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Analysis of Cooling Gas Compressors Discharge Silencer

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Abstract - *Discharge silencer are used with rotary compressors to avoid flow fluctuations of gas. In this paper an industrial problem of discharge silencer is discussed. A discharge silencer connected to a cooling gas compressor developed cracks at various locations on its body and supports. The vibration and noise levels of the system were very high. To find out the root causes of the high level of vibration measured on the discharge silencer, it was decided to carry out a CAE based investigation along with comparative study of the actual vibration on site measurements. Hyper-mesh with shell elements is used for meshing the various design models and modal analysis of the structure is done using Radioss as a solver. The response of the discharge silencer under dynamic loading condition through transient dynamic analysis is done using Ls-Dyna as a solver. Hyper-view is used for post-processing of the modal analysis results.*

Keywords – *Finite element modeling, Modal analysis, Transient analysis, Vibrations.*

I. Introduction

In many process plants discharge silencers are used in conjunction with rotary compressor to avoid flow fluctuations in the lines leading to process plant. This also aids controlling temperature and pressure in the plant. This discharge silencer is essentially for impedance matching of the downstream process flow to the upstream pulsating flow from the compressor. The discharge silencer is connected to the cooling gas rotary compressor. The compressor gas is consists of carbon mono-oxide with some impurities like water vapor, hydrogen, nitrogen, etc. and operational temperature is approximately 70^{o}C. The compressor runs at a speed of 520rpm. The compressor discharge pressure is around 1.2 to 1.9 bar, which is the inlet pressure of the discharge silencer.

The cooling gas compressors discharge silencer face problem of high vibration. The problem of cracks into the silencer developed because of these high vibrations. So in order to reduce the vibration and avoid cracks the study of modal parameters is very important. To get modal parameters modal analysis using Finite Element Analysis is best method, which discussed in this paper. With the help of transient analysis, the effect of the inlet pressure changing with respect to time is also discussed in this paper. [1]

II. Literature Review

Modal analysis for filament wound pressure vessels filled with fluid was done Z. Wu a, W. Zhou, H.Li. [2] This work done provides a preliminary research and information of the modal properties for a filament wound pressure vessel filled with and without fluid. Their work is aimed to contribute to the study on vibration-based damage detection and structural health monitoring. Due to the characteristic of fluid filled vessel, the effect of the invariant fluid to the dynamic response of the structure is limited.

The problem of free and forced vibrations of cylindrical shell submerged in acoustic medium was developed by Bleich and Baron which subsequently have been analyzed by Junger[3]. In these investigations, where isotropic shells were concerned, the effect of fluid on motion of cylinder shells has been described. Brown made a survey of the hydrodynamic response of fluid-coupled coaxial cylinders under small displacements.

A. Di Carluccio, G. Fabbrocino, E. Salzano, G. Manfredi[4] did analysis of pressurized horizontal vessels under seismic excitation. In this approach, displacements and rotations of the shell and dynamic pressure of the fluid are modeled by a hybrid finite element method. Pressures were primary loadings that were applied either internally or externally over the surface of the vessels; applied forces act either at local points or throughout the mass of the vessel. Dynamic analyses of cylindrical vessels designed according to National and International Codes are discussed and relevant outcomes for seismic protection of components are highlighted.

III. Finite Element Modeling Of Discharge Silencer:

The model in IGES format is imported in the hyper-mesh and meshing is done. The CGC discharge silencer has a single inlet duct and two outlet ducts as shown in Fig 1. The radial outlet duct is in closed position. The discharge silencer is divided into inlet and outlet chamber with the help of partition plate. The inlet and outlet chamber is connected with the help of transfer tubes.

The basic idea behind a finite element method or finite element analysis is a building of complicated object with simple blocks or dividing complicated object into small and manageable pieces. Finite element analysis has a great power, but when the analyst using this and similar methods the disadvantages of computer solutions must require to be kept in mind. The computer solutions do not know how the materials properties and geometrical features influence the stresses. Errors in input data by the analyst can produce wildly incorrect results. The meshed model of CGC discharge silencer is shown in the Fig 1. The meshed model of transfer tubes is shown in Fig 2.

