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Potential of Savings in the Design of Industrially Adequate Pipework for Reciprocating Compressors

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ABSTRACT

Standards stating "good practice" have had an undeniable influence on the demand for extensive dynamic analysis in the design of pipe systems for reciprocating compressors. The merits of such an approach have to be acknowledged in complex systems or when difficult service conditions must be taken into account. But in many cases it is possible to use empirical analytical methods which are less expensive but still give satisfactory results. To draw attention to this fact two typical examples of delivered compressor plants are given. The potential of savings is of the order of several $10000 per case. The selection of the most economical approach should be made rather on grounds of sound engineering judgement than by uncritical use of written codes.

INTRODUCTION

To all those who are familiar with the engineering of industrial compressor plants it is well-known that a pipework system is not always easy to design because of gas pulsations and consequent mechanical vibrations which occur mainly in the neighbourhood of the machines.

However techniques have been developed and their application has been recommended in literature, which help to analyse a system during the design stages. Such guide lines are specified in the API 618 1) standard as an example. Although this code is intended for the refinery industry it is often used for other applications as well. Hence most manufacturers of reciprocating compressors who are in international business find in inquiries a clause "as per API 618" quite often.

In general this clause is then to be applied not only to the design of the compressor but also to the system engineering work mostly to be included in a quotation which consequently should involve API 618 pulsation and vibration control methods.

BRIEF SURVEY OF API BASIC TECHNIQUES FOR CONTROL OF PULSATIONS AND VIBRATIONS

If we go through the pertinent paragraphs of API 618 we find descriptions of three different "Design Approaches", a key which helps to identify the recommended appropriate approach for a given case together with a statement that the purchaser should specify the type of design approach.

What do they impose on the supplier of such services?

Approach 3:

- Mathematical modeling or simulation by eletric analog techniques of the compressor cylinder action on the gas in the equipment system involved.

- Consideration of mechanical and acoustical system responses.

- Determination of the amplitudes and the spectral frequency distribution of pressure pulsations.

- Calculation of pulsation induced forces and mechanical reponse of the piping system and the pertinent supporting structures.

- Determination of vibration amplitudes, frequencies and consequent cyclic stresses and optimize design to satisfy API 618 limits.

Approach 2:

Same as approach 3 but without mechanical analysis of pipe system and supports.

Approach 1:

- Calculation of pulsation suppression devices to meet given pulsation levels using proprietary and/or empirical analytical techniques.

- Analysis of pipework for critical lengths to avoid resonance with acoustical harmonics.

It is easy to recognize that the amount of work and the cost involved vary considerably according to the kind of approach to be applied. Let us therefore have a closer look at the principles on which selection as per API 618 is based.

In a first step a case is classified in one of two groups according to various distinctive features such as:

discharge pressure
power input
likelihood of interactions with adjacent machines
alternative service with gases of different molecular weights

Within each of the two individual groups further criteria apply such as number of stages and number of individual cylinders which help to find the numerical designations of the recommended approach. We show this in Table 1.

For further considerations two relevant results of the above process of selection are relevant and should be kept in mind:

a) Machines having at least two compressor stages qualify invariably for Approach 3 regardless of power and pressure or other criteria.

b) Approach 1 is only applicable for machines which do not have more than one stage and not more than two cylinders.

SIGNIFICANCE OF PROPRIETARY TECHNIQUES AND COMMERCIAL ASPECTS

In the second part of our paper we should like to elaborate on the extent to which such engineering standards really are used in practice in our work. We therefore have analysed our own engineering methods from beginning 1985 through to the end of 1988.

We first picked out of our reciprocating compressor program the family of the labyrinth piston compressor orders which we shipped to customers during the above mentioned period.

From the total number of orders 88 % fall within the definition of API 618 Approaches 2 and 3, and 12 % would qualify for Approach 1.

We show this statistically in Figures 1 and 2 which also inform about the types of gas for which the machines are used. It is furthermore of interest to know that all these orders had imposed at least a part of the responsibility for the engineering of pipework on us. This is not unusual and is one of the reasons for which we have been keeping our physical means and specialized human resources at a level which enables us to make pulsation and vibration analysis independently, either on a classical analog simulator or by means of digital methods. In this context it is of some interest to note that a small part of only 14 % of the "Approach 3 and 2 candidates" have really gone through an analysis to the depth as recommended by API 618.

