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Procedia Manufacturing 7 (2017) 316 – 326

Procedia MANUFACTURING

International Conference on Sustainable Materials Processing and Manufacturing, SMPM 2017, 23-25 January 2017, Kruger National Park

Harmonic Analysis of Cavitation in Engine Cooling Fluid Due to Piston-Cylinder Assembly Forces

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Abstract

The problem with cavitation and possible pitting in internal combustion diesel engines is studied here. Both theoretical and numerical simulation studies are carried out in order to understand the noise and vibration sources using a force equation. Noise and vibration are modeled in both 2D and 3D using the finite element analysis software ANSYS. A model pressure variation is carried out using the software with finite element analysis and fluid – structure interaction analysis interface. Some relevant results have been shown with the presence of negative pressures leading to the realization that the models developed predicts the presence of cavitation in the cylinder liner surface, indicating that cavitation can occur.

Peer-review under responsibility of the organizing committee of SMPM 2017

Keywords: Cavitation; cooling fluids; internal combustion diesel engines; Ansys; vibrations; Acoustics; Fluid-structure interaction

1. Introduction

In its operation, Diesel engines tend to be widely used due to its fuel economy. They operate at high peak pressures leading to greater noise and vibration. The noise generated in the cylinder chamber causes the cylinder liner to vibrate. The effect of the piston side thrust on the cylinder liner, known as the piston slap cannot be neglected. The result of these activities in the cylinder causes the cooling fluid around the cylinder to undergo vibration, leading to the production of cavitation in the cooling chamber. The activities of these vibrations and noise generate bubbles that implodes along the fluid-solid interface thus the cooling fluid and the cylinder liner interface. The impinging force on

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Peer-review under responsibility of the organizing committee of SMPM 2017
doi:10.1016/j.promfg.2016.12.078

the solid surface produces forces that creates damaging cavities on the cylinder liner called pitting. The formation of these cavities is due to a continuous series of high frequency pressure pulsation in the liquid.

The effects of cavitation are very worrying. Cavitation can even lead to the destruction of the cylinder liner. A modeled fluid-structure interaction featuring cavitation at the interfacing domain boundary as applied on the cylinder liner and cooling fluid is considered assuming negative pressure values indicating presence of cavitation. The modelled parameters included the entry pressure of the fluid, the exit pressure, the point loads and the temperature of the heating surface. For the fluid – structure assembly, one end of it was clamped and the other left to vibrate freely.

2. Literature Review

There is a lot of literature on cavitation since it causes a lot of material failure. Cavitation in fluid machinery leads to wear of their components, Pohl et al [1]. Cavitation has negative effects such as failure of components, reduction of efficiency, noise and vibration as stressed by Hattori et al [2]. Young [3] explains cavitation as the formation and activities of bubbles or cavities in a liquid. These bubbles may be suspended in a liquid or may be trapped in tiny cracks either in the liquid's boundary surface or in solid particles suspended in the liquid. The expansion of these bubbles may be affected by the reduction of the ambient pressure by static or dynamic means. These bubbles grow and implode, causing damaging effect on the solid interface. Lee et al [4] attributed to the fact that the collapse of individual cavitation bubbles in the vicinity of the material surface was due to cavitation erosion. Acoustic cavitation takes place when sound waves, produced by pressure variation in the liquid, take place. Vibratory cavitation also takes place when the liquid is at rest and or flows with a very small velocity. Here, many cycles of cavitation in a given time period can be noticed, Young [3]. Repeated collapses apply local cycle pressure loads to the surface which leads to damage and loss of material, Pohl et al [5].