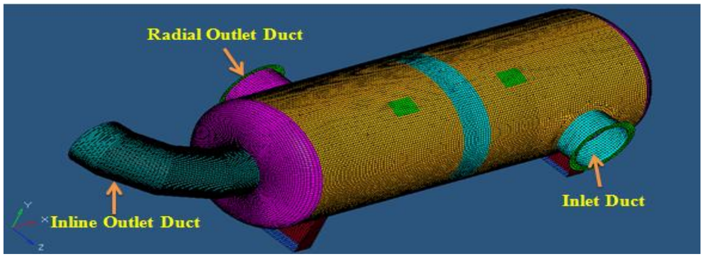


Fig 1: Meshed model of CGC discharge silencer.

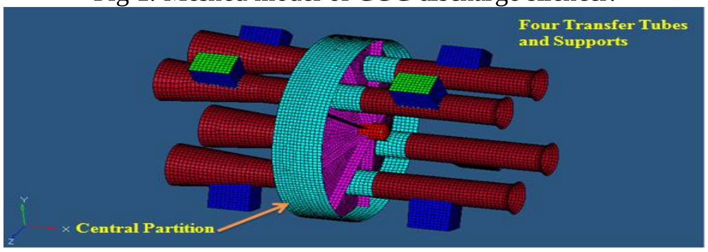


Fig 2: Meshed Model of Transfer Tubes.

IV. Finite Elements Analysis Of Discharge Silencer

Two types of analysis were carried out for the discharge silencer to see the behavior of the structure under loading condition.

a) Modal analysis
b) Transient analysis.

In order to perform any type of analysis the boundary conditions plays important role. The inaccurate modeling of the loads and restraints on a model is one of the most significant sources of errors in FE modeling. So giving accurate boundary conditions is very much important in any type of analysis.

4.1 Modal Analysis

The corresponding boundary conditions to perform modal analysis are given. For modal analysis only constrained are required. Fig 3 shows the applied boundary conditions to CGC discharge silencer. In X- direction sliding support is kept free about a 5mm. This free movement is for thermal expansion.

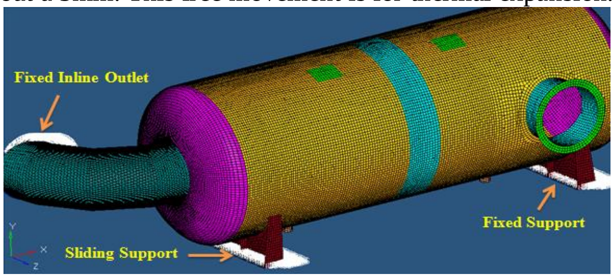


Fig 3: CGC Discharge Silencer Boundary Conditions.

4.2 Transient analysis

The response of the system under application of the load the transient dynamic analysis of the CGC discharge silencer is carried out for 1 sec. The compressor rotates at 520 rpm (8.5 Hz). As this was a twin lobe type compressor, for every rotation of the compressor, four pressure pulsations were available at the inlet. Thus, pressure pulsation at the inlet was of 34 Hz (4*8.5 Hz). Pressure is considered to be varying sinusoidal with mean value of 1.2 bar over amplitude of 360 mbar at 34 Hz. Pressure is applied on the exactly opposite face of the inlet on discharge silencer assuming that flow will strike at this region. The outputs at different locations of the discharge silencer are taken. Fig 4 shows the transient pressure curve for one second.

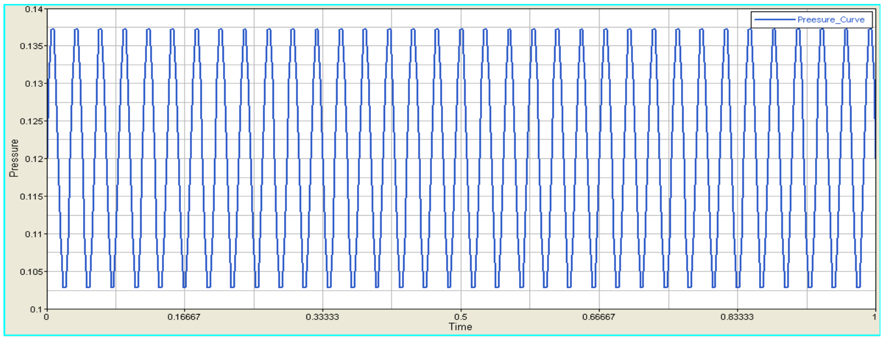


Fig 4: Transient Pressure Curve.

