

Thermo Coupled Stress Analysis of Exhaust Manifold Assemblage using ABAQUS

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Abstract— *Comprehension of the basic conduct in the beginning period of outline can help to lessen the quantity of virtual models and the time taken for their assessment. For an engine development program, the engine exhaust manifold system is a mind boggling framework subjected to thermo mechanical burdens. The investigation of such a framework is discriminating because of the many-sided quality of meeting the basic and stream prerequisites. In this paper the Thermal Stress Index (TSI) is created for down to earth reason. Especially the TSI is appeared to be extremely powerful in discovering an ideal shape with imperatives in outline of the exhaust manifold system. Therefore, the objective for discovering an ideal state of the exhaust manifold system is to minimize the TSI, and philosophy received to address a failure of an Exhaust complex amid physical testing on an engine test bed. A non-linear thermo-auxiliary quality investigation of the exhaust manifold system gathering of the four cylinder four stroke engine is done. The FE model comprises of the exhaust manifold system get together, head and mounting bolts. The contact interaction between the parts is likewise modeled. The assembly is subjected to bolt preloads, for reproducing the mechanical assembly. The thermal loads utilized as a part of the reenactment are computed through CFD (Input). The model considers a temperature reliance of a heat exchange coefficient on wall temperature in cooling sections. For the structural analysis temperature subordinate non-linear material properties and contact non-linearity is considered.*

Keywords— *thermal stress; mechanical assembly; Abaqus; CFD.*

I. INTRODUCTION

the exhaust manifold system mounted on the cylinder head of an engine gathers a gas depleted from an engine, and sends it to a catalyst converter. The exhaust manifold system assumes a vital part in the execution of an engine framework [1]. Especially, the effectiveness of discharge and fuel utilization is firmly identified with the exhaust manifold system. The exhaust system is under a thermal fatigue delivered by expanding and diminishing temperature, which prompts a crack of the exhaust manifold system [2]. Automotive engine works under severe thermo mechanical loading conditions. Working temperature increases up to 800°C from ambient temperature and extensive thermal stress is prompted by temperature gradient and geometrical imperatives. This

thermo mechanical coupling is one of basic issues in automotive designing [3]. Engine exhaust manifolds systems are generally subject to severe thermal cycles amid operation and upon shutdown. Thermal expansion and constriction of the complex is obliged by its collaboration with the engine head to which it is catapulted [4]. These requirements administer the thermo-mechanical exhaustion life of the complex. The starting of assembly strategy comprises of catapulting the flanges of the manifold to the engine head with endorsed bolt drives that deliver uniform axial bolt stresses. Under consequent working conditions, for example, thermal cycling and creep, these bolt powers may expand or unwind conceivably changing normal pressures and bringing about sidelong slippage between the engine head and the manifold flanges.

II. OBJECTIVE

This task is persuaded by the way that under thermal loads the exhaust manifold system joint with the engine head unwind or increase causing on the joint to be feeble and the gasses attempt to escape at the interface between the flange and the engine head. Subsequently the point of this task is to perform Non-linear contact thermal mechanical coupled stress of engine exhaust manifold system assemblage utilizing ABAQUS/STANDARD and study the impact of thermal loading on the joint slippage and the joint design criteria.

Problem Statement

The problem scenario consists of THREE steps:

1. Apply prescribed bolt loads to fasten the exhaust manifold to the engine head
2. Subject the assembly to the steady-state operating temperature distribution
3. Return the assembly to ambient temperature conditions

General approach in engineering domain to simulate the joint slippage is to apply the bolt preload and study the shear capacity and shear load. For exhaust manifold generally the thermal loads won't be considered for joint behavior study. In this project, study has been conducted

to analyze the exhaust manifold joint with the engine head considering the thermal loads.