The engine generates mechanical power but also generates waste heat since they are not efficient. Cooling is done to prevent the engine from burning in its own heat. Most engine liquid coolants contain water and about 30 percent ethylene glycol. For an effective running of the engine the cooling liquid is kept at 70 and 75 degrees centigrade. Chen et al [6] stated that many researchers have already investigated the principle of motion and impact between the piston and inner wall of the internal combustion engines (IC engines). Zinchenko [7] and Meier [8] are the pioneers in the study of noise induced by piston slap. Ungar and Ross [9] studied the dynamics of lateral piston motion across the cylinder clearance spaces and developed relationship between the side thrust force and the inertial force and the combustion force. They found that the times of occurrence of the piston slap in one cycle are dependent on the operating conditions. Haddad [10] and Haddad and Fortescue [11] appear to be the earliest researchers to use a computer to simulate the motion between piston and liner on basis of a simplified mathematical model. Cho et al [12] and Geng [13] used the simple model to estimate the impact force induced by piston slap but the detailed parameters were not considered.

2.1. Force Equation

Cho et al [12], Ungar and Ross [9] derived a force equation relating the piston cylinder assembly based on certain assumptions.

Figure 1 below indicates the work done by Cho, Ahn and Kim which show the schematic diagram and force equation of the engine noise generation.

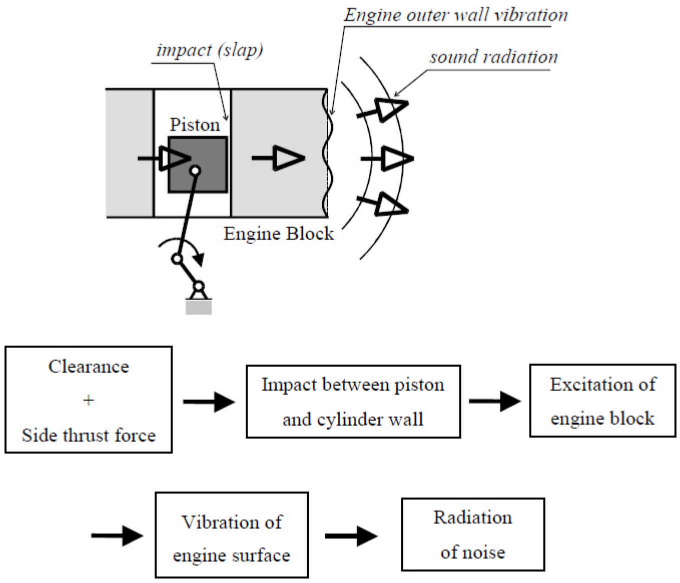


Fig.1. Schematic diagram of generation of engine noise/vibration related to piston slap [12]

A two dimensional lumped parameter model is shown in figure 2 below which describes the motion in translation along the X and Y – axis. The Z axis motion and angular motion are neglected since they are negligible and can be ignored.

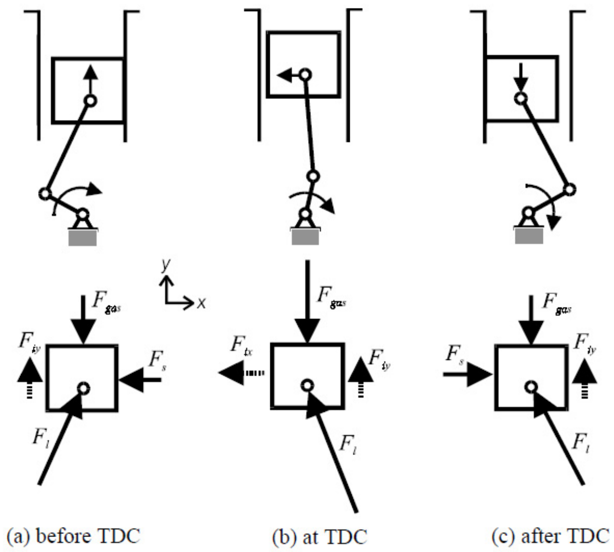


Fig.2. Schematic diagram of generation of engine noise/vibration related to piston [12]

A slider crank mechanism indicating the motion of the piston in the cylinder with its related equations according to Ungar and Ross [9] is shown below.

