ZIBELINE INTERNATIONAL PUBLISHING
ISSN: 1024-1752
CODEN : JERDFO

Journal of Mechanical Engineering Research and Developments (JMERD)
DOI : http://doi.org/10.26480/jmerd.04.2019.09.13

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RESEARCH ARTICLE

MECHANICAL STRESSES ANALYSIS IN CYLINDER LINER FOR PERKINS 1306 DIESEL ENGINE

Abdul Wahab Hassan Khuder*, Ammar Ali AL-Filfily, Khalid M. Sowoud

Middle Technical University, Baghdad, Iraq, Engineering Technical College-Baghdad, Department of Power Mechanics Engineering.
****Corresponding Author Email: akhuder@toc.edu.iq***

ARTICLE DETAILS

Article History:

Received 14 April 2019
Accepted 24 May 2019
Available online 17 Jun 2019

ABSTRACT

A cylinder liner is defined as a cylindrical part that is fitted inside the engine block to form a cylinder. It is a vital component of cylindrical engine. The most important functions of cylinder liner are to form a sliding surface for the piston to obtain a smoothly reciprocating motion, resist the wear form the piston and piston rings and sustain high pressure and high temperature. However, continuous exposure to high thermal stress, mechanical stress and friction inside combustion chamber may cause failure and reduce cylinder liner life cycle. Therefore, the main objective of this research is to study the stresses due to action of gas pressure, major thrust force of the piston, and thermal load in wet cylinder liner of Perkins 1306 diesel engine as a real case. Two materials namely cast iron, grades C4 28-48 and cast alloy steel, C4 35-56, grade 38XMIOA with three different thicknesses (t_1= 13.13, t_2= 9.2 and t_3= 6.93) mm are considered. The finite element package ANSYS (19.1) has been used as a numerical method to calculate the stresses and deflection in 3D liner model for the three parts of cylinder sleeve. The PTC MATHCAD 4 is used to formulate the analytical solution of the selected case and the results are compared to the numerical results obtained from ANSYS. The numerical results show that the maximum deflection occurs at a point of applied piston load in which it is 0.135 mm for cast iron and 0.109 mm for steel. The deflection in circumferential is very small varied in range (-0.9 to 0.9) % of maximum deflection, and the maximum axial deflection for cast iron and steel are 0.66 mm and 0.67 mm, respectively. In addition, confirming the reliability of developed PTC MATHCAD 4 program by consider one verification case. These results show that a good agreement between the numerical and analytical methods.

KEYWORDS

Cylinder wet liner, CFD, Perkins 1306 engine, Mechanical stresses.

1. INTRODUCTION

As the current, the biggest trend in automotive diesel engine development is improving engine efficiency on all important aspect such as design, material, fuel economy and combustion system. Heavy duty diesel engine has a wide application. Since 1910s have been used in heavy trucks, submarines, ships, construction equipment, electrical generators…etc. Compression ignition CI or diesel engine is one of the two types of IC engine, in which the combustion take place inside the engine block. The engine cylinder consists of many important components such as piston, rings and liner (sleeve).

A cylinder liner is defined as a cylindrical part that is fitted inside the engine block to form a cylinder, as shown in Figure 1. A cylinder sleeve is a vital component of cylindrical engine. The most important functions of cylinder liner are to provide the surface for the piston to sliding resist the wear form the piston and piston rings, to resist high pressure and high temperature and must having a high thermal conductivity. Cylindrical liners are subjected to substantial mechanical load and thermal stresses. In order to increase the efficiency and life cycle of the liners, the designing and material selection must be optimizing. The cylindrical part in an engine in which the piston moves upward and downward, and may be a separated liner or integrate part of the cylinder block. The first type is commonly used in CI or diesel engine, it has an advantage when the excessive wear occurs in cylinder it can be replaced, whereas, the second type, commonly used in gasoline engine and has disadvantage it cannot be replaced and the cylinder block must be re-bored.

Generally, the cylinder liners in IC engine may be classified into three main types:
(A) Dry Cylinder Liners.
(B) Wet Cylinder Liners.
(C) Finned Cylinder Liners.

