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DESIGN AND VIBRATION ANALYSIS OF TWO STROKE CRANKLESS GASOLINE ENGINE

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Abstract**—** Noise and vibration have attracted attention because of the growing demand for low noise engines. Analyzing and predicting engine noise and vibration are important in generating a design in accordance with demand and regulations. If an engine will be used in a motor vehicle, certain noise levels have to be observed. The vibration of the cylinder block, which is excited by the main bearing force, combustion force, and piston side force, primarily generates airborne noise. The relationships among the designs of the engine structure, the excitation forces, the vibration of the engine, and the noise produced by engines are the main factors to consider in modelling and designing. This project focused on opposed two stroke engine structure design subjected to vibration using finite element analysis. A model of the engine is constructed by using Creo parametric software and performed a modal analysis to arrest vibration of the engine structure.

Keywords — Cylinder, Engine, Creo.

I. INTRODUCTION

Noise and vibration have attracted attention because of the growing demand for low-noise engines. Analyzing and predicting engine noise and vibration are important in generating a design in accordance with demand and regulations. The vibration of the cylinder block, which is excited by the main bearing force, combustion force, and piston side force, primarily generates airborne noise. Structural analysis focuses on the static and dynamic area and demonstrates the structural mode shape, the resonant frequencies, and the response of the structure to the internal forces of the engine design. The modal analysis approach is the process of describing a structure in terms of its natural characteristics, such as frequency, damping, and displacement patterns (mode shapes), that facilitate structural vibration. The natural frequencies of structures should be estimated at the design stage, and if required, the structure should be stiffened so that the natural frequencies would be increased above the frequency of the excitation forces to prevent resonance from occurring in the system. The prediction of vibration for crank less engine is also important in determining the stability of a structure that allows it to withstand the pressure during the combustion process. A number of previous studies have provided a thermodynamic analysis of the combustion process in the engine, but limited research has been conducted to evaluate the vibration of the crank less engine design structure.

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II. LITERATURE SURVEY

This study focuses on engine structure design subjected to vibration using finite element methods. The finite element method (FEM) have been employed in numerous studies. The application of FEM in vibration analysis are promising, especially at the early design stage. Modifying and analyzing the design is easier and more cost-effective than building a prototype. A complex structure can be configured and modeled using these methods so that response at any desired point of the structure can be determined easily. The diagnosis methodology is based on the cauterization of working conditions by means of acoustic and vibration measurements and relating the data to the pressure inside the cylinder. The dynamic behavior of a current production 1.6 liter gasoline engine with the objective of reducing low frequency radiated noise from the cylinder block. Theoretical development of the engine balance motion and frequency response was also conducted. From the simulation and finite element analysis, the force response pattern of the engine vibration was determined and then compared with its natural frequency. The vibration data were used as the input data for noise analysis using the boundary element method. The integration of the finite element and the boundary element determined the noise-frequency data of the engine structure toward the occurrence of engine noise. The relationships among the designs of the engine structure, the excitation forces, and the vibration of the engine are the main factors to consider in modeling and designing a new engine. These relationships can be determined from the integration of structural and creo parametric analysis.

III. WORKING PRINCIPLE:

The crank less linear generator generally consist of three subsystem: Combustion Chamber, linear generator and return unit (normally a gas spring), which is coupled through a connecting rod. In the combustion chamber, a mixture of fuel and air is ignited, increasing the pressure and forcing the moving part in the direction of the gas spring. The gas spring is compressed, and, while the piston is near the BDC, fresh air fuel are injected into the combustion chamber, expelling the exhaust gases. The gas spring pushes the moving parts assembly back to the TDC, compressing the mixture of air and fuel that was injected and the cycle repeats.

The linear generator can generate a force opposed to the motion, not only during expansion but also during compression. The magnitude and the force profile affect the piston movement, as well as the overall efficiency.

Methodology

A two stroke crank less gasoline engine has to be meet the stringent design requirements for automobiles. The Aluminum Alloy (Al6061) was chosen for reference and the components will be analyzed. The material properties of all materials considered from design considerations. The analysis carried out by using CREO PARAMETRIC 3.0.

