Exhaust System Optimization of Passenger Car for Maximizing Fuel Efficiency through HyperWorks

Abbreviations: SUV – Sports Utility Vehicle, DOF- Degrees of Freedom, mm – Millimeter, rpm – revolution per second,

Keywords: Optimization, HyperStudy, Exhaust System

Abstract
In recent development, newer technologies are progressing towards enormous light weight systems with optimized performance. It is due to the consumer demand of higher fuel efficiency with superior performance. In-order to that, an optimization approach towards Exhaust system weight reduction is presented in this paper. Even though most of exhaust system parts are made of sheet metal the total system weighs up to 15-20 Kg in SUV's and much higher in Euro V vehicles. As per Indian market SUV system is analyzed, optimized and also it is validated using HyperWorks. The total weight reduction of more than 2Kg is achieved using Optistruct.

1. Introduction

Optimization technique based on the finite element method has become one of the global research interests in the field of automotive industry. The main objective is to examine how and when structural optimization process is utilized in the design hierarchy. Application of technology should be evaluated for possibilities and limitations to the areas of improvement in design and quality cycle.

During design phase of vehicle exhaust system, it is always challenging to optimize the design for achieving higher stiffness, strength and concurrently weight reduction. There have been various types of the optimization methods that were developed and have successfully been used in the vehicle exhaust design. Recently, in the course of designing vehicle exhaust systems, various optimization methods were attempted for optimal structural design, especially the pipe bends, thickness, material and hanger rods and flanges, etc. It is very important that these resonators, pipes and hanger rods should have proper stiffness and strength to provide high durability and NVH characteristics to a vehicle, while maintaining light weight.
1.1 Background - Sharda Motor Research & Development

SMIL has its Research and Development facility at MWC, Chennai in 2009. Our R & D capabilities include defines, design and develop exhaust components and integrate into full system. Our product development capabilities include computational modeling, virtual testing, and key life validation tests. Computational Fluid Dynamics (CFD), Finite Element Analysis (FEA), and Engine Simulation Modeling (WAVE) used up front to design, develop, and validate exhaust system for Gasoline and Diesel operated vehicles. On-road, Off-road, and Non-road vehicle exhaust system will be developed and validated for global OEMs and engine manufacturers. This pro-active development methodology will provide comparative and competitive advantage to OEMs. Emission control systems such as Three Way Catalyst (TWC), Diesel Oxidation Catalyst (DOC), Diesel Particulate Filter (DPF), Lean NOx Trap (LNT), Selective Catalytic Reduction of NOx (SCR) and Hydro Carbon Trap (HC trap) are being developed. Modular exhaust system development includes tailpipe noise reduction and acoustic optimization. SMIL R & D is also setting up the design validation facility in our Research & Development which includes engine dynamometer, hot vibration shaker, flow bench, transmission loss kit, salt spray chamber and anechoic room.

1.2 Objective

The main task is to investigate how and when structural optimization can be used at Sharda Motor Industries Ltd - R&D Center in the design process. This includes an evaluation of possibilities and limitations and to areas where this technology is applicable. The end result is a sensible methodology of when and how to perform an optimization using a commercial software suite. Guidelines for different aspects of the process like values of various parameters and options will also be determined.

The main purpose of this paper is to present an analytical optimization tool developed to assist design engineer to achieve Stiffness, Weight, and stress reduction in an exhaust system with the most potential and without any compromise in packaging and manufacturing. All the required steps will be presented in a Flowchart.

This program has the following functions:
  • Modal Frequency Analysis
  • Static Analysis

1.3 Scope and Limitations

The focus of this work is to develop a practical and robust method when using Topography and stress optimization in the design process and this is where most of the effort will be put. The theory and mathematics behind the different optimization methods will not be investigated in any greater detail. Boundary conditions, loads and material parameters are considered to be known, the derivation of them are beyond the scope of this work.