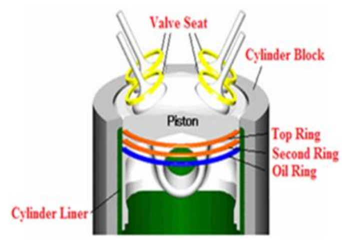

Figure 1: Cross-section of a cylinder in IC engine [1].

Generally, the cylinder liners in IC engine may be classified into three main types:
(D) Dry Cylinder Liners.
(E) Wet Cylinder Liners.
(F) Finned Cylinder Liners.

(A) Dry Liner Type

A dry liner designed to be strong enough to extremely high pressure and high temperature. Dry sleeve does not come in contact with the coolant liquid, as shown in Figure 2-a. It has relatively thin wall and it is material is composition dominantly including high –grid material such as Cast iron and ceramic- Nickel. Dry liners are used in practically all types of engines, the most aluminum automotive engine blocks used dry Gray iron cylinder sleeve in their engine.

(B) Wet Liner Type

Wet cylinder liners, are thicker than the dry liners and must be hard to withstand the full working pressure of the combustion gases, in this type of liner do not have integral cooling passage, the water jacket is formed by the liner and separate jacket which is a part of the block, and the coolant comes in contact with wet liner through the passage known as a water jacket as shown in Figure 2-B.There are two types of static seal must be provide at both the crankshaft and combustion chamber in order to prevent the leakage of coolant liquid into the oil pan sump, or combustion chamber.

(C) Finned Liner Type

Finned cylinder liner are also made from toughed material, as like that for dry and wet cylinder sleeves, and commonly used for air-cooled engine. It is used the surrounding air as a cooling medium. However, this type of liners is characterized by integrated tiny fins that allow the cylinder to cool down from the overflowing incoming air, as shown in Figure 2-C.

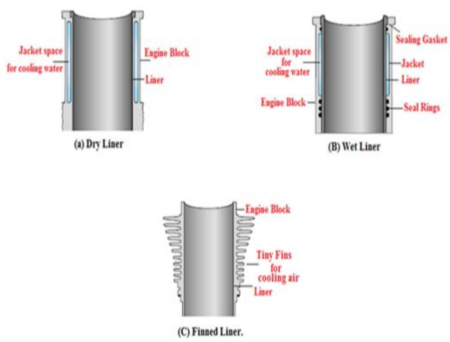

Figure 2: Cross-section of Cylinder Liners: (a) Dry liner, (B) Wet liner, (C) Finned liner.

Although, there are several work were done dealing with the thermal and mechanical stresses in cylinder liners. But there are a lack of using analytical equations that take into account the effect of thrust piston force on the wall of the cylinder liner that cause bending stresses in addition of pressure force in the cylinder sleeve. The effects of cylinder wall material on the performance of the diesel engine were studied [2]. Three different liner material Gray Cast iron Grade 60, Cast SS17PH and Inconel 713C are investigated. The numerical analysis (CFD) was carried out for wet liner 3D model by using Soild Work 2012. The results show that the tensile strength of Inconel 713C and Cast iron SS17 is higher than the Cast iron Grade 60. So, these materials from the point of mechanical factors can resist the high pressure, whereas, the Inconel 713C material has good thermal conductivity when compared to the other tested material. A studied numerically the effect of thickness of cylinder liner on the hoop stress, maximum thermal stress and combined stress, and consequently, the effect of these stresses on cylinder linear geometry for SANTRO vehicle engine of Hyundai Motors as a real case [3]. Finite element method as analytical procedure was done by using ANSYS and NASTRAN software. Two cases were taken for liner optimization: Case (1) if the liner thickness is reduced up to 1.5 mm from 2.31 mm results an increasing in hoop, maximum thermal and combined stresses, whereas, in case (2) taken thickness 5 mm from 2.31 mm the results show that these stresses reduced if the thickness is increased.