III. METHODOLOGY

The contact pair algorithm in Abaqus/Standard, which is shown in below, is built around the Newton Raphson technique. Abaqus/Standard inspects the condition of all contact associations toward begin of every addition to set up whether slave nodes are open or close. On the off chance that a node is close, Abaqus/Standard figures out if it is sliding or staying.

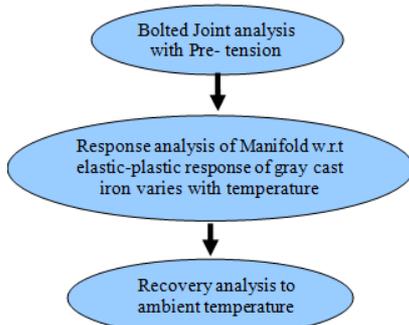


Fig. 1: Flowchart of manifold assemblage analysis

Abaqus/Standard applies an imperative for each close node and expels limitations from any node where the contact state changes from close to open.

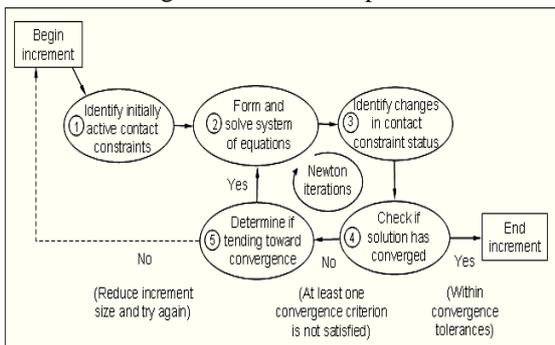


Fig. 2: Parameters of manifold assemblage analysis

Abaqus/Standard then completes an emphasis and redesigns the design of the model utilizing the computed revisions.

A. Model of Exhaust Manifold Assemblage:

The exhaust manifold assemblage being analyzed consists of a four tube exhaust manifold with three flanges, bolted with seven bolts to a small section of the engine head.

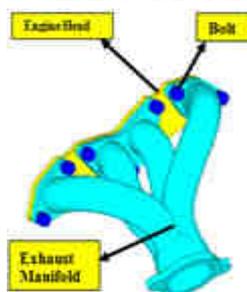


Fig. 3: C3D8I:8 nodes linear brick incompatible mode

The first-order quadrilateral continuum elements in Abaqus/Standard of type CPS4I, CPE4I, CAX4I, CPEG4I, and C3D8I, as well as the related hybrid elements, are enhanced by incompatible modes to improve the bending behavior. In addition to the displacement degrees of freedom, incompatible deformation modes are added internally to the elements. The primary effect of these degrees of freedom is to eliminate the parasitic shear stresses that are observed in regular linear continuum elements if they are loaded in bending. In addition, these degrees of freedom eliminate artificial stiffening in bending due to Poisson's effect. In regular linear continuum elements, the linear variation of the axial stress due to bending is accompanied by a linear variation of the stress perpendicular to the bending direction, which leads to incorrect stresses and an overestimation of the stiffness. The incompatible modes prevent such a stress from occurring. All three structural components (manifold, head and bolts) are modeled with three dimensional continuum elements. The model consists of 7450 first-order brick elements with incompatible deformation modes, C3D8I and 282 first-order prism elements, C3D6. The C3D6 elements are selected to represent the bending of the manifold walls with only one element through the thickness of the tube walls. The bottoms of the bolt heads form contact bearing surfaces, with the top surfaces of the manifold flanges lying directly beneath them. In addition, the bottoms of the manifold flanges form contact bearing surfaces with the top of the engine head. Each of these surfaces is defined in Abaqus with the SURFACE option. Respective mating surfaces are paired together with the CONTACT PAIR option. Normal pressures will be transmitted through these contact pairs as a result of the bolt tightening forces in Step 1. The forces carried by the bolts will vary as they respond to the thermal cycling of the assembly in subsequent steps. These fluctuations in bolt loads will result in varying normal pressures transmitted across the contact pairs. Lateral slip of the mating components will occur if lateral forces developed in the system surpass the critical frictional shear stress limit.

