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Intake Manifold Noise Prediction Due to the Combined Effects of Combustion Loads and Fluid Flow

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Abstract— Reduced engine noise has contributed greatly to the comfort of today's passenger vehicles. The trend towards lighter vehicles has led to massive increase in the use of plastic parts, especially for engine components such as intake manifolds and intake air pipes. The primary purpose of using a plastic material instead of conventional aluminum cast for intake manifold is to reduce its weight and cost. The engine power can be increased with the help of improved interior surface roughness and lowered air temperature. The increased usage of plastics for air intake manifold (AIM) production, in place of metallic materials, made the NVH optimization more complicated. In recent years, automotive engine manufacturers are increasingly focusing their attention on noise generated by plastic air intake manifolds (AIMs).

The main objective is to predict the B12D engine intake manifold noise due to the combined effects of combustion loads and fluid flow pressure at various engine speeds (2000rpm-6400rpm). The meshing of intake manifold was done in finite analysis software HYPERMESH. The post processing was done in NASTRAN software to get the noise levels in AIM. The analytical results were validated by using experimental results.

Keywords— Air Intake Manifold, Noise, NVH, Finite Element Analysis, HYPERMESH, NASTRAN.

I. INTRODUCTION

An intake manifold is a piece of metal tubing that directs air into an internal combustion engine. The intake manifold itself has no moving parts, but it performs an important part of making an engine run. An engine needs constant airflow to mix with gasoline to form an explosively combustible fuel that will create power to move a vehicle. Intake manifold is the pathway between throttle body of carburetor and cylinder head. It contains two parts. Intake manifolds consist typically of a plenum, to the inlet of which bolts the throttle body, with the individual runners feeding branches which lead to each cylinder. Piston action creates a vacuum in the manifold, which draws outside air through the throttle body and into the manifold.

Air intake manifold is subjected to various modes of excitation: structure-borne vibration from the engine (to which it is attached) and air borne sound and air-rush noise from within the intake manifold itself. The intake manifold responds to this excitation by vibrating at its natural frequencies. The motion of vibrating manifold is transmitted to the surrounding air as sound energy. An air intake manifold is subjected to mainly three types of loads.

1) *Structure-Borne Noise*: Structure-borne noise is produced from the engine to which it is attached. To satisfy performance and emission requirements, the combustion process takes place very rapidly over a relatively short period when the piston is close to the top of its travel. For the four stroke cycle this rapid pressure rise in each cylinder is repeated every other rotation of the crank shaft. Due to the high impact loads developed in the combustion process, the engine cylinder gets vibrated. Since intake manifold is closely attached to the engine cylinder head, it is also vibrates. So obviously sound produced from the vibrating component.

2) *Air-Borne Noise*: Piston action creates a vacuum in the manifold, which draws outside air through the throttle body and into the manifold. When inlet valve is open, due to the pressure difference between the engine cylinder and intake manifold, the air-fuel mixture is flowing into the cylinder. At the time of inlet valve closes, the fast moving incoming air-fuel mixture is abruptly halted by the closing of the inlet valves. The resulting pressure pulse then propagates against the mean flow back to the intake manifold where it radiates noise.

3) *Mechanical Noise*: The primary noise source within a reciprocating engine is that produced the piston-crank mechanism as components traverse running clearances and create impacts. Such impacts occur at main bearings, big end bearings and, more especially, between the piston and their respective liners, commonly known as PISTON SLAP.

In the case of piston slap, the side force on the piston changes direction a number of times throughout its 720 degree operating cycle as shown in. The number and location of traversals being dependent upon speed and load. In large, slow speed diesel engines, the biggest impact usually takes place around the firing top dead centre.

II. OBJECTIVE

The objective is to predict the vibro-acoustic properties of B12D air intake manifold due to the combined effects of combustion loads, bearing loads, piston slap load and fluid flow pressure by using the finite element analysis software NASTRAN.

III. ANALYTICAL FEA ANALYSIS

The FEA model of the entire powertrain in conjunction with the intake manifold was built using Hypermesh, a Commercial software. The model included the proper Combustion and valve train forces.

Fig 1 Meshed model of intake manifold

A. Bearing Loads:

To apply mean bearing forces build RBE3 spiders with the dependent node at the center of the main bearing and the independent points connected to the bulkhead nodes, where the bearing would normally rest. The spider should be distributed around the bearing as shown in the figure below.

Fig 2 Bearing load representation

Fig 3 Representation of bearing loads with RBE3 elements

It is not necessary to connect every node on the bearing area of the bulkhead but it does not matter if all the nodes are connected. All nodes connecting to the RBE3 elements should be in global coordinates.