The numerical computation by using (FEM) of the temperature distribution in the wet cylinder liner of diesel engine was carried out [4]. Their results show that the cylinder liner heated up the fastest during the first 20s and the temperature start to stabilize at 40s and then it change in small range about 0.7^{o}K/s. In this work a numerically investigated four basic materials that are almost used for liners manufacturing (Cast iron, Steel, Aluminum alloy and Titanium alloy), analysis were carried out for solid 3D model by using (CFD) ANSYS 15 workbench [5]. Their results show that the Titanium alloy is lighter, equivalent stress and total deformation is less than that the cast iron used in FORD3600 as a real case model. Also, the result of nitriding process for both the inner and outer diameter of the cylinder liners shows a very good improvement in thermochemical properties such as wear and hardness resistance.

A studied numerically the most suitable materials (Carbon Steel, Low Alloy Steel, Cast PH Stainless Steel, Wrought PH Stainless Steel, and Cast Nickel-Chromium Alloy) for wet cylinder liner, by using Cambridge Engineering Software (CES) and by compared these five materials, they found that Cast Nickel-Chromium Alloy is the best among these material [1]. A group researchers in their research, they selected Titanium alloy (Grade 4) as cylinder liner alternative material in marine engine [6]. Designing and analysis are carried out by using ANSYS 14.0 software. The results show that the weight, equivalent stress (von mises) and total deformation is slightly less than that cast-iron alloy which is currently used. An examined in detail, the effect of various surface coating materials such as Ceramic, Nickel chrome alloy and Aluminum alloy on the, heat flux, thermal displacement, thermal stresses, thermal gradient and nodal temperature of the diesel engine cylinder liner. Results show that the best coating among these three materials is aluminum alloy (AL_2O_3) [7].

In the present research, the stress due to the action of gas pressure, side thrust force of the piston and thermal stresses in **Perkins 1306** diesel engine cylinder wet liner, as shown in Figure 3, are calculated with addition to the deflection as a real case model using two types of materials namely cast iron and alloy steel with three different thicknesses (t_1= 13.13, t_2= 9.2 and t_3= 6.93) mm are considered.

2. METHODOLOGY

Methodologies of the present work start with the product selection (Perkins 1306 wet liner), identification and collection of dimensions data from already existing engine cylinder liner. After that creation of 3D model using ANSYS software for two different materials (cast iron, grades C4 28-48 and cast alloy steel, C4 35-56, grade 38XMIOA) were done, fine mesh generation using 3D element namely Shell181 with suitable boundary conditions and loads applied to the model are used to simulate the real cylinder liner in ANSYS to obtain a reliable results for optimum design.

3. DESIGN OF CYLINDER LINER

The thickness of a liner wall chosen during the design is checked by the formula used for computing cylindrical vessels [8]:

t_d = 0.5D ($\sqrt{(\sigma_z + 0.4p_z)/(\sigma_z - 1.3p_z)}$-1) --------------- (1)

Where:

D: cylinder wall diameter (mm)
σ_z : the permissible extension stress (σ_z=50-60MPa for cast iron, σ_z = 80-100 MPa for steel.)
P_Z: is the gas pressure equal to 4 (MPa)
In this paper the design the stresses are only due to loads such as maximum gas pressure and temperature gradient in the sleeve, as shown in Figures 4-a & 4-b.

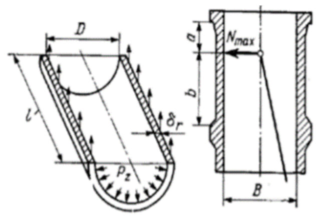

Figure 4-a: Show hoop and bending stresses in cylinder

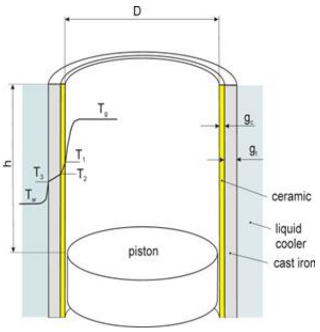

Figure 4-b: Show temperature gradient through cylinder liner

3.1 Longitudinal stress (σ_L) [8]

σ_L= P_Z D/4t --------------- (2)

The value of σ_L is determined mainly for load carrying liners of air cooled engines in which cylinder element ruptures are less probable because of the walls reinforced by ribs.