Whether another manufacturer would have worked on a comparable basis has not been investigated, but we have analysed another group of our own products. This branch is producing reciprocating compressors having oil lubricated or dry running cylinders and pistons with positively contacting piston rings. These machines including balanced opposed and vertical ones are designed to serve the process gas segments in general. The relationship here is:

15 % of the candidates have really been processed as per API - Approach 2 or 3.

It is obvious that in our opinion the real needs for an in-depth dynamic study cannot be identified by the uncritical application of a code.

A competent manufacturer will consider addional means as well, such as :

- comparison with other cases, taking advantage of similarities
- accumulated experience, standardization
- selective application of analytical work
- negotiated compromises
- innovation

For better understanding we present an example:

A labyrinth piston compressor plant for oxygen built in 1981 (Fig. 3 and 4)

The operating conditions as per design are the following:

volume at suction	6330 m^3/h
suction conditions	1,04 bar abs/22,6°C
discharge pressure	25,90 bar abs
shaft speed and input	420 rpm/1044 kW
number of cylinders	4
number of stages	3

Based on the above premises and API 618 - recommendations this case would fulfil Approach 3 criteria.

In fact a much more modest analysis was selected.

The following critical components were calculated:

1) Volumes of 6 individual pulsation dampeners with a residual pressure amplitude of 2% peak to peak as a target.

The formula used was: $$V = \frac{V_z \times y}{(1+\frac{\Delta p}{100})^{1/\varkappa} - 1} \qquad 2)$$

V_z displaced volume of all cylinders per stage
Δp pressure amplitude in percent of line pressure
y coefficient taking into account:
- double or single action of pistion
- relative lengths of stroke during opening phase of valve (suction or discharge).

$\varkappa$ isentropic exponent

2) Diameters of pipe sections between cylinders and pulsation volumes.

They were calculated according to the following formula:

$$F_R = F_P \times \frac{v_P}{v_G}$$

v_G = velocity of gas in pipe
v_P = linear piston speed
F_P = total of piston areas which are simultaneously active in relation to the pipe considered
F_R = section of pipe considered

The velocity of gas is selected primarily according to the density, but other parameters as well are to be considered, here the chemical reactivity of oxygen for example. We generally adhere to rather conservative v_G - values, which in this case are 16, 14 and 7 m/s in the three stages 1, 2 and 3.

3) Lengths of pipes between cylinders and dampeners.

To avoid critical pipe lengths in resonance with acoustical harmonics we endeavoured to have short pipes. "Short" in this case means:

. L to be considerably shorter than $\frac{a}{4f}$

L = lengths of path of gas from valve to pulsation dampener
f = number of actions per second of piston
a = velocity of sound

In our example, with all pistons double acting,

L critical ($= \frac{a}{4f}$) was approx. 6 to 7 m.

Thinking back to terminology of API 618 we had been working in this example somewhat along the lines of Approach 1 with a very positive result. This plant went into operation with no complications.

Cost estimates for the above example: extra engineering money spent for work in connection with analysing pulsation and vibration control is below $3000.--. In contrast to this a full size API Approach 3 would have generated extra engineering hours equivalent to approx. $40'000 - 60'000. The customer would also spend man-hours on the survey of processing and critical approval of results.

A CASE FOR ENHANCED INVESTMENT IN THE ANALYSIS OF PIPEWORK

Whereas the above example may be typical for a majority of cases we also meet such problems where extensive analytical work is a necessity. For instance when difficult conditions accumulate as described hereafter.

The order encompassed a packaged unit for the compression of natural gas boil-off from suction conditions 1.04 bar abs/-160°C to a discharge pressure of 23.3 bar abs, volume flow 5250 m^3/h. The final installation was in an existing LNG terminal with severe restrictions on site work such as welding, mainly for safety reasons. The low gas temperature called for heavy insulation of pipework. It was to be applied after arrival at site, before initial start-up. This would have rendered any subsequent corrective work impossible.