B. Piston-side And Combustion Loads:

Piston side and combustion are also applied at a single point and a similar RBE3 spider should be built to distribute the load. For the piston side load, the dependent node of the RBE3 spider should be on the bore center line and close to or exactly at the piston pin center when the piston is at the TDC of that cylinder. Independent nodes of the RBE3 spider should be connected to the cylinder liner nodes spreading approximately $\pm 15^\circ$ from the center and also one element up and down as shown in the figures below.

Fig 4 Representation of loads using RBE3 element

The dependent node for the RBE3 spider for the combustion load should also be on the bore centerline. However, its location along the bore centerline is arbitrary. It can be placed at any convenient location as shown in the figures below. The independent nodes should be connected to the combustion chamber nodes to evenly distribute the load as shown in figures below. It is not necessary to connect all the nodes in the combustion chamber yet it does not matter if all are connected. All nodes connecting to the RBE3 elements should be in global coordinates.

C. Engine Mountings:

In order to obtain correct rigid body response of a powertrain (or an engine) to engine forcing the engine and transmission mounts should be modeled. Unless the calling procedure specifies otherwise, the engine and transmission mounts should be modeled using the default values specified below.

For the NVH analysis, The Mount brackets, when their models are available should be included in the model. Otherwise, RBE3 links, as shown in the figure below should be constructed to attach the linear springs that represent the mount stiffness and damping. Use CELAS2 cards to specify stiffness and damping for engine mounts. Use of CELAS1 requires an additional property card which is usually an unnecessary complication. Alternative method of specifying mount stiffness would be to use CBUSH and PBUSH cards where stiffness and damping can be specified for all DOF on the same card.

When modeling the engine mounts do not make zero size springs connected to coincident nodes. This makes debugging the model more difficult. Springs should be always visible when model is displayed as shown in figure below.

Fig 5 Representation of engine mounts

D. Material Properties:

For all the components in the engine, the material properties are needed to run the analysis. In Hypermesh, the material card was created and the

properties are given. The material properties are shown in the table 1.

TABLE I
COMPONENT MATERIAL PROPERTIES

com	Mat	E	μ	ρ	C
Head	Al	73.5	0.32	2.75	0.09
Block	Al	12	0.26	7.2	0.15
Intake manifold	Plastic	10	0.39	1.4	0.06
Front cover	Al	73.5	0.32	2.75	0.05
Oil pan	Steel	200	0.29	7.8	0.3
Bearing cup	Cast iron	100	0.3	6.8	0.15
IM bracket	Steel	200	0.3	7.7	0.05

E. Input Data:

For the noise analysis, the main bearing loads, combustion loads and piston side load for 5500 rpm are graphically represented below. Like this the loads are applied to the engine to predict the noise levels in the intake manifold.

Fig 5 crank angle vs bearing load 1

Fig 6 crank angle vs bearing load 2

Fig 10 frequency vs combustion load

Fig 7 crank angle vs bearing load 3

Fig 11

frequency vs piston side load

Fig 8 crank angle vs bearing load 4

Fig 9 crank angle vs bearing load 5

IV. RESULTS AND VALIDATION

The noise in B12D intake manifold at the particular engine speed 4500 rpm was predicted for the application of combustion loads, bearing loads and fluid flow pressure. The noise level in the intake manifold is 98 dBA at 5500 rpm. The same procedure was followed for the remaining engine speeds. The overall maximum noise obtained is 96 dBA at 5100 rpm. And the maximum noise level 92 dBA obtained at 4500 rpm.

Fig 12 sound power vs frequency at 4500 rpm

Fig 13 sound power vs frequency at 5100 rpm.

The maximum sound power was predicted as 98 dBA at 5500 rpm. The same procedure was adopted for the all engine speeds from 2000 rpm to 6400 rpm with the engine firing order variation up to 30th order. The pseudo sound power was predicted for all the engine speeds shown in the graph below.

Fig 14 sound power vs frequency at 5500 rpm

Fig 15 sound power vs engine speed.

The experimental results of B12D intake manifold and the analytical FEA results were correlated with the experimental data for the B12D intake manifold. The maximum sound power predicted through the experiment was 101 dBA at 5500 rpm and the results obtained with analytical was 98 dBA at 5500 rpm. So the obtained results matched up to 2% accuracy.

Fig 16 Experimental result

CONCLUSION

It is possible to predict the pseudo sound power level of air intake manifold due to the combined effects of combustion loads, piston side load, bearing loads and fluid flow pressure at a time. Without any CFD software, both the fluid flow noise and mechanical noise predicted was predicted by using structural software.

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