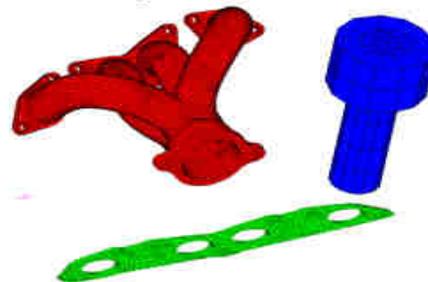


Fig. 4: Mesh model of manifold assemblage

A friction coefficient of 0.2 is used between all contacting surfaces. Contact conditions are not necessary between

the bolt shanks and the holes in the manifold flanges because of the design clearance between them. Contact between the bolt shanks and the holes in the engine head are not modeled.

Table 1. Material properties

Parts	Young's Modulus	Poisson's ratio
Engine Manifold	138 Mpa	0.283
Engine Head	138 Mpa	0.283
Bolt	207 Mpa	0.3

B. Boundary conditions

It is assumed that the engine head is securely fixed to a stiff and bulky engine block, so the nodes along the base of the head are secured in the direction normal to the base (the global x -direction) but are free to move in the two lateral directions to account for thermal expansion. It is also assumed that the bolts are threaded tightly into the engine head, with the bolt threads beginning directly beneath the section of engine head modeled. Therefore, the nodes at the bottom of the bolt shanks are shared with the nodes of the surrounding engine head elements and are also secured in the global x -direction.

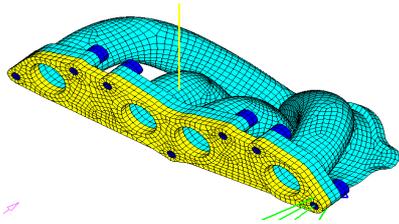


Fig. 5: Boundary conditions of manifold assemblage

The manifold flanges are sandwiched between the top of the engine head and the base of the bolt heads using the contact pair option. The line of action of the bolt forces (bolt shank axes) is along the global x degree of freedom. Soft springs acting in the global y - and z -directions are attached to the outlet end of the manifold and to the two ends of the head to suppress rigid body motions of the manifold and head, respectively. These springs have no influence on the solution., contact definition with small sliding, which assumes that although two bodies may undergo large motions, there will be relatively little sliding of one surface along with other surface.

C. Loading conditions

In the analysis a concentrated clamping load of 20 kN is applied to each of the pre-tension nodes in node set BOLTS, the concentrated load is removed and replaced by a “fixed” boundary condition that will hold the pre-tension section length changes fixed. STEP: 1 the analysis each of the seven bolts is tightened to a uniform bolt force of 20 kN. In subsequent steps the variation of the bolt loads is monitored as the bolts respond to the thermal loading on the assembly as a whole. The “prescribed

assembly load” capability of Abaqus is used. For each bolt we define a “cut,” or pre-tension section, and subject the section to a specified tensile load. As a result, the length of the bolt at the pre-tension section will change by the amount necessary to carry the prescribed load, while accounting for the compliance of the rest of the system. In the next step the prescribed bolt loads are replaced by the condition that the length changes calculated in the previous step remain fixed. The remainder of the bolt is free to deform. The same procedure is used for all seven bolts. First, pre-tension sections are defined as “cuts” that are perpendicular to the bolt shank axes by using the SURFACE option on the faces of a group of elements within each bolt shank, as shown in Fig 4. The line of action of the bolt force is in the direction that is normal to this surface. Next, each bolt is assigned an arbitrary, independent node that possesses one degree of freedom (dof 1), to which the bolt force will be applied. These nodes are called the “pre-tension nodes” (all seven bolt pre-tension nodes are placed into a node set named BOLTS). The spatial position of a pre-tension node is irrelevant. Finally, each surface is associated with the appropriate pre-tension node using the PRE-TENSION SECTION option.