One further handicap is to be seen in the thermal conditions to which the pipework near the compressor was subject. As a consequence considerable shrinking and expansion during transient starting and shut-down process had to be taken into account. This was rendered more difficult by the application of low temperature grades of materials which typically have high coefficients of heat expansion.

We show in a picture the three stage Laby compressor packaged together with all appurtenances as it left our works. (Fig. 5)

As to be expected this gas pipe system was subject to an in-depth pulsation study in combination with an analysis of the mechanical dynamic response. The stresses in critical areas had to be analysed taking into account dynamic as well as heat deformations, all this in parallel with the design work in progress.

To illustrate we pick out the suction pipe upstream of the compressor and, as one result of the combined analysis, the system of supports to absorb all static, dynamic and the needs for heat deformation (including a fixed point which had been defined by the customer in regard to his adjacent pipework). (Fig. 6)

Again we can report a trouble-free operation of this pipe system right from start-up. This demonstrates in a more demanding case the high virtues of analytical tools which engineers have today. The high investments however are justified only if one is willing to invest an equal amount of care in the mechanical details involved, which often are less spectacular and risk being overlooked. (Fig. 7)

CONCULUSIONS

Tools are available to simulate the dynamic behaviour of compressor pipework. Commercial aspects prevent engineers from making unreserved use of these means. Uncritical application of codes to regulate such conflicting interests can impede proprietary techniques which offer a potential to minimize investments. Sound workmanship during mechanical construction is a complementary obligation to all investments in any analytical work.

REFERENCES

1) API Standard 618 (1986)
2) Lyman F. Scheel "Gas and Air Compression Machinery" (N.Y. 1961)

Table 1:

Criteria to find numerical designation of approach as recommended by API 618:

group classification	< 1000 psig < 500 HP			> 1000 psig > 500 HP		
2 or more stages	-	3	3	-	3	3
one stage	1	1	2	2	2	3
number of cylinders	1	2	>2	1	2	>2

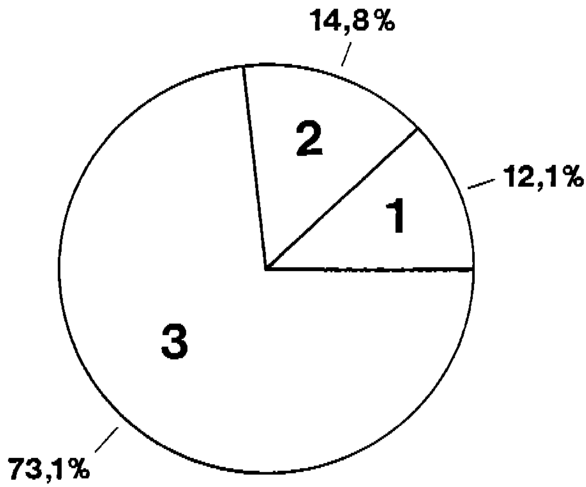


Fig. 1: Appropriate design approaches 1,2, and 3 according to API 618 guidelines when applied to Labyrinth-Piston Compressor orders 1985 to 1989.

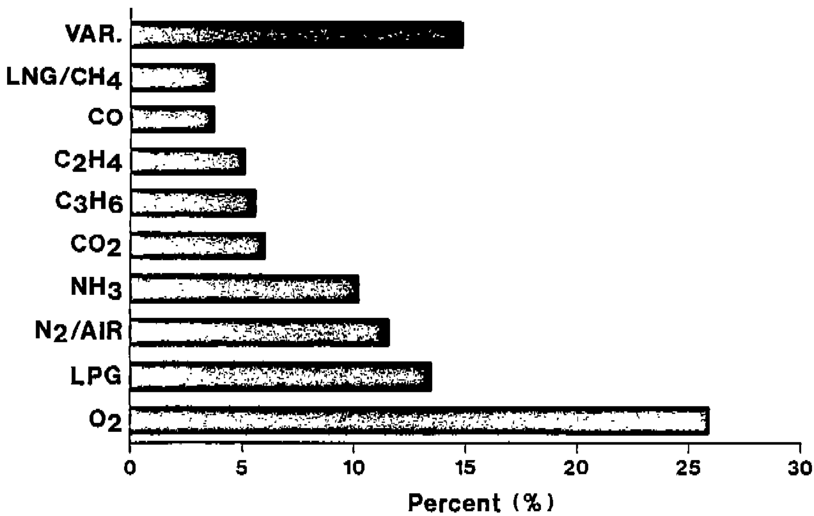


Fig. 2: Gas segments for which the orders 1985 to 1989 were used.

