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INFLUENCE OF SUCTION LINE ON COMPRESSOR PERFORMANCES

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1 ABSTRACT

This paper deals with an experimental application of the supercharging phenomenon in the small reciprocating compressors.

The idea is to exploit the resonance condition of the gas contained within the suction line in order to make the suction phase more efficient.

The paper begins with a brief explanation about the basic theory, then takes into account the outcome of the tests and eventually offers the key to limit the noise emission.

2 INTRODUCTION

The technical evaluation of reciprocating compressor performances is based on the analysis of a few parameters like the specific capacity (cooling capacity over displacement), the efficiency and the noise.

The trend of the market is to require a global optimization of the product , especially under the thermodinamic point of view.

To increase the specific capacity many practical solutions have been implemented but other potential sources of improvement can be exploited.

Among them there is the so called *supercharging effect* that can increase to about 10 % the mass flow in the cylinder by introducing a resonance of the intake line.

The purpose is to find the length of the intake pipe so that, during the opening cycle of the suction valve, an overpressure takes place and aids the intake phase.

In other words, the pipe length must be calculated in order to produce pressure waves that in one cycle arrives at the open end and comes back at the valve with a positive amplitude.

In the following sections the tube has been handled by considering the valve side as a close end and modelling the pipe length in such a way as to reach the resonance condition with a frequency double the fundamental frequency of the valve opening.

One must understand that the resonance in the gas tube could produce more noise.

For this reason, it is has been necessary to design new suction mufflers ,helped by a computer software calculating the Insertion Loss of the system.

#3 BASIC THEORY

The gas behaviour in the suction line has been handled by considering it as a separated element in the whole system.

The object of this study is to tune the natural resonance of the gas tube at 100 Hz, i.e. double of the fundamental harmonic.

The meaning of the tuning is to produce an overpressure and then an extra boost of the gas located in front of the suction valve.

The theory that gives the final formula to size the intake pipe is based on the *electromechanics analogy*.

The principle of this theory is to reduce the gas tube to a mechanical system composed of mass,spring and damper.

In a complete mechanical system (fig.1) the forces balance gives the expression

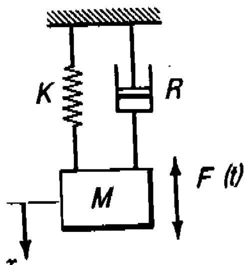


Fig. 1

$$Ft() = Fi + Fr + Fk \qquad (1)$$

F(t) = forcing force

Fi = inertia force

Fr = dissipative force

Fk = elastic force

The equation (1) can be also written as :

$$F(t) = M\,du/dt + Ru + K\int u\,dt \qquad (2)$$

where u = vibration velocity

M = mass

Rm = damping factor or mechanical resistance

K = spring stiffness = $1/Cm$ with Cm = compliance

The solution of the equation (2) describes the oscillating motion.

Then a mechanical system can be reduced to an equivalent electric circuit in the following way.

A general electric circuit (fig.2) is composed of a generator (electromotive force), a resistance , an inductance and an electric capacity.

Applying the Ohm's Law and taking the derivative with respect to the time the equation is

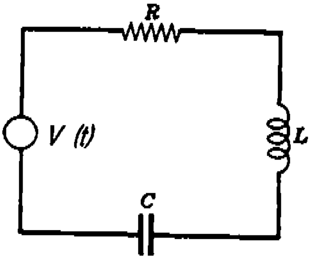


Fig. 2

$$V(t) = L\,di/dt + Ri + 1/C\int i\,dt \qquad (4)$$

where $V(t)$ = electromotive force

i = current

L = inductance

R = resistance

C = capacitance

The solution of the equation (4) gives the current i as a function of the time.