A portion of the Abaqus model definition section defining the pre-tension section is shown below:

```
ELSET, ELSET=BCUT1, GENERATE19288,
19307
SURFACE, NAME=BOLT1BCUT1, S2
NODE, NSET=BOLTS99991, 21.964 , -139.80
, -12.425...99997, 21.964 , 137.38 , -12.226
PRE-TENSION SECTION, SURFACE=BOLT1,
NODE=99991
```

STEP: 2 Over the course of a step in which a load is replaced by a boundary condition, CF1 is ramped down, while RF1 is ramped up to replace it. Therefore, the total force across the bolt is the sum of the concentrated force (CF1) and the reaction force (RF1) on the pre-tension node. This total force is available as TF1. Additionally in this step of the analysis nodal temperatures depicting the steady-state temperature distribution in the manifold are read from an external file. Each of the nodes in the model has its temperature ramped up from the initial ambient temperature of 300 K to its final steady-state temperature. These nodal temperatures are interpolated to the element integration points so that the correct temperature-dependent plasticity data can be used in the constitutive calculations. STEP: 3 the nodal temperatures are ramped back down to the initial ambient temperature of 300 K.

The criteria for preventing slip in a bolted joint is that the normal load between the abutment contact surfaces time the co-efficient of friction should always be greater than the actual shear load between the abutment contact surfaces that results from the application of external load. The parameters CFN and CFS seem to be ideal choices for providing the data from which to make this calculation. CFN gives the total normal force across each contact pair in the abutment joint interface and CFS gives the total shear force across the same contact pairs.

IV. FEA SIMULATION

The nonlinearities in this project are the result of changing contact definitions, frictional slip and stick and temperature dependent plasticity. As a result of frictional sticking, the ends of the two outer manifold flanges have expanded outward relative to one another. Plastic yielding conditions result since thermal expansion of the remainder of the manifold is constrained by this limited lateral flange motion.

CFN AND CFS OUTPUT-%SHEAR CAPACITY METHOD

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V. RESULTS AND DISCUSSIONS

The nonlinearities in this project are the result of changing contact definitions, frictional slip and stick and temperature dependent plasticity.

A. Displacement magnitude-Thermo mechanical loading

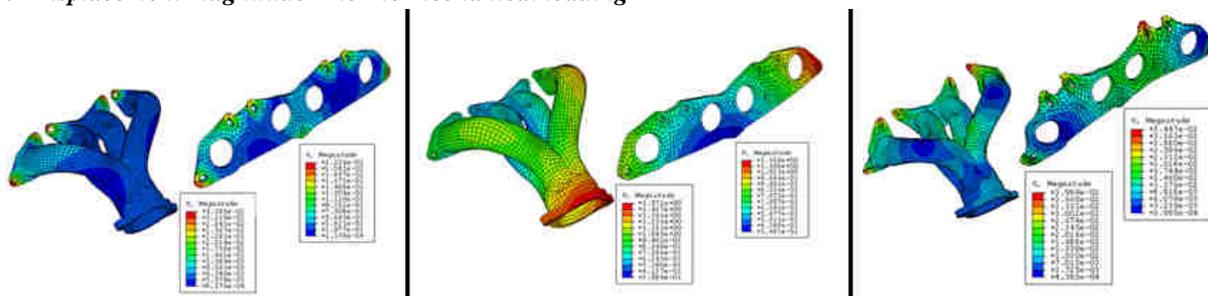


Fig. 6: Prescribed bolt load, thermal load and return ambient temperature

B. Temperature (bolted joint design guide)-Thermo mechanical loading

Table 2. Mechanical loading of joint failure criteria

No of Bolts	Shear load (N)	CFN (N)	Co-efficient of Friction	Shear Capacity	% Shear capacity/Shear load
Bolt-1	32.56	20000	0.2	4000	122.85
Bolt-2	86.35	20000	0.2	4000	46.32
Bolt-3	93.07	20000	0.2	4000	42.07
Bolt-4	69.95	20000	0.2	4000	57.18
Bolt-5	98.81	20000	0.2	4000	40.48
Bolt-6	93.69	20000	0.2	4000	42.69
Bolt-7	28.37	20000	0.2	4000	140.01