Fig. 3:

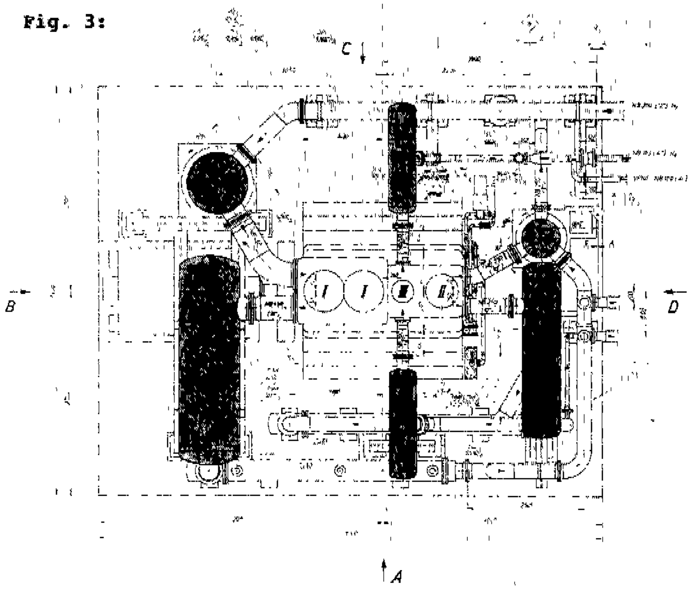
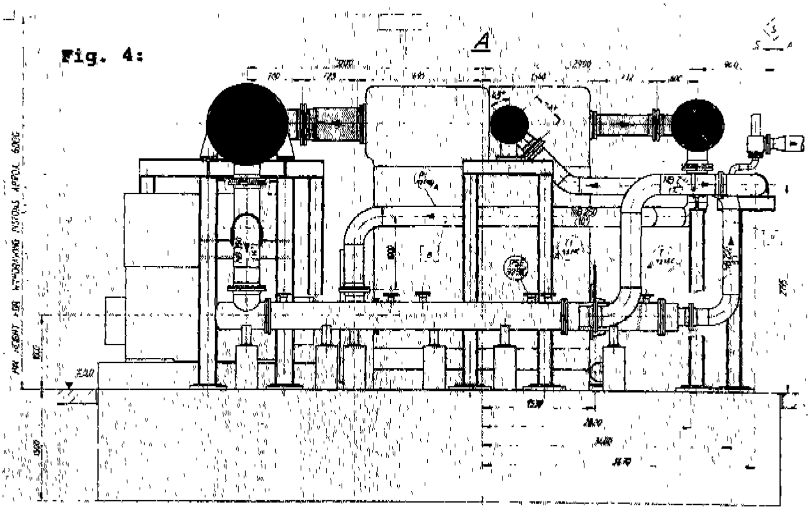


Fig. 4:



Pipe system to Labyrinth-Piston Compressor built for dry industrial gas in 1981.

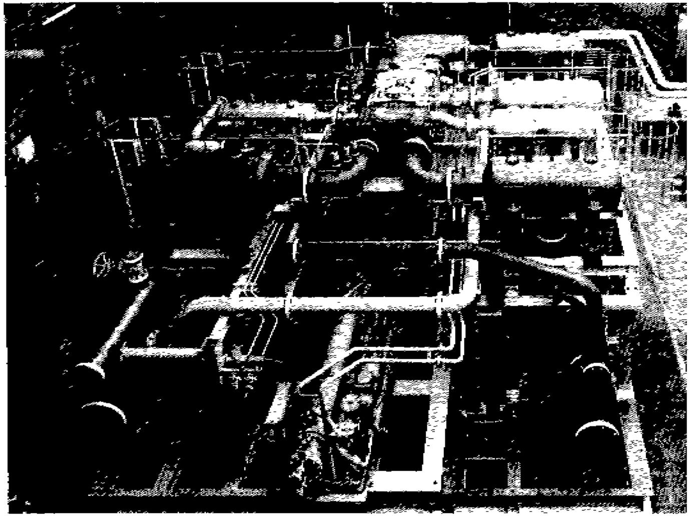


Fig. 5: Labyrinth-Piston Compressor Package for LNG boil-off, state of pipe system before insulation.