This pressure curve is applied on the opposite face of the inlet where the fluid flow will strike as shown in Fig 5. For transient dynamic analysis the support conditions are kept same as used for modal analysis. That is in X-direction sliding support is kept free about a 5mm. This free movement is for thermal expansion and the fixed support is constrained in all degrees of freedom.

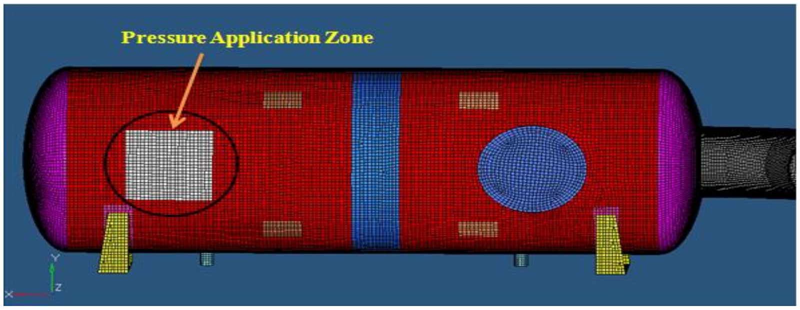


Fig 5: Transient Pressure Curve Application.

As the pressure is applied at reason opposite to inlet, so the pressure application zone is critical. The response of the structure is measured at pressure application zone as shown in Fig 6. Points P1, P2 and P3 shows the out measurement locations for transient dynamic analysis. The solution for dynamic transient analysis is obtained by using Ls-Dyna as a solver. The output for the analysis is in the form of velocity peaks verses time.

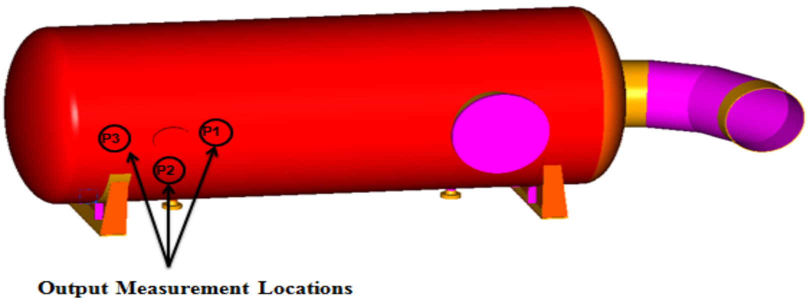


Fig 6: Output Measurement Points.

V. Results And Discussion

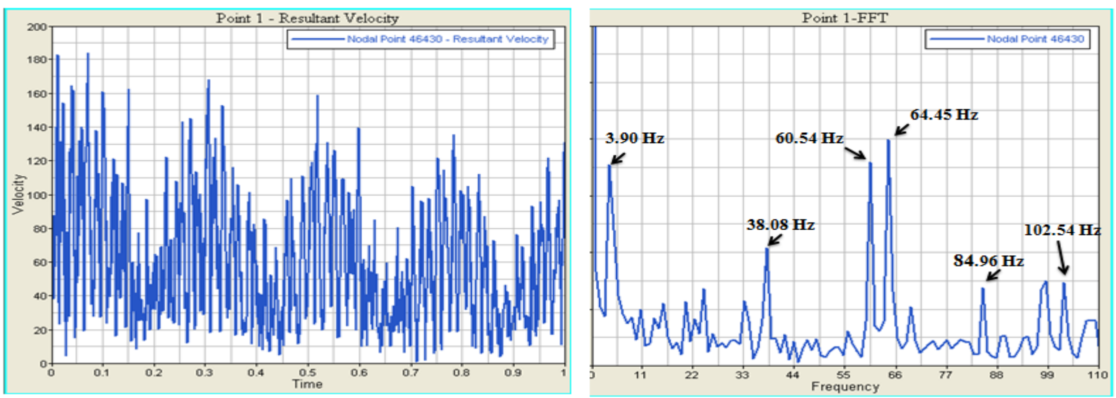
Following table shows the values of first twenty mode shape frequencies obtained from modal analysis.

