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Pulsation and Vibration Study Of Reciprocating Compressor According To API 618 5th Edition

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Abstract: *The compressor package contains the reciprocating compressor, pulsation dampers, gas coolers and the connected pipe system which are often the heart of an installation and should be operate smoothly and reliably. The compressor piping vibrations can contribute to fatigue failure of the system or entire package which can lead to unsafe situations for human being as well as environment, loss of capacity and increase in maintenance as well as repair cost and to avoid this situation compressor piping vibration analysis to be carried out at a very early stage of the design of an installation. The pulsation analysis should be carried out before the piping vibration analysis. The guidelines for the pulsation analysis are given in API 618 Approach 2 and the guidelines for the vibration analysis are given in API 618 Approach 3. The original system layout is checked with respect to pulsations with all operating cases that are characterized by steady-state operating conditions. The different gas properties and operation cases are considered with valve unloading cases as operation cases. The measures are proposed to reduce pressure pulsation by installation of orifices. The shaking forces are used in the subsequent vibration study. After the analysis of pulsation results, find out the worst case for the vibration study means we used the shaking forces induced by pressure pulsation excite the mechanical piping system of worst case. The Vibration Study determines the effect on the mechanical piping system and proposes measures to avoid stresses possibly leading to deformation or rupture by fatigue. The finite element program ANSYS is used for modeling of the mechanical system. The model is built of several types of basic piping elements (e.g. pipes, beams, elbows, T-pieces) connected at node points. The modifications are proposed to meet agreed criteria of vibration. This paper demonstrated that by properly analysising the compressor piping vibration in an accurate and economic way using Pulsim and Ansys software. The accuracy of the analytical solution is validated by means of experimental results by using B & K Analyser for the measurement of compressor piping vibration.*

Keywords: *ANSYS, API 618, APL LANGAGUGE, FFT ANALYSER, PULSE, PULSIM*

I. Introduction

The high vibrations were reported at thee piping of compressor which was build according to API 618 Standard. This compressor is using to compress Hydrogen gas from 19 bar to 70 bar in two stages for feeding a hydrocracker in a refinery. The flow rate of this compressor is varying from 226 kg/hr to 1136 kg/hr. The compressor rotation speed is 742 RPM. The motor power for this 2 crank compressor is 355 kW.

The objective of this project is to reduce the vibration of piping system of reciprocating compressor. To accomplish this, the following specific objectives are defined and completed.

- Fundamentals of Pulsation and Mechanical Vibration Theory
- Pulsation Analysis as per API 618 5th Edition approach 2 : To reduce the pulsation across the piping system by using orifices at different locations
- Vibration Analysis as per API 618 5th Edition approach 3 : To reduce the Vibration across the piping system by using supports at different locations
- Compare the results with before and after pulsation and vibration analysis plotting graphs in Velocity-Time Domain.

II. LITERATURE OVERVIEW

This literature review summarises, interprets and critically evaluates existing "literature" in order to establish correct knowledge of subject. The many experts present these views about the pulsation and vibration in different literatures. Following are literature review is to go through the main topics of interest.

Shelley Greenfield and Kelly Eberle (2008) summarized an interesting literature overview of the new API standard 618 (5th edition) and its impact on reciprocating compressor package design. The new API 618 5th Edition includes many improvements over the 4th Edition in the specification for engineering studies to minimize pulsation and vibration. This provides the changes of The Pulsation, Vibration, Torsional, Skid and Engineering studies. Also this gives the Summary of API 618 standard and key changes in the 5th edition.

James D. Tison and Kenneth E. Atkins (2008) described the control philosophy for the pulsation and vibration as per new fifth edition of API 618 for reciprocating compressors. This paper provide the user with a working knowledge of good engineering practices for pulsation and vibration control of reciprocating machinery in relatively high mole weight gases as well as an in depth understanding of the postposed changes in API 618 and differing design philosophies. The conclusions for this paper are A) The intent of API 618 at the Third Edition was that Design approach 3 meant effective pulsation control. This usually required that reactive filtering be used in relatively high mole weight systems. B) The Fourth Edition attempted to define the steps required to quality a piping system, in the event that the allowable pulsation levels were exceeded. This created confusion and led to systems being designed with less emphasis on pulsation control, and justified with mechanical response calculations of questionable validity. C) The fifth Edition will clarify the confusion that resulted from the addition of the language concerning mechanical forced response calculations. The User will now be able to determine if the systems mattes Design Approach 3 by the use of the technically sound pulsation and shaking force control philosophy or through the use of the higher risk philosophies based on mechanical forced response calculations.

Paul Alves (2006) studied the acoustical and mechanical Analysis of reciprocating compressor installation as per API 618 4th Edition. This paper gives basic about the Pulsation, Dynamic Forces, Resistive elements, Reactive elements, analysis cycle. Also he explained different clause which are mentioned in API 618 for Pulsation and Vibration analysis. We can easily understand API 618 M3 is applicable for pulsation analysis and API 618 M4 to M8 are applicable for vibration analysis.

Enzo Giacomelli et al. (2006) studied the forced response of cylinder manifold for reciprocating compressor applications by considering Crosshead guides, distance pieces, cylinder flanges, joint, supports etc for the analysis. The study performed frequencies and amplitudes of pulsation induced shaking forces defined by acoustical simulation, internal gas forces in the cylinder, and unbalanced mechanical forces and moments allows a proper forced response analysis of the cylinder manifold system. The study applied these forces to the finite element model to calculate the relevant vibrations and stress amplitudes by performing a harmonic analysis. The paper covered the existing procedures, application experience and recommendations for properly considering the applied loads.

J. C. Wachel and J. D. Tison (1999) investigated a wide variety of vibration and failure problems occur in reciprocating machinery and piping systems. Excessive piping vibration problems usually occur when a mechanical natural frequency of the piping system or compressor manifold system is excited by a pulsation or mechanical excitation source since reciprocating compressors and pumps generate high pulsation forces, vibration and failure problems in these systems are common. As per this paper, whenever high vibrations are encountered in reciprocating compressors, pumps or piping, it is necessary to determine if vibrations and dynamic stresses are acceptable.

W. W. von Nimitz (1982) described pulsation and vibration control requirements in the design of reciprocating compressor and pump installations. The paper provided the basis for evolving improved methods for assuring the reliability of reciprocating compressor and pump installations at the design stage. This contains new methods for sizing surge volumes, maximum allowable pulsation levels at compressor valves and in the piping systems, and improved pressure drop criteria based on performance. The introduction of the paper explained the nature of the problem, previous work, purpose, and the contribution of the paper. The contents of each section may be provided to understand easily about the paper.

III. FUNDAMENTALS OF PULSATION AND MECHANICAL VIBRATION THEORY

In order to understand how to control pulsation and vibration in positive displacement machinery systems, it is imperative that one understands the differences between acoustical and mechanical concepts. In 2.1, the acoustic issues, along with acoustic control techniques, will be addressed. In 3.2, the elements of the mechanical system will be explained along with the concept of acoustic-mechanical coupling and the mechanical techniques for controlling vibration.

Overview of Pulsation Concepts

Pressure variations that result from oscillatory flow of positive displacement machinery are the subject of this section. These variations in pressure are referred to as pulsation. The pulsation occur in systems handling both gases and liquids. High vibration, support degradation and fatigue failures caused by dynamic forces induced by the pulsation are the most common problems resulting from pulsation. In order to reduce the possibility of detrimental pulsation and vibration at the design stage, it is necessary to understand several technical concepts Excitation mechanisms are first addressed in 2.1.1. Acoustic response is then explained in 2.1.2. In 2.1.3, the most common results of excessive pulsation are reviewed and the concept of acoustic-mechanical coupling is explained.

General Brief about pulsation

Pulsations are the pressure and flow variations in gases and liquids that propagate in the pipe systems and fluid machinery. Every pulsation consists of a pressure pulsation wave and a flow pulsation wave. It found that where the pulsation is high, flow pulsation is low and vice versa. Always pulsation propagate with speed of sound id the gas. Pulsation should be controlled in order to avoid dangerous vibration & fatigue in the pipe system, ensure the integrity of the pipe system, get optimum performance of machinery like compressor and pump, achieve high flow meter accuracy, Control Noise.

Effect of Pulsation

The pulsation that results in high shaking forces can cause excessive vibration in a piping system. However, excessive vibration can occur even in cases where the dynamic forces are low if an excitation frequency is close to, or coincides, with a mechanical natural frequency. In this case, vibration will be amplified, typically a factor of 5 through 10 compared to the off-resonance condition. The amplitude at resonance is limited by the damping of the system. The Pulsation cause pipe vibrations and subsequently failures due to fatigue of the material, reduce compressor efficiency, produce noise, cause errors or inaccuracy in the flow metering, reduce the lifecycle of the compressor valves.

Pulsation Control Methods

The pulsation control in compressor piping systems can be accomplished by application of the basic acoustic elements of acoustical compliance (volume), acoustical iterance (choke tube), and resistance (pressure drop). These elements can be used individually or combined in various manners to achieve pulsation control. The pulsation suppression devices range from single surge volumes (empty bottles) to acoustic filters (bottles with internals or utilizing secondary volumes), often used in conjunction with orifice plates. The user should understand that this discussion is not intended to enable one to design these elements themselves.

Spread of Pressure Wave

Excitation Sources

In systems utilizing positive displacement machinery, the flow of gas or liquid is not steady. Instead, the fluid moves through the piping in a series of flow pulses (dynamic or time varying), which are superimposed upon the steady (average) flow. As an example, the magnitude and shape of the flow pulses through the compressor valves in a reciprocating compressor cylinder are determined by physical, geometrical and mechanical characteristics of the compressor (rotational speed, bore, stroke, loading, compression ratio, etc.). These flow pulses act as excitations which create pressure and flow modulations (acoustic waves) that move through the process fluid as it moves through the piping system. Generally, the predominant pressure and flow modulations generated by a reciprocating compressor are at frequencies which can be modeled as one-dimensional waves. An important part of the acoustic analysis is the development of a compressor model that accurately predicts the dynamic flow excitation (flow versus time) delivered by the compressor. Some simplified examples are shown in Fig.: 3.1, Fig.: 3.2, Fig.: 3.3, and Fig.: 3.4.