3.2 The Bending stress [8]

σ_b= Mb / W --------------- (3)

The Bending Moment: - The stresses caused by normal force N_{max} acting on the load carrying liner are usually determined in engines with separate cylinders applied at the center of the piston pin. The bending moment of force N_{max} is

$M_b = N_{max}\, ab / (a + b)$

Where N_{max} is the maximum value of the normal force determined from the dynamic analysis,

a: is the distance from the piston pin axis to T.D.C., mm.

$$\sigma_b = Mb/W \qquad ------------- (4)$$

Where W is the resistance moment of the liner transverse section m^3

$$W = (0.1\,(D_1^4 - D^4))/D_1 \qquad -------------- (5)$$

D_1 and D: are the outer and inner diameters of the cylinder liner.

3.3 Circumferential stress or hoop stress (σ_{ex})

$$\sigma_{ex} = Pz\, D/2t \qquad -------------- (6)$$

Where Pz is the maximum gas pressure equal to P = 4 MPa

D: is the cylinder bore equal to 127 (mm).

t: is the cylinder liner wall thickness equal to 3.82 mm.

3.4 Thermal stress [9]

During engine operation, there is a substantial temperature differences between the outer and inner surfaces of the cylinder liner that causes thermal stresses:

$$\sigma_t = \frac{E\alpha_e \Delta T}{2(1-\nu)} \qquad --------------- (7)$$

Where E is the modulus of material elasticity, MPa

α_e: is the coefficient of linear expansion

ΔT: is the temperature difference

ν: is Poisson's ratio

The extension stresses on the liner outer surface are associated with the plus sign and the compression stress on the inner surface, with the minus sign.

The total stresses due to the gas pressure and temperature differences are:

On the outer surface of the cylinder liner

$$\sigma_o = \sigma_{ex} + \sigma_t \qquad --------------- (8)$$

on the inner surface

$$\sigma_i = \sigma_{ex} - \sigma_t \qquad --------------- (9)$$

The total stress σ_{total} in a cast iron liner should not exceed 100 to 130 MPa, and 180 to 200 MPa in a steel liner.

3.5 Sample of calculation

Thickness of cylinder liner:

$t_1 = (D_1 - D_i)/2 = 13.37$ mm

$t_2 = (D_2 - D_i)/2 = 9.2$ mm

$t_3 = (D_3 - D_i)/2 = 6.93$ mm

The design thickness of the liner wall is

$$t := 0.5 \cdot D_i \cdot \left(\left(\sqrt{\frac{(\sigma_z + 0.4 \cdot P_{max})}{(\sigma_z - 1.3 \cdot P_{max})}} \right) - 1 \right) \qquad ---------- (10)$$

t = 3.433 mm

The liner wall thickness is chosen with certain safety margin, as $t_i > t_d$

Let us adopt $t_1 = 13.37$ mm

The extension stress in the liner due to maximum gas pressure

$$\sigma_{Hoop} := \frac{P_{max} \cdot D_i}{2 \cdot t}$$

$\sigma_{Hoop} = 17.053$ MPa

The temperature stresses in the liner

$$\sigma_{thermal} := \frac{E \cdot \alpha \cdot \Delta T}{2 \cdot (1-\nu)} \qquad --------------- (11)$$

$\sigma_{Thermal} = 10$ MPa

The total stresses in the liner caused by gas pressure and temperature difference are summarized in Table (1):

Table 1: Shows the mechanical and thermal stresses in different location of cylinder liner

Thickness (mm)	Stresses (MPa)			
	σ_{ex}	σ_{long}	σ_{th}	σ_b
t_1= 13.37	17.053	8.527	80.67	1.815
t_2= 9.2	24.783	12.301		1.336
t_3= 6.93	32.9	16.45		0.879

In addition, the total stresses due to combination of four types of stresses in the outer and inner surface of cylinder liner are tabulated in Table (2):

Table 2: Shows the total stresses on the outer and inner surface of the cylinder liner

Thickness (mm)	Stresses (MPa)		
	$\sigma_{\sum hoop-inner_t}$	$\sigma_{\sum hoop-outer_t}$	$\sigma_{\sum axial\_t}$
t_1= 13.37	-63.614	97.72	10.342
t_2= 9.2	-55.884	105.449	13.727
t_3= 6.93	-47.766	113.567	17.33

4. NUMERICAL SI MULATION

General purpose finite element software ANSYS 19.1 is used to compute the mechanical and thermal stresses for the present cylinder liner. The main steps to do the analysis are

1. Build the models using Pre-Processor module built in ANSYS
2. Apply the suitable boundary condition and operating loads to the model.
3. Solve the model using solution processor module.
4. Study the deformations and stresses due to the applied loads.
5. Optimize a design by selecting a suitable thickness for the three parts of cylinder liner.

