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Pipeline Vibration Reduction in Reciprocating Compressors

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ABSTRACT: Pipeline vibration due to reciprocating compressors is a major problem faced by Oil and Gas industries worldwide. We present a method to reduce the pipeline vibration of 2HD/1 opposed-balanced reciprocating compressor. Vibrational readings are recorded for two of the reciprocating compressors in the plant (ONGC-Uran); Compressor A with major vibration in discharge pipeline and compressor B with vibration within acceptable limits so as to find the source of vibration in compressor A. As per the layout, pipeline design of reciprocating compressor A is done in SolidWorks software. Vibrational analysis of the pipeline by varying the external support locations is carried out using Ansys software. From the study, it is understood that the pressure pulsation in pulsation dampener is the primary reason for pipeline vibration. A suggestion for the vibration reduction is made by adding supports to alter the frequency of the pipeline. We conclude that for overall system performance and vibration reduction, anti-vibration measures as suggested should be implemented.

KEYWORDS: Vibration, Ansys, Harmonics, Pulsations, Frequency, SolidWorks

1 INTRODUCTION

Reciprocating compressor (RC) is the most widely used type of compressor found in industrial applications and is a crucial machine in gas transmission pipelines, petrochemical plants, refineries, etc. due to a high pressure ratio achievement [1]. A reciprocating compressor is a positive displacement compressor that uses pistons driven by a crankshaft to deliver gases at high pressure. The intake gas enters the suction manifold, then flows into the compression cylinder where it gets compressed by a piston driven in a reciprocating motion via a crankshaft, and is then discharged. They can be either stationary or portable, can be single or multi-staged, and can be driven by electric motors or internal combustion engines [2-4]. Reciprocating compressors are capable of giving large pressure ratios but the mass handling capacity is limited or small. Reciprocating compressors may also be single acting compressor or double acting compressor. Single acting compressor has one delivery stroke per revolution while in double acting there are two delivery strokes per revolution of crank shaft [5].

2 BACKGROUND THEORY

When gases are compressed by any of the usual methods now employed commercially it is frequently found that several undesirable effects of pressure variation appear in the system associated with the compressor or in the compressor itself. All of these undesirable effects are connected in some manner with pulsation phenomena which appear as a result of the reciprocating action of the compressor. In certain systems where rotary or other types of compressors are employed, as contrasted with the reciprocating type, similar pulsation phenomena are evident. However, the present discussion will be confined to the phenomena which appear in systems utilizing reciprocating compressors, and in particular those which are employed in the compression of lean gas for transmission in pipelines, or for other purposes such as recycling or re-pressuring operations [6]. Reciprocating compressors emit pulsations by virtue of their design. Pulsations travelling away from and to the compressor cylinders will set up standing wave patterns that result in unbalanced pressure forces in the piping system. These unbalanced forces can result in extreme levels of vibration on the compressor and associated piping [7].

2.1 SOURCE OF PULSATIONS

Reciprocating compressors generate flow modulations that in turn generate pressure pulsations. The flow modulations come about as a result of intermittent flow through the suction and discharge valves, as well as geometry effects due to the (finite) length of the connecting rod.

Figure 1 shows a schematic of a compressor cylinder. The suction flow (QS) enters the cylinder, and the discharge flow (QD) exits the cylinder. The velocity of the piston, shown in Figure 1, is approximately sinusoidal in shape. The deviation of the actual piston motion from the sinusoidal shape is due to the finite length of the connecting rod. As the ratio of the connecting rod length to the crank radius (L/R) is increased, the shape becomes more closely sinusoidal. The pressure pulsation generated by the compressor is proportional to the flow (QS or QD) modulation [8, 9].

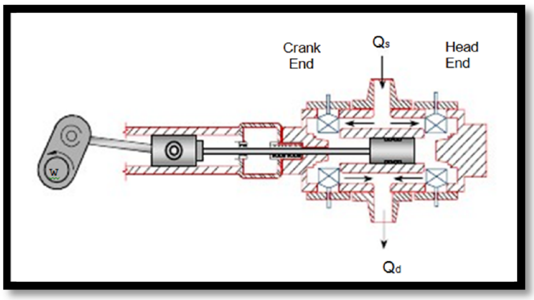


Fig 1: Double acting reciprocating compressor cylinder [9]