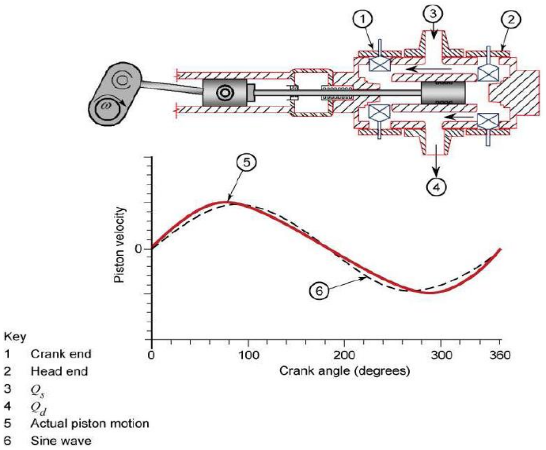


Fig.: 3.1 — Piston Motion and Velocity for a Slider Crank Mechanism

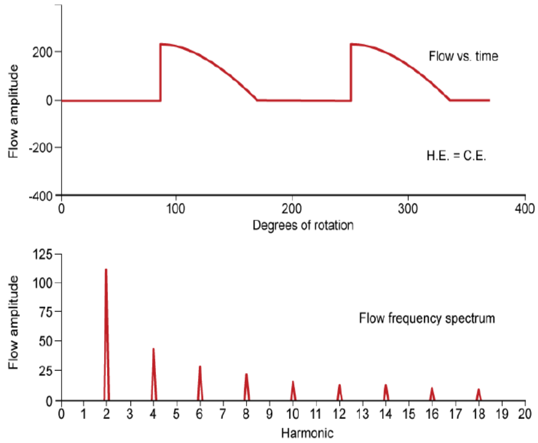


Fig.: 3.3 — Symmetrical, Double Acting Compressor Cylinder with Rod Length/Stroke = ∞ and No Valve Losses

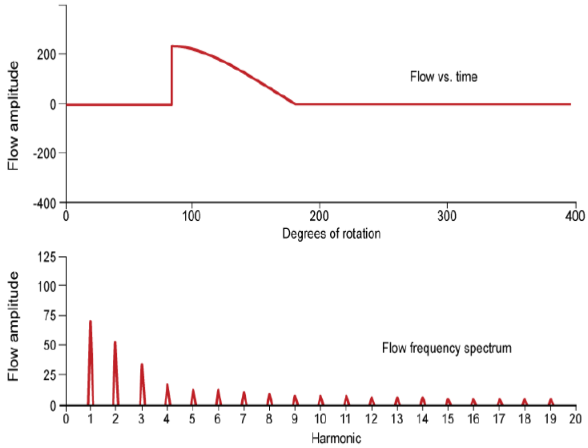


Fig.: 3.2 — Single Acting Compressor Cylinder with Rod Length/Stroke = ∞ and No Valve Losses

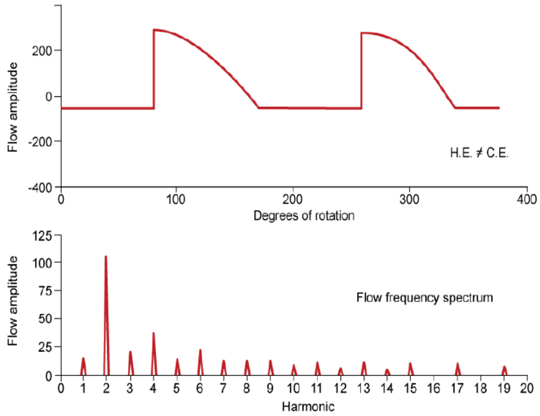


Fig.: 3.4 — Unsymmetrical, Double Acting Compressor Cylinder with Rod Length/Stroke = 5 and No Valve Losses

Acoustic Response and Resonance

The flow pulses caused by the reciprocating action of the compressor or pump create pressure pulses or waves that move through the piping system as shown in Fig.: 5.

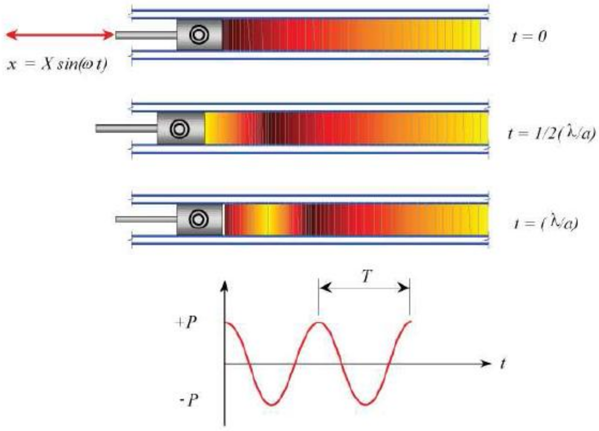


Fig.: 3.5 — Traveling Wave in Infinite Length Pipe

While the flow pulse frequencies generated by the compressor are a function of the mechanical properties of the compressor, the acoustical response in the piping is a function of the mechanical properties of the compressor, the thermo physical properties of the gas, and the acoustical network defined by the attached piping. When a particular harmonic of running speed is near or coincident with an acoustical natural frequency, the acoustic response (dynamic pressure amplitude) is amplified. These resonances that occur when an excitation frequency coincides with a natural frequency can be simple organ-pipe type resonances or complex modes involving all of the piping. For simple constant diameter lines with open and/or closed boundary conditions, specific pipe lengths determine acoustical natural frequencies. If a line length coincides with integer multiples of one half or one quarter of the wavelength, depending on the combination of open or closed end conditions, an acoustical resonance can be excited. End conditions are defined as either open or closed. For half wave resonances, both end conditions must be the same, i.e. open-open or closed-closed. For quarter wave resonances, the end conditions must be opposite, i.e. one open end and one closed end. Examples of these configurations are shown in Fig.: 6 and Fig.: 7, and are defined by following Equations.

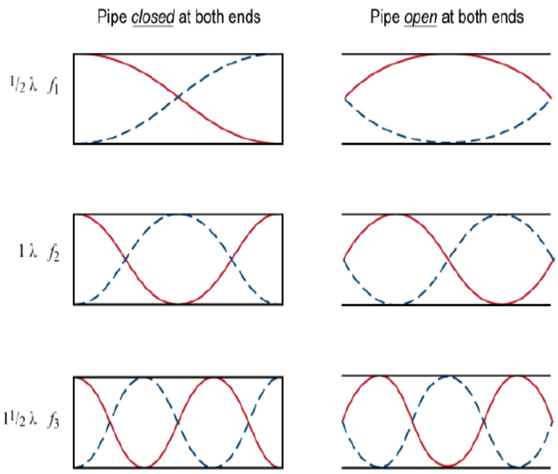


Fig.: 3.6 — Mode Shapes of Half Wave Responses

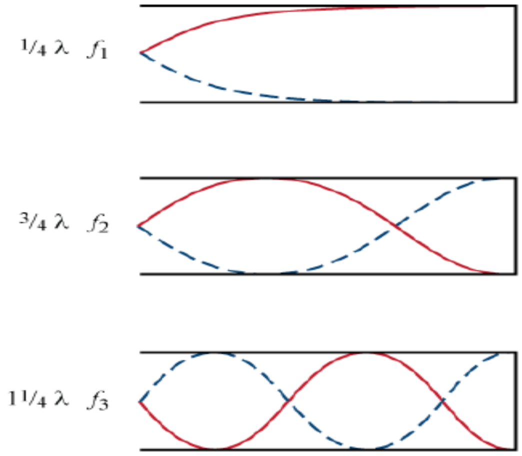


Fig.: 3.7 — Mode Shapes of Quarter Wave Responses

Formula for half wave (closed-closed and open-open acoustic response frequency):

$$f = \frac{na}{2L}$$

Formula for quarter wave (open-closed acoustic response frequency):

$$f = \frac{na}{4L}$$

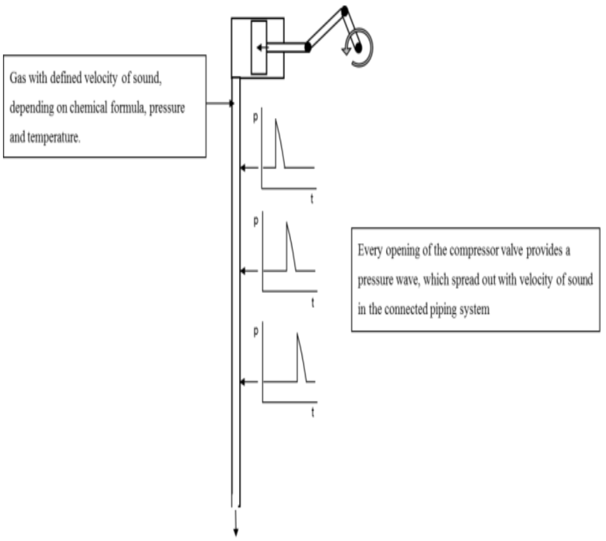


Fig.: 3.8 — Pulsation Wave Form

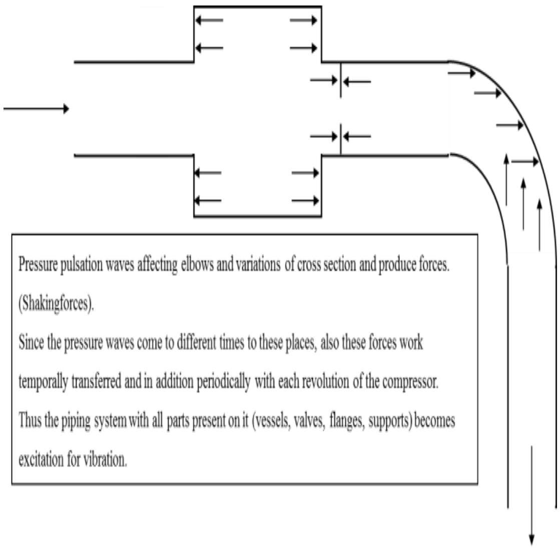


Fig.: 3.9 — Propagation of Pulsation Wave

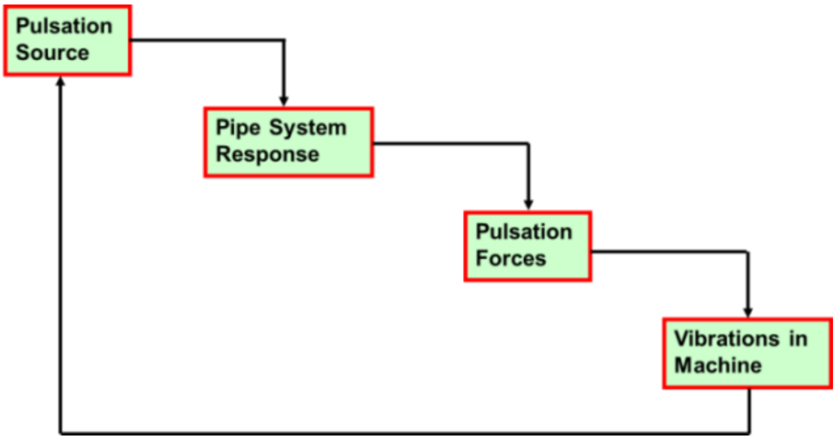


Fig.: 3.10 — Pulsations and Vibrations – the mainline

Pulsation and Vibration Study – The Mainline

Pulsation study

The Pulsations Study defined the height of the pulsations and describes measures to reach the pulsation limits.

The pulsation limits are most according API 618, Approach 2. The calculation program used for this analysis is Pulsim 3.1 which is developed by M/S TNO, Netherland. The more details about this will discuss in chapter III.

Vibration study

Vibrations are mechanical oscillation in a plant, which can occur in pipes, vessels and beam structures. The vibration provides cyclic stress and can lead to fatigue break of the piping system.

In the vibrations study the dynamic behavior of the piping system is investigated and the necessary measures are described to reach the vibration limits. Base of the vibration study is the previous performed Pulsation Study. The Limits are most according API 618, Approach 3. The calculation program used for this analysis is ANSYS. The more details about this will discuss in chapter IV.