4.1 The real case cylinder liner

The cylinder liner of Perkins 1306 Diesel Engine has been taken as a real case to study the effect of mechanical and thermal loads on different types of stresses induced in cylinder liner. The technical specification of this engine is shown in Table (3).

Table 3: Selected technical data for Perkins 1306 Diesel Engine

No.	Specification	Units	Value
1	Type/Model	---	1306C-E87TAG 4
2	Combustion principle	---	4-Stroke
3	No. of cylinder	---	6-in line
4	Engine power output at rated rpm	kWm	205
5	Horsepower	Hp	292
6	Engine speed	rpm	1500
7	Bore	mm	116.6
8	Stroke	mm	135.9
9	Comparison ratio	---	16.9:1

4.2 Convergence study

The hoop and axial stresses of cylinder liner under pressure only are shown in the equations 2 and 6. To make a verification study of the numerical method with the analytical solution, a cylinder model was built and it was subjected to internal pressure only and the results is shown in Table (4).

Table 4: Verification the theoretical and numerical results.

Stress	Theory	ANSYS	Error percentage (%)
6 hoop (MPa)	63.5	62.216	2
6 Long. (MPa)	31.75	31.58	0.5

It was clear that the difference between analytical solution and the one given by ANSYS differ in just 2% for hoop stress and 0.5 for longitudinal stress.

4.3 Mesh model

The domain is meshed using a three-dimensional element namely Shell 181 as shown in Figure (5). The convergence study is done until the suitable numbers of elements and nodes are reached as illustrated in Figures 5-a & 5-b.

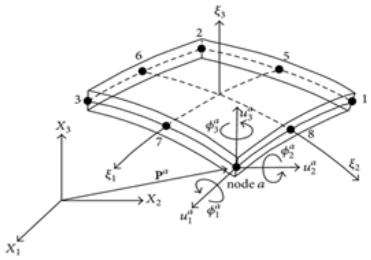

Figure 5-a: Shell 181 elements.

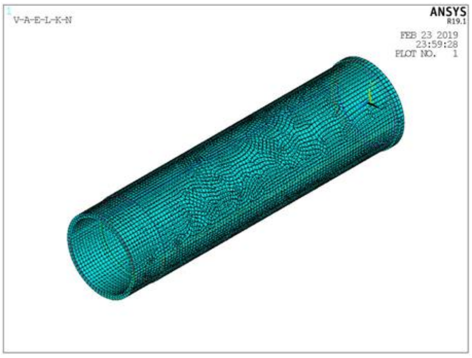

Figure 5-b: Mesh used in current study for the cylinder

4.4 Boundary conditions

The suitable boundary conditions and loads applied to the model are used to simulate the real cylinder liner to obtain reliable results. In real case the model is fixed at the top and free at the bottom with sliding boundary condition applied at the third part of the model as shown in Figure 6.

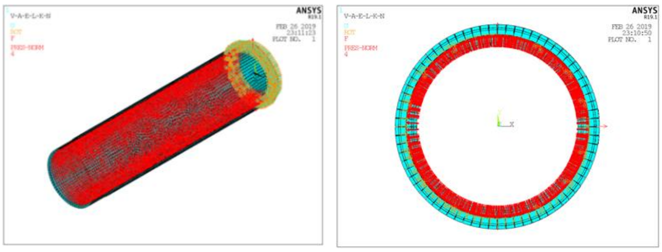

Figure 6: Applied boundary condition for the present work, Isometric view and side view

5. RESULTS AND DISCUSSIONS

The effect of hoop, longitudinal, bending and thermal stresses in the three main parts of cylinder liner is illustrated in Figure 7 by using the PTC Mathcad software. It is clear that the effect of thermal stresses is the dominant stress in which it reached a value of 80 MPa and it is constant regardless the value of thickness because the thermal stresses does not depend on thickness. The second high stresses is the extension stress with the value of nearly 32 MPa at thickness 6.93 mm and it was double the value of longitudinal stress. This is agreed with the mechanics of materials formula [10]. The effect of bending stresses show a low impact on the total stresses compared with other stresses in the cylinder liner.