IV. MATERIAL SELECTION

Aluminum Alloy (Al6061) is one of the most widely used alloys in the 6000 Series. This standard alloy, is one of the most versatile of the heat-treatable alloys, and is popular for medium to high strength requirements. It has very good corrosion resistance and very good weld ability although reduced strength in the weld zone. It has medium fatigue strength.

CHEMICAL COMPOSITION:

Mg	Si	Fe	Cu	Cr	Zn	Ti	Mn	Al
0.8-1.2	0.4-0.8	0.0-0.7	0.15-0.40	0.04-0.35	0.0-0.25	0.0-0.15	0.0-0.15	96-98.61

Design Of Two Stroke Crank Less Gasoline Engine

The crank less engine has a straight rod that moves back and forth with a piston and 2 stroke combustion chamber on either ends. The central portion of the rod has a magnets attached which move part stationary coil to generate electricity.

This creates an opposed 2 stroke, every stroke is a power stroke generating electricity. Used in place of a fuel cell, you still power the wheels with an electric motor. The engine could approach 50% efficiency which is about twice as efficient as standard internal combustion engine.

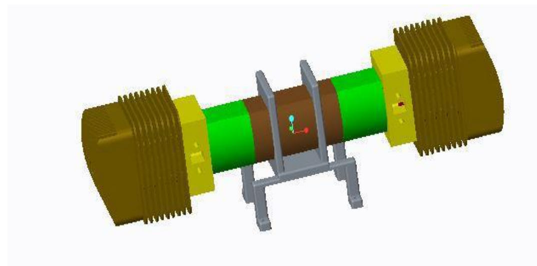

Fig.1 Model of two stroke crank less gasoline engine

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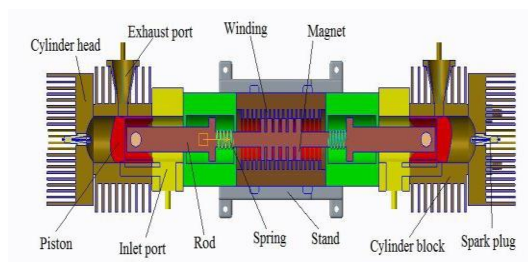
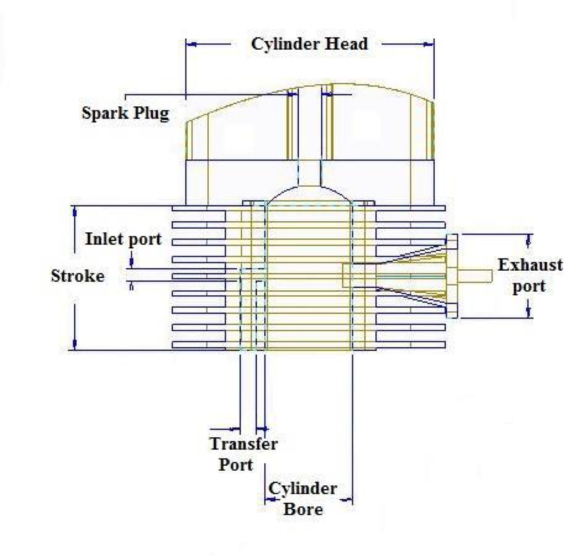


Fig.2 Cut section view of two stroke crank less gasoline engine

1.	Engine type= Air cooling 2-stroke engine
2.	Fuel type = Petrol
3.	Bore x Stroke = 88mm x 80mm
4.	Displacement = 120cc
5.	Maximum power at =33.333Hz
6.	Compression ratio = 10:1
7.	Density of petrol = 737.22kg/m^3
8.	Temperature = 288.855k

Table 7.1 Engine specifications

V. DESIGN OF A CYLINDER:



Diameter of the cylinder, D = 38 mm.
Brake power, BP = 5KW = 5000 W.
Speed, N = 2000 rpm.
Mean effective pressure, Pm = 0.35 N/mm2.
Mechanical efficiency, ηm = 80 % = 0.8.

LENGTH OF THE CYLINDER:

l= Length of the stroke in mm = 1.5 D
= 1.5(38) = 57 mm.
Length of the Cylinder, L = 1.15 l
= 1.15(57) = 65.55 mm
L = 70 mm (approx.).