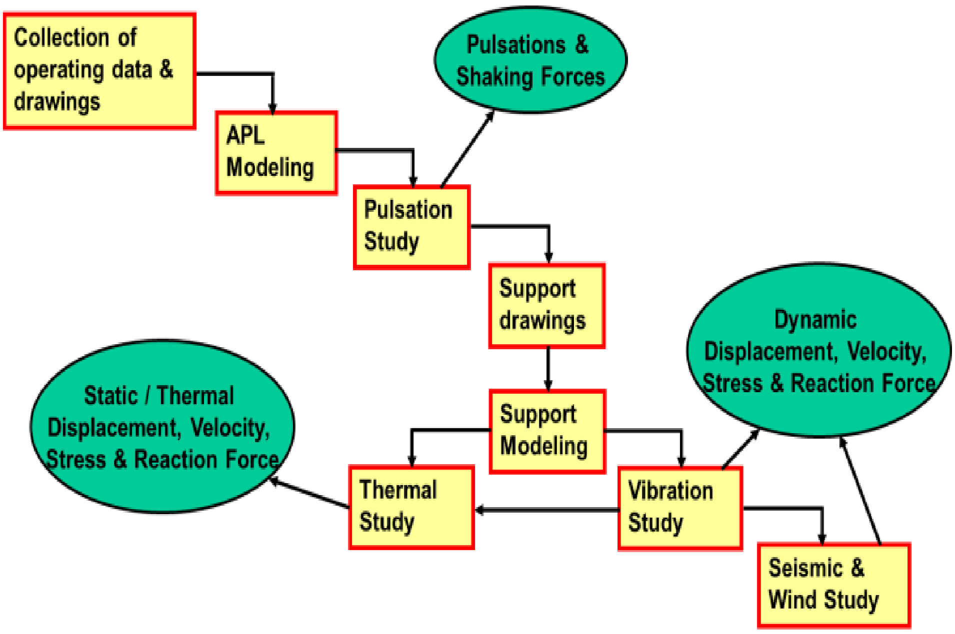


Fig.: 3.11 — Pulsation and Vibration Study – the mainline

IV. STUDY OF PULSATION ACCORDING TO API 618 (ANALYTICAL METHOD)

According Design Approach 2 of API Standard 618 5th edition 2007, an acoustic simulation is required to evaluate and control pulsation in piping systems at reciprocating compressors. The pulsation report contains a summary of the pressure pulsations in the piping system as well as modification proposals to reduce pulsation to the agreed level. The original system layout will be checked with respect to pulsation. All operating cases that are characterized by steady-state operating conditions will be checked, thus transient cases (e.g. start-up-cases) are excluded. Different gas properties and operation cases are considered. Valve unloading cases are considered as operation cases. Measures are proposed to reduce pressure pulsation, e.g. installation of orifices, changes in piping length or diameter. Shaking forces are used in the subsequent vibration study.

Used Documents for Pulsation Study

The exact gas data means gas composition and gas properties like velocity of sound, density, pressure, temperature flow for which compressor is going to use is considered for the pulsation simulation. Also the Isometric drawings of piping and compressors are used for the pulsation simulation.

Table : 4.1 — Gas Properties / Input Data for Pulsation Simulation

		Load case:	Unit	1	2	3	4	5	6
				EOR 100%	SOR 100%	Low Purity alt design case	EOR 50%	SOR 50%	Low Purity alt design case 50%
Index	**Name**								
1	*1st stage suction*	VOS	[m/s]	1358.8	1358.8	876.9	1358.8	1358.8	876.9
		density	[kg/m3]	1.521	1.521	3.531	1.521	1.521	3.531
		FFF	[-]	0.005	0.005	0.005	0.005	0.005	0.005
		pressure	[bar]	19.8	19.8	19.8	19.8	19.8	19.8
		temperature	[°C]	40	40	40	40	40	40
		flow	[kg/h]	486	487	1136	226	227	530
		wave length	[m]	109.9	109.9	70.9	109.9	109.9	70.9
2	*1st stage discharge*	VOS	[m/s]	1506.5	1504.4	960.6	1510.8	1508.7	963.4
		density	[kg/m3]	2.466	2.457	5.831	2.499	2.491	5.923
		FFF	[-]	0.005	0.005	0.005	0.005	0.005	0.005
		pressure	[bar]	39.21	38.98	39.24	39.97	39.73	40.09
		temperature	[°C]	107	106	100	109	108	102
		flow	[kg/h]	486	487	1136	226	227	530
		wave length	[m]	121.8	121.6	77.7	122.2	122.0	77.9
3	*Cooler 1-2*	VOS	[m/s]	1440.9	1439.8	924	1444.5	1443.3	926.2
		density	[kg/m3]	2.669	2.658	6.260	2.708	2.696	6.365
		FFF	[-]	0.005	0.005	0.005	0.005	0.005	0.005
		pressure	[bar]	38.74	38.515	38.74	39.49	39.255	39.58
		temperature	[°C]	73.5	73	70	75	74.5	71.5
		flow	[kg/h]	486	487	1136	226	227	530
		wave length	[m]	116.5	116.4	74.7	116.8	116.7	74.9
4	*2nd stage suction*	VOS	[m/s]	1372.2	1372	885.5	1374.8	1374.7	887.2
		density	[kg/m3]	2.917	2.900	6.772	2.962	2.945	6.895
		FFF	[-]	0.005	0.005	0.005	0.005	0.005	0.005
		pressure	[bar]	38.27	38.05	38.24	39.01	38.78	39.07
		temperature	[°C]	40	40	40	41	41	41
		flow	[kg/h]	486	487	1136	226	227	530
		wave length	[m]	111.0	110.9	71.6	111.2	111.2	71.7
5	*2nd stage discharge*	VOS	[m/s]	1516.5	1506.9	969.3	1514.6	1505.8	968.1
		density	[kg/m3]	4.436	4.327	10.465	4.448	4.401	10.494
		FFF	[-]	0.005	0.005	0.005	0.005	0.005	0.005
		pressure	[bar]	70.2	67.68	70.2	70.2	68.68	70.2
		temperature	[°C]	100	96	94	99	95	93
		flow	[kg/h]	486	487	1136	226	227	530
		wave length	[m]	122.6	121.9	78.4	122.5	121.8	78.3

Calculation and Interpretation of Analytical Results

The study of the gas pulsation is carried out with the digital simulation program PULSIM which has been developed by the TNO/TPD Institute in Delft (NL). A system of pipes is built of several complicated parts and elements which must be simplified for calculations. For example volumes of a cylinder casing between valves and connecting flange, heat exchangers, separators, dampers, etc. The data of the gas might also change during operation. In order to take these uncertainties into account, we make several calculations at different velocities of sound. The range of variation is ± 12 % and it is varied in steps of 2%. The results are presented in tables.

Admissible limits

The main requirements of API618 are as follows:

[API618, section 7.9.4.2.5.2.2.2] "For systems operating at absolute line pressures between 3.5 bar and 350 bar (350-35'000 kPa), the peak-to-peak pulsation level of each individual pulsation component shall be limited to that calculated by (in SI units):"

$$P_1 = \sqrt{a/350}\left(\frac{400}{\sqrt{P_L \cdot D_I \cdot f}}\right)$$

With the following equation for the frequency f

$$\boxed{f = \frac{rpm \cdot N}{60}}$$

For absolute pressures less than 3.5 bar, the peak-to-peak levels of individual pulsation components need only meet the levels calculated for an absolute pressure of 3.5 bar.

Outcome of Pulsation Study for 1st Stage Suction

With the original system layout the pulsations are too high. The dominant order of pulsation is primarily the 1st & 2nd order. The 1st stage suction side showed slightly increased pulsation, the worst case was the EOR 50% regulation of case 4 which reached 4.3% ptp. At the state of realization a change in piping was not practicable; orifices were used to dampen the pulsation to the possible minimum. The residual pulsation is still above the API limit. With orifices the worst case reached values 1.4 times higher than the API Limit. For the calculation of the vibration analysis we use the case with the highest pulsation with its shakingforces. With this procedure we ensure that the increased pulsation does not affect the piping stability. The vibration study however showed throughout acceptable values for the suction system. A reason for this is the relatively small shaking forces due to the relatively low pressure pulsation. No modifications of the piping were made on the suction side. The original system and orifices with only a small pressure drop gives satisfying pulsation results. With the recommended modification no. 5, we achieve satisfied results. The following table shows the highest value of the pulsation at different deviation of velocity of sound for design case at some representative nodes..

Table : 4.2 — A Few Suction Calculation Results for Different Load Cases

Case No.	Case 1	Case 1	Case 4	Case 4	Case 6	Case 6
System No.	1	1	4	4	6	6
Load case	EOR 100%	EOR 100%	EOR 50%	EOR 50%	Low Purity alt design case 50%	Low Purity alt design case 50%
Modification No.	0	5	0	5	0	5
Orifice RO-FAD1-I	No orifice	33.00 mm	No orifice	33.00 mm	No orifice	33.00 mm
Orifice RO-FAD1-O	No orifice	92.00 mm	No orifice	92.00 mm	No orifice	92.00 mm
Node 120	0.54/-	0.34/-	3.0/2.3	1.9/1.4	1.4/+	0.88/+
Node 370	1.5/-	1/-	3.2/1.1	2.1/+	4.2/1.3	2.1/+
Node 810 (Cyl flange)	1.4/-	1.3/-	4.3/+	3.4/-	3.0/-	3.0/-
Recommended		X		X		X

Following are the recommended orifices with locations :

Table : 4.3 — Recommended Orifices for Suction Side

Orifice	Location	Orifice ID [mm]	Line-inner Dia [mm]	Pressure drop [%] Design 100%	(EOR 100%)
RO-FAD1-I	1st stage Suction Dampner FAD1 inlet	33.00	78.00 (3")	0.81%	0.160 bar
RO-FAD1-O	1st stage Suction Dampner FAD1 outlet	92.00	92.00 (4")	0.00%	0.000 bar

Outcome of Pulsation Study for Interstage

With the original system layout the pulsations are too high. The dominant order of pulsation is primarily the 1st & 2nd order. The interstage showed high pulsation, the worst case was the EOR 50% regulation of case 14 which reached 6.0% ptp. At the state of realization a change in piping was not practicable; orifices were used to dampen the pulsation to the possible minimum. For the calculation of the vibration analysis we use the case with the highest pulsation with its shakingforces. With this procedure we ensure that the increased pulsation does not affect the piping stability. The vibration study however showed throughout acceptable values for the interstage system. A reason for this is the relatively small shaking forces due to the relatively low molecular weight. No modifications of the piping were made on the interstage side. The original system and orifices with only a small pressure drop gives satisfying pulsation results. With the recommended modification no. 5, we achieve satisfied results. The following table shows the highest value of the pulsation at different deviation of velocity of sound for design case at some representative nodes.

Table : 4.4 — A Few Interstage Calculation Results for Different Load Cases

Case No.	Case 1	Case 1	Case 4	Case 4	Case 6	Case 6
System No.	11	11	14	14	16	16
Load case	EOR 100%	EOR 100%	EOR 50%	EOR 50%	Low Purity alt design case 50%	Low Purity alt design case 50%
Modification No.	0	5	0	5	0	5
Orifice RO-FAD2-I	No orifice	31.00 mm	No orifice	31.00 mm	No orifice	31.00 mm
Orifice RO-FAD2-O	No orifice	31.00 mm	No orifice	31.00 mm	No orifice	31.00 mm
Orifice RO-FAD3-I	No orifice	35.00 mm	No orifice	35.00 mm	No orifice	35.00 mm
Orifice RO-FAD3-O	No orifice	35.00 mm	No orifice	35.00 mm	No orifice	35.00 mm
Node 2070 (Cyl flange)	3.5/-	3.9/+	6.2/+	4.4/+	3.7/+	5.2/+
Node 2390	3.3/2.6	1.3/+	3.8/1.7	1.6/+	4.0/2.8	1.6/+
Node 3030	1.1/+	0.22/.	1.3/+	0.57/-	0.63/-	0.27/-
Node 3190 (Cyl flange)	3.7/+	1.4/-	2.6/-	2.1/-	3/-	3.1/-
Recommended		X		X		X

Following are the recommended orifices with locations :

Table : 4.5 — Recommended Orifices for Interstage

Orifice	Location	Orifice ID [mm]	Line-inner Dia [mm]	Pressure drop [%] Design 100% (EOR 100%)	
RO-FAD2-I	1st stage Discharge Dampner FAD2 inlet	**31.00**	92.00 (4")	0.37%	0.145 bar
RO-FAD2-O	1st stage Discharge Dampner FAD2 outlet	**31.00**	67.00 (3")	0.32%	0.126 bar
RO-FAD3-I	2nd stage Suction Dampner FAD3 inlet	**35.00**	67.00 (3")	0.15%	0.057 bar
RO-FAD3-O	2nd stage Suction Dampner FAD3 inlet	**35.00**	67.00 (3")	0.15%	0.057 bar

Outcome of Pulsation Study for 2nd Stage Discharge

With the original system layout the pulsations are too high. The dominant order of pulsation is primarily the 1st & 2nd order. The discharge stage showed high pulsation, the worst case was the Low Purity alt design case 50% regulation of case 26 which reached 4.3% ptp. At the state of realization a change in piping was not practicable; orifices were used to dampen the pulsation to the possible minimum. For the calculation of the vibration analysis we use the case with the highest pulsation with its shakingforces. With this procedure we ensure that the increased pulsation does not affect the piping stability. The vibration study however showed throughout acceptable values for the interstage system. A reason for this is the relatively small shaking forces due to the relatively low molecular weight. No modifications of the piping were made on the discharge stage. The original system and orifices with only a small pressure drop gives satisfying pulsation results. With the recommended modification no. 5, we achieve satisfied results.

