AIN SHAMS UNIVERSITY
FACULTY OF ENGINEERING
MECHATRONICS ENGINEERING DEPARTMENT

Analysis of Pressure Pulsations in Pipeline Networks to Prevent Vibration Induced Failures

A Thesis submitted in partial fulfillment of the requirements of the

M.Sc. in Mechanical Engineering

By

Ahmed Ali Mohammed Hassan Okasha

B.Sc., Mechanical Engineering, Mechatronics Section
Ain Shams University, 2010

Supervised by

Associate Prof. Dr. Tamer Elnady

Associate Prof. Dr. Adel Elsabbagh

Cairo – (2014)

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Statement

This thesis is submitted as a partial fulfillment of M.Sc. degree in Mechanical engineering, Faculty of Engineering, Ain Shams University.

The author carried out the work included in this thesis and no part of it has been submitted for a degree or qualification at any other scientific entity.

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Abstract

Sound and Vibration problems in Oil and Gas industry have been a major concern in the last decade. There is a need to solve performance problems caused by fluid machines in pipeline networks. Fluid Machines such as compressors and pumps produce pressure pulsations. These pulsations turn into shaking forces, which in turn excite vibrations in the connected piping system. High vibrations beyond the endurance limit of the material may cause damage to pipes, supports, and equipment. Moreover, if the source pulsation frequency coincides with one of the natural frequencies of the piping network, the network will incorporate resonance and consequently vibrations will be magnified. This might cause a damage to the foundation if the vibrations are not well controlled.

In this thesis, the indirect multi-load source characterization technique, commonly used to characterize internal combustion engine noise, is applied to characterize the acoustics of a reciprocating compressor and a screw compressor working at different operating conditions. The source characteristics are described by the source strength and the source impedance in the frequency domain. A linearity test for each source is carried out. If the source is linear, the two-port theory is valid and can be applied to predict the pressure pulsation at any given point in the system using the measured source data. The two-port technique is utilized to model the receiving system where the network can be divided into elements; each is described by a transfer matrix.

For the reciprocating compressor case, a pilot plant consists of pipes, bends and terminated by a vessel is constructed. The pressure pulsations are measured in different positions inside the network for two cases and are compared to simulations. One of these cases involves a pulsation suppression device that is designed according to the standards to attenuate the pressure pulsations in the pipeline network.

Keywords: Multi-load method - Source characterization - Linear source data - Piping vibration control.

Acknowledgment

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Contents

Chapter 1: Introduction

In Oil and Gas Industry, there are usually complex pipeline networks from the well to the storage vessels (Figure 1-1). This configuration can be found onshore and/or offshore installations. Majority of vibration problems occur in pipeline networks are attributed to pressure pulsations [1]. Fluid Machines such as compressors and pumps produce acoustical, pressure, or dynamic pulsations. Some of piping network elements such as pipe bends and pipe reducers act as a coupling mechanism. At these elements, reflection occurs and standing waves might be formed. The pulsation standing wave amplitude is dependent on the geometry, and the frequency, amplitude, and phase of the initial pulsation wave. According to these parameters, the standing wave will be either greater, or less than the initial travelling pulsation wave [1]. Shaking forces are then generated from the pulsation standing waves. The generated shaking forces therefore excite vibrations at bottles, scrubbers, piping and the foundation.

Figure 1-1. A complicated pipeline network installation at onshore upstream Oil/Gas plant. Web. 12 June 2014. https://www.mottmac.com/oil-and-gas/onshore-upstream.

If these vibration levels exceed the permissible limits, and are not well controlled, they can lead to various problems such as machinery downtime, damage to pipes, supports, foundations, explosions, and fires. Figure 1-2 shows the sequence of fatigue failure caused by the reciprocating compressor. As shown, for the reciprocating compressor, there is other forces generated plus the pulsation forces. However, in the research work here, the focus is on the pulsation forces only.

The consequences of excessive vibration might lead to catastrophic failures. For instance, a piping failure at a petrochemical plant in 1974 caused over of $114,000,000 damage costs [2]. Depending on the extent of the problem, the plant might be shut down for days or even months. Hundreds of thousands of dollars might be spent on parts replacement or on maintenance in the production line.