Table 1: CGC Discharge Silencer Mode Shapes Frequency

Mode Number	Frequency (Hz)	Mode Number	Frequency (Hz)
1	28.55	11	80.4
2	46.70	12	85.06
3	53.52	13	87.29
4	54.64	14	92.05
5	56.01	15	94.49
6	63.36	16	97.25
7	64.04	17	97.72
8	68.51	18	100.69
9	72.77	19	104.38
10	78.12	20	105.86

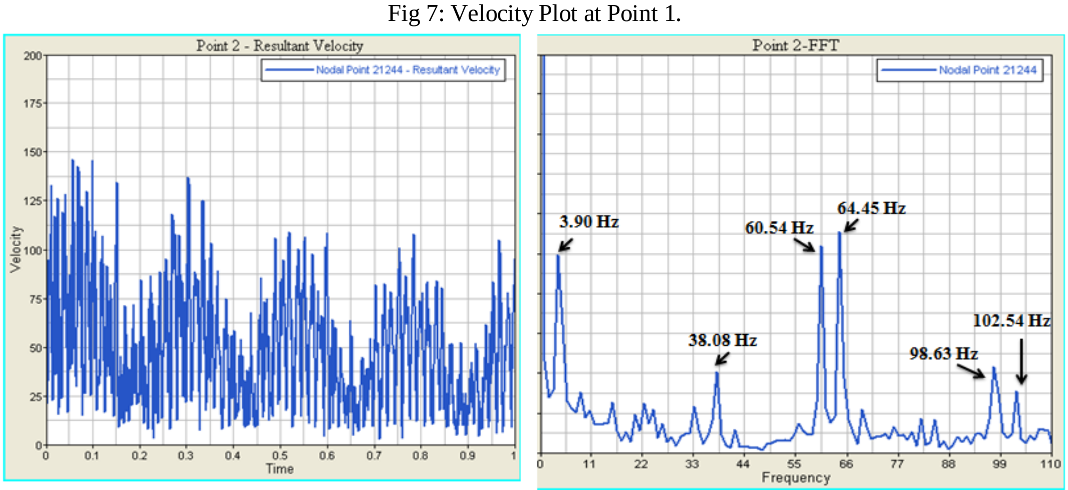
As shown in Table 1 the frequency range for first twenty modes of the cooling gas compressor discharge silencer is 28 Hz to 106 Hz.

The dynamic transient analysis output is shown in Fig 7 to Fig 9. The output obtained is in time domain, so to convert time domain data into frequency domain FFT is done. Conversion of the data from time domain to frequency domain is done by using hyper-graph tool. The resultant velocity is plotted on Y-axis. The time and frequency is plotted on an X-axis.



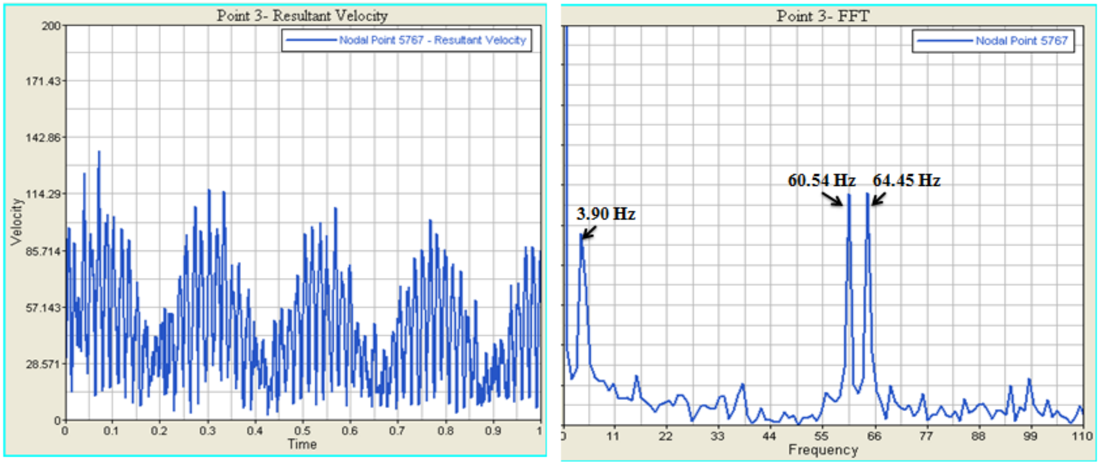
a) Point 1- Resultant Velocity **b) Point 1- FFT**

Fig 7: Velocity Plot at Point 1.



a) Point 2- Resultant Velocity **b) Point 2- FFT**

Fig 8: Velocity Plot at Point 2.



a) Point 3 – Resultant Velocity **b) Point 3 – FFT**

Fig 9: Velocity Plot at Point 3.

