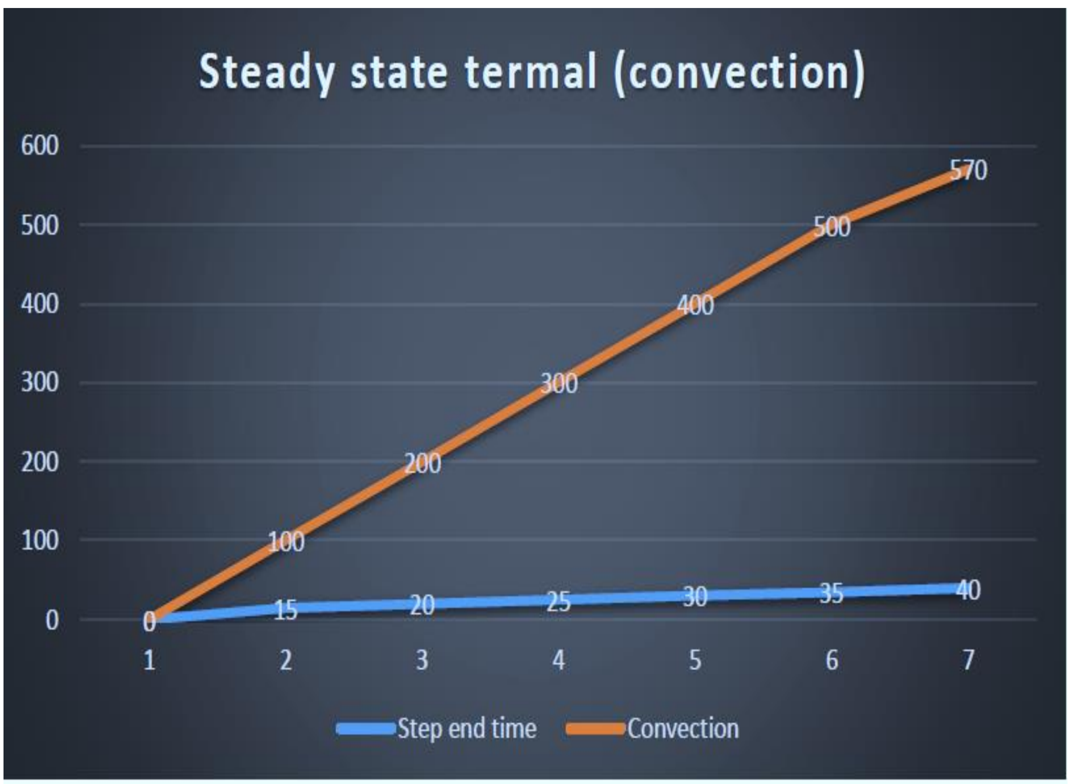

Figure 18 Steady state thermal (convection) vs time