THICKNESS OF THE CYLINDER:

The maximum pressure (P) in the Engine cylinder is taken as 9 to 10 times the mean effective pressure (Pm),
P = 9 Pm
= 9(0.35) = 3.15 N/mm2
Thickness of the Cylinder head, Th = D√ (CP/σt)
= 3.3 mm
(Take, C=0.1, σt = 42 Mpa = N/mm2)

9.1.3 SIZE OF THE STUDS FOR THE CYLINDER HEAD:

Force acting on the cylinder head, = π/4 x D2 x P
= π/4 x 382 x 3.15
= 3571 N.
Let us take, ns = 4.5
(Take, σt = 65 Mpa = 65 N/mm2, dc = 0.84d)
Resisting force offered by all the studs = ns x π/4 x dc2 x σt
= 162 d2
162 d2 = 3571
d = 5 mm.
The pitch circle diameter of the studs (DP) is taken D + 3d,
DP = 38 + 3(5) = 53 mm.
Pitch of the studs = π x DP / ns
= π x 53 / 4.5 = 37 mm.

PISTON HEAD (OR) CROWN:

Thickness of piston head, Th = √ 3PD2/16σt
= √ 3x5x382 / 16x38
= 6 mm.
(Take, σt = 38 Mpa = 38 N/mm2)
Cross sectional area of the cylinder, A = π/4 x D2
= 1134 mm2.
Indicated power, IP = Pm x L x n / 60
= 0.75 x 0.057 x 1134 x 2000 / 60
= 1616 W
= 1.616 KW.
Brake power, BP = IP x ηm
=1.616 x 0.8
= 1.3 kW.
Heat flowing through the piston head, H = C x HCV x m x BP
= 0.05 x 42 x 103 x 41.7 x10-6 x 1.3
(Take, C = 0.05) = 114 W.
Thickness of the piston head on basis of heat dissipation,
Th = H / 12.56 K (Tc – TE)
= 114 / (12.56 x 46.6 x 220) = 0.9 mm.
= 6 mm.
Take Larger of the two values.
(Take, K = 46.6 W/m/◦c, Tc – TE = 220 ◦c)

RADIAL RIBS:

The radial ribs may be four in number. The thickness of the ribs varies from Th / 3 to Th / 2.
Thickness of the ribs, Tr = 6/3 to 6/2
= 2 to 3 mm.
Let us adopt, Tr = 3 mm.

PISTON RINGS:

No. of Rings, nr = 2.
Radial thickness of the piston rings, T1 = D √ 3Pw / σt
= 38 √ 3 x 0.035 / 90
= 1.3 mm.
(Take, Pw = 0.035 N/mm2, σt = 90 Mpa)
Axial thickness of the piston rings, T2 = 0.7 T1 to T1
= 0.9 to 1.3 mm.
Let us adopt, T2 = 1 mm.
Minimum axial thickness of the piston rings, T2 = D / 10 nr
= 38 / 10 x 2

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Let us take the larger value, T2 = 2 mm.
Width of the top land, b1 = Th to 1.2 Th
= 6 to 1.2 x 6
Let us adopt, b1 = 7 mm.
Width of the other ring land, b2 = 0.75 T2 to T2
= 0.75 x 2 to 2
Let us adopt, b2 = 2 mm.
Gap between the free ends of the ring, G1 = 3.5 T1 to 4 T1
= 3.5 x 1.3 to 4 x 1.3
Let us adopt, G1 = 5 mm.
Gap between the ring in the cylinder, G2 = 0.002 D to 0.004 D
= 0.002 x 38 to 0.004 x 38
Let us adopt, G2 = 1 mm.

PISTON BARREL:

Radial depth of piston ring grooves is about 0.4 mm.
Radial thickness of the piston rings (T1), b = T1 + 0.4
= 1.3 + 0.4
= 1.7 mm.
Maximum thickness of the barrel, T3 = 0.03 D + b + 4.5 = 0.03 x 38 + 1.7 + 4.5 = 7.34 mm.
Piston wall thickness towards the open end, T4 = 0.25 T3 to 0.35 T3
= 0.25 x 7.34 to 0.35 x 7.34
Let us adopt, T4 = 2 mm.