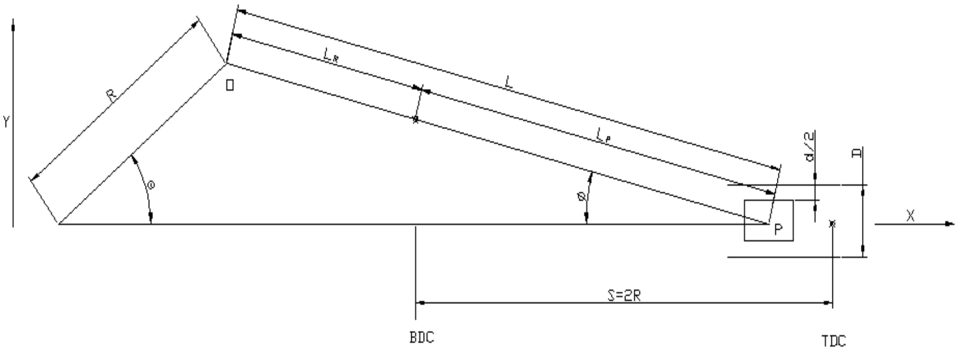


Fig.3. The component parts and its motion [9]

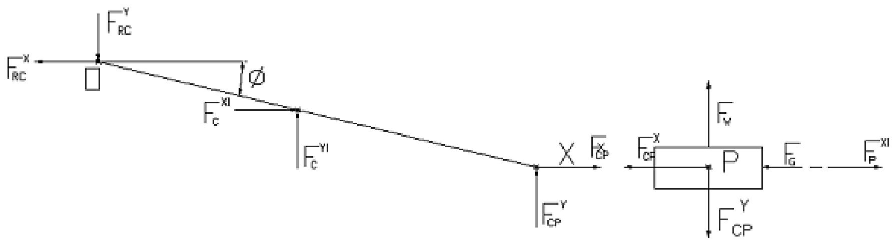


Fig.4. Free body diagram: shows the inertia and forces on its mechanism, Ungar and Ross [9]

2.2. Piston and connecting rod geometry and forces

Taking the piston as a first approximation to be constrained to move only along its ideal kinematic axis of motion, then the calculation of the co-ordinates of the piston P and of the centre of gravity, C, of the connecting rod in terms of the angles θ and ϕ, the corresponding acceleration components may then be obtained by double differentiation with respect to time, using the rotational speed as θ to be constant with the presently considered degree of accuracy.

The angle ϕ is related to the crank angle θ as defined in the figure 4 by

$$\sin\phi = \gamma \sin\theta$$
$$\gamma = \frac{R}{L} \qquad (1)$$

The length of the connecting rod is L. This is greater than the crank Radius, therefore $\gamma^2 > 1$. This leads to the equation mentioned below (Considering the approximations made)

$$\cos\phi \approx 1 - \left(\frac{\gamma^2}{2}\right)\sin^2\theta \qquad (2)$$

$$\tan\phi \approx \gamma \sin\theta \left(1+\left(\frac{\gamma^2}{2}\right)\sin^2\theta\right) \tag{3}$$

$$\phi \approx \gamma \sin\theta + \frac{1}{2}\left(\gamma \sin\theta\right)^3 \tag{4}$$

$$\ddot{\phi} \approx -\gamma\omega^2 \cos\phi \sin\theta \left(1-\gamma^2 \cos 2\theta\right) \tag{5}$$

The inertia forces are as shown below where,

The superscript i denotes inertia, x and y denote co-ordinates,

c- Centre of gravity,

p- Piston

Thus:

$$F_p^{xi} = -M_p \ddot{x}_p = \omega^2 M_p R\left(\cos\theta + \gamma\cos 2\theta\right), \tag{6}$$

$$F_c^{xi} = -M_c \ddot{x}_c = \omega^2 M_c R\left(\cos\theta + L_R \gamma^2 \cos 2\theta\right), \tag{7}$$

$$F_c^{xi} = -M_c \ddot{y}_c = \omega^2 M_c \left(L_p \gamma \sin\theta\right), \tag{8}$$