Table : 4.6 — A Few Discharge Calculation Results for Different Load Cases

Case No.	Case 1	Case 1	Case 4	Case 4	Case 6	Case 6
System No.	21	21	24	24	26	26
Load case	EOR 100%	EOR 100%	EOR 50%	EOR 50%	Low Purity alt design case 50%	Low Purity alt design case 50%
Modification No.	0	5	0	5	0	5
Orifice RO-FAD4-I	No orifice	23.00 mm	No orifice	23.00 mm	No orifice	23.00 mm
Orifice RO-FAD4-O	No orifice	23.00 mm	No orifice	23.00 mm	No orifice	23.00 mm
Node 4070 (Cyl flange)	1.7/-	2.9/-	1.5/-	2.0/-	2.3/-	3.5/-
Node 4170	0.93/-	0.87/-	0.92/-	0.87/-	0.99/-	0.78/-
Node 4490	1.4/1.5	0.61/-	2.2/1.3	0.96/-	3.5/2.5	1.2/+
Node 4640	1.2/+	0.37/-	2.2/1.4	0.83/-	3.4/2.4	1.0/+
Node 4950	1.4/*	0.83/-	2.4/1.6	1.2/+	4.2/2.7	2.0/*
Recommended		X		X		X

We recommend to install the following orifices:

Table : 4.7 — Recommended Orifices for Discharge Side

Orifice	Location	Orifice ID [mm]	Line-inner Dia [mm]	Pressure drop [%] Design 100% (EOR 100%)	
RO-FAD4-I	2nd stage Discharge Dampner FAD4 inlet	**23.00**	67.00 (3")	0.36%	0.252 bar
RO-FAD4-O	2nd stage Discharge Dampner FAD4 outlet	**23.00**	49.00 (2")	0.29%	0.204 bar

Sketches of Piping for the Nodes of Pulsation Study

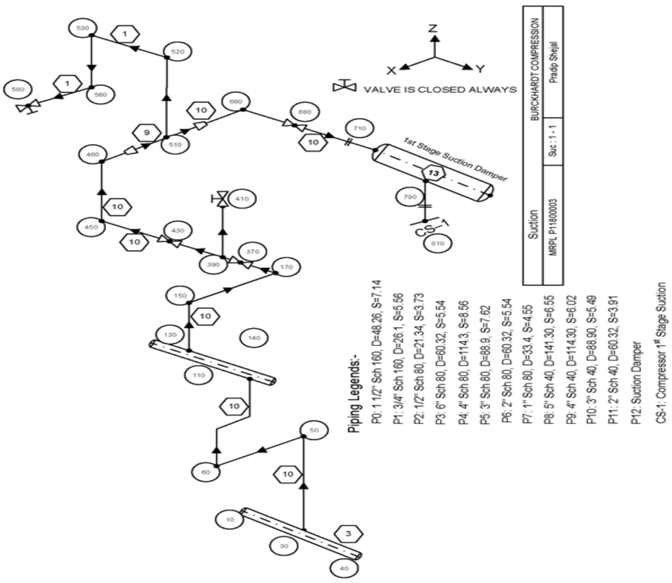


Fig.: 4.1 — Sketch of Suction Piping for Pulsation Study

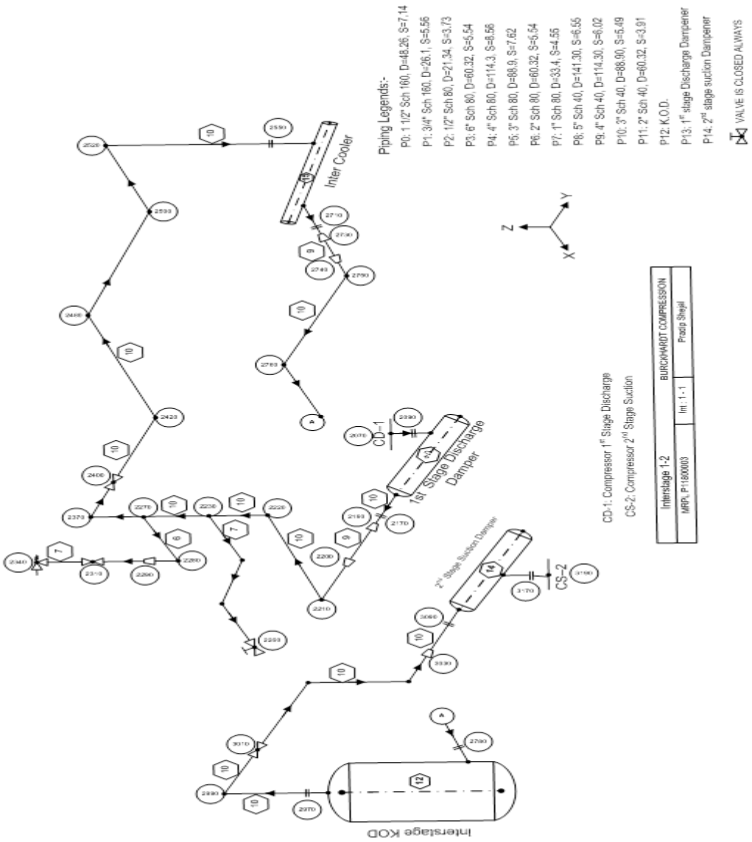


Fig.: 4.2 — Sketch of Interstage Piping for Pulsation Study

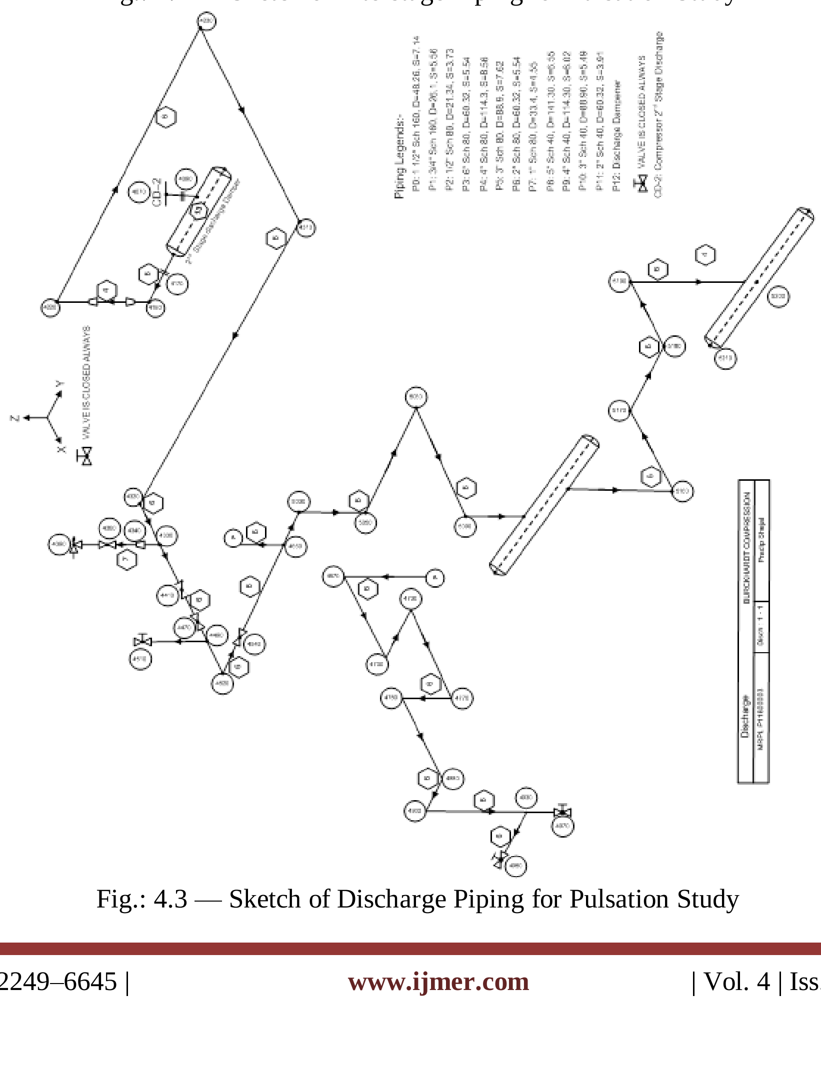


Fig.: 4.3 — Sketch of Discharge Piping for Pulsation Study

V. STUDY OF PIPING VIBRATION ACCORDING TO API 618

In this chapter the results for the vibration study for the compressor type can be found. The vibration study gives an overview of the dynamic behaviour of the piping systems. The natural frequencies and the corresponding modes are determined. In a second step the response of the system towards shaking forces which are caused by the pressure pulsation is calculated. The resulting displacement, velocity and bending stress values should not exceed certain limits.

The most critical case of the pressure pulsation is used for the dynamic analysis. Uncritical parts of the piping are not further investigated.