As a result of frictional sticking, the ends of the two outer manifold flanges have expanded outward relative to one another. Plastic yielding conditions result since thermal expansion of the remainder of the manifold is constrained by this limited lateral flange motion. The curves contain the values of the total forces for the pre-tension nodes in node set BOLTS. The load carried by the bolts increase significantly during the heat-up step.

Table 3. Thermo-Mechanical loading of joint failure

No of Bolts	Shear load (N)	CFN (N)	Co-efficient of Friction	Shear Capacity	% Shear capacity/Shear load
Bolt-1	18.3	20042	0.2	4008.40	219.04
Bolt-2	284.2	19997	0.2	3999.40	14.07
Bolt-3	161	19993	0.2	3999.00	28.84
Bolt-4	82.22	20003	0.2	4000.60	48.66
Bolt-5	478.3	19871	0.2	3974.20	8.31
Bolt-6	496.6	19981	0.2	3996.20	8.03
Bolt-7	31.65	19998	0.2	3996.60	77.44

The loads do not return precisely to the original bolt load specification upon cool down because of the residual stresses, plastic deformation and frictional dissipation that developed in the manifold. However since the bolt load increases with respect to time during thermal loading, the thermal loading should be considered while designing the engine head and exhaust manifold joint.

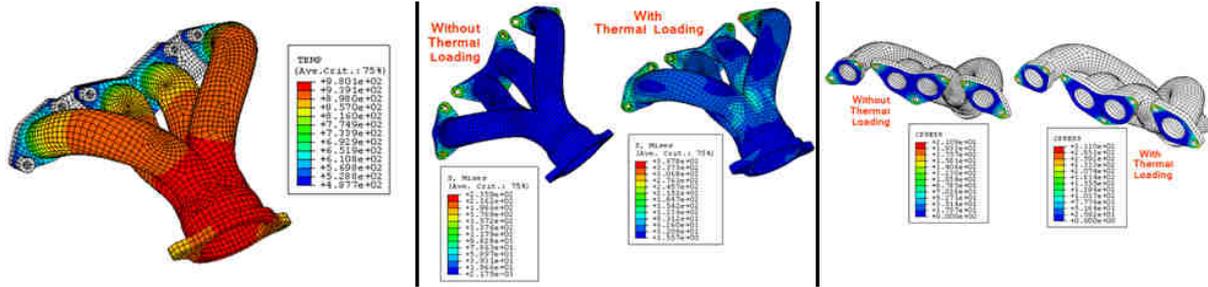


Fig. 7: Temperature analysis, stress vonmises and contact pressure

C. CFN and CFS output-%

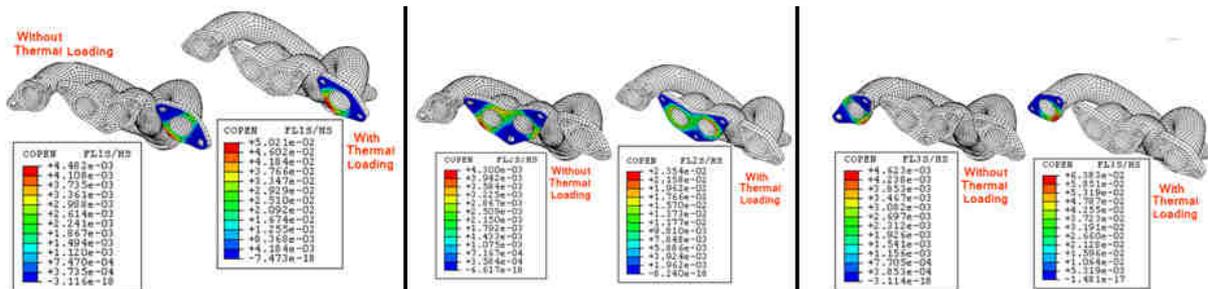


Fig. 8: Comparison of COPEN flange 1, 2 and 3

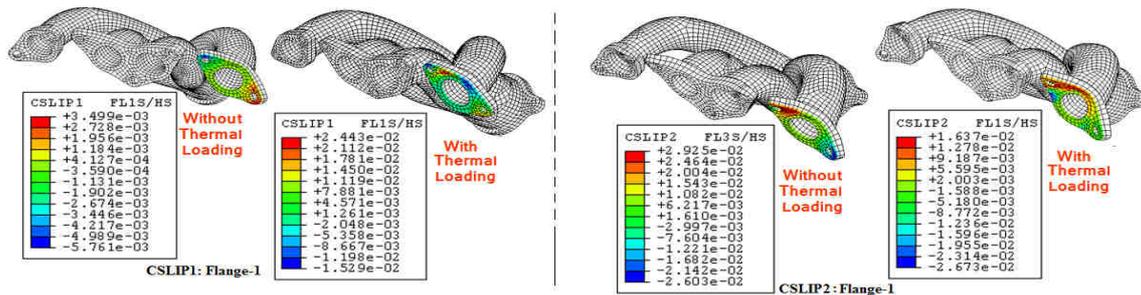


Fig. 9: CSLIP 1 and 2 Flange-1(Comparison with and without thermal load)

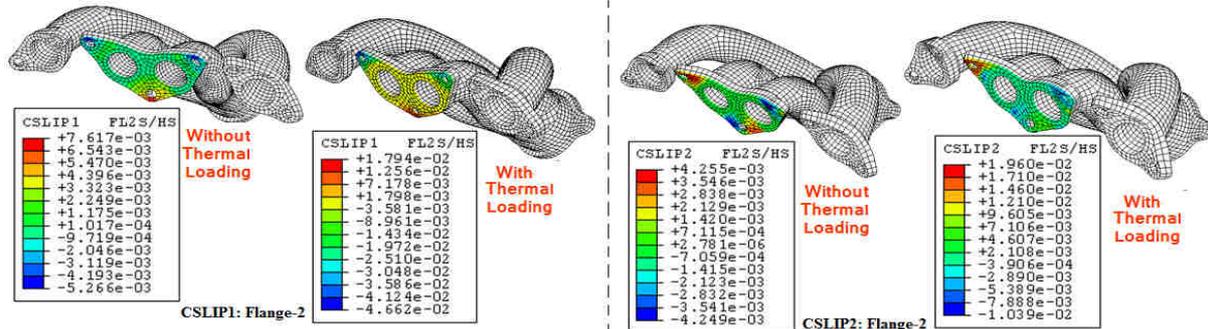


Fig. 10: CSLIP 1 and 2 Flange-2(Comparison with and without thermal load)