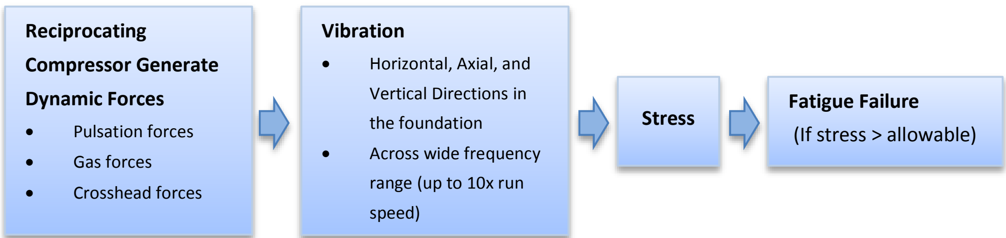

Figure 1-2. Vibration is the leading cause of machinery downtime (Due to excessive stress on different piping network elements)

The problem might be even worse. A pipeline network has several Mechanical Natural Frequencies (MNFs). At these MNFs, the structure reaches its maximum flexibility or minimum stiffness. If the source excitation frequency coincides with one of the piping networks' natural frequencies, the resulting excitations are magnified from 10 to 100 times [3]. This will lead to cyclic mechanical strain, and consequently will cause fatigue failure in the piping network, if their levels are beyond the endurance limit of the material. As reported in reference [4], "*ten million cycles of stress levels in excess of the endurance limit will cause a fatigue failure*". For example, vibrations beyond the limit, at 15 Hz will reach 10 million cycles in approximately seven days. It is very clear, that for higher frequencies, the fatigue failure will occur much faster and consequently is more dangerous.

Pulsation amplitudes can exceed 10% of line pressure and the shaking forces can be significant. As basic rule of thumb, piping vibrations can usually be controlled with normal good engineering practices with regard to clamps and supports, if the pulsation induced shaking forces are less than 2225 N (peak-to-peak) in ground level piping, and 890 N (peak to- peak) for elevated piping, such as in pipe racks, offshore platforms,… etc. Since the shaking force is proportional to the pressure pulsation amplitude and the projected flow area of the pipe, pulsation levels must be controlled to achieve acceptable vibrations. There are four methods for evaluating acceptable vibration levels in piping systems. These methods include "Allowable Vibration Amplitude *Vs* Frequency", "Vibration Displacement Amplitude - Stress Relationship", "Vibration Velocity Amplitude - Stress Relationship", and "Measured Dynamic Strain" [4] .

High Pulsations have several undesired impacts on metering accuracy, system pressure drop, and valve performance. Pressure pulsations are accompanied by intermittent flow. Excessive flow fluctuation at flow metering devices causes unstable or inaccurate readings. Metering devices readings are usually used to determine the amount of Oil/Gas to be sold. Thus, inaccurate readings can result in substantial financial losses. Another consequence of the intermittent flow is producing more pressure drop due to the squared relationship between flow rate and pressure drop. In addition, improper valve openings, and closing events and valve flutter are typical indications of high pulsations [1].

As indicated in reference [5], static and dynamic pressure drop should be kept into consideration while calculating the reciprocating compressor performance. Dynamic pressure drop might exceed the static pressure drop in some cases. Therefore, it is important to determine accurately the dynamic pressure drop. Howes [6] ensures that dynamic pressure drop is important to consider when there is a large oscillating flow at flow resistance elements. An increase in pressure drop leads to a decrease in compressor capacity or to an increase in required power. Thus, reducing the flow fluctuations is important for minimizing the pressure drop.

1.1. Pulsations in compressors

There are two broad categories of compressors as shown in Figure 1-3, which are positive displacement and dynamic compressors. Dynamic compressors impart a velocity to the fluid, which is then converted to pressure. Positive displacement compressors use pistons, screws or vanes to produce compressed air through reducing its volume.

In this thesis, we are concerned with studying the sources of pulsations in both reciprocating compressors and screw compressors (marked red in Figure 1-3). Each source has different pulsation generation mechanism that will be discussed briefly in the following subsections.