2.3. Side thrust action on piston

Now the equations of dynamic equilibrium for the piston in the X-direction and for rotation of the connecting rod about the pin O:

From the set of equations mentioned above the side thrust force F_{cp}^{y}, the connecting rod exerts on the piston can be solved. This is as follows:

$$\psi_s = \frac{F_{cp}^{y}}{M_p R\omega^2 \gamma} = \left[\psi_G - \left(\psi_p + \psi_c\right)\right]\sin\theta \tag{9}$$

where

$$\psi_G = \frac{F_G \tan\varphi}{M_p R\omega^2 \gamma \sin\theta} = \frac{F_G}{M_p R\omega^2}, \tag{10}$$

$$\psi_p = \frac{F_P^{xi} \tan\varphi}{M_p R\omega^2 \gamma \sin\theta} = \frac{F_G}{M_p R\omega^2}, \tag{11}$$

$$\psi_c = \frac{\left(F_c^{yi} + F_c^{xi}\tan\varphi\right)L_R\cos\varphi + \left(r^2 + L_R^2\right)\ddot{\varphi}M_c}{L\cos\varphi.M_p R\omega^2\gamma\sin\theta} = \mu L\left(V + \cos\theta\right) \tag{12}$$

In which r denotes the radius of gyration of the connecting rod about an axis through its centre of gravity (and parallel to the crank shaft), and

$$\mu = \frac{M_c}{M_p}, L = \frac{L_R}{L}, \rho = \frac{r}{l} \tag{13}$$

$$V = \frac{\left(L - 2L^2 - \rho^2\right)}{Lr} \tag{14}$$

ψ_s represents a dimensionless side thrust force quantity. ψ_G, ψ_P and ψ_c are made by the gas force, the piston inertia and the connecting rod inertia respectively.

The appropriate equation in 12 was obtained using the inertia force equation (6-8) and the small angle approximations of equation 1, which holds for the usually applicable inequality $\gamma^2 << 1$. $M_p R\omega^2$ may be used as centrifugal force that the piston would exert if it were a point mass attached to the rotating crankshaft at the crack radius.

$\psi_G(\theta) =$ ratio of the gas force at the crank angle θ to its centrifugal force and ψ_p = Ratio of the piston inertia force to this centrifugal force.

The lateral piston motion is induced when $\sin\theta = 0$ and also for $\psi_G = \psi_p + \psi_c$.

This piston slap occurs when $\theta = 0$ (top dead center and bottom dead center), $\theta = \pi$

3. Finite Element Analysis

Models were created using the geometry of the system assembly. Modeling and simulation were carried out using ANSYS. For this problem, use was made of a two dimensional beam and a three dimensional beam. Initially a job was created with its attendant job name. In the pre-processor domain, modeling was picked and then area selected to create the job dimensions. Areas 1 and two were created, one lying on the other, signifying the region for the grey cast iron (cylinder liner) and the cooling fluid (liquid).

For the solid model, loads of magnitude 65N and 45N were applied at points A and B. Also, the temperature from the heating surface was kept at 2000C. One end of it was clamped and the other end left to vibrate freely as studied by Cho et al [12]. For the fluid, an entry liquid pressure and temperature of 3 bars and 750C respectively were used with an exit pressure of 1.86 bars. The flow was also taken into consideration.

3.1. Harmonic Analysis with Fluid-Structure Interaction

Harmonic Analysis with the frequency limits taken as 0 and 100 Hz. The effect of temperature is significant which lays foundation to thermal analysis of the present system. Thermal analysis has been coupled with Acoustic analysis & Structural analysis. Thermal analysis is performed first, taking thermal element 'Quad4node 55'.While performing the acoustic analysis 'Quad4node 42' and '3d acoustic 30' for solid and fluid was used respectively. The latter are performed together and the thermal analysis results are inserted while declaring the boundary conditions for the coupled analysis. The analysis below also takes into consideration the thermal effect on Cavitation (as shown in the 3-D figures). To understand the effect of temperature, a comparison is shown between 2-D (without thermal analysis) and 3-D (with thermal analysis) in the following figures. Here, thermal analysis has been coupled with acoustic analysis & structural analysis, which are performed together.