2. Process Methodology

What if” studies to understand the effects of changing geometrical parameters or to change a design parameter to avert failure and improve the product design is essentially known as design optimization. Design optimization provides a robust and systematic methodology by carefully studying the effects of various design variables and improves the design by varying the design variables.
1. Stiffness values of the isolators
2. Hanger rods
   a. Type
   b. Location
   c. Geometry
3. Pipe
   a. Bend angle
   b. Material
   c. Thickness

2.1 Inputs
- CAD model of Exhaust system for BS III Emission regulation
- Material data for each components (BOM data)

2.2 Engine Details
- No of cylinders = 4
- Displacement = 2498 cc
- Maximum speed = 3800 rpm
- Idle speed = 800 rpm
- Natural frequency from Idle speed = 800/60 x 4/2 = 26.67 Hz
- Natural frequency from Engine maximum speed = 3800/60 x 4/2 = 126.67 Hz

2.3 Optimization Flow Chart

Step 1: Existing Design Analysis as per SMIL Standard
   (Stress, Weight, Stiffness values, Natural frequency & Static balance)

Step 2: Optimizing parameters
   (Stiffness values, Hanger rods, Pipe)

Step 3: Hyper study – Using Optimized parameters
   (Stress, Weight, Stiffness values, Natural frequency & Static balance)

Step 4: Reanalysis the optimized model as per SMIL Standard & Comparison
   (Stiffness target, Static durability, Resonance condition)

Step 5: Existing and optimized proposal comparison
   (Stress target, Weight, Natural frequency estimation, 1G and 4G Durability target)

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2.4 Base Model

Figure 1. Base Model nSUV Exhaust System

2.5 FE Model Description & Assumptions

- FE Model developed with 4 noded Quad Shell elements for most of the components.
- Weld between the components is modeled with one row of solid elements and penta elements in order to simulate the weld region.
- The connection between flanges (bolt holes) is modeled with rigid elements.
- Flexible bellows and rubber isolators are modeled with Bush element i.e. CBUSH in Nastran and stiffness values are defined as per reference data.
- Dynamic stiffness of flexible coupling has been assumed as per reference data
  - $K_1 = 5.98$ N/mm, $K_2 = 1.8$ N/mm, $K_3 = 1.8$ N/mm,
  - $K_4 = 4.5 \times 10^6$ Nmm/rad, $K_5 = 15738$ Nmm/rad, $K_6 = 15738$ Nmm/rad
- For Rubber isolators as below,
  - Hanger 1, $K_1 = xxx$ N/mm, $K_2 = xxx$ N/mm, $K_3 = xxx$ N/mm,
  - Hanger 2 & 3, $K_1 = xxx$ N/mm, $K_2 = xxx$ N/mm, $K_3 = xxx$ N/mm,
  - Hanger 4, $K_1 = xxx$ N/mm, $K_2 = xxx$ N/mm, $K_3 = xxx$ N/mm
- Mass of catalytic converter inner components has been assumed as 0.5 kg and modeled as CONM2 with RBE3 element.
Mass of flexible bellow has been assumed as 600 gm and modeled with CONM2 as half mass at inlet and outlet with RBE2 rigid elements.

Figure 2. Boundary Condition

2.6 Modal Analysis

To determine the natural frequency and mode shapes for Full Exhaust system and also to find the critical locations of failure under given boundary conditions

<table>
<thead>
<tr>
<th>Modes</th>
<th>Description</th>
<th>Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1&lt;sup&gt;st&lt;/sup&gt;</td>
<td>Exhaust system translation about Y-axis</td>
<td>3.83</td>
</tr>
<tr>
<td>3&lt;sup&gt;rd&lt;/sup&gt;</td>
<td>Exhaust system translation about Y-axis</td>
<td>4.87</td>
</tr>
<tr>
<td>4&lt;sup&gt;th&lt;/sup&gt;</td>
<td>Vertical bending of exhaust system</td>
<td>7.68</td>
</tr>
<tr>
<td>5&lt;sup&gt;th&lt;/sup&gt;</td>
<td>Vertical bending of exhaust system</td>
<td>9.96</td>
</tr>
<tr>
<td>6&lt;sup&gt;th&lt;/sup&gt;</td>
<td>Vertical bending of exhaust system</td>
<td>20.43</td>
</tr>
<tr>
<td>7&lt;sup&gt;th&lt;/sup&gt;</td>
<td>Exhaust system translation about Y-axis</td>
<td>22.09</td>
</tr>
<tr>
<td>8&lt;sup&gt;th&lt;/sup&gt;</td>
<td>Rotation of exhaust system about X-axis</td>
<td>28.30</td>
</tr>
<tr>
<td>10&lt;sup&gt;th&lt;/sup&gt;</td>
<td>Vertical bending of exhaust system</td>
<td>57.05</td>
</tr>
<tr>
<td>12&lt;sup&gt;th&lt;/sup&gt;</td>
<td>Rotation of exhaust system about X-axis</td>
<td>97.83</td>
</tr>
<tr>
<td>14&lt;sup&gt;th&lt;/sup&gt;</td>
<td>Vertical bending mode of catalytic converter</td>
<td>120</td>
</tr>
<tr>
<td>16&lt;sup&gt;th&lt;/sup&gt;</td>
<td>Vertical bending of exhaust system</td>
<td>151.7</td>
</tr>
<tr>
<td>17&lt;sup&gt;th&lt;/sup&gt;</td>
<td>Rotation of exhaust system about X-axis</td>
<td>192.2</td>
</tr>
</tbody>
</table>
2.6.1 Results and Observations