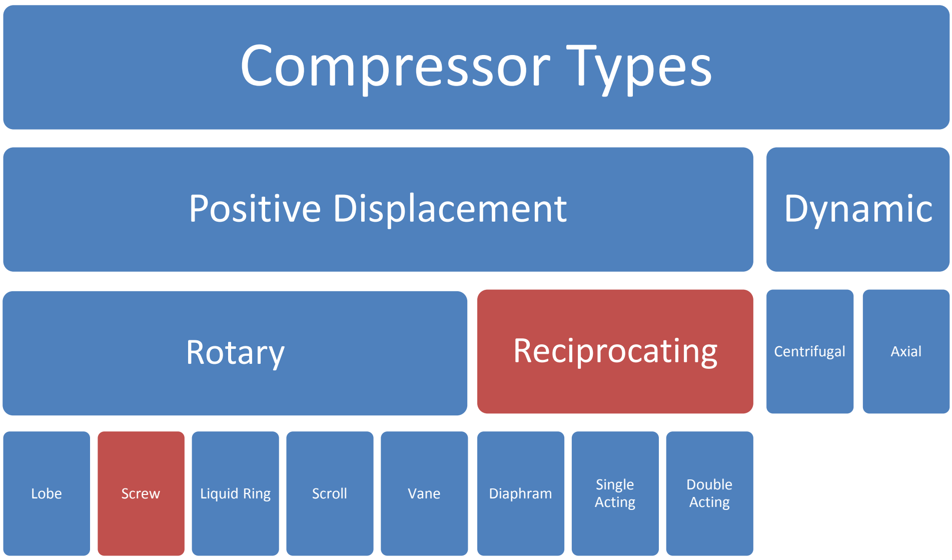

Figure 1-3. Compressors types include two broad categories, positive displacement and dynamic compressor.

1.1.1. Pulsations in reciprocating compressors

Pressure pulsations are generated by the intermittent flow that comes through the discharge valve. Regions of compressions are formed and released into the attached piping network. The discharged pressure waves are sent into the system regularly, and are always traveling away from the source at the speed of sound of the fluid [1].

Figure 1-4 shows a schematic for a single-cylinder reciprocating compressor. Ideally, piston velocity is a sine wave motion. The finite length of the connecting rod is behind the deviation from this motion. As the ratio of the connecting rod length L to the crank radius R increases, the motion becomes close to the sinusoidal shape. As flow is equal to the piston velocity multiplied by the piston area, and pressure pulsation is proportional to the flow, the pulsation has the same wave shape of the piston velocity [3].

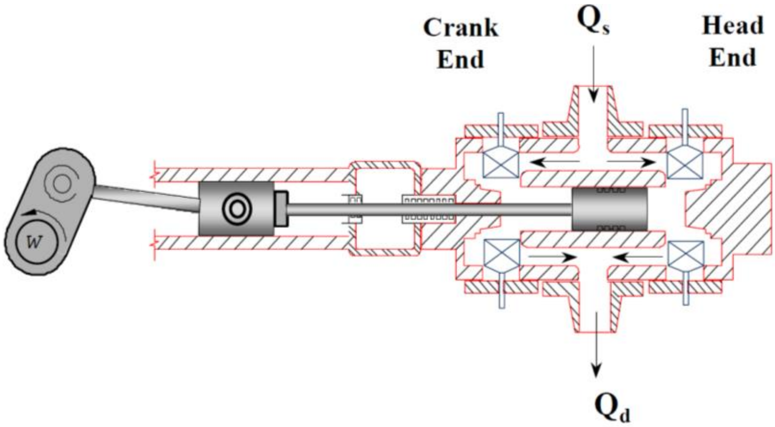

Figure 1-4. Reciprocating compressor schematic drawing where Q_s is the suction flow and Q_d is the discharge flow. Photo from reference [3]

For a "perfect" double-acting cylinder, every one cycle will have two non-identical flow slugs. Accordingly, odd harmonics cancel each other out. The two slugs are not identical because of the differences in head/crank end clearance and finite length connecting rod. Figure 1-5 show the flow amplitude for a double-acting compressor cylinder in the angle domain, and Figure 1-6 shows the frequency spectrum for the angle domain curve. The flow curve in the angle domain show sudden change in the flow while this is not the case in reality.

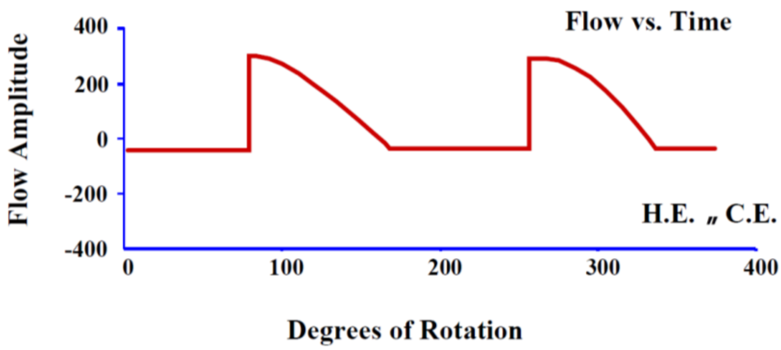

Figure 1-5. Flow curve for unsymmetrical double-acting compressor cylinder discharge valve. Photo from reference [3]