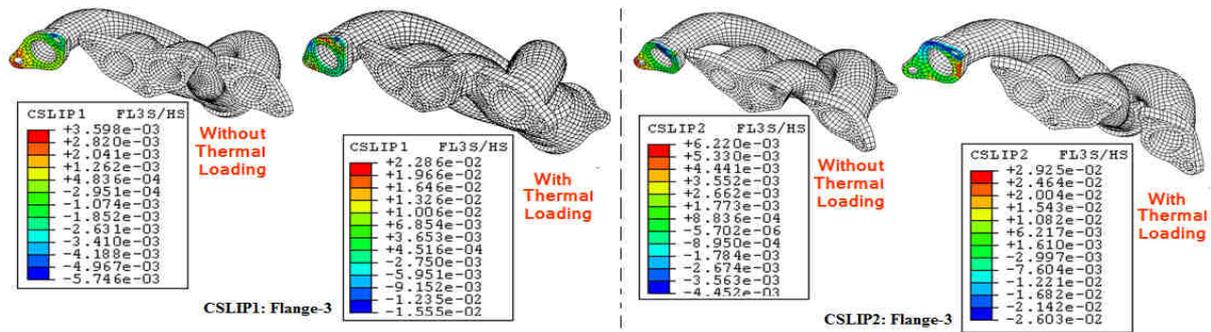


Fig. 11: CSLIP 1 and 2 Flange-3(Comparison with and without thermal load)

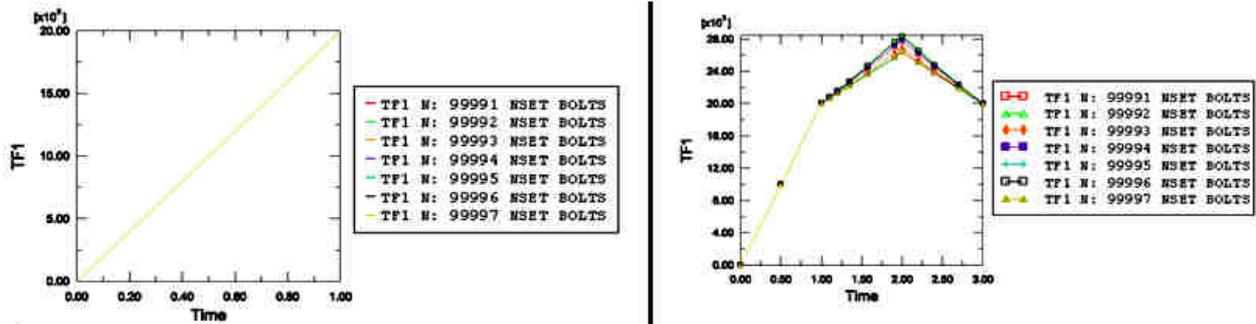


Fig. 12: Bolt forces without thermal loading and with thermal loading

VI. CONCLUSION

By thermo-mechanical coupling analysis of exhaust manifold assemblage, we can get the increase in stress vonmises by 55.9% and contact pressure by 44.5% due to thermal load than the mechanical load, contact between the abutments opens more under thermal loading than the bolt preloading, the joint slippage is more under thermal loading than the bolt preloading. It is also observed that the joint between engine manifold and engine head is meeting the safe joint acceptance criteria since shear capacity is more than the shear load under both the loading. The joint is passing the safe acceptance criteria and the joint slippage within the acceptance. The bolt load increases with respect to time during thermal loading.

REFERENCES

[1] Gopaal, M.M.M.K. Varma and L.S. Kumar, Jr. (2014), “Thermal and Structural Analysis of an Exhaust Manifold of Multi Cylinder Engine,” *International Journal of Mechanical Engineering and Technology* © IAEME, **5**(12), 16-26.
 [2] Ch. Dattu, S.N., M.P. Varma, and B.S. Sundar, Jr. (2014), “Thermal Analysis on 4-1 Tubular Type IC-Engine Exhaust Manifold through Ansys,” *International Journal of Advanced Mechanical Engineering* © RI Publications, **4**(7), 755–762.
 [3] Bin, Z., Y. Hu, F. Yan, and C. Wang, Jr. (2013), “The Impact of Temperature Effect on Exhaust Manifold Thermal Modal Analysis,” *Research*

Journal of Applied Sciences, Engineering and Technology, **6**(15), 2824–2829.

[4] Sweta, J. and A. Agrawal, Jr. (2013), “Coupled Thermal-Structural Finite Element Analysis for Exhaust Manifold of an Off-road Vehicle Design Engine,” *International Journal of Soft Computing and Engineering*, **4**(13), 226–230.