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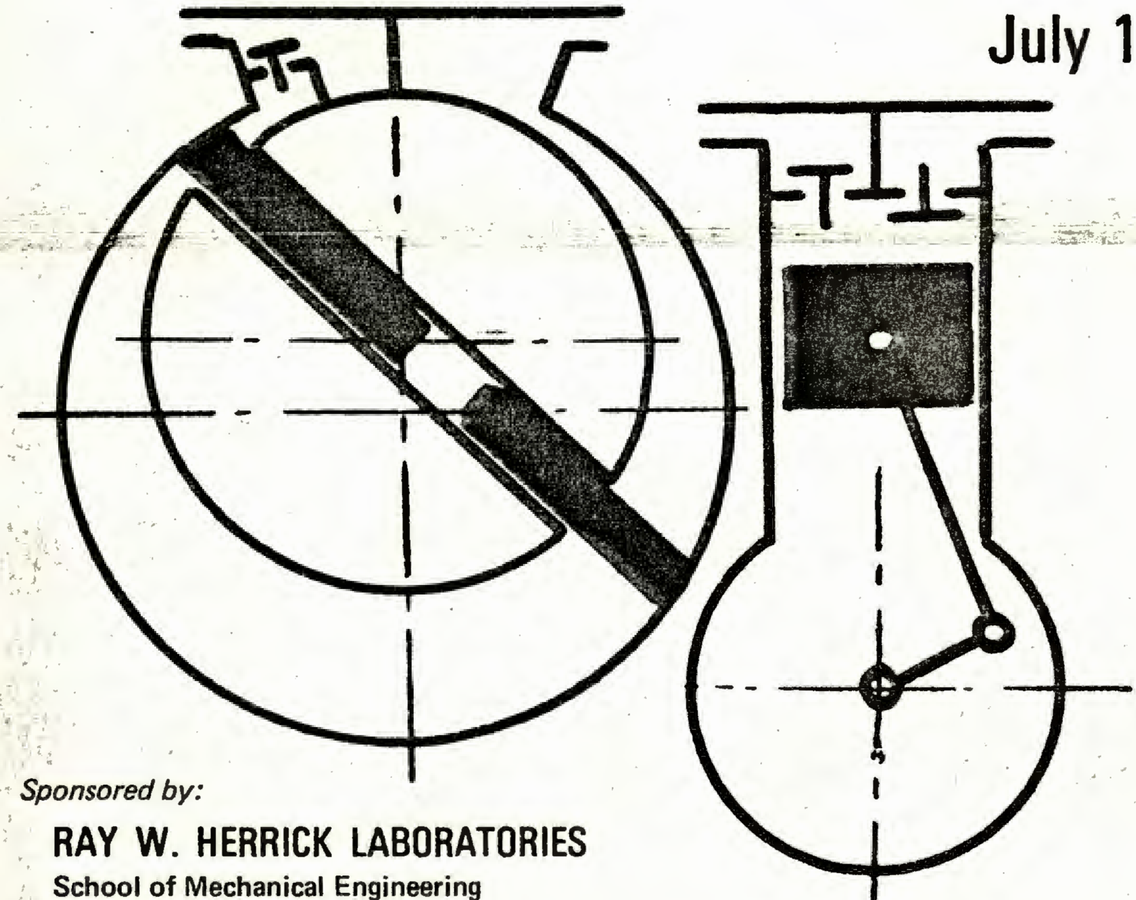

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A REVIEW OF COMPRESSOR LINES PULSATION ANALYSIS AND MUFFLER DESIGN RESEARCH

PART I - PULSATION EFFECTS AND MUFFLER CRITERIA

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With continuing demand for sound and rational methods for compressor design, it has become vital to take into account the effects of pulsations in suction and discharge systems of reciprocating gas compressor installations. The purpose of the present paper is to discuss and review the various aspects of these harmful uneven flows. Part I of the paper deals with the pulsation causes, its nature and effects and the satisfactory means of reducing them. A summary of analytical studies of pulsating flows is presented in Part II of the paper.

COMPRESSOR VIBRATIONS & NOISE

In a reciprocating compressor and its attached piping, vibrations and noise are induced mainly because of the following exitations: rotary or reciprocating unbalance, exitation due to cylinder gas reaction on piston and crankshaft, and exitation due to the pulsating flows in the suction and the discharge lines. All three of these arise from the movement of reciprocating piston. Imperfect mechanical operation like bearing knock and piston slap and electric motor noise also make contribution to the overall compressor noise. Here, we are concerned only with the exitations due to gas pulsations. Their effect on the compressor performance and operation is very pronounced and its prediction and control has become an important step in the modern compressor designs.