If the equations (4) and (2) are compared, the following correspondence can be done

Mechanical System	**Electric Circuit**
Forcing Force $F(t)$	*Electromotive Force $V(t)$*
Vibration velocity u	*Current I*
Mass M	*Inductance L*
Mechanical Resistance Rm	*Resistance R*
Elastic Constant $K = 1/Cm$	*Inverse Capacitance $1/C$*

By considering the effective values and introducing the impedance term Z we have for the electric circuit

$$Z = V / I = \sqrt{R^2 + X^2} = \sqrt{R^2 + (\omega L - 1/\omega C)^2} \quad (5)$$

where X is the Reactance $= \omega L - 1/\omega C \quad (\omega = 2\pi f)$

The condition

$$X = 0 \qquad \omega L = 1 / \omega C$$

is called resonance condition.

Similarly, for the mechanical system, we can write

$$Zm = F / U = \sqrt{Rm^2 + (\omega L - 1/\omega X)^2}$$

where Zm is the mechanical impedance

The correspondence between the two systems belonging to two different fields can be expanded to a third system.

The acoustic-model is represented by the gas tube and the equivalent mechanical system can be obtained.

The tube is open at one side and disturbed at the other side.

The disturbance is caused by the valve motion and produces continuous pressure changes in the pipe.

The gas tube presents a mass, an elastic constant due to the property that behaves like a spring and a damping factor, or resistance, taking into account the energy dissipation (fig.3).

Applying the equation (2) to the new model is

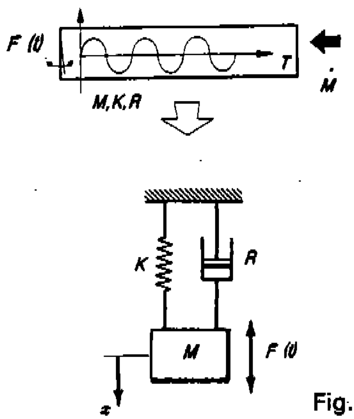


Fig. 3

$$f(t) = M\,du/dt + Rmu + K\int u dt \quad (7)$$

where f(t) = valve motion disturbance

u = vibration velocity

M = mass of the gas tube

Rm = mechanical resistance

K = gas stiffness = 1/Cm

If we consider the pressure p, the previous equation becomes :

$$\Delta p(t) = (M / S) du/dt + (Rm/ S) u + (1/ C mS) \int u dt \qquad (8)$$

and with the effective values

$$P = j\omega(M / S) U + (Rm / S)U - j U / (\omega Cm S) =$$

$$= (Rm / S)U + jU(\omega M / S - 1 / (\omega CmS)) \qquad (9)$$

Introducing the term Q, acoustic flowrate,

$$Q = US$$

we may write the equation (9) in the form

$$P = (Rm / S^2)Q + jQ(\omega M / S^2 - 1 / (\omega CmS^2)) \qquad (10)$$

Now we can define

Ra = Rm/S^2 = Acoustic Resistance

Ma = $M/S^2 = \rho L/S$ = Acoustic Mass

Ca = $CmS^2 = V/(\rho c^2) = SL/(\rho c^2)$ = Acoustic Compliance

For analogy, the acoustic impedance Za is

$$Za = P / Q = \sqrt{Ra^2 + Xa^2}$$

where Xa = Acoustic Reactance = $\omega Ma - 1/(\omega Ca)$

The resonance condition arises when the acoustic reactance Xa is = 0,then

$$\omega Ma = 1 / \omega Ca$$

$$\rho\omega L/S = \rho c^2 / (\omega LS)$$

$$\omega = c/L \qquad f = c / (2\pi L) \qquad (11)$$

Then the pipe length will be

$$L = c/(2\pi f)$$

4 EXPERIMENTAL PROCEDURE. RESULTS

The experimental application of the theory has been faced with a preliminary bench test, with its aim to confirm the theoretical data.

In this test , an open compressor with a variable length of the suction pipe was taken and a manometer set in the discharge line.