PISTON SKIRT:

Maximum side thrust on the cylinder due to gas pressure (P),
R = μ x π x D2 x P / 4
= 0.1 x π x 382 x 5 / 4
= 567 N.
Side thrust due to bearing pressure on piston barrel (Pb), R = Pb x D x l
= 0.45 x 38 x l
17 l = 567
l = 567 / 17
l = 35 mm.
Total length of the piston,
L = Length of the skirt + Length of the ring section + Top land.
L = l + (4 T2 + 3 b2) + b1
= 35 + (4 x 2 + 3 x 2) + 7
= 56 mm.

PISTON PIN:

Take, Pb1 = 25 N/mm2.
Load on the pin due to bearing pressure = Bearing pressure x Bearing area.
= Pb1 x do x l1
= 25 x do x 0.45 x 38
(Take, l1 = 0.45 D) = 428 do N.
Maximum load on the piston due to gas pressure (or) Maximum gas load,
= π x D2 X P / 4
= π x 382 X 5 / 4
= 5671 N.
We find that, 428 do = 5671
do = 5671 / 428 = 13.25 mm
do = 13 mm.
The inside diameter of the pin (di) is usually taken as 0.6 do,
di = 0.6 x 13 = 8 mm.
Let the piston pin be made of heat treated alloy steel for which the bending stress (σb) may be takes as 140 Mpa. Now, let us check the induced bending stress in pin.
Maximum bending moment at the centre of the pin, M = P x D / 8
= 5671 x 38 / 8 = 26937.25
= 27 x 103 N-mm.
We also know the maximum bending moment (μ),
27 x 103 = π/32 (do4 - di4/do) σb
27 x 103 = π/32 (134-84/13) σb
σb = 146 N/mm2.
σb = 146 Mpa.
Since, the induced bending stress in the pin is less than the permissible value of 140 Mpa (140 N/mm2), therefore the dimensions for the pin as calculated above (do = 13 mm and di = 8 mm) are satisfactory.

VI. VIBRATION ANALYSIS ENGINE STRUCTURE VIBRATION ANALYSIS:

A crank less engine is a two-stroke engine without a rotary component, such as a crank and a camshaft, normally found in a typical engine design. In the engine, two horizontally opposed pistons are mounted on a common connecting rod, which is allowed to oscillate freely between the two end-mounted cylinders. As the piston moves in either direction, one cylinder undergoes the expansion process while the other undergoes the compression process.

In this study, combustion in one cylinder with compressed pressure at another cylinder has been selected for the crank less engine design. The compressed pressure in the other cylinder forces the piston back and forth in an alternating manner. However an engine with a single combustion cylinder and rebound device tends to become imbalanced. This situation occurs when pistons subjected to pressure in both cylinders generates vibration, which will be examined further in the dynamic analysis of the engine structure.

Engine structure simulation:

In normal mode analysis, no loading configurations were employed, which is known as a free-free situation. The normal mode simulation determines the natural frequencies and the mode shapes of the engine. Thus, the output data from the normal mode analysis was used for vibration analysis of the engine structure using CREO PARAMATRIC 3.0 software. The data input for frequency and force response analysis were taken from a previous study conducted on combustion, another combustion chamber, and friction force data. An additional force datum for the bushing or bearing element is defined by two axis systems, one on each of the two bodies connected by this element. An axis system is created on each body to define the location and orientation of the force. Another additional input datum is the translational spring damping actuator (TSDA), which defines a spring damper- actuator force element between two bodies.

A mesh mapping set is used to transfer node-based data from the source mesh to another target mesh, such as a set of modes calculated on a fine structural mesh that should be used on a coarser surface mesh in a creo parametric calculation.

The transfer vector set is designed to set up and manage transfer data vectors defining a model. The basic parameters of the set are the physical type of structural inputs, structural outputs, and creo parametric outputs. A transfer vector defines the relationship between a physical quantity at an output location (i.e., the response) and a physical quantity at an input location (i.e., the load).

VII. RESULTS AND DISCUSSIONS

The FEM of the structure normal mode analysis produced four significant modes. Mode 1 was at 423.42 Hz, mode shape showed the engine structure undergoing a twisting phenomenon. For the second mode, the frequency was at 426.49 Hz, where the structure tilted in the –*x* direction,. The third mode of the engine had an eigenvector value equal to 4 at a frequency of 558.5 Hz.