As shown in Table 1 and Fig 7 to Fig 9 the frequencies such as 60.54 Hz, 64.45 Hz, 84.96 Hz, 98.63 Hz, and 102.54 Hz where velocity peaks found are matching with the mode shape frequencies of the discharge silencer. The multiple of excitation frequency is also matching with normal mode frequency; hence resonance is created which causes high level of vibration. While doing the transient analysis critical assumption is made regarding the damping factor. As the damping factor of the structure changes with the change in natural frequency, therefore for transient analysis the value for damping factor needs to be considered. The fluid damping is also present; hence application of the exact damping factor is difficult. In this analysis the interest is in the frequency values where the velocity peaks are present. The application of damping factor just changes the value of velocity peaks, but it does not affect on the value of the frequencies where the velocity peaks are observed. Therefore assumption of any value of damping factor does not make any change in the frequency values at which velocity peaks present.

VI. Experimental Verification Of Results

In order to see the behavior of the discharge silencer under dynamic loading, the spectral peaks measurement is done on site. The compressor running at a speed of 520 rpm, the pressure of fluid inside silencer was 1.2 bar. It varies between amplitude of 360 mbar. The flow of fluid inside silencer was 60000Nm^3/hr to 110000Nm^3/hr. The velocity peaks measured at those locations where the velocity peaks are obtained in transient dynamic analysis. The peaks are shown in Fig 10 and Fig 12. Velocity peaks were measured using CSI 2130 Machinery Health Analyzer.

Fig 10 shows frequency verses RMS velocity plot at point 1. The velocity peaks were observed at 104 Hz. The maximum amplitude of RMS velocity is 69.1 mm/sec. Fig 11 shows frequency verses RMS velocity plot at point 2. The velocity peaks were observed at 104 Hz. The maximum amplitude for RMS velocity is 97.6mm/sec. The frequency verses RMS velocity plot for point 3 is shown in Fig 12. The maximum amplitude of 16.5 mm/sec was observed at peak frequency of 104.1 Hz.

Experimental results are shown in following Fig 10 to Fig 12. These Fig shows that the velocity peaks are at a frequency of 34.70 Hz, 86.74 Hz, 95.44 Hz, and 104.10 Hz. The frequency at which velocity peaks observed during experiment matches with the frequencies at which velocity peaks get obtained by simulation.

Table 2: Comparison of Peak Frequencies

Sr. No.	Simulated Peak Frequencies (Hz)	Experimental Peak Frequencies (Hz)	Percentage Error
1	38.08	34.70	9.74
2	84.96	86.74	2.05
3	98.63	95.44	3.34
4	102.54	104.10	1.49

Table 2 shows the comparison between peak simulated peak frequency and experimental peak frequencies. The peak frequencies observed in both methods are matching as shown in table 2.

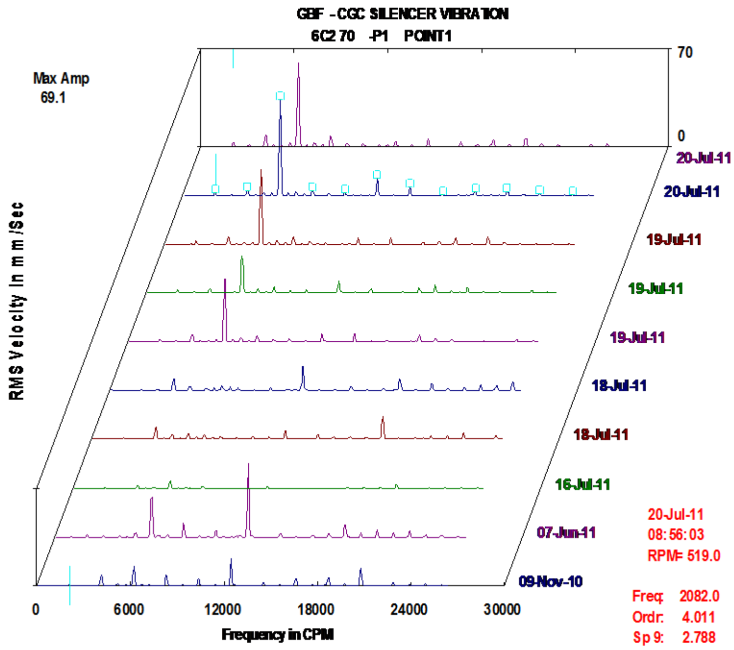


Fig 10: Frequency Vs RMS Velocity (Point 1).