General Information

Modelling the vibration system

The system is modelled within the Finite Element Program ANSYS. It consists of a piping modelling tool, with several different elements and modelling possibilities. Elements are the connection between different nodes. The vibration study consists in general of the following ANSYS elements:

- BEAM4 : supporting structures
- COMBIN14 : spring supports
- PIPE16 : straight pipes & flanges / valves with specific mass and flexibility factors
- PIPE18 : pipe elbows

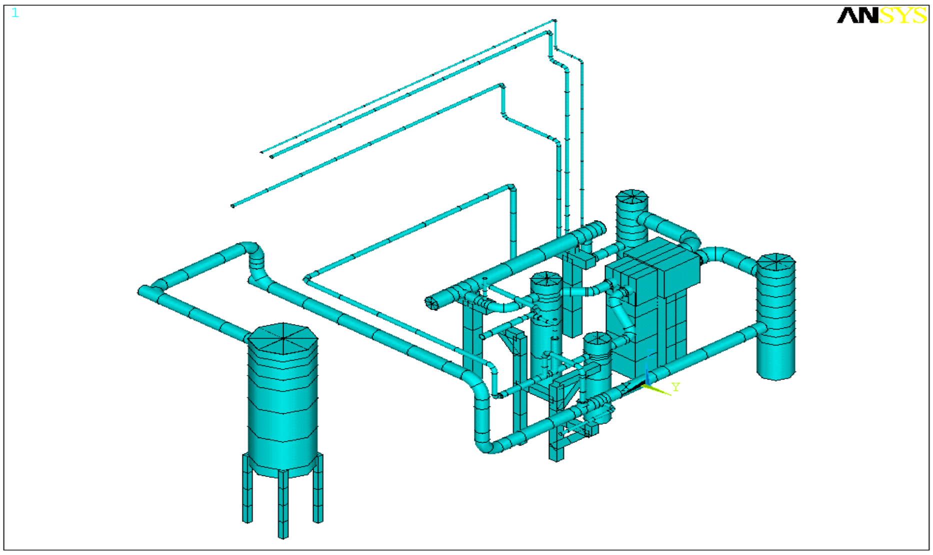


Fig.: 5.1 — Typical Finite Element Model

On the compressor skid our supports are modelled with beam elements, like in fig.: 1. On the customer side of the piping, supports are modelled by restricting at least one degree of freedom of a node. On request we model even the customer piping with supporting structures. Without restriction a node has six degrees of freedom, translations and rotations in each direction. These are the following four typical types of supports:

- Rest Support restricting only translation in vertical direction
- Directional Guide restricting translation in vertical and transversal directions
- Fixed Support restricting translations in all directions
- Anchored Support restricting translations and rotations in all directions

If friction forces are high enough, a support can possibly restrict an additional direction. For cases with a slight excess in vibration we check if this additional restriction can be furnished by the expected friction. For static thermal analyses, with in general higher force values than for vibration calculations, the friction is neglected.

Calculation procedure

The calculation for the vibration analyses is carried out in two steps.

The first step is the calculation of natural frequencies and mode shapes. This indicates which excitation frequencies will cause resonance effects, and which part of the system will be the most affected. This gives a general overview of the vibration behavior of the system.

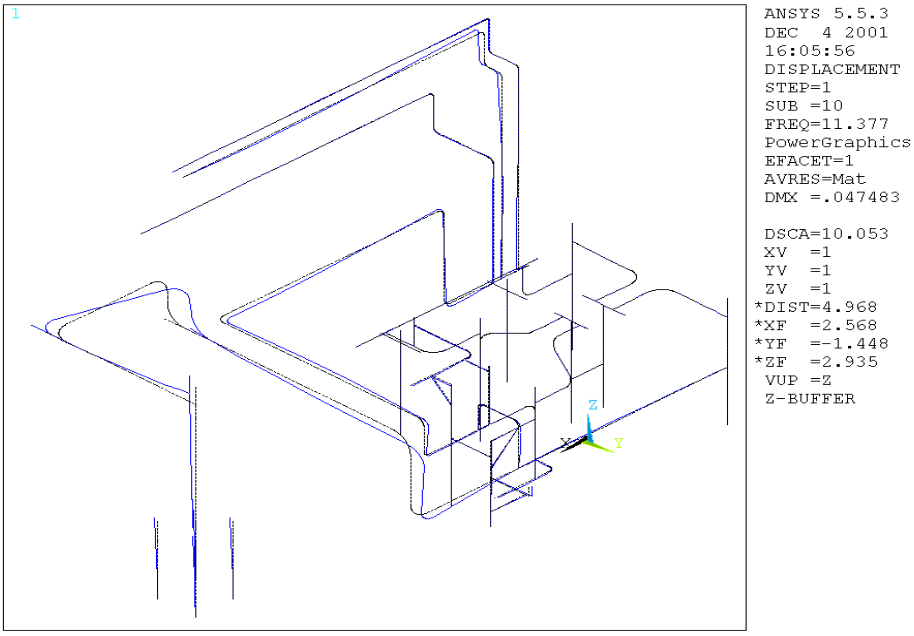


Fig.: 5.2 — Typical Finite Element Model (Wireframe)

In step two we perform a dynamic response analysis. The excitation comes from the shaking forces, which are a result of the pressure pulsation in the piping system. The forces are calculated with the PULSIM program (see pulsation study) and are given as harmonics of the compressor speed for the first twelve orders. Shaking forces occur in case of a change of the flow area or direction (inlet/outlet of a volume, elbows, orifices,...). We use the "worst case" shaking forces that means the forces from the calculated deviation of velocity of sound that delivered overall the highest values.

The results of this calculation are the dynamic displacements, vibration velocities and bending stresses of the pipes as well as the reaction forces on the supports. The peak-to-peak values are given in tables, and for nodes of special interest the time functions are plotted over one crank revolution. The peak-to-peak values are compared to the maximum allowable limit.

Admissible limits

According API 618 5th ed. the stress in the pipes should not exceed the endurance limit of the material used. For steel pipes the peak to peak cyclic stress should be less than 179 N/mm^2 including all stress concentration factors. Normally use a much lower limit of 45 N/mm^2, because we do not know all locations of stress concentration, especially at welded pipes, and we allow for inaccuracies to a certain degree. Stress intensification factors at pipe elbows and tee pieces are taken into account, according to ANSI B31.1. An additional limit we use, which is not limited by API, is the maximum displacement value of 1 mm. A pipe vibrating at such a high level, may cause problems and will generally leave a bad impression of plant operation, even if stress limits are not exceeded. Moreover, we survey the vibration velocities at all nodes, which should not exceed 30 mm/s, according to our experience. The support loads are given as information only, and have to be checked individually by the customer, if supports could carry the calculated reaction force.

Guide to the Sketches

The node numbers for the following sketches are sorted by the following criterions:

- Node 1 - Node 84 and Node 118 - Node 374 for straight pipes
- Node 85 - Node 117 for the compressor model
- Node 375 - Node 440 for support constructions
- Node 441 - Node 647 for bends
- Node 648 - Node 713 for additional nodes near T connections

The numbers arise in the direction of the flow in the pipes.
Further details are given on the sketches.