- It has been observed from modal analysis that the natural frequency obtained are not matching with Idle and Engine excitation frequency.
- Vertical bending modes are observed on modes 4, 5, 6, 10, and 16.
- Translation modes in Y-axis are observed in modes 1, 3 and 7.
- Exhaust system rotation about X-axis are observed in modes 8 and 12.
- It has been concluded that resonance will not occur because natural frequencies from engine and exhaust system are not coinciding.
- The critical locations of failure to be identified by referring strain energy plots and further dynamic analysis has to be done as per assumed loading condition.
- The frequency may vary based on modification of thickness of pipes and baffle plates.

2.7 Static 1G Analysis

To determine the displacement, stress levels and hanger forces for the applied static load of 1G gravity load in vertical direction of exhaust system i.e. Z-axis. For boundary condition, use figure 2 as reference.
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Figure 4. Loading Condition

[Diagram showing a 1G (9810 mm/s²) load in the Z-direction]

Figure 5. Maximum System Displacement of 5.06mm

[Diagram showing maximum displacement of 5.06 mm in a pipe after flexible bellows]

Figure 6. Hanger Forces and Hanger Displacements

[Diagram showing hanger forces and displacements: Hanger 1 - HD 3.8mm, HF 26.51N; Hanger 2 - HD 2.86mm, HF 49N; Hanger 3 - HD 2.56mm, HF 45N; Hanger 4 - HD 1.5mm, HF 8.4N]
2.7.1 Results and Observations

- It has been observed that Von-mises stress is within the material yield stress limit for the applied static gravity load 1g in vertical direction, i.e. Z-direction.
- The maximum displacement level for the whole model is observed in pipe after flexible coupling and muffler inlet pipe as 5.06 mm.
- Hanger forces on muffler hanger rods (Hanger 2 & 3) are higher when compared to muffler inlet pipe hanger rod (Hanger 1) and tail pipe hanger rod (Hanger 4).
- Hanger displacement is maximum on muffler inlet pipe and then on hanger 2, 3 and tail pipe.
- It has been concluded that hanger forces are within the specified limit as < 50 N and hanger displacement are less than specified limit of 5 mm as per reference data.

2.8 Hanger Stiffness Analysis

To determine the stiffness values in terms of frequency for all hanger rods.

Boundary condition: Weld regions of the hanger rod are constrained in all degrees of freedom.

Target Criteria: > 500Hz in vertical direction

2.8.1 Results and Observations

![Hanger Rod Mode Shape Plots](image)

- All hanger rods meets the target criteria
- Considering weight optimization, all solid hangers can be converted to hollow types
- Need to confirm resonance frequency of the system, hanger displacement of the system and road impact durability test (static 4G)
2.9 Static 4G Analysis

To determine the stress levels and bending moment for the applied static load of 4G gravity load in vertical direction of exhaust system i.e. Z-axis. For boundary condition, use figure 2 as reference.

![4G (39240 mm²) in Z-direction](image)

Figure 8. Loading Condition

2.9.1 Results & Observations

- It has been observed that Von-mises stress are within the material yield stress limit (100MPa) for the applied static gravity load 4G in vertical direction, i.e. Z-direction.
- From stress values, it is clearly evident that the utilization of pipe’s material and thickness is significant less.
- Moreover, optimization of pipe diameter will also be considered after confirming the flow and back pressure analysis for structural durability.