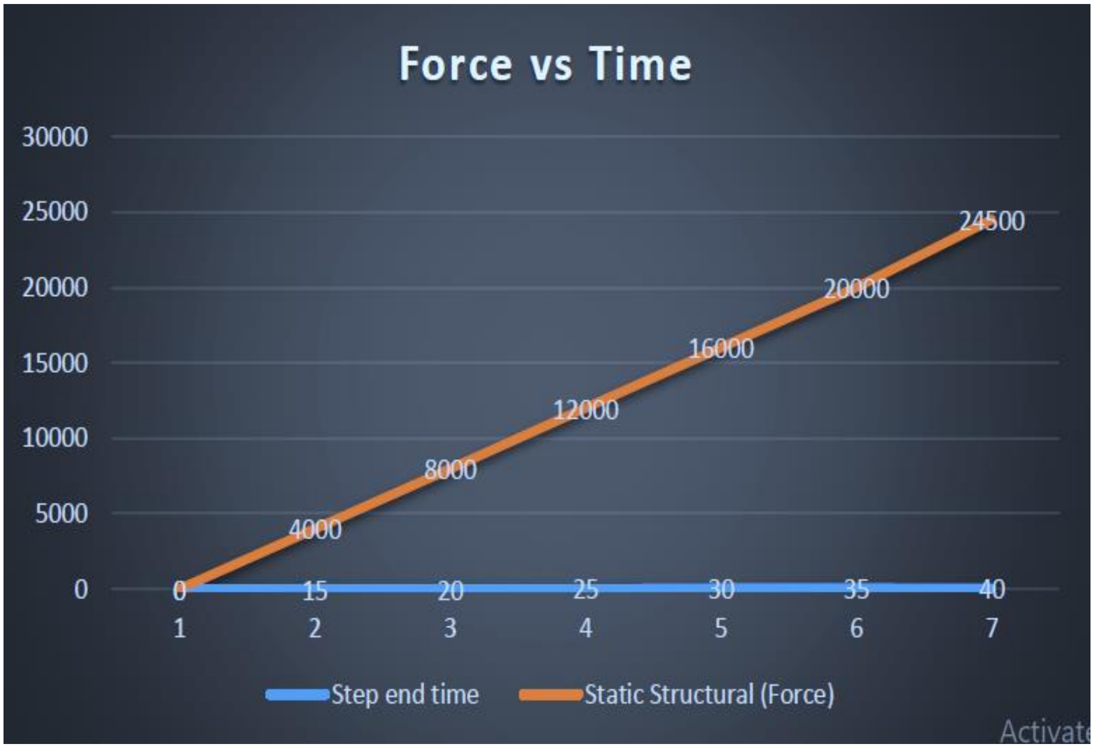

Figure 19 Force vs time

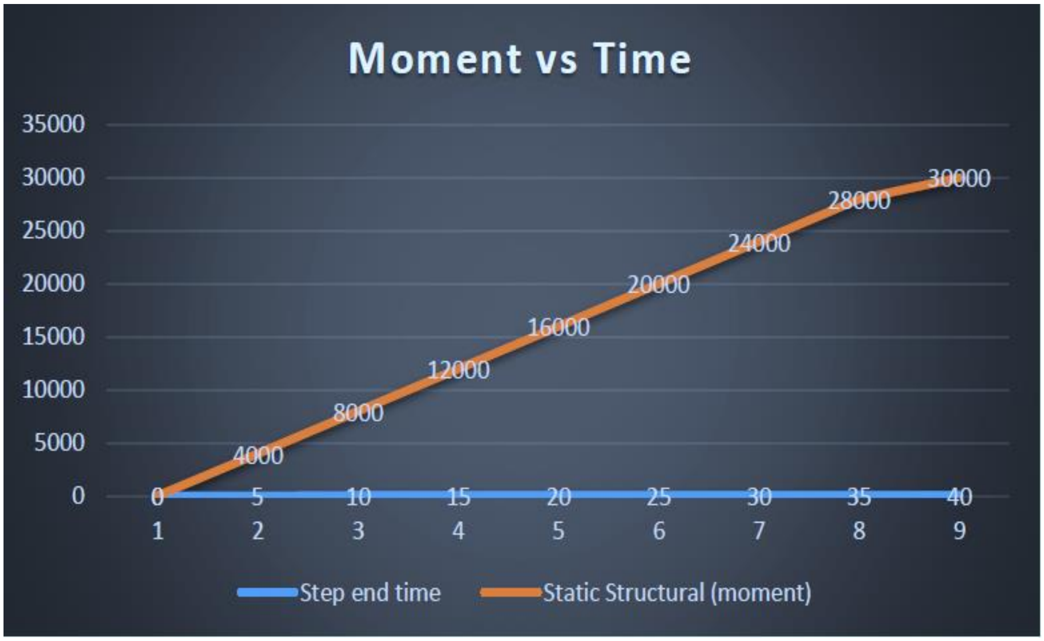

Figure 20 Moment vs Time