Thus, a valve allowed to control the the flow coming out from the compressor and discharged in.the air (fig. 4)

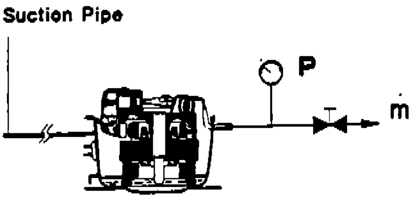


Fig. 4

Since the test was carried out in air, the pipe length was sized with reference to the corresponding sound speed.

Then , after the reaching of a steady temperature in the suction pipe, a pressure measurement has been done for each pipe length.

The resonance phenomenon was expected with an increase of pressure in the discharge pipe , as a consequence of higher flow rate.

In fig. 5 the pressure changes versus the pipe length are reported and where the value is highest the best condition are achieved.

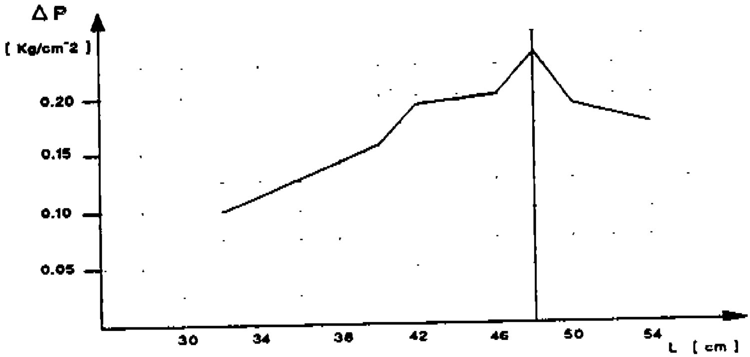


Fig. 5

Then, considering the R12 application,the pipe length was fixed.

The tests were carried out at the calorimeter and special prototypes , having a transducer near the suction valve and different pipe lengths, were used.

The transducer allowed a continuous monitoring of the gas pulsations and the corresponding graphs are reported below.

In fig.6, pulsations of the standard configuration of suction mufflers are represented and it can be noticed how flat the pressure is before the valve opening , due to the mufflers and passages damping.

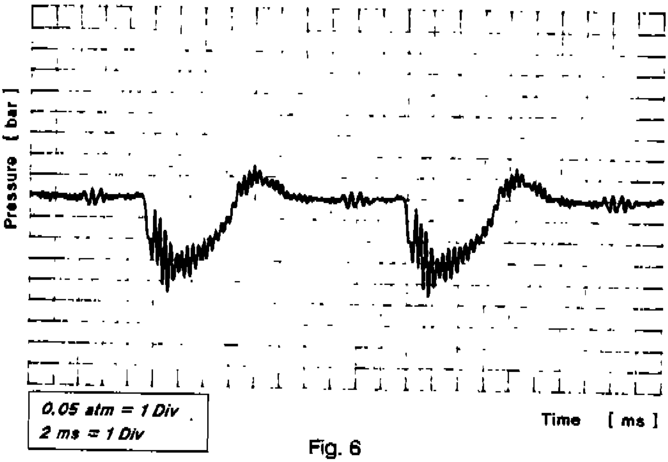
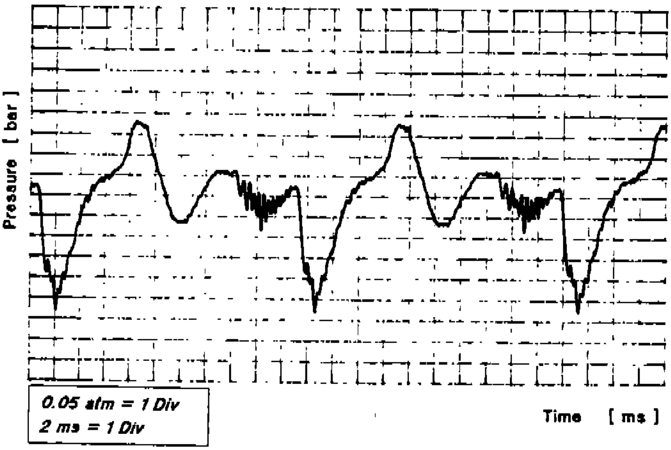


Fig. 6

If a pipe is taken, a completely different outline is obtained and this is because the resonance conditions arise., even if no tuned with the fundamental frequency.