Fig. 5. (a) To start with a solid structure with required dimensions is taken as shown above; (b) The above figure is the meshing of the solid element

The Thermal Element chosen to perform thermal analysis is solid- “Quad4node 55”. The meshing of the area is as shown below with 3*20 elements. The material properties of gray cast iron in SI units are 20 for thermal conductivity, 547 for specific heat along with emissivity of 0.6. For better results radiation is also considered.

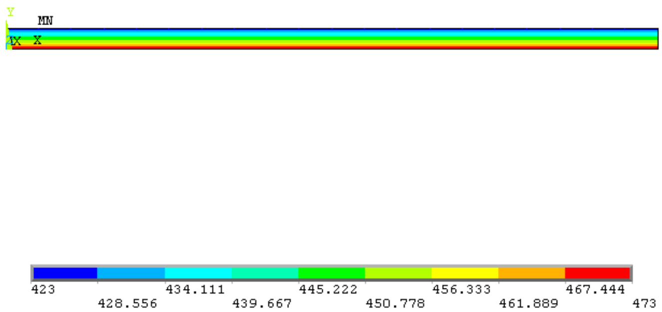


Fig. 6. Figure above is the temperature distribution along the solid structure. The temperatures range from 423K to 473K.

Note: Before we move to Analysis using FLUID-STRUCTURE Interaction we replace the thermal element Quad4node 55 with structural element Quad4node 42 with the same dimensions and similar mesh. The material properties are also changed as per the structural requirements of the acoustic analysis.

3.2. Acoustic Analysis

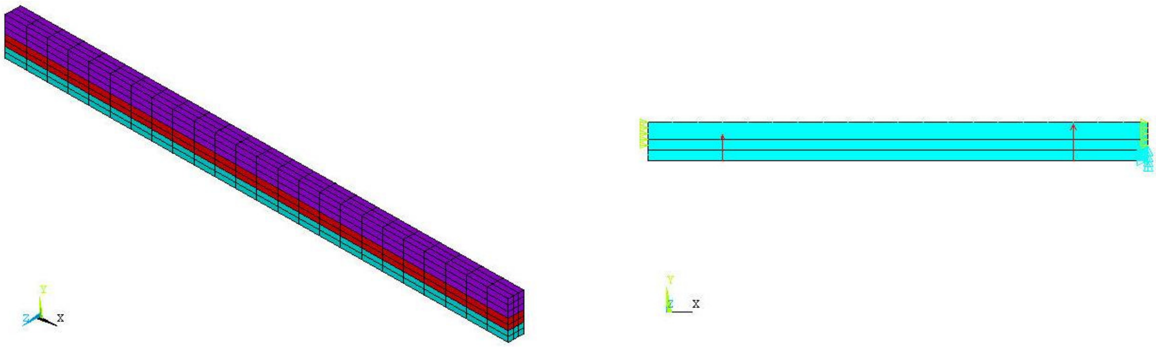
An acoustic analysis, available in the ANSYS Multiphysics and ANSYS Mechanical programs only, usually involves modeling the fluid medium and the surrounding structure. Typical quantities of interest are the pressure distribution in the fluid at different frequencies, pressure gradient, particle velocity, the sound pressure level, as well as scattering, diffraction, transmission, radiation, attenuation, and dispersion of acoustic waves. A coupled acoustic analysis takes the fluid-structure interaction into account.

The ANSYS program assumes that the fluid is compressible, but allows only relatively small pressure changes with respect to the mean pressure. Also, the fluid is assumed to be non-flowing, uniform mean density and mean pressure are assumed with the pressure solution being the deviation from the mean pressure not the absolute pressure.