2.10 Optimized Model

After studying through the base model, optimizing parameters are clearly evident.
They are
- Isolator stiffness
- Hanger rods
- Pipe: Material, thickness and diameter
- Flange

2.10.1 Isolator Stiffness

For base model, three different configurations of isolators are used for same hanger rod diameter. In order to avoid confusion and utilizing isolator to its fullest potential, we have tried to optimize with single isolator to check its performance.

<table>
<thead>
<tr>
<th>Description</th>
<th>Hanger 1</th>
<th>Hanger 2&amp;3</th>
<th>Hanger 4</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Base Model</strong></td>
<td>![Base Model 1]</td>
<td>![Base Model 2]</td>
<td>![Base Model 3]</td>
</tr>
<tr>
<td><strong>Stiffness values</strong></td>
<td>$K_x = xxx \text{ N/mm}$</td>
<td>$K_x = xxx \text{ N/mm}$</td>
<td>$K_x = xxx \text{ N/mm}$</td>
</tr>
<tr>
<td></td>
<td>$K_y = xxx \text{ N/mm}$</td>
<td>$K_y = xxx \text{ N/mm}$</td>
<td>$K_y = xxx \text{ N/mm}$</td>
</tr>
<tr>
<td></td>
<td>$K_z = xxx \text{ N/mm}$</td>
<td>$K_z = xxx \text{ N/mm}$</td>
<td>$K_z = xxx \text{ N/mm}$</td>
</tr>
<tr>
<td><strong>Optimized model</strong></td>
<td>![Optimized Model 1]</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Stiffness values</strong></td>
<td>$K_x = xxx \text{ N/mm}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>$K_y = xxx \text{ N/mm}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>$K_z = xxx \text{ N/mm}$</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*Note: Confidential data*

![Figure 10. Isolator Configuration](image)

2.10.2 Hanger Rods

For base model, the main intent is to reduce the stress and weight of the hanger rods. As such, we have tried to change the hanger rods from solid to hollow. If this doesn’t suffix, geometrical changes are required to meet the target.
### 2.10.3 Flange

In order to optimize from solid flange of 9.5mm thickness to 3.2 mm thickness available flange, we have to study the strength of the sheet metal flange connecting two exhaust pipes. We need to check the structural strength of the solid flange under static loading condition which is yet to be confirmed by RLDA analysis.

<table>
<thead>
<tr>
<th>Analysis</th>
<th>Hangers</th>
<th>Base Model</th>
<th>Optimized Model</th>
<th>Result</th>
<th>Target</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hangar Stiffness Frequency</td>
<td>Hanger 1</td>
<td>545 Hz</td>
<td>630 Hz</td>
<td>Ok</td>
<td>&gt; 500 Hz</td>
</tr>
<tr>
<td></td>
<td>Hanger 2</td>
<td>1049 Hz</td>
<td>1236 Hz</td>
<td>Ok</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Hanger 3</td>
<td>1049 Hz</td>
<td>1236 Hz</td>
<td>Ok</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Hanger 4</td>
<td>729 Hz</td>
<td>865 Hz</td>
<td>Ok</td>
<td></td>
</tr>
</tbody>
</table>

*Figure 11. Stiffness Comparison: Base Model Vs Optimized Model*

*Figure 12. Loading Condition*
The flange study has been done for the loading condition as 500 N vertical loads on both pipe ends and 1G gravity load.

- The stresses are crossing the material yield or UTS limit for the applied load of 500 N.
- The stresses are well below the yield limit for gravity load.
- Flange optimization has to be done based on design target and design variable.