Sketches of Piping for Vibration Study

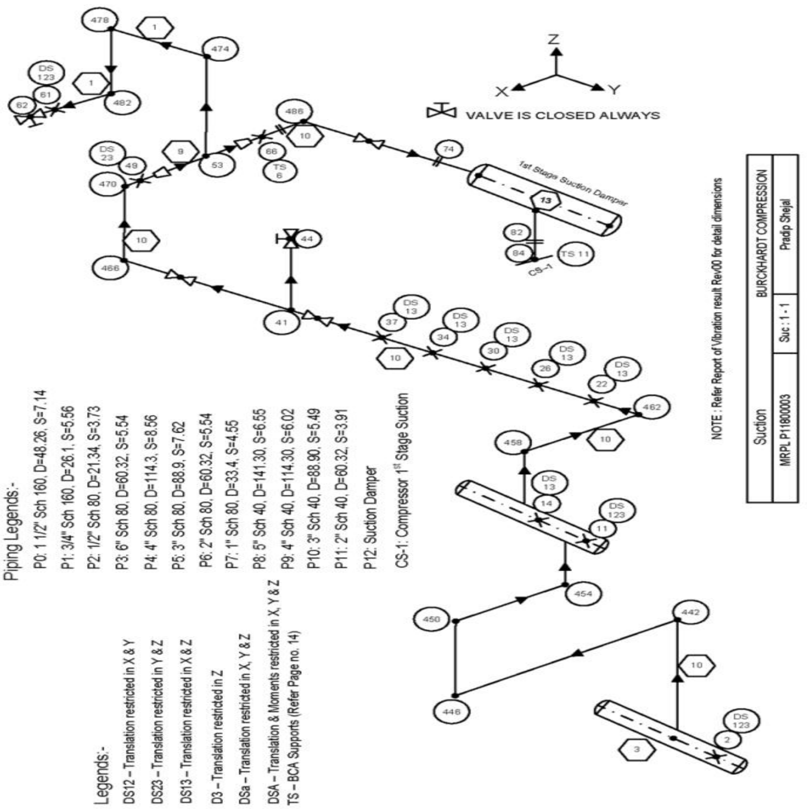


Fig.: 5.3 — Sketches of Suction Piping for Vibration Study

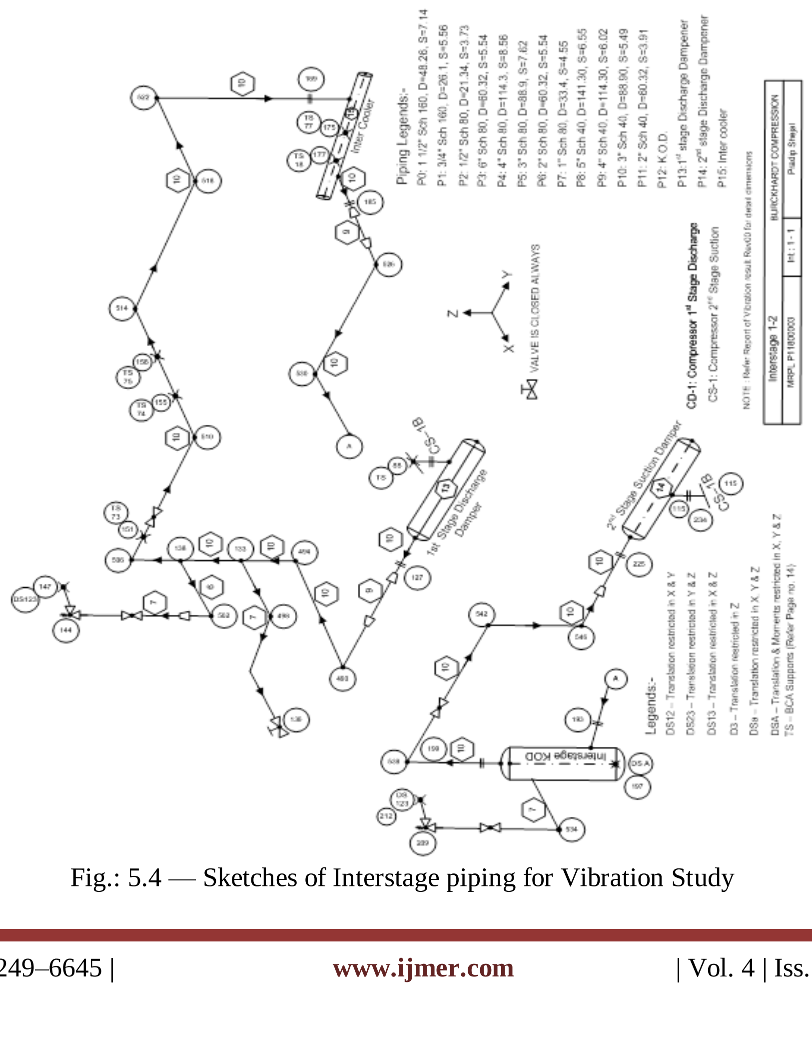


Fig.: 5.4 — Sketches of Interstage piping for Vibration Study

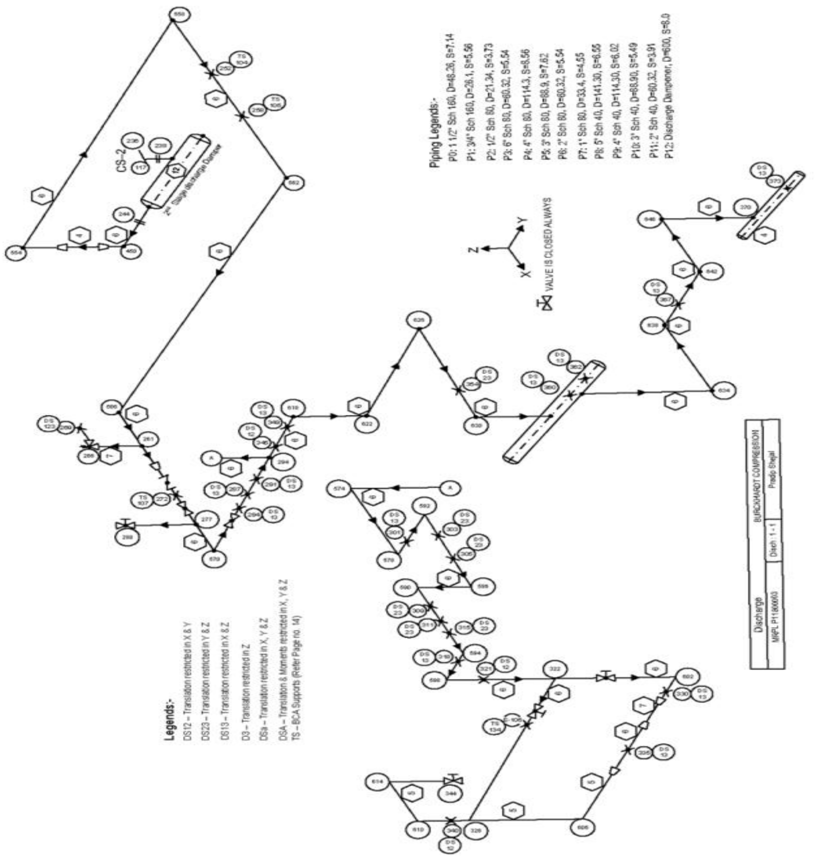


Fig.: 5.5 — Sketches of Discharge Piping for Vibration Study

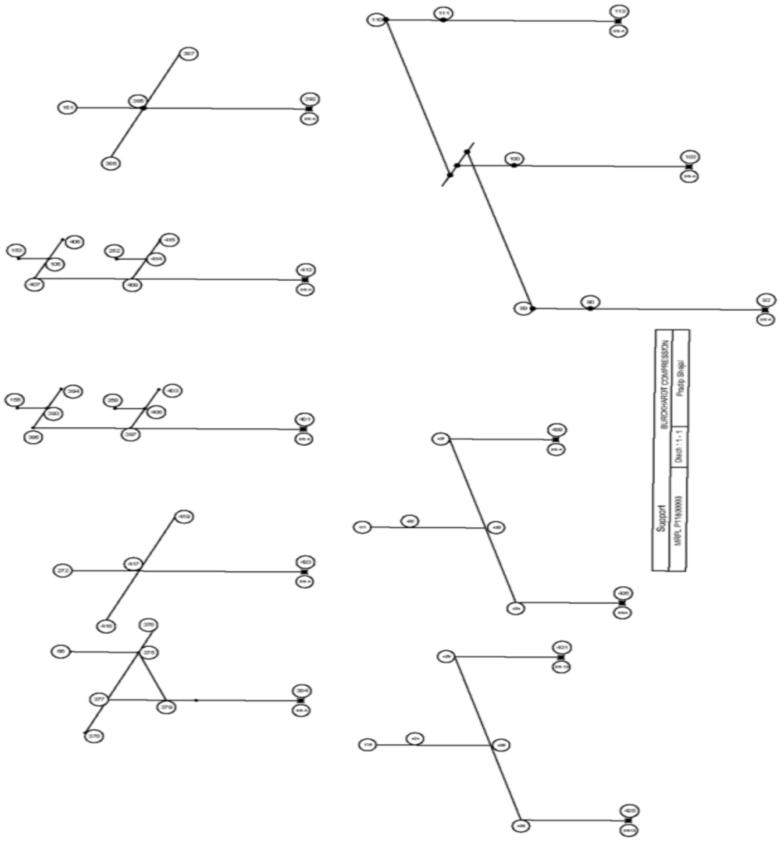


Fig.: 5.6 — Sketches of Support and Compressor

Detailed Results

Summary of Results

The investigation of the piping showed no severe problems with vibrations. One reason for this is the reduced pulsation level which is achieved from the modifications recommended in the pulsation study.

The model for this vibration study includes the suction, interstage and discharge side. Therefore natural frequencies could be calculated once for whole system. For the dynamic response analysis we choose deviation of the velocity of sound with the highest pulsations of the worst case. These are the following load cases:

- for suction : Case 4, EOR 50% - Sys 4 Mod 5, variation of velocity of sound +12%
- for Interststage : Case 4, EOR 50% - Sys 14 Mod 5, variation of velocity of sound +12%
- for discharge : Case 6, Low Purity alt design case 50% - Sys 26 Mod 5, variation of velocity of sound +12%

The model contains the suction side starting at 6”-P-551093-B2A1-IT existing header, suction piping, interstage piping with cooler, KOD and PSV line, the discharge side with PSV-line and with the exiting header 4”-P-551107-B4A1-IH40. Relevant sidelines are also included in the calculation.

No change of piping layout is required except orifices which are already discussed for the pulsation study and also following support should be added into the piping layout.

- Support : X & Y at Node 345 : Refer Sketch
- Support : X & Z at Node 367 : Refer Sketch

After several calculations with the support modification all vibration results are within the limits.

The supports with its restricted direction of the last version are indicated in the attached isometrics. At the piping parts it was necessary to implement additional supports to achieve satisfied results. The details of all supports (blocked translations and rotations) in the calculation model can be found in the sketches of the system. These were discussed with the customer, including blocked translations and rotations for all supports.

The highest displacement occurs at the node 145 for the safety valve 554 for interstage piping before the interstage cooler EA-45555. The detailed picture of the displacement for this node (Appendix A4-1) shows influences of the 1st order. The model analysis of the part at node 145 shows resonance for a frequency around 12.942 Hz (16th eigenmode on fig.: 14). This value is close to the compressor frequency of the 1st order (12.37 Hz). All values of displacement at this nodes are within the limit.

The highest velocity occurs at the node 297 in the side line of the main discharge piping around Y-direction. The detailed picture of the displacement for this node (Appendix A4-1) shows influences of the 1st order. The model analysis of the part at node 297 shows resonance for a frequency around 12.882 Hz (15th eigenmode on fig.: 13). This value is close to the compressor frequency of the 1st order (12.37 Hz). All values of velocity at this nodes are within the limit.

The highest stress occurs at the node 140 in the safety valve 554 piping for interstage piping before the interstage cooler EA-45555. The detailed picture of the displacement for this node (Appendix A4-1) shows influences of the 1st order. The model analysis of the part at node 145 shows resonance for a frequency around 12.942 Hz (17th eigenmode on fig.: 14). This value is close to the compressor frequency of the 1st order (12.37 Hz). All values of stress at this nodes are within the limit. The highest reaction force occurs at the node 92 for the 1st stage cylinder support near to the 2nd stage discharge damper Z- direction. The detailed picture of the displacement for this node (Appendix A4-1) shows influences of the 1st order. The model analysis of the part at node 145 shows resonance for a frequency around 12.942 Hz (16th eigenmode on fig.: 14). This value is close to the compressor frequency of the 1st order (12.37 Hz).

An overview of the results is shown in following table 5.1.

Table : 5.1 — Maximum values for the dynamic response analysis

Dynamic response analysis	Operating Case: EOR 50% & Low Purity alt design case 50% (worst deviation for each stage) (with modified supports)	
		Limits
Lowest natural frequency [Hz]	1.6074	-
Max. displacement [mm] (abs value) Peak to peak worst case (node)	0.9685 (145)	1
Max. velocity [mm/s] RMS (node/direction)	28.27 (297/Y)	30
Max. bending stress [N/mm²] Peak to peak (node)	16.24 (140)	45
Max . reaction force [N] Peak to peak (node/direction)	1216.2 (92/Z)	-

Many supports are described to be loose. Therefore the horizontal stiffness is extremely low. In reality the friction between pipe and supports will give an additional stiffness. That means, in reality lower displacements can be expected.

Plot of the model of calculation

Following are the few plots of the compressor package which is modelled in Ansys.

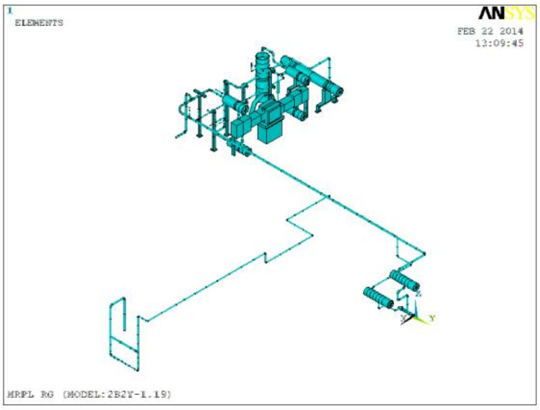


Fig.: 5.7 — Display of Elements for compressor package

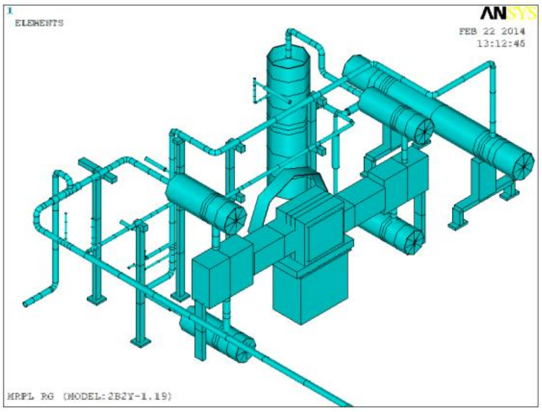


Fig.: 5.8 — Display of Elements for Compressor System

Plot of the model of calculation – Boundary Conditions

Following are the few plots of the compressor package with boundary conditions.

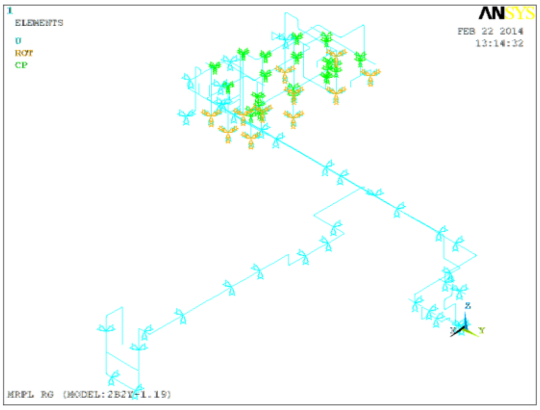


Fig.: 5.9 — Boundary Conditions for compressor package

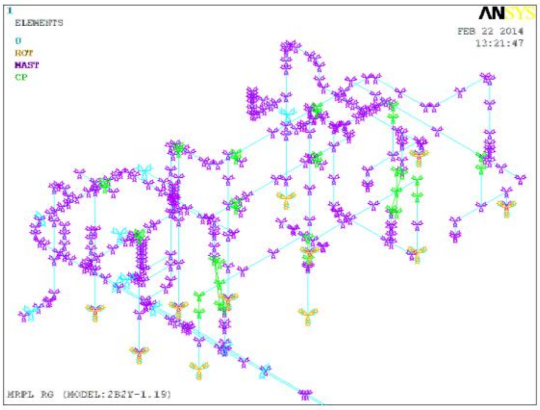


Fig.: 5.11 — Boundary Conditions with Master Nodes for Compressor System

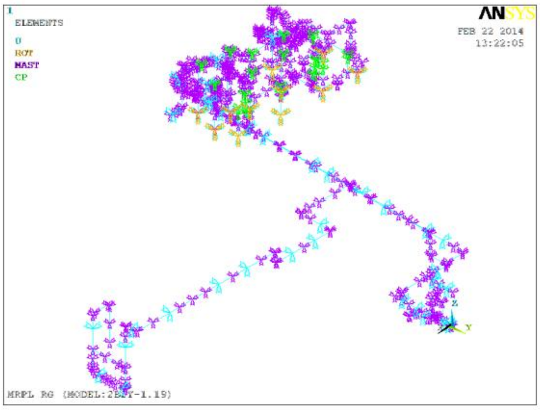


Fig.: 5.10 — Boundary Conditions with Master Nodes for compressor package

Natural frequencies and natural modes of the system

Table : 5.2 — The first 179 eigenmodes (out of 219) of the piping system model

SET	Frequency [Hz]
1	1.6074
2	5.1959
3	5.8717
4	5.9192
5	6.2108
6	6.9121
7	6.9535
8	8.6063
9	8.8348
10	9.5087

The first 10 eigenmodes (out of 219) of the piping system model. The shapes of some Eigenmodes are plotted on the following pages. Some Eigenmodes have to be considered as unrealistic since some boundary conditions are assumed. Therefore only selected Eigenmodes are plotted.