(a) (b)

Fig.7. (a) The above figure demonstrates the Geometric Model with areas corresponding to fluid (on top, A2) and solid (at the bottom, A1); (b) The 2D mesh of the geometric model

The material models include the properties of the fluid, solid and also elements at the interface. The finite Element model is shown above (fig. 7a) with elements of the fluid on top, elements of the interface at the center and solid at the bottom. It is important to create similar mesh for both solid and fluid so that they could be merged accurately as shown above (fig. 7b). After merging the fluid and solid elements, an interface is created by merging lines 3 and 5 in the mesh.

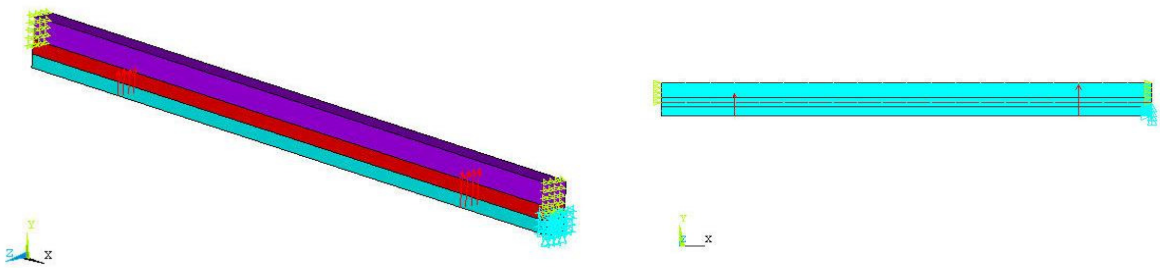


(a) (b)

Fig. 8. (a) The 3D mesh of the geometric model; (b) Demonstrates the various boundary conditions in the 2D

After creation of the mesh, the boundary conditions for both the fluid and the solid are specified. Then we associate the line at the center and the nodes attached to it with the properties of the third material as declared along with the impedance. The effect of temperature is taken from the thermal analysis file with extension .rth, which cannot be seen in the figure.

The following figures (fig. 9) demonstrate the various boundary conditions



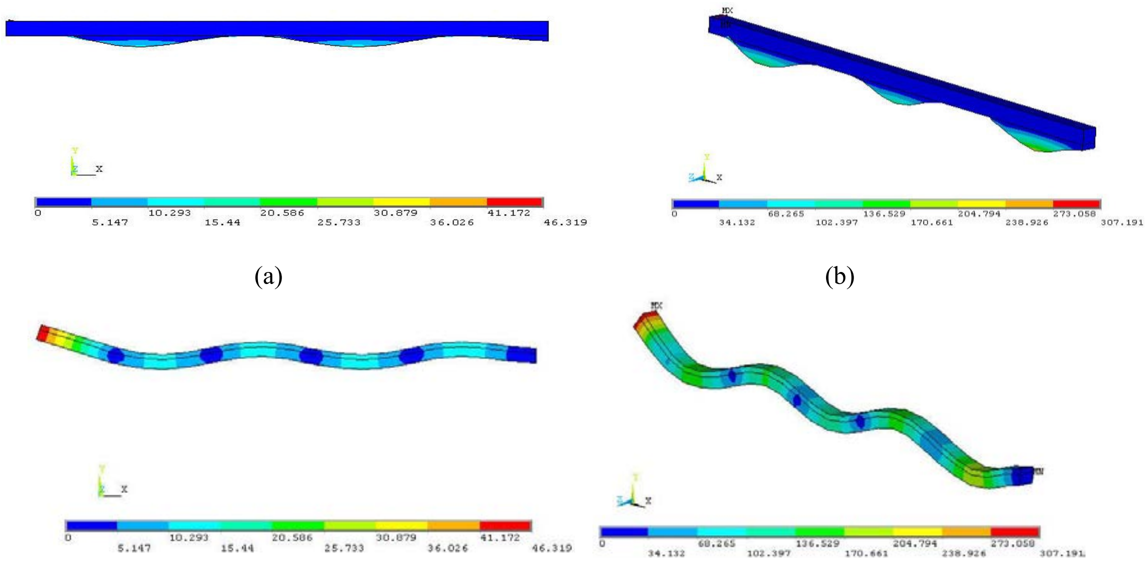
(a) (b)

Fig. 9. (a) Demonstrates the various boundary conditions in the 3D; (b) All the boundary conditions and impedance shown in particular in the 2D.