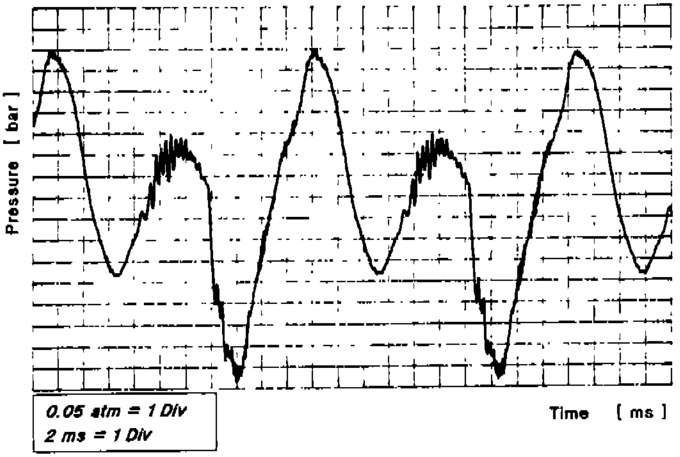
Consequently , the amplitude of the gas pulsations are bigger and the resonances waves can be noticed (fig. 7-11).The closer the gas resonance frequency is with the double of the fundamental harmonic, the higher are the amplitudes of the gas pulsations.

The best situation can be considered with the length between 24 cm and 27 cm.



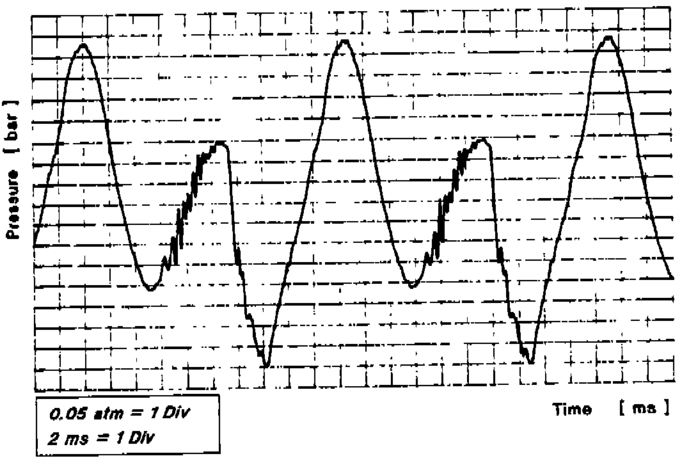
Pipe Length = 12 cm

Fig. 7



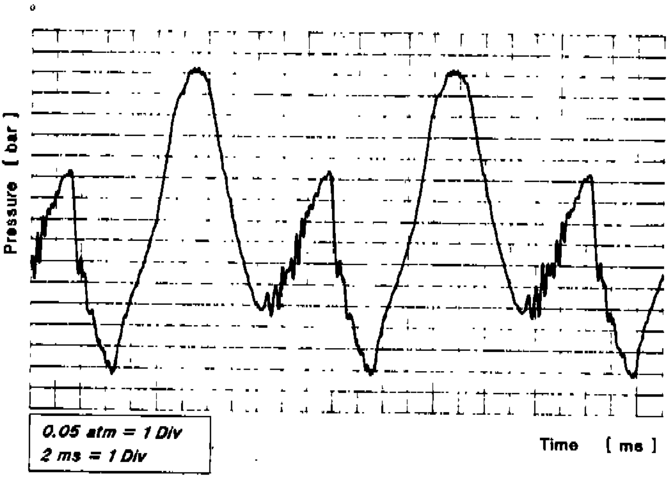
Pipe Length = 21

Fig. 8



Pipe Length = 24 cm

Fig. 9



Pipe Length = 27 cm

Fig. 10

#5 NOISE CONTROL

By experience , the noise range affected by the suction line is below 1000 Hz and it plays an important role in the overall noise generation mechanism of reciprocating compressors.