### 2.10.4 Pipe

- From virtual analysis and benchmarking similar vehicles, we have observed pipe thickness, diameter and material can be optimized. In order to change these parameters, back pressure and flow analysis is required.
- From Flow analysis prediction: Reducing the intermediate pipe diameter from 55.56 to 50.8 mm increase the pressure drop from 10.8 mbar to 20.66 mbar. It is within the target, so we can reduce the pipe diameter.
- Reducing the tail pipe diameter from 55.56 mm to 50.8 mm increase the pressure drop to increase from 25.76 mbar to 38.78 mbar which is within the target.
- The tail pipe noise has to be checked since the velocity is increasing in the tail pipe.
- Comparison of targets for existing and optimized model are shown below

#### TABLE 2
**BACK PRESSURE COMPARISON: BASE MODEL VS OPTIMIZED MODEL**

<table>
<thead>
<tr>
<th>S.No</th>
<th>Criteria</th>
<th>Target achieved in Existing Model</th>
<th>Target achieved in Optimized Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Back Pressure</td>
<td>209 mbar</td>
<td>231.88 mbar</td>
</tr>
</tbody>
</table>

*Figure 13. Von Mises Stress Plots*
So changing the pipe diameter to the desired one has no issue with flow analysis, but clarification is required for its structural strength.

### TABLE 3
**MATERIAL & PIPE PROPERTIES COMPARISON: BASE MODEL VS OPTIMIZED MODEL**

<table>
<thead>
<tr>
<th>Component</th>
<th>Material</th>
<th>Existing Diameter/ Thickness (mm)</th>
<th>Material</th>
<th>Optimized Diameter/ Thickness(mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet flange</td>
<td>FE410</td>
<td>9.5 mm Thk</td>
<td>FE410</td>
<td>9.5 mm Thk</td>
</tr>
<tr>
<td>Catcon Inlet pipe</td>
<td>Stainless Steel</td>
<td>1.5 mm</td>
<td>Stainless Steel</td>
<td>1.5 mm</td>
</tr>
<tr>
<td>Catcon inlet and outlet cone</td>
<td>Stainless Steel</td>
<td>2 mm</td>
<td>Stainless Steel</td>
<td>1.5 mm</td>
</tr>
<tr>
<td>Catcon shell</td>
<td>Stainless Steel</td>
<td>1.5 mm</td>
<td>Stainless Steel</td>
<td>1.5 mm</td>
</tr>
<tr>
<td>Catcon outlet pipe</td>
<td>Stainless Steel</td>
<td>1.5 mm</td>
<td>Stainless Steel</td>
<td>1.5 mm</td>
</tr>
<tr>
<td>Pipe 1</td>
<td>Ф55.56 &amp; 1.5 mm</td>
<td>Ф50.8 &amp; 1.5 mm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pipe 2, Pipe 3</td>
<td>Ф55.56 &amp; 1.5mm</td>
<td>ACMS</td>
<td>Ф50.8 &amp; 1.2mm</td>
<td></td>
</tr>
<tr>
<td>Intermediate pipe flange, muffler inlet flange</td>
<td>FE410</td>
<td>9.5 mm (Solid)</td>
<td>Stainless Steel</td>
<td>3.2 mm (Sheet metal)</td>
</tr>
<tr>
<td>Muffler Inlet pipe</td>
<td>Ф55.56 &amp; 1.6 mm</td>
<td>ACMS</td>
<td>Ф50.8 &amp;1.2 mm</td>
<td></td>
</tr>
<tr>
<td>Muffler inlet connection pipe</td>
<td>Ф55.56 &amp; 1.2 mm</td>
<td>Ф50.8 &amp; 1.2 mm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Muffler end plates A &amp; B</td>
<td>Ф48.6 &amp; 1.6 mm</td>
<td>Ф48.6 &amp; 1.2mm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Baffle plates A &amp; B</td>
<td>Stainless Steel</td>
<td>1.2 mm &amp; 1.2 mm</td>
<td>ACMS</td>
<td>1 mm &amp; 1.2 mm</td>
</tr>
<tr>
<td>Perforated tube 1,2,3 &amp; perforation region</td>
<td>Stainless Steel</td>
<td>1.2 mm &amp; 0.9 mm</td>
<td>ACMS</td>
<td>1 &amp; 0.8 mm</td>
</tr>
<tr>
<td>Muffler outlet pipe</td>
<td>Ф55.56 &amp; 1.6 mm</td>
<td>Ф50.8 &amp;1.2 mm</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
| Muffler shell, Tail pipe | Ф55.56 & 1.2 mm | Ф50.8 &1.2 mm | Simulate to Innovate
2.11 Performance Validation

Resonance frequency, Static 1G and Static 4G analysis will determine the natural frequency displacement, stress levels and hanger forces for the optimized model and help us compare with the base model.