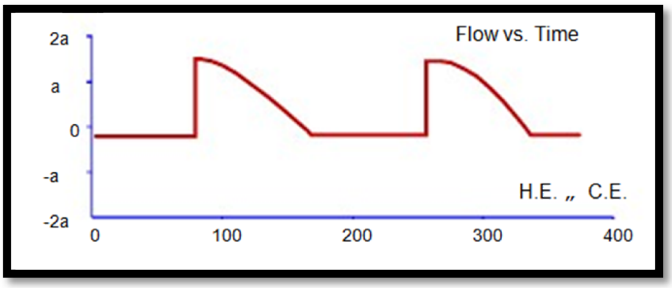
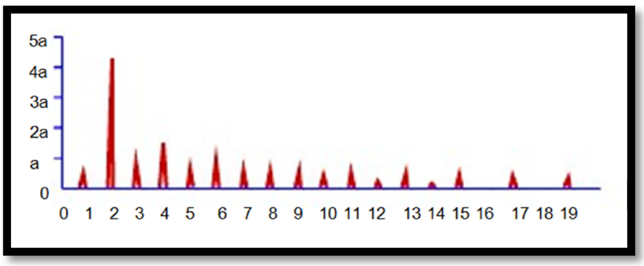


Fig 2: Unsymmetrical double acting compressor cylinder [9]

For a "perfect" double acting cylinder (symmetrical head end and crank end flows), the flow versus time contains two identical flow "slugs" 180 degrees apart in time. Therefore, the odd harmonics (in this idealized case) cancel, so that the nonzero cylinder flow excitation occurs at even harmonics of running speed (2, 4,…). Actual cylinders have piston rods, differences in head end/crank end clearance volumes and finite length connecting rods, so that the two "flow slugs" generated each revolution are not identical (Figure 2). Therefore, even in double acting operation, the cylinder will, in general, produce flow excitation at all harmonics of running speed as shown in Figure 3. These flow harmonics act as excitations to the piping acoustics, and the acoustic resonances of the piping will amplify pulsation at particular frequencies [9, 10].

Flow Amplitude



Harmonics

Fig 3: Flow frequency spectrum for double acting cylinder [9]
Where "a" is a positive integer different for different compressors

3 DESIGN OF PULSATION DAMPENER AND PIPLINE SYSTEMS

The LPG Unit of ONGC Uran Plant consists of four reciprocating compressors two in each unit that is LPG Unit-1 and LPG Unit-2. In LPG Unit-2, reciprocating compressor (B) is working completely fine without any vibration in its pipeline system contrary to the reciprocating compressor (A) which shows visible and dangerous level of vibration when its switched on. The compressor (A) is therefore is not used and the load is completely is taken by compressor (B) without any stand by. The input parameters and conditions are same for both the reciprocating compressors with almost similar external support locations in pipeline systems.

Vibrational readings were taken at different locations of both the compressors to find out the source of maximum vibrations. The measurements were taken using Emerson's CSI 2130 Health Analyzer kit. The readings were compared and it was observed that the major vibrations were observed in the pipeline systems. Figure 4 and 5 shows the location of major vibrations (marked in red).

Fig 4: Points of heavy vibration (on-site location)

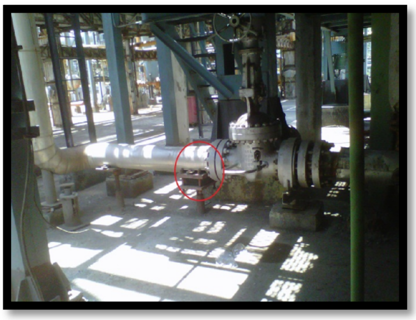


Fig 5: Points of heavy vibration (on-site location)

As the problem was seen in the pipeline systems, the study was conducted from the pulsation dampener. The design of the pipe layout was made according to the dimensions measured on-site. Both the designs are made in SolidWorks software.