Note: Before we move to solution options we consider the frequency interval for harmonic analysis between 0 and 100 Hz with load step of 20.

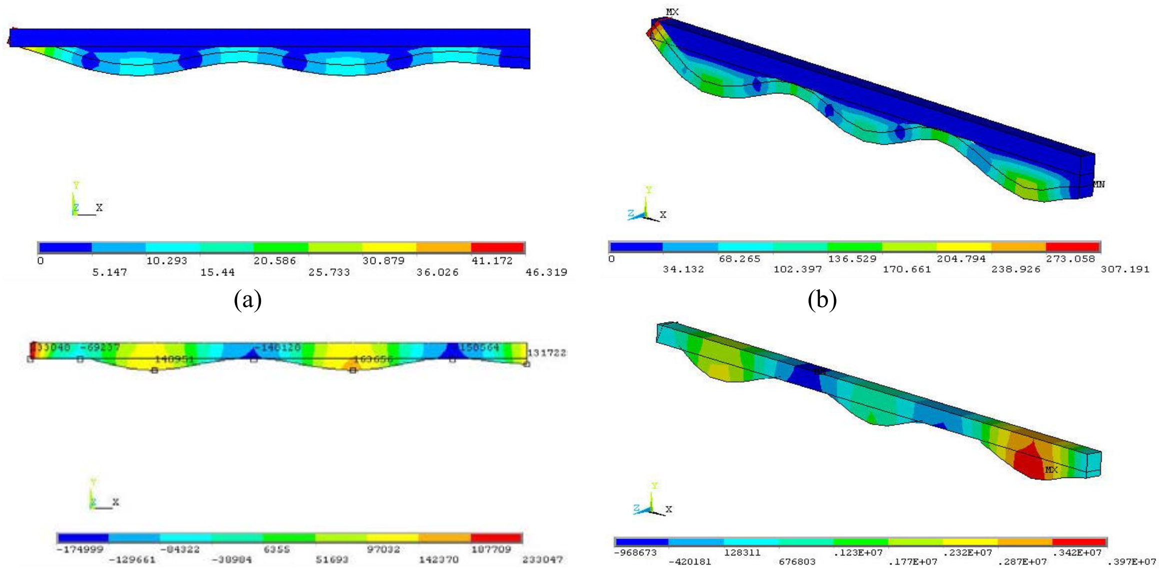
4. Results

Following (fig. 10 – 12) are results of the harmonic analysis performed starting with the displacement in the fluid due to the vibration in the solid structure on account of the loads applied at the bottom.



(d)

Fig. 10. (a). Figure demonstrating the displacement in the fluid in the 2D; (b). Figure demonstrating the displacement in the fluid in the 3D; (c).Figure demonstrating the displacement in the Solid in 2D; (d). Figure demonstrating the displacement in the Solid in 3D



(c) (d)