Plot of the mode shapes for the most important frequencies

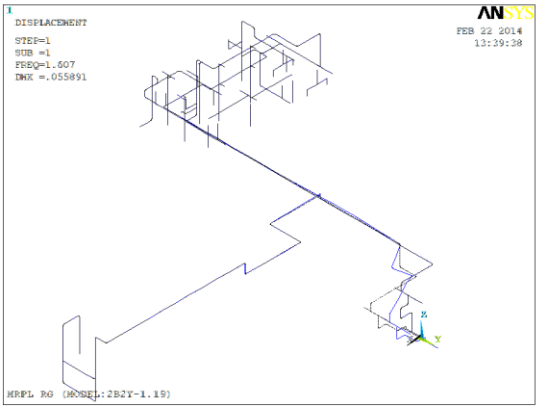


Fig.: 5.12 — Plot of the 1st Eigenmode (1.607 Hz)

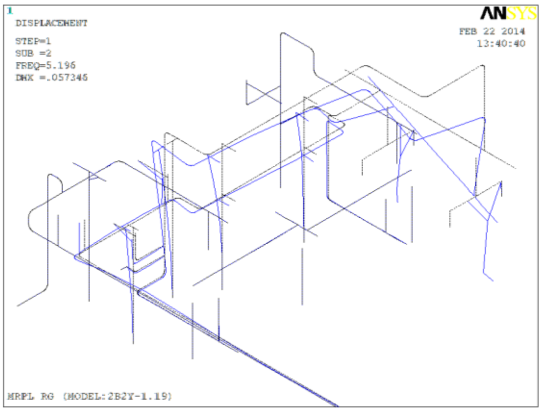


Fig.: 5.13 — Plot of the 2nd Eigenmode (5.196 Hz)

Plot of dynamic responses at selected nodes

Table : 5.3 **-** Dynamic displacements Peak-to-Peak in mm

Nodes	UX	UY	UZ	ABS
145	0.8888	0.0315	0.3887	0.9706
144	0.788	0.0001	0.3887	0.8786
143	0.6874	0.0295	0.3887	0.7902
142	0.5468	0.062	0.3886	0.6737
297	0.0908	0.61	0.2325	0.6591
298	0.091	0.5822	0.217	0.6279
598	0.3515	0.4971	0.0296	0.6095
577	0.0911	0.5115	0.3093	0.6047
578	0.0931	0.5061	0.3167	0.6043
599	0.3601	0.4822	0.0352	0.6028

Plot of dynamic displacements at selected nodes Peak-to-Peak in mm

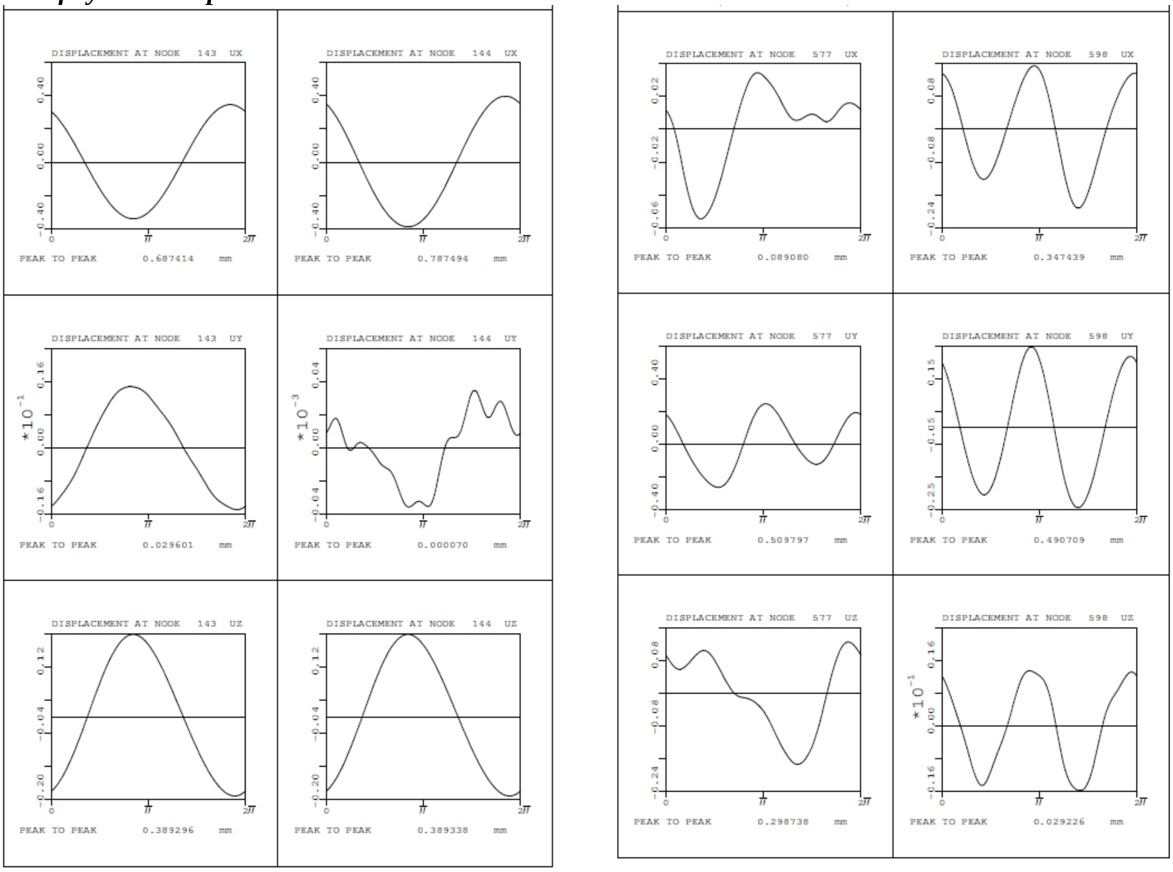


Fig.: 5.14 — Plot of Dynamic Displacements At Selected Nodes Peak-To-Peak in mm

Plot of dynamic responses at selected nodes

Table : 5.5 - Dynamic velocities RMS in mm/s

Nodes	UX	UY	UZ
297	3.37	28.32	7.47
298	3.38	26.38	6.49
595	13.45	25.64	0.98
318	0	25.64	0
597	14.72	25.64	1.19
598	16.98	25.43	1.63
594	15.85	24.7	1.1
599	17.42	24.67	1.91
145	24.56	0.87	10.66
622	23.96	9.28	1.64

Plot of dynamic velocities at selected nodes Peak-to-Peak in mm/s

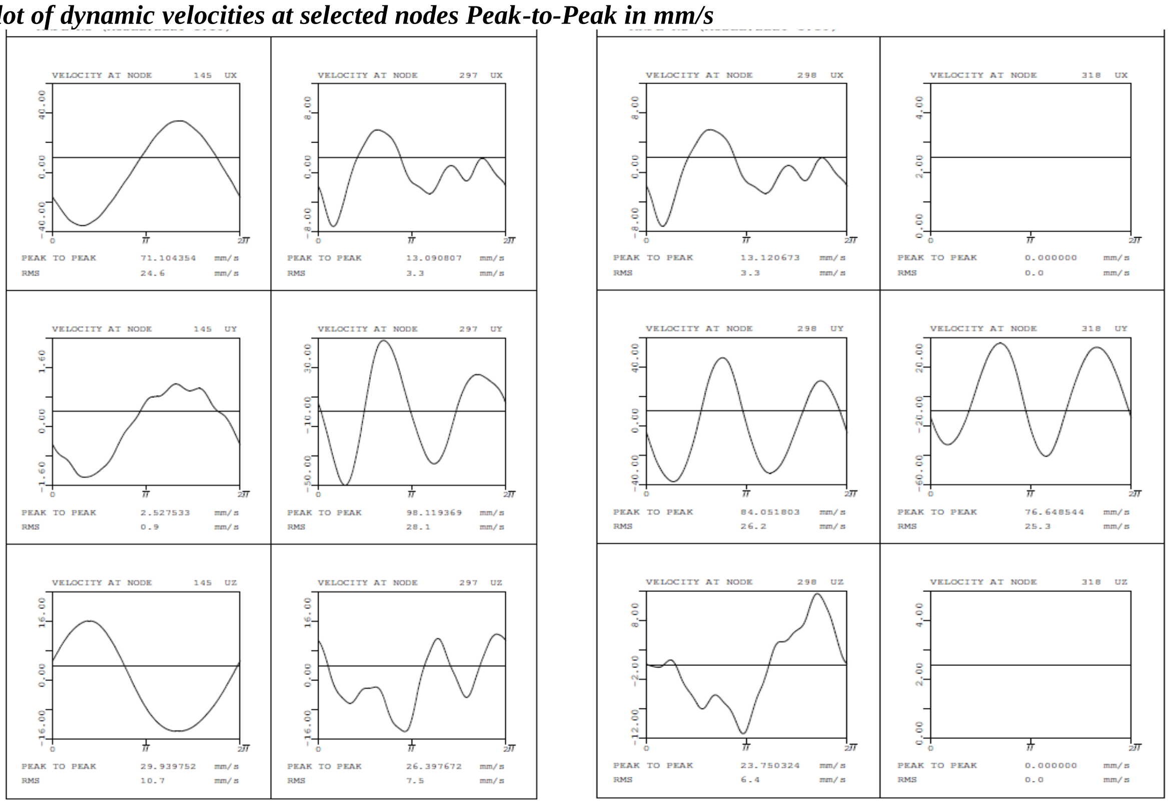


Fig.: 5.15 — Plot of Dynamic Velocities At Selected Nodes Peak-To-Peak in mm/s

Plot of dynamic responses at selected nodes

Table : 5.6 - Dynamic bending stresses Peak-to-Peak in N/mm^2

Node	Stress	Element
140	16.29	129
582	13.26	490
141	13.12	129
144	13.01	134
501	12.96	449
583	12.59	490
581	12.5	489
138	11.77	548
593	11.5	495
361	11.38	585