A support for the investigations is provided by existing computer simulation programs.

In our case, a Fortran program allows the calculation of the frequency spectrum of noise ,due only to the suction gas pulsations.

The main aim of this study is to find out a new geometry of the mufflers , when the long suction pipe is applied.

The acoustic properties of the mufflers can be represented by the INSERTION LOSS , defined as :

$$I.L. = W1 / W2 \quad or \quad I.L. = 10 \, Log \, W1 / W2 \qquad (db)$$

where W1 = sound power of the system without mufflers

W2 = sound power of the system with mufflers inserted.

The mufflers have been reduced to an equivalent cylindrical pipe with a length equal to the physical distance between the input and the output of the muffler and a radius calculated considering the volume of the pipe equal to the real volume.

In the following graphs, three different suction lines are compared.

The first concernes the standard configuration (fig.11), whereas the others two are with the long pipe,the former without any specific design (fig. 12), and the latter with two Helmotz Resonators to damp a few frequency peaks (fig.13).

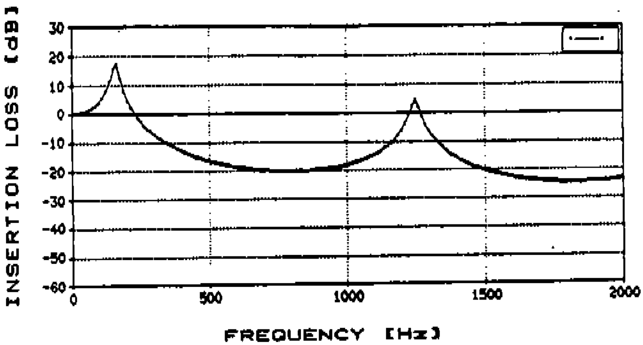


Fig. 11

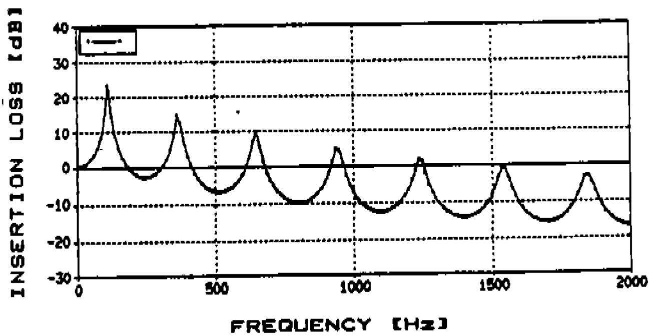


Fig. 12

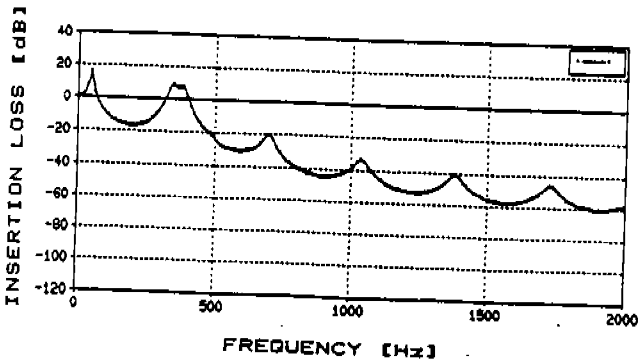


Fig. 13

#6 CONCLUSION

The outcome of this study have shown the validity of the supercharging effect.

The great increase of the specific capacity has been followed by a limited increase of input power and therefore the efficiency of the compressors improved.

Particular attention must be dedicated to the suction muffler design in order to dampen the noise without influencing the resonance phenomenon.

Further investigations have been planned to optimize this application.

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