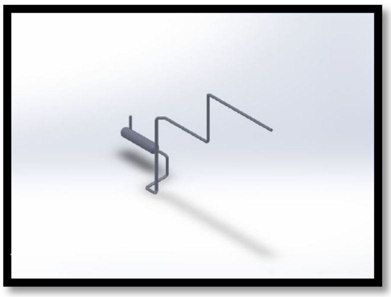


Fig 6: pulsation dampener and pipline

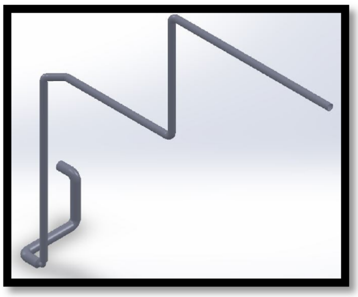


Fig 7: Pipline layout

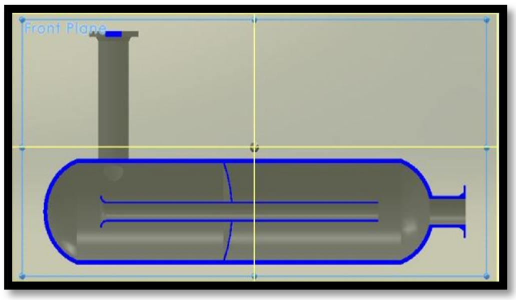


Fig 8: Sectional view- Pulsation dampener

Figure 6 shows the pulsation dampener and the complete piping layout designed in SolidWorks software. The pipeline layout consists of 8'' diameter pipe as well as 6'' diameter pipes which are connected with the help of a reducer as shown in Figure 7. The sectional view of the pulsation dampener can be clearly visible from figure 8. It consists of the baffle plate which divides the dampener into two chambers. The choke pipe is fixed symmetrically to allow the flow of gases between the chambers. Inlet of gas flow is little offset to provide enough time for the pulsation to die down.

4 MODEL ANALYSIS

The problem of vibration in pipeline is solved by providing the external supports to the discharge pipelines of the compressor. The principle of shifting the mechanical natural frequency of pipe line away from the harmonics of forcing frequencies is being used to solve the problem.

Formulae

Pulsating frequency (f_p) = M*(N/60) Hz [

- N = RPM of Motor = 327
- M = 1 for single acting
  = 2 for double acting [9].

The pulsating frequency is calculated to be 10.9 Hz and has been used for the analysis to find the accurate support locations.

Input parameters

Element Type = Solid 185
Young's Modulus of Elasticity = 111GPa
Poisson's ratio = 0.26
Density = 7.2 g/cm^3
Mesh type = Hexahedral

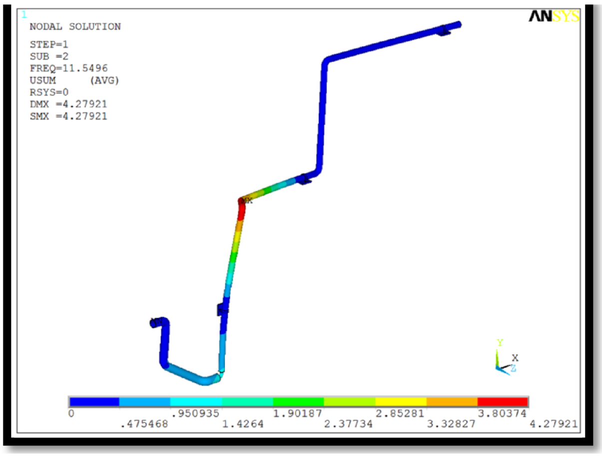


Fig 9: Current support location model analysis

The natural frequency of the current pipeline system was found to be 11.54 Hz which is very close to the forcing frequency 10.9 Hz (Figure 9). The analysis shows large amplitude of vibration (the red colour) which is exactly the location of visible vibration in the on-site reciprocating compressor pipeline structure. Hence the support locations have to be modified to alter the natural frequency of the pipeline to completely avoid the occurrence of vibration due to acoustic resonance.

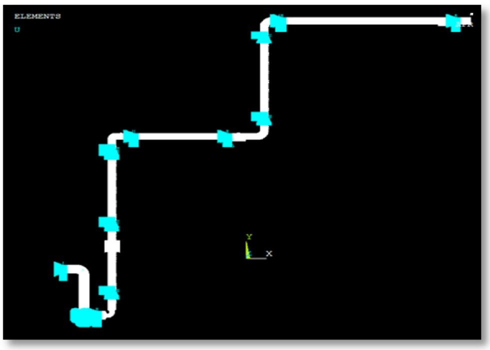


Fig 10: Modified support locations

***** INDEX OF DATA SETS ON RESULTS FILE *****

SET	TIME/FREQ	LOAD STEP	SUBSTEP	CUMULATIVE
1	53.773	1	1	1
2	53.957	1	2	2
3	112.58	1	3	3
4	126.73	1	4	4
5	128.97	1	5	5
6	132.16	1	6	6
7	133.38	1	7	7
8	135.51	1	8	8
9	146.43	1	9	9
10	147.06	1	10	10