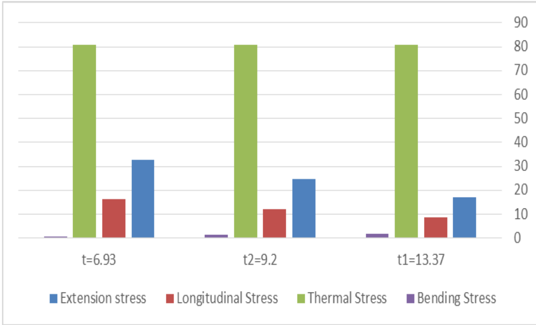

Figure 7: Mechanical and thermal stresses as a function of thickness at different sections of cylinder liner.

Furthermore, it has been found that when the thickness decreased from 13.37 to 9.2 mm the hoop stress increased by 12% for inner and 7.9% for outer hoop stress, while the axial stress increased by 39.7%. The maximum stresses have been obtained with the minimum thickness in the cylinder liner, as shown in Table 2, the increment percentages are 24.9%, 16.2% and 80% for inner hoop stress, outer hoop stress and axial stress respectively. Figures 8 to 10 represent the von Mises, radial and hoop stresses for the real case cylinder liner of CI and subjected to mechanical and thermal loads. In Figure 8, the von Mises stress shows that the maximum stresses occur at the region of stress concentration area which is near the sliding boundary condition while the von Mises stress in the middle part of the cylinder liner is approximately 100 MPa which indicate a good agreement with the analytical solution (113 MPa).

The radial stresses in Figure 9 show a very low level of stresses and that also agreed with the theory of thin cylinder since it is a plane stress problem. The hoop stresses in Figure 10 reflect the big influence of this stresses in the middle part of the cylinder liner since it reach a value of about 136 MPa. In Figures 11 and 12, the radial deflection reach a maximum value of -0.66 mm and the axial deflection shows a value of 0.667 mm and it occur, as expected, at the free end of cylinder liner. Figures 13 to 17, show the same general behaviour of CI cylinder liner (Figures 8-12) except there are a high level of von Mises and hoop stresses in which the stresses are double in the middle part of cylinder liner, however the radial and axial deflection show no significant difference in their value.

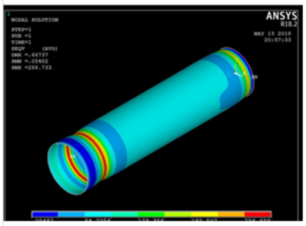

Fig.(8) Shows the Von-Mises stress of the cylinder liner (CI material) under thermal and mechanical

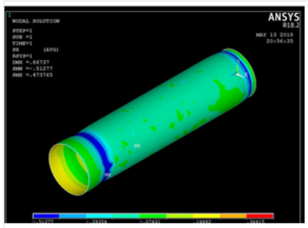

Fig.(9) Shows the radial stress of the cylinder liner (CI material) under thermal and mechanical loads.

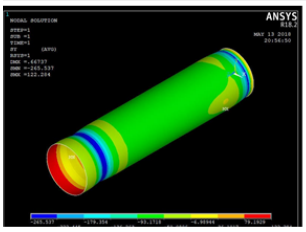

Fig.(10) Shows the hoop stress of the cylinder liner (CI material) under thermal and mechanical loads.