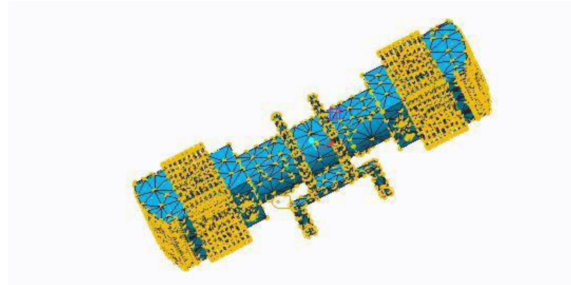

Fig.3 Mesh view of two stroke crank less gasoline engine.

MODE NUMBER	FREQUENCY (Hz)	MODE SHAPE
1	423.42	Twisting
2	426.49	Tilting (-x direction)
3	531.40	Bending (x direction)
4	534.08	Bending and tilting (y direction)

Table 11.1 Data from the normal mode analysis of engine structure.

Stress Mode

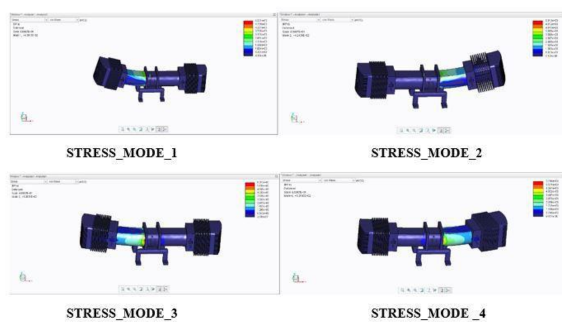

Fig.4 Engine 4-modes stress view.

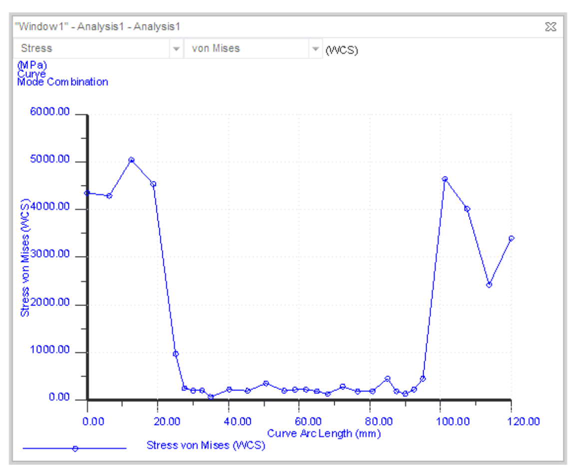

Fig.5 Stress in generator length

STRAIN VIEW

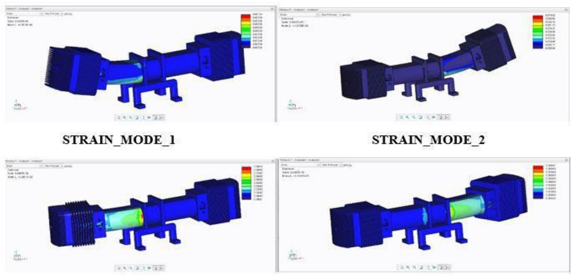

Fig.6 Engine 4-modes strain view.

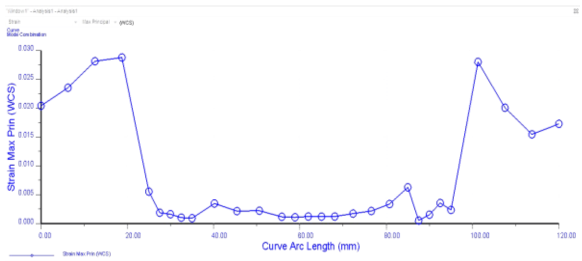

Fig.7 Strain in generator length

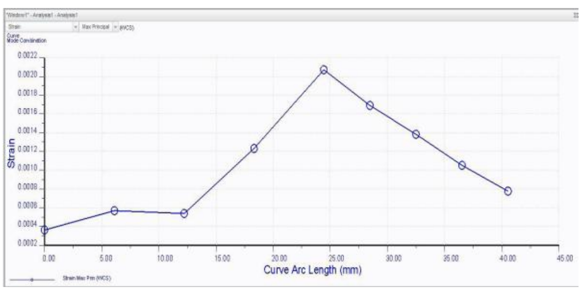

Fig.8 Strain in left side cylinder head length

The mode shape showed that the structure was bent in the *x* direction.

