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An Empirical Method for Designing Discharge and Suction Pipes of Small Reciprocating Air Compressors

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INTRODUCTION

Design procedures for piping systems of large compressors usually take detailed account of the pulsating flow in order to achieve reliability and good efficiency. The mathematical models for handling the pulsating flow are well established (see, for instance, Ref. 1) but require extensive computer programs for numerical solution.

The designer of small compressor systems, therefore, usually has to do without detailed mathematical analysis; and this often results in piping and valve failures and in low operating efficiency. A simple empirical design method for avoiding such problems is outlined in this paper. This method is based on the author's own extensive measurements of pressure pulsations in compressor pipes (Ref. 2) and on various published data.

DESIGN OF THE DISCHARGE PIPE

The first objective is to assure that the length of the pipe is such that there is no gas column resonance at the normal operating speed. Such resonances would lead to excessive pressure pulsations which cause efficiency losses and can cause valve failures and excessive pipe vibration.

In contrast to large compressor systems, the discharge pipes of small compressors are usually so short that their lowest natural frequencies are well above the operating speed. For the simple system shown in Figure 1(a), the lowest natural frequency of the gas column between the compressor and the receiver can be estimated from Eq. (1):

$$n_e = \frac{15\ a}{L} \qquad (1)$$

where n_e is the lowest (quarter wave length) natural frequency, in cycles per minute.

a is the speed of sound, in m/s.

L is the pipe length, in m.

The lowest natural frequency depends only on the length of the pipe and on the speed of sound which, in turn, depends only on the temperature of the gas but is independent of the pressure. For a discharge gas temperature of 150°C, which is typical for air compressors, the speed of sound is 412 m/s so that Eq. (1) reduces to $n_e = 6180/L$. This relationship is shown graphically in Figure 1(b). This figure and Eq. (1) should be very useful for avoiding resonances in simple piping systems. For more complicated, branched piping systems, see Ref. 2 and Ref. 3.