Fig. 6: Suction pipe upstream of 1st stage with supports, temperature span from ambient (standstill) to operating conditions: +30 to -160 ^{o}C (+86 to -256 ^{o}F).

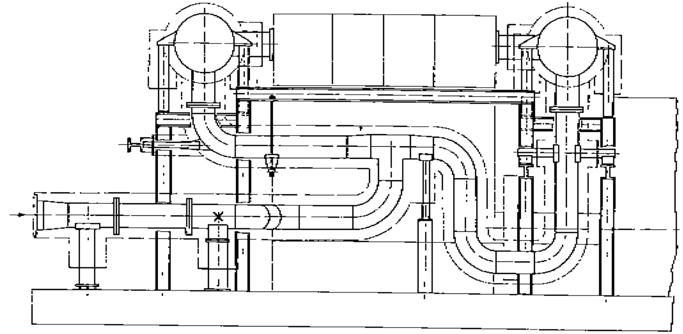
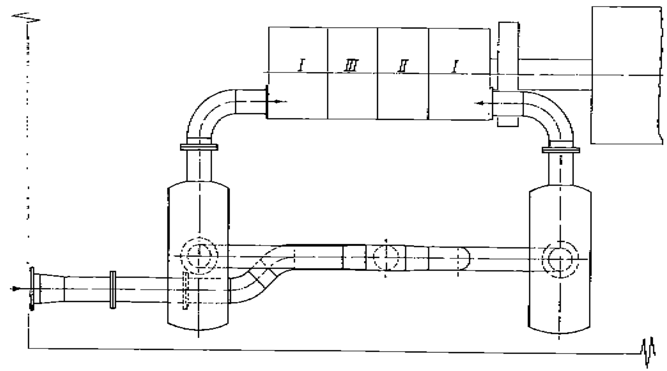


Fig. 6:



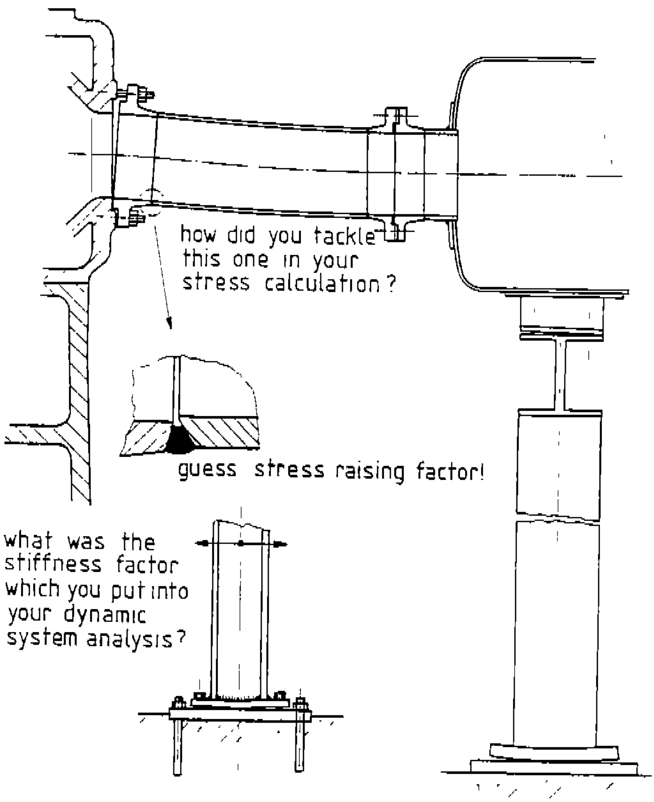


Fig. 7: Workmanship is a complementary factor for success.