Mode 4, indicated the highest eigenvector value of 23, which occurred at another combustion chamber with a frequency of 534.08Hz. The structure tended to bend in the *y* direction. The output data of the normal mode analysis are summarized in Table 1.

DISPLACEMENT VIEW

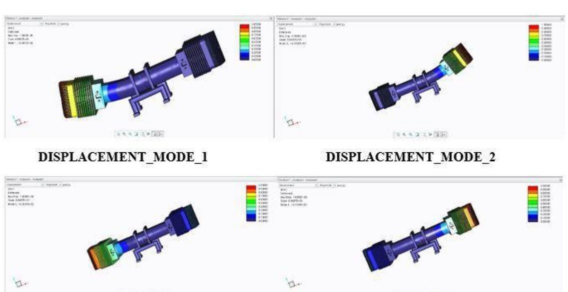

Fig.9 Engine 4-modes displacement view.

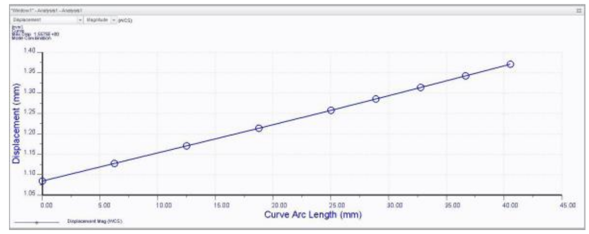

Fig.10 Displacement in right side cylinder head length

The modal data for the engine natural frequencies can be used as an input for vibration analysis using the boundary element approach. Based on these data, structural changes and modifications can be made. For example, if the operated engine frequency was equal to 426.49 Hz, This graph shows that the engine frequency determined from the loading function had a maximum amplitude at 534.08Hz.

MODE	STRAIN	STRESS (Mpa)	DISPLACEMENT (mm)
1	0.08104×10^{3}	3.193	1
2	0.2945×10^{3}	3.509	1
3	0.3×10^{3}	6.043	1
4	0.3×10^{3}	5.743	1

Table: 11.2 Result for model analysis.

The structure tended to bend in the *y* direction. The output data of the normal mode analysis are summarized in Table 1. The modal data for the engine natural frequencies can be used as an input for vibration analysis using the boundary element approach. Based on these data, structural changes and modifications can be made. For example, if the operated engine frequency was equal to 426.49 Hz, This graph shows that the engine frequency determined from the loading function had a maximum amplitude at 534.08Hz.

To prevent the occurrence of resonance, the combustion chamber has to be modified. Creo software was used to create the boundary element mesh of the engine structure. This mesh was used to determine from the outer surface of the vibrated structure. From the integration of the normal mode and the frequency response analysis using creo software, for the engine requirement, the minimum running frequency was at 50 Hz.

VIII. CONCLUSION

By using the finite element approach has been presented to determine the vibration properties of the engine structure. The vibration properties obtained from the normal mode results indicate that the engine structure has four natural frequencies, with the highest value at mode 4. The engine also has four different mode shapes, and is twisted at mode 1, tilted in the $-x$ direction at mode 2, bent toward the x direction at mode 3, and bent and tilted in the $-y$ direction at mode 4. The results from the frequency response analysis demonstrated that the highest engine response occurred at 534.08Hz. The maximum strain obtained at the cylinder head one is 22mm from the left side of the head. For the opposite cylinder head the maximum strain energy is 26mm from the right side of the head. In the generator portion the maximum strain is obtain at 19mm from the right side of the generator. The maximum stress obtained at the cylinder head one is 22mm from the left side of the head. For the opposite cylinder head the maximum stress energy is 27mm from the right side of the head. In the generator portion the maximum stress is obtain at 14mm from the right side of the generator. The maximum displacement obtained at the cylinder head one is 48.5mm from the left side of the head. For the opposite cylinder head the maximum displacement energy is 48.5mm from the right side of the head. In the generator portion the maximum displacement is obtain at zero from the right side of the generator.

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