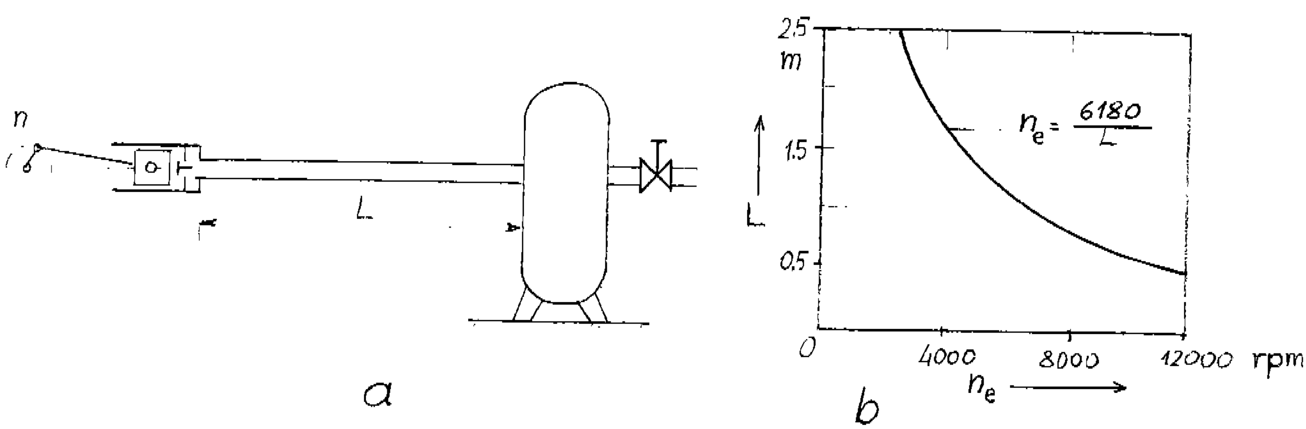

Fig. 1

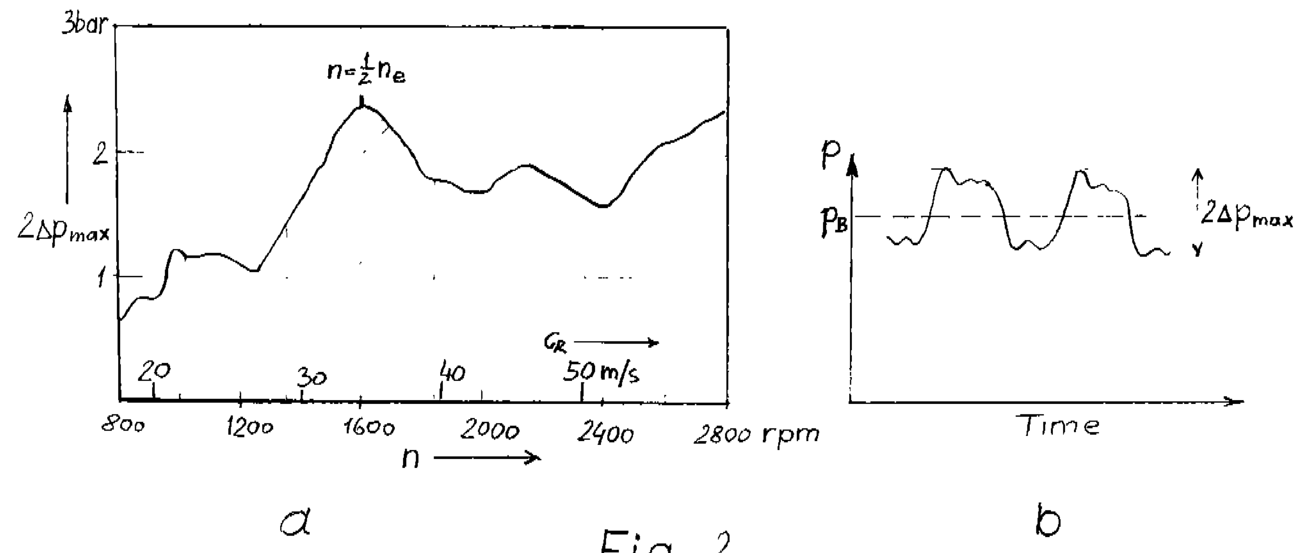

Fig. 2

The actual operating speed of the compressor, n, should not be within ± 15% of n_e. Since gas resonances in the discharge line can also be excited by the second harmonic component (and to a lesser extent higher harmonics) of the pulsating gas flow of the compressor, speeds for which n/n_e = 1/2 or 1/3 etc. should also be avoided.

Figure 2(a) shows the peak-to-peak pressure pulsation amplitude in a discharge pipe of a small single stage V-type air compressor as a function of speed. In the cross-hatched speed range, pulsations are excessive due to resonance. In this particular case, the piping design was poor both with respect to length and with respect to diameter. The resonance occurs at n/n_e = 1/2 and is discussed in more detail in Ref. 2. The curve in Figure 2(a) shows that, outside the resonant speed range, the pulsation amplitude is roughly proportional to speed. Figure 2(b) shows a typical non-resonant pressure wave form which results from the presence of many harmonic components.

The second objective in discharge piping design is to keep the peak-to-peak amplitude of the pressure pulsations, $2\Delta p_{max}$, within reasonable limits. Experience shows that $2\Delta p_{max}$ should preferably not exceed 10% of the absolute pressure, p_B, in the discharge receiver and certainly not exceed 20% of p_B for both reliability and efficiency reasons.

This part of the proposed design procedure is based on the empirically established fact (see Ref. 2 and 3) that pressure amplitudes, for given mean pressure conditions, are roughly proportional to compressor displacement and inversely proportional to piping cross sectional area. The proposed off-resonance design procedure thus consists of the following three steps:

1st Step: Specify the maximum acceptable peak-to-peak pulsation amplitude $2\Delta p_{max}$ in the pipe as a fraction of the absolute pressure p_B in the discharge receiver:

$$y = \frac{2\Delta p_{max}}{p_B}; \text{ (usually, } y = 0.1 \text{ to } 0.2)$$

2nd Step: Calculate the mean piston speed c_m in m/s:

$$c_m = \frac{s\ n}{30}$$

where s is the stroke in m.

n is the compressor speed in revolutions per minute.