2.11.1 Modal Analysis

<table>
<thead>
<tr>
<th>Modes</th>
<th>Result modal frequency from FEA (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Base Model</td>
</tr>
<tr>
<td>1\textsuperscript{st}</td>
<td>3.83</td>
</tr>
<tr>
<td>3\textsuperscript{rd}</td>
<td>4.87</td>
</tr>
<tr>
<td>4\textsuperscript{th}</td>
<td>7.68</td>
</tr>
<tr>
<td>5\textsuperscript{th}</td>
<td>9.96</td>
</tr>
<tr>
<td>6\textsuperscript{th}</td>
<td>20.43</td>
</tr>
<tr>
<td>8\textsuperscript{th}</td>
<td>28.30</td>
</tr>
</tbody>
</table>
2.11.2 Static 1G Analysis

<table>
<thead>
<tr>
<th></th>
<th>Base Model</th>
<th>Optimized Model</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Force (N)</td>
<td>Displacement (mm)</td>
</tr>
<tr>
<td>Hanger 1</td>
<td>26.51</td>
<td>3.8</td>
</tr>
<tr>
<td>Hanger 2</td>
<td>50</td>
<td>2.86</td>
</tr>
<tr>
<td>Hanger 3</td>
<td>45</td>
<td>2.56</td>
</tr>
<tr>
<td>Hanger 4</td>
<td>8.4</td>
<td>1.5</td>
</tr>
</tbody>
</table>

2.11.3 Static 4G Analysis

![Figure 15. Von Mises Stress Plots](image)

Maximum Von-mises stress of 83.56MPa occurring in pipe 1
Maximum Von-mises stress of 73.32MPa occurring in pipe 5

3. Result Summary

- From Modal analysis, it has been observed from, the natural frequencies of the optimized model is not matching the engine excitation at idle and max speed. Moreover, there is clear difference in frequency range of 3 Hz from engine resonant frequency.
- It has been observed that Von-mises stress are within the material yield stress limit for the applied static gravity load 1G in vertical direction, i.e. Z-direction.
The maximum displacement level for the whole model is observed in pipe 1 and pipe 2 as 5.29 mm. Hanger forces on hanger rods 2 and 3 are higher when compared to muffler inlet pipe hanger rod (hanger 1) and tail pipe hanger (hanger 4). Hanger displacement is maximum on muffler inlet pipe hanger rod (hanger 1) and then on muffler hanger rod (hanger 2&3) and tail pipe. It has been concluded that hanger forces are within the specified limit as < 50 N and hanger displacement are less than specified limit of 5 mm as per reference data. From Static 4G analysis, that Von Mises stress are clearly increased when compared to base model, the utilization of change in pipe thickness, material and diameter are within the specified target limit. Total mass of the existing model as 19.0 kg and optimized as 16.1 kg after modifying the diameter of pipe 2 and tail pipe as 50.8mm. In the case of hanger forces and displacement for the optimized model, it is higher when compared to existing model but within the specified limit. Conclusive evidence will be annexed, if analysis like transient response with RLDA is performed.

4. Benefits Summary

The design, optimization and selection of full exhaust system are iterative processes. Due to the sensitivity of the application, it is generally tough to correctly predict the effect of a certain modification on the system. Hence, with the conventional methodology the productivity as well as accuracy of the results is compromised. With these optimization techniques, it gives more flexibility to the designer to choose the concept as per the design requirements of the exhaust assembly. The time spent on building, analyzing and reading results of redundant design iterations is effectively saved with the improved methodology. The new process is more of a top-down approach wherein all suggested features are added together in the beginning and then, features are removed based on aesthetics and manufacturability. This approach was found to be less time consuming. Further it has reduced the time required to arrive at the best design (Eliminated Intermediate Iterations), thus shortening the product design cycle time.

5. Challenges

The addition of pipe diameter for optimization would certainly help in reducing the cycle time for iteration process. Moreover, Addition of standard pipe thickness would effectively increase the result quality and comparison.

6. Future plans

We are planning to deploy and discretize advance methodology in Optistruct module in other projects to reduce weight and product design cycle time.

7. Conclusion

The combination of the simulated modeling and modified method of feasible parameter is powerful tool for the optimization of simple problem such as exhaust performance.

ACKNOWLEDGEMENTS

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References

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