Fig 11: Model shapes and frequencies

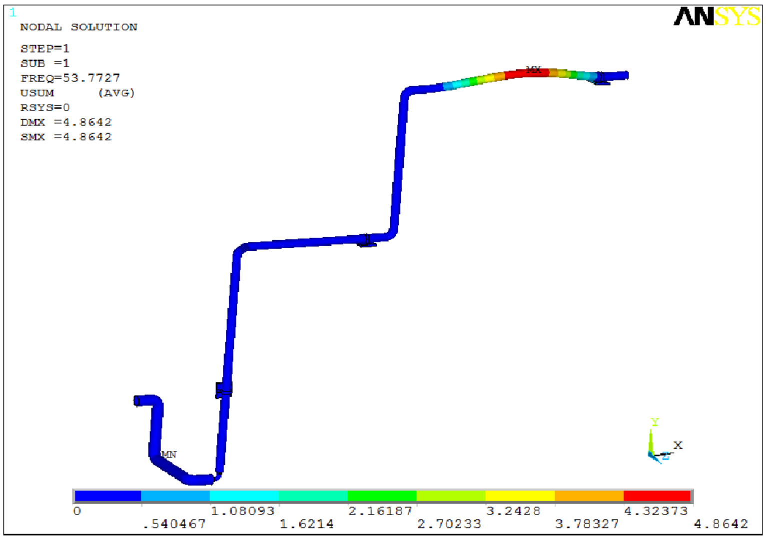


Fig 12: Modified support locations analysis (a)

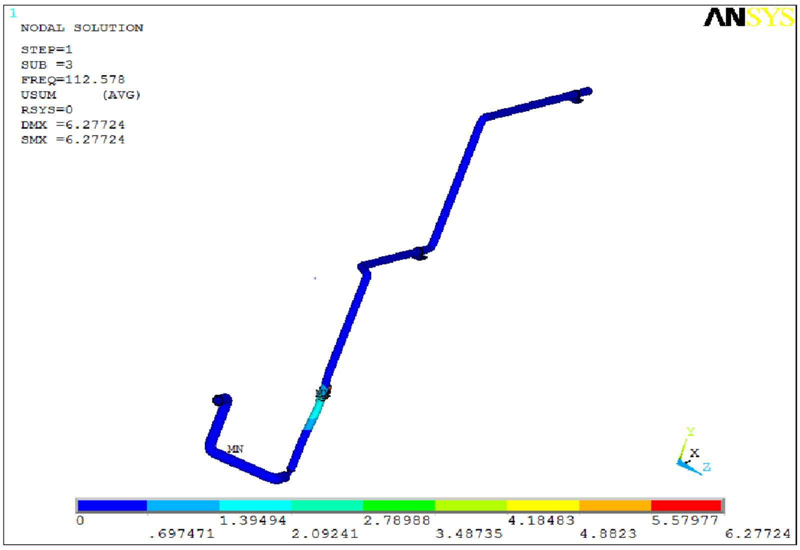


Fig 13: Modified support locations analysis (b)

The modified support locations are shown in figure 10. It can be seen that the natural frequency of the pipeline has changed from 11.54 Hz to 53.77Hz (figure 11) which is nearly 90% away from the forcing frequency (fourth harmonic) of the pipeline structure. Figure 12 and 13 shows the deformation in the pipeline and we can conclude that the vibration is reduced to considerable extent in the system.

5 REMARKS AND CONCLUSION

The following conclusions has been derived from the model analysis

- Natural frequency of pipeline set-up after support modifications = 53.77 Hz
- Pulsation forcing frequency = 10.9 Hz
- A safety factor of 90% from the nearest (fourth) harmonic is obtained in the modified support structure to ensure resonance does not occur.
- After analyzing all the possible natural frequencies in between the each set of harmonics to prevent the acoustical resonance, the natural frequency of 53.77 Hz obtained from optimization of support locations avoids complete vibration in the piping.

And hence the support locations as specified should be implemented to bring the vibration within acceptable limits. For reaching more accurate and optimum results, the forcing frequency obtained at the outlet of the pulsation dampener calculated using flow analysis should be used for modal analysis. However the flow analysis requires very fine meshing. The pressure time transient output can be converted into amplitude frequency by doing a FFT transformation. The frequency thus obtained can be further used to do the model analysis for more accurate results.

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