3rd Step: Calculate the required pipe cross sectional area A_R in m^2:

$$A_R = \frac{c_m\ \pi\ A_{Kb}\ k\ C'\ f}{2\ y\ a} \qquad (2)$$

where A_{Kb} is the cylinder cross-section in m^2

k is the isentropic coefficient ($k = 1.4$ for air)

C' is an empirical factor, depending on compressor stage pressure ratio $\Pi = p_2/p_1$, see Figure 3

f is a factor which takes into consideration the number z of cylinders working into the discharge pipe. From experiments, $f = \sqrt{z}$

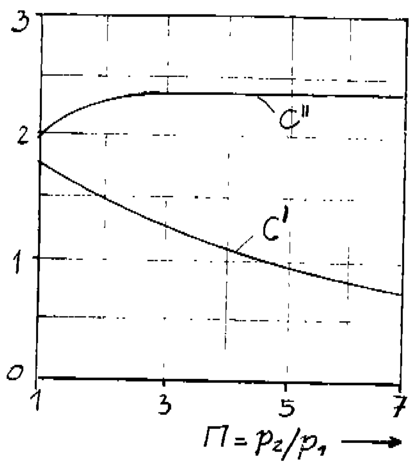

Fig. 3

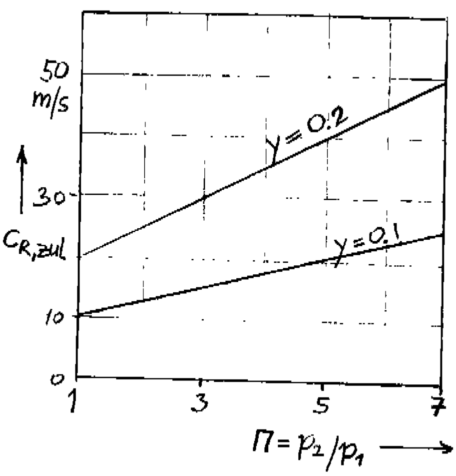

Fig. 4

Eq. (2) can be transformed so that one can calculate immediately the permissible velocity factor, $c_{R,zul}$ in m/s for values of a = 412 m/s, k = 1.4, and f = 1 (i.e. z = 1) one gets:

$$c_{R,zul} = c_m \frac{A_{Kb}}{A_R} = \frac{187\,y}{C'} \qquad (3)$$

$c_{R,zul}$ is the mean piston speed multiplied by the ratio of the area of the piston to the required area of the pipe. Figure 4 shows $c_{R,zul}$ for $2\Delta p_{max}$ = 10% and 20% of the discharge receiver pressure p_B (i.e. for y = 0.1 and 0.2).

Empirical values of C' for use in Eq. (2) and (3) are given in Figure 3. These values are applicable to piping systems which are not in resonance. If there is a gas resonance in the pipe, the pressure pulsation amplitude can be estimated from Eq. (2) by substituting C" for C' and solving the equation for y. An empirical curve for C" is also given in Figure 3. It should be emphasized, however, that resonances are very undesirable and should be avoided by proper selection of the pipe length as described above.

Two numerical examples will illustrate the use of the entire procedure.

Example 1

Design of the discharge pipes of a two-stage air compressor with the following data:

speed:	n = 1500 rpm
stroke:	s = 80 mm
bore:	1st stage, D_I = 160 mm, 2nd stage, D_{II} = 100 mm
discharge pressure:	1st stage, 3 bar absolute 2nd stage, 9 bar absolute
length of pipes:	1st stage to heat exchanger, 0.5 m 2nd stage to vessel, 1.3 m

Design of 1st Stage Discharge Pipe:

- Check resonance frequency (fig. 1):
  n_e = 12000 cpm; no danger of resonance.
- Specify maximum pressure pulsation amplitude: y = 0.1 (a low value!); i.e., we allow $2\Delta p_{max}$ = 0.1 x 3 = 0.3 bar
- Calculate mean piston speed
  $c_m = s\,n/30 = 0.08 \times 1500/30 = 4$ m/s
- From Figure 3, C' = 1.25 for $p_2/p_1 = \Pi = 3$
  $$c_{R,zul} = \frac{187\,y}{C'} = \frac{187 \times 0.1}{1.25} = 15 \text{ m/s}$$
- Calculate required pipe area:
  $A_R = A_{Kb}\, c_m/c_{R,zul} = 0.0053 \text{ m}^2 = 53 \text{ cm}^2$
- Required pipe diameter: $D_{I,R}$ = 82.6 mm

Design of 2nd Stage Discharge Pipe:

- Check resonance frequency (fig. 1):
  n_e = 5000 cpm; no danger of resonance.
- y = 0.1, $2\Delta p_{max}$ = 0.1 x 9 = 0.9 bar
- A similar calculation as for the 1st stage gives $D_{II,R}$ = 51 mm dia

Example 2

Check the piping design for an existing small air compressor:

Discharge Pressure: 7 bar (absolute)

speed: $n = 3000$ rpm
stroke: $s = 50$mm, bore: 50mm
pipe length: $L = 1.1$ m
pipe diameter: $D = 15$ mm

- $n_e = 6000$/min (fig. 1) This is dangerous, because $n = 1/2\ n_e$! Pipe must be shortened !
- $c_m = 5$ m/s
- $c_R = c_m \quad A_{Kb}/A_R = 50$ m/s
- from Eq. (3), assuming that pipe is shortened
  $y = c_R \quad C'/187 = 50 \times 0.75/187 = 0.2$

This means that pulsations will be very high ($2\Delta p_{max} = 0.2 \times 7 = 1.4$ bar) even if pipe is shortened to avoid resonance. If pipe is not shortened, resonance will result in even higher pulsation amplitudes which can be estimated by using C" instead of C' in Eq. (3):

$$y = c_R \quad C''/187 = 0.65$$

$$2\Delta p_{max} = 0.65 \times 7 = 4.5 \text{ bar !}$$

Pressure pulsations of this magnitude are likely to be destructive.

Conclusion: Pipe must be shortened and area should be doubled (to get a lower y, as y = 0.2 is too high)

DESIGN OF THE SUCTION PIPE

Frequently, the suction and discharge pipes of a compressor stage are made of equal size. This is acceptable provided that the discharge line diameter is properly chosen so that y_d is equal or smaller than 0.1. The relative pulsation amplitude in the suction pipe will then be somewhat larger than that in the discharge pipe because the intake flow to the cylinder extends over a longer portion of the stroke than the discharge flow and thus excites the gas in the suction pipe more effectively. The absolute pulsation amplitude in the suction pipe, however, will be lower than the absolute pulsation amplitude in the discharge pipe because the absolute pressure on the suction side is lower than on the discharge side.

If desired, Eq. (2) can be used to separately calculate the suction pipe diameter required to meet a specified suction pressure pulsation limit. For this purpose $C' = 1.75$ should be used as read from Figure 3 for $\Pi = 1$.

It should also be remembered that the speed of sound in air at 30°C, which is typical for suction conditions, is approximately 350 m/s. Thus, Eq. (2) and (3) reduce (for f = 1) to:

$$A_R = A_{Kb} \quad c_m/c_{R,zul} = A_{Kb} \quad c_m/91\ y \qquad (4)$$

$$c_{R,zul} = 91\ y \qquad (5)$$

Because of the lower speed of sound, Figure 1(b) cannot be used for checking the suction pipe length for resonances. Instead, Eq. (6) should be used to calculate the lowest natural frequency in cycles per minute:

$$n_e = \frac{5255}{L} \qquad (6)$$

For two-stage compressors, the effective length of the suction pipe of the second stage for use in Eq. (6) is usually the length from the inter-cooler to the suction valve. This is based on the assumption that the inter-cooler has a large enough volume to decouple the discharge pipe of the first stage from the suction pipe of the second stage. If this is not the case, the two pipes are acoustically coupled and a more sophisticated analysis is required to determine the lowest natural frequency.

It is generally desirable to provide a suction cavity of substantial volume in the cylinder head in order to minimize pressure pulsations. If this is done, the pipe length for use in Eq. (6) should be the "effective" pipe length which can be estimated by adding a length correction to the actual pipe length. This length correction is that length of pipe which contains a gas volume equal to the volume of the suction cavities in the cylinder head.

As far as the design of the suction pipe of the first stage is concerned, some comments on the selection of suction filters may be useful. Frequently, such filters are attached directly to the cylinder head. The filter has to be large enough to prevent excessive pressure drop. Pressure drop data found in the catalogs of filter suppliers are usually based on uniform flow. If a single filter handles the flow for many cylinders, the flow will, indeed, be more or less uniform. If the filter handles the flow of a single cylinder, however, the flow will be pulsating. For such applications, appropriate pressure drop information should be obtained from the filter manufacturer.

REFERENCES:

[1] Benson, R.S., and Uçer, A. S.: Pressure Pulsations in Pipe Systems with Multiple Reciprocating Air Compressors and Receivers. Journal Mechanical Engineering Science, Vol. 15, No. 1, 1973.

[2] Böswirth, L.: Messungen an einem schnellaufenden Kolbenverdichter. Konstruktion Vol. 26 (1974) No. 4, p. 143-151.

[3] Frenkel, M. I.: Kolbenverdichter, translated from Russian by W. Christian, Berlin 1969.