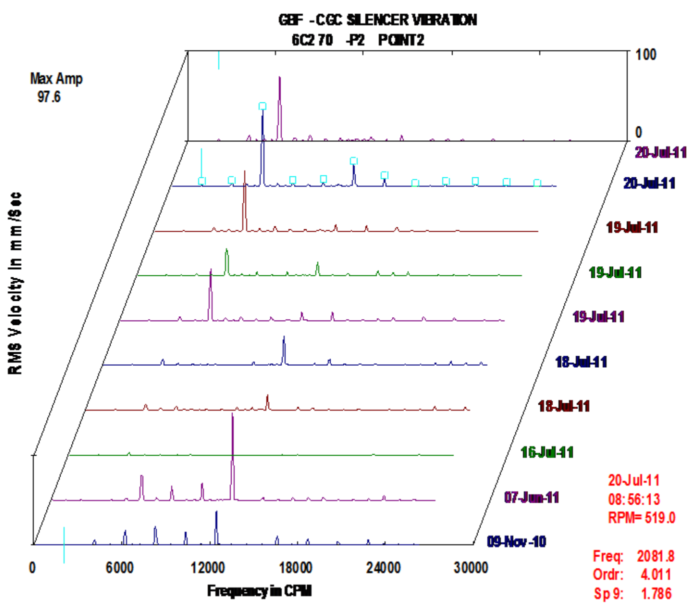


Fig 11: Frequency Vs RMS Velocity (Point 2).

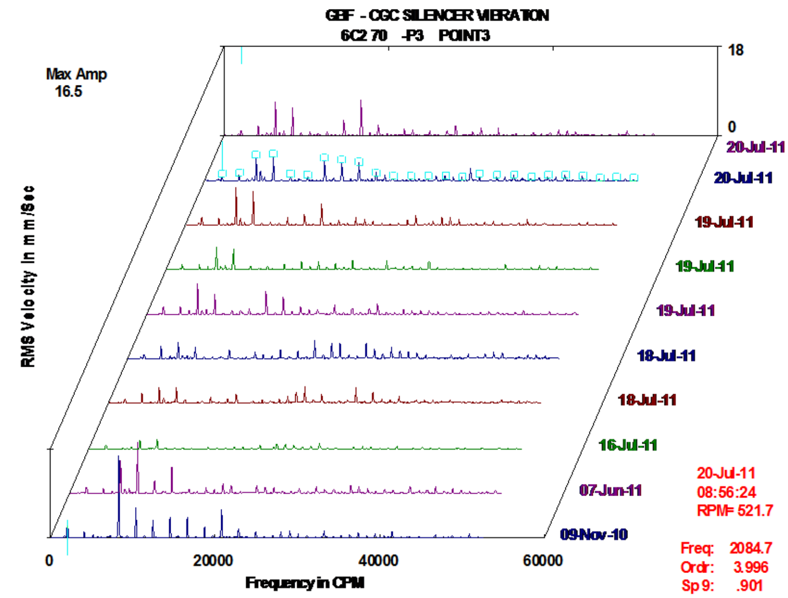


Fig 12: Frequency Vs RMS Velocity (Point 3).

VII. Conclusion

The design of cooling gas compressors discharge silencer is analyzed. The purpose of analysis was to verify the vibrations experienced in each design were a result of fluid flow inside the silencer. When modal results of the design matched with experimental results in an attempt to correlate the analytical results with experimentally obtained results. Some of the modal frequencies were matching with fluid intake frequency. Hence it was concluded that the vibrations were essentially result of fluid structure interaction. Transient response analysis of this design was carried out and have fair amount of correlation with experimental results.

Future scope of this work is the complete design of the discharge silencer will be carried by using ASME code of pressure vessels. Then carry out dynamic analysis of newly designed discharge silencer to see its behavior under excitation.

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