Plot of dynamic bending stresses at selected nodes Peak-to-Peak in N/mm²

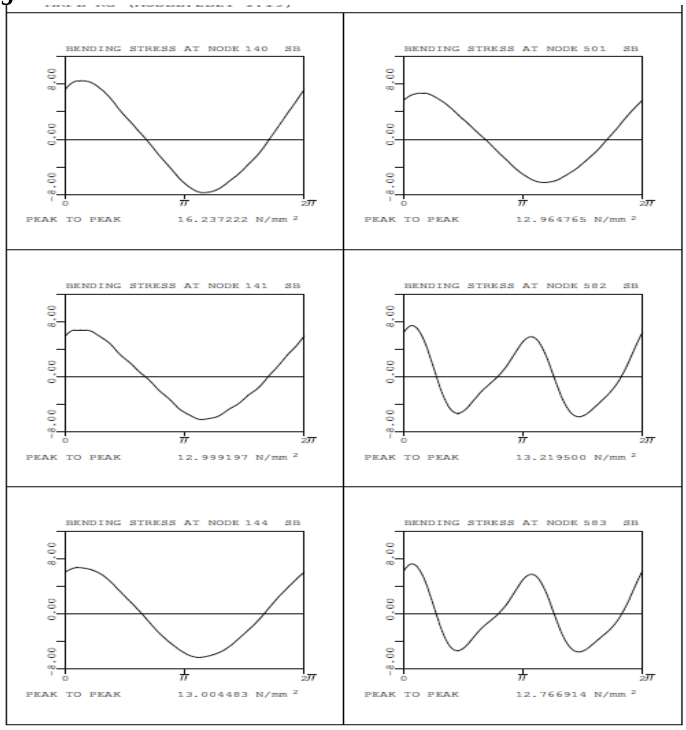


Fig.: 5.16 — Plot of Dynamic Bending Stresses At Selected Nodes Peak-To-Peak in N/mm^2

VI. EXPERIMENTATION

Experimental set-up and Instruments

The arrangement of the experimental setup used for the measurements is shown below

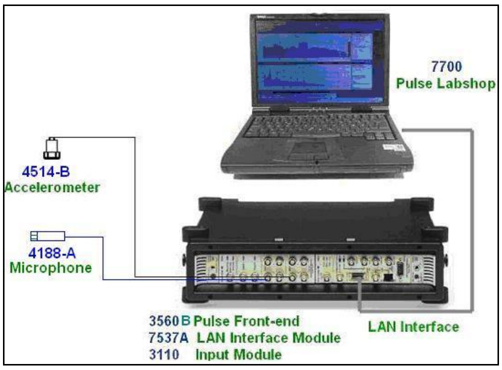


Fig.: 6.1 — Arrangement of Experimental Setup for Vibration Measurements

Table : 6.1 - List of Instruments used

Sr. No.	Instrument / Software	Remarks
1	B & K (Denmark) Make 5 channel PULSE Data Acquisition System (Analyzer) with BNC Connectors – Type 3560B with LAN Interface	Data Acquisition System
2	B & K (Denmark) Make IEPE Accelerometer – Type 4514B	Transducer for Vibration measurement
3	B & K (Denmark) Make Pulse FFT & CPB Analysis Labshop – Type 7700	Vibration & Noise Analysis Software

Experimental Procedure

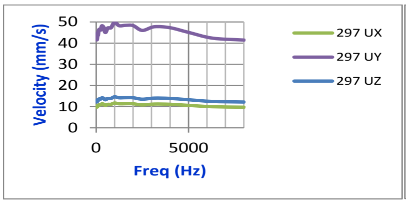
The vibration measurements were carried out at rated load (EOR case) since EOR case is the design case for this compressor. The Experimental set-up and Instruments details are shown in above chapter VI.
The measurements locations at different Nodes for vibration are shown in chapter V Sketches of Piping for Vibration Study.

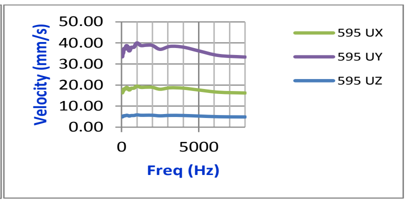
Vibration Measurements:

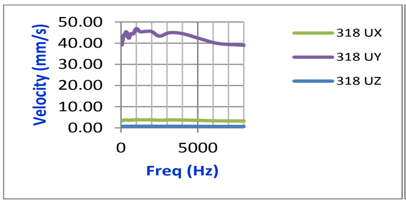
Only few nodes 297, 595, 318, 145 and 599 are considered for vibration measurements, which are having very high vibrations. The vibration measured in all three directions (vertical, horizontal & axial) in accelerations for 1/3rd Octave center frequency over a range of 0 Hz to 8 KHz.

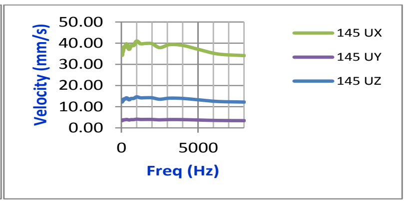
Result Analysis and Discussion

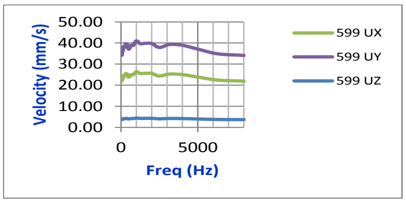
Results before Vibration Study (Experimental)









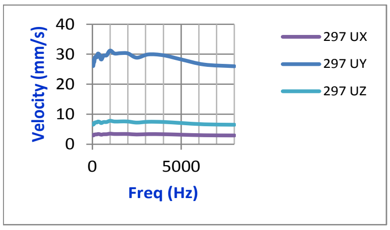


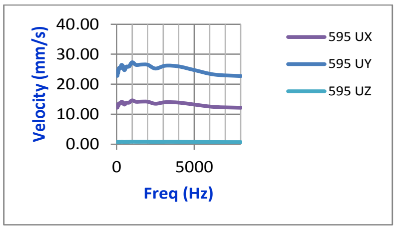
Nodes	UX	UY	UZ
297	10.09	45.33	13.21
595	18.93	36.47	0.84
318	0.38	42.85	0
145	34.64	2.89	36.15
599	23.84	37.15	0.34

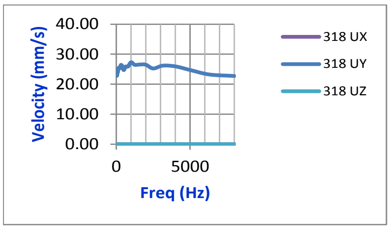
Table : 6.3 – Mean Values Before Vibration Study

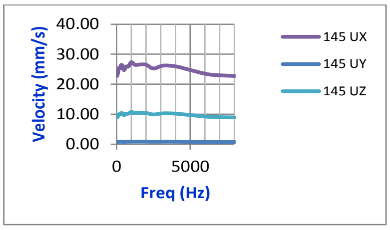
The above graphs are the experimental results before Vibration Study means before the modifications into the support. The graphs shown for the five nodes and found that the velocities are the much higher than the specified limit i.e. 30 mm/s in API 618. The above table 6.3 is shown the mean values of the graphs.

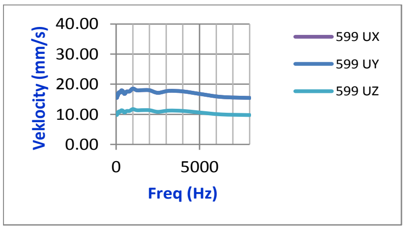
Results after Vibration Study (Experimental)











Nodes	UX	UY	UZ
297	3.21	27.52	6.88
595	12.90	24.08	0.77
318	0	24.08	00
145	24.08	0.77	9.46
599	16.34	16.34	0

Table : 6.3 – Mean Values after Vibration Study

The above graphs are the experimental results after Vibration Study means after the modifications into the support. The graphs shown for the five nodes and found that the velocities are the much lower than the specified limit i.e. 30 mm/s in API 618. The above table 6.4 is shown the mean values of the graphs.

Results after Vibration Study (Analytical)

Table : 6.4 - List of Nodes with Maximum Velocities RMS in mm/s after Pulsation and Vibration Study (Analytical)

Nodes	UX	UY	UZ
297*	3.26	28.14	7.55
298	3.27	26.24	6.37
595*	13.29	25.29	0.97
318*	0	25.29	0
597	14.54	25.29	1.18
598	16.78	25.08	1.61
145*	24.55	0.87	10.66
594	15.66	24.36	1.09
599*	17.21	24.33	1.89
577	3.28	22.97	10.39

*- nodes are used for comparison with experimental results

The model for this vibration study includes the suction, interstage and discharge side. The model contains the suction side starting at 6”-P-551093-B2A1-IT existing header, suction piping, interstage piping with cooler, KOD and PSV line, the discharge side with PSV-line and with the exiting header 4”-P-551107-B4A1-IH40. Relevant sidelines are also included in the calculation.

The natural frequencies could be calculated once for the whole system. For the dynamic response analysis we choose the deviation of the velocity of sound with the highest pulsations of the worst case. The details about the worst case are mentioned in chapter V. No change of piping layout is required except orifices which are already discussed for the pulsation study and also following support should be added into the piping layout.

- Support : X & Y at Node 345 : Refer Sketch
- Support : X & Z at Node 367 : Refer Sketch

After several calculations with the support modification all vibration results are within the limits.

The supports with its restricted direction of the last version are indicated in the attached isometrics. At piping parts it was necessary to implement additional supports to achieve satisfied results. The details of all supports (blocked translations and rotations) in calculation model can be found in the sketches of the system (chapter V).

VII. CONCLUSION

The investigation of the piping showed no severe problems with vibrations. One reason for this is the reduced pulsation level which is achieved from the modifications recommended in the pulsation study. The combination of the acoustic simulation with a mechanical analysis as defined in Design Approach 3 of API 618 is the content of the vibration study. Shaking forces induced by pressure pulsation excite the mechanical piping system. The vibration study determines the effect on the mechanical piping system and proposes measures to avoid stresses possibly leading to deformation or rupture by fatigue.

The finite element program ANSYS is used for modeling of the mechanical system. The model is built of several types of basic elements (e.g. pipes, beams, elbows, T-pieces) connected at node points. Modifications are proposed to meet agreed criteria of vibration as per API 618 5th edition 2007).

Since shaking forces are a result of the previous pulsation calculation, vibration study can only be done in combination with a preceding pulsation study.

Also the accuracy of the analytical solution had been validated by means of experimental results by using B & K Analyser for the measurement of compressor piping vibration.

REFERENCES

[1.] Shelley Greenfield and Kelly Eberle, New API Standard 618 (5TH ED.) And Its Impact On Reciprocating Compressor Package Design, Compressor TechTWO, 2008, pp. 55-67.

[2.] Paul Alves, Acoustical And Mechanical Analysis Of Reciprocating Compressor Installation - API 618, Compressor TechTWO, 2006, pp. 64-69.

[3.] James D. Tison and Kenneth E. Atkins, The New Fifth Edition of API 618 for Reciprocating Compressors – Which Pulsation and Vibration Control philosophy should you use?, Engineering Dynamics Incorporated Seminar Manual, San Antonio, Texas, 2008.

[4.] J. C. Wachel and J. D. Tison, Vibrations In Reciprocating Machinery And Piping Systems, Engineering Dynamics Incorporated Seminar Manual, San Antonio, Texas, 1999.
[5.] Enzo Giacomelli et al., Forced Response of Cylinder Manifold For Reciprocating Compressor Applications, 8TH Biennial ASME Conference on Engineering Systems Design and Analysis, Torino, Italy, 2006.
[6.] W. W. von Nimitz , Pulsation And Vibration Control Requirements In The Design Of Reciprocating Compressor And Pump Installations, International Compressor Engineering Conference, Purdue University, Indiana, USA, 1982.
[7.] Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services. API Standard 618, Fourth Edition , June 1995.
[8.] Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services. API Standard 618, Fifth edition, December 2007
[9.] http://www.betamachinery.com/
[10.] http://www.tno.nl/
[11.] Ansys Software