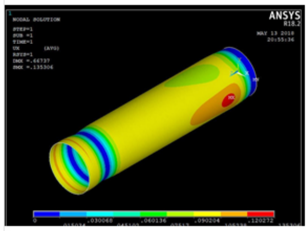

Fig.(11) Shows the radial deflection of the cylinder liner (CI material) under thermal and mechanical

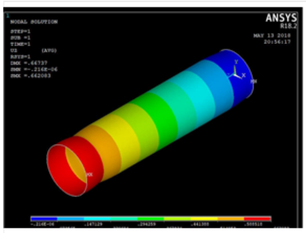

Fig.(12) Shows the axial deflection of the cylinder liner (CI material) under thermal and mechanical

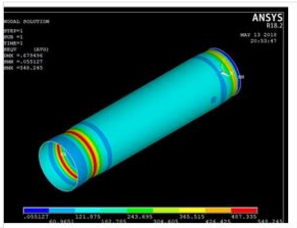

Fig.(13) Shows the V on-Mises stress of the cylinder liner (alloy steel material) under thermal and

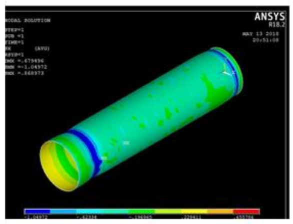

Fig.(14) Shows the radial stress of the cylinder liner (alloy steel material) under thermal and mechanical

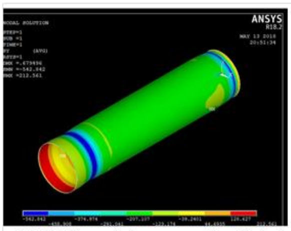

Fig.(15) Shows the hoop stress of the cylinder liner (alloy steel material) under thermal and mechanical

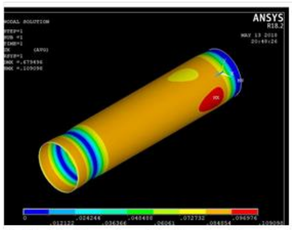

Fig.(16) Shows the radial deflection of the cylinder liner (CI material) under thermal and mechanical

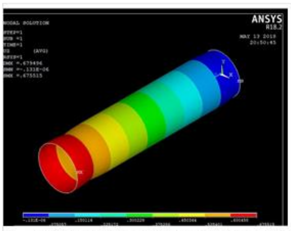

Fig.(17) Shows the axial deflection of the cylinder liner (CI material) under thermal and mechanical

6. CONCLUSIONS

The objective of this research is to study the stresses due to the action of gas pressure, major thrust force of the piston, and thermal load in wet liner cylinder of Perkins 1306 diesel engine as a real case. Two materials namely cast iron, grades C4 28-48 and cast alloy steel, C4 35-56, grade 38XMIOA with three different thicknesses (t1= 13.13, t2= 9.2 and t3= 6.93) mm are considered. The finite element package ANSYS (19.1) has been used as a numerical method to calculate the stresses and deflection in 3D liner model for the three parts of cylinder sleeve. The PTC MATHCAD 4 is used to formulate the analytical solution of the selected case and the results are compared to the numerical results obtained from ANSYS. The following conclusion may conclude from the present research:

1. The results of verification case show that a good agreement between simulation and analytical methods. Since, the difference is 0.5% and 2% for longitudinal and hoop stress, respectively.
2. The thermal stress plays a dominant stress in which it reached a value of 80.67 MPa compared to hoop and longitudinal stresses of 32.9 MPa and 16.45 MPa respectively along the cylinder liner.
3. It has been found that when the thickness decreased from 13.37 to 9.2 mm the hoop stress increased by 12% for the inner layer and by 7.9% for the outer layer of cylinder liner.
4. It is evident that the maximum radial deflection occurs at a point of applied piston load in which it reaches values of 0.109 mm and 0.135mm for steel and cast-iron cylinder sleeves, respectively. This result is acceptable as the modules of elasticity for steel is larger than for cast iron.
5. The hoop stress at the mid length of cylinder is approximately 100 MPa in the cast iron cylinder sleeve, while the analytical solution from the PTC MATHCAD reveals a result of 105 MPa, and that reflect a good agreement between the analytical and numerical methods.
6. The maximum value of radial deflection is -0.66 mm at the point of applied piston thrust and that for axial deflection is 0.667 mm and it occur, as expected, at the free end of cylinder liner.
7. The effect of bending stress developed form the thrust piston load is very small compared to other stresses caused by pressure and thermal loads in the cylinder liner.

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