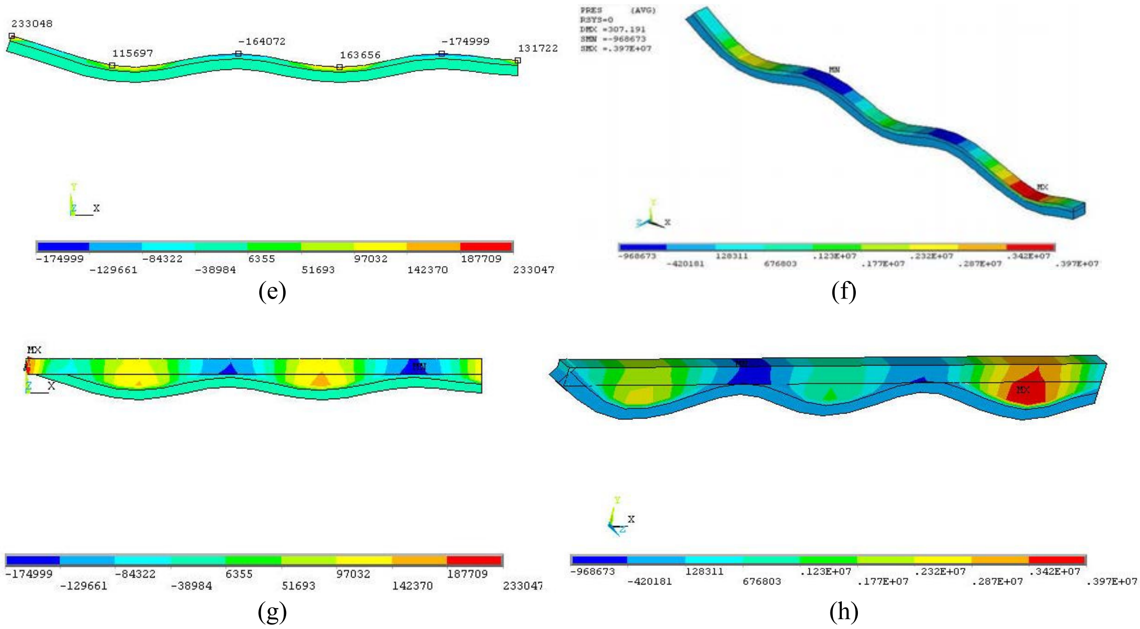


Fig. 11. (a). Following is the displacement of the coupled structure in 2D ; (b). Following is the displacement of the coupled structure in 3D; (c) Figure demonstrating the changes in pressure in the fluid after the total analysis in 2D; (d). Figure demonstrating the changes in pressure in the fluid after the total analysis in 3D; (e). The above figure shows the pressure drop below zero in the solid, where there is a chance for development of a cavity in 2D; (f). The above figure shows the pressure drop below zero in the solid, where there is a chance for development of a cavity in 3D; (g). Pressure drop in the whole system in 2D; (h). Pressure drop in the whole system in 3D

The above figures (Fig. 11) show the pressure drop below zero in the fluid, which is an evidence of presence of cavitation.

The Following (fig. 12) is the FE Model of the assembly for an elemental indication of the cavitation location on the solid structure

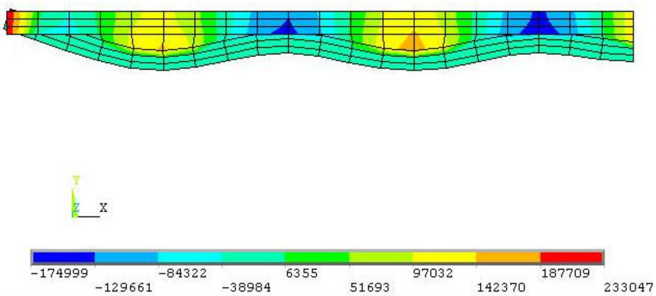


Fig. 12. Pressure drop in the whole system with elemental description

5. Conclusion

The occurrence of cavitation in IC diesel engines need not be overlooked. The damaging effects of cavitation have material and financial effect on users. In this paper consideration was given to cavitation, force equation on the piston cylinder assembly of an IC diesel engine. Data taken from other works were used to model a 2D and 3D simulation using ANSYS.

Harmonic analyses were considered for both thermal and acoustic consideration with respect to Grey Cast Iron and the cooling fluid. The effect of the thermal properties led to higher values of pressure variation in magnitude. The results indicated certain negative pressure values on the pressure variation, predicting cavitation can occur. The effect of cavitation can be reduced by reducing the clearance between the piston and the cylinder liner. Also wrapping the piston skirt with Teflon pad on the trust side.

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