NATURE OF FLOW IN PIPING

Causes of Pulsating Flow

The fluctuating flow is caused by the reciprocating action of the piston. The velocity profile of the piston can be given by the following expression: (also refer Fig. 1)

$$u_p = R\omega\,(\sin \omega t + \frac{R}{2L} \sin 2\omega t) \qquad (1)$$

where u_p is the piston velocity, R is the crank radius, L is the connecting rod length, t is the time and ω is the circular frequency and is given by the formula, $\omega = \frac{2\pi N}{60}$ rad/sec where N is the compressor rpm. In the absence of valves, the passage between the compressor cylinder and the piping is unobstructed and the velocity, u, of the fluctuating gas, induced or discharged in the piping will be,

$$u = \frac{A_p}{A}\, u_p \qquad (2)$$

where A is the area of pipe, A_p is the area of piston. Thus, the alternating motion of piston inherently causes alternating flow velocities. However, the passage between cylinder and pipings is not unobstructed, it is blocked by the suction and discharge valves. The valves are uni-directional and open only when there is substancial pressure difference (sufficient to overcome valve inertia and stiffness) across them and since are open only for a small fraction of the total cycle, valves in fact amplify the alternating flows. Electrical analog of the cylinder, and the valves is shown in Fig. 2. The piston is analogous to an electrical generator, the piston inertance is analogous to a source impedance, working space to a variable capacitor and valves can be interpreted as rectifiers. Valve, apart from amplifying the flows, can also generate pressure pulsations because of its vibrations. The resulting pressure will mainly be in the natural frequency range of the valve.

Flow Nature

Gas fluctuations in both suction and discharge lines consist of a periodic train of pulses. The instantaneous values of fluid variables are given by the addition of both mean and fluctuating flows, as shown below:

$$p_t = p_o + p \quad ; \quad \rho_t = \rho_o + \rho$$
$$u_t = u_o + p \quad ; \quad T_t = T_o + T \qquad (3)$$

where p, T, u & ρ represent the fluid variables namely pressure, temperature, velocity and density respectively. Subscript t means instantaneous total value; o represents the mean quantities and the variables without any subscript represent the fluctuating quantities. The pressure pulses in both the suction and the discharge lines travel through and ahead of the fluid with the speed of sound and are reflected back by the discontinuities and the end conditions. With the frequent opening of valves, pressure in lines are altered rapidly and are functions of the spatial coordinates and time. But in general a one dimensional model, $p = p(x, t)$, is sufficient for fairly accurate analysis. Here, x is the longitudinal coordinate along the lines. Mean flow velocities (10 - 25 m/sec, or 40 - 50 ft/sec) are very small compared to the sonic velocities (for air = 340 m/sec or 1115 ft/sec). The fluctuating pressures are generally small compared to the mean pressures. Under these circumstances, it is obvious that the problem can be treated by the acoustic theory. (For details, refer Part II)

Flow Representation

The wave shape of the fluctuating fluid variables depends upon the gas properties and the geometry of the suction or discharge systems. For any wave shape, the state of pulsation at any arbitrary point can be expressed in the form of the fourier series.

$$p = \sum_{n=0}^{\infty} (A_n \cos n\omega t + B_n \sin n\omega t) \qquad (4)$$

$$Q = \sum_{n=0}^{\infty} (C_n \cos n\omega t + D_n \sin n\omega t) \qquad (5)$$

Where Q is the volume velocity and is equal to the product of cross-sectional area and the fluctuating or particle velocity u. A_n, B_n, C_n and D_n are the fourier coefficients and n is the order of harmonic. Equation (4) and (5) can also be written in terms of amplitudes P_n & Q_n and phase angles ϕ_n & ψ_n as,

$$p = \sum_{n=0}^{\infty} P_n \cos (n\omega t - \phi_n) \qquad (6)$$

$$Q = \sum_{n=0}^{\infty} Q_n \cos (n\omega t - \psi_n) \qquad (7)$$

From the above expression, it is clear that only the multiples of fundamental frequency ($\omega = \frac{2\pi N}{60}$) or harmonics occur in the system. No matter what the shape of mass flow through valve is, it can always be broken down into components of fundamental frequency and its harmonics by the fourier analysis. If the suction or discharge system characteristics are known, then pressures at the valves and in the piping can also be calculated.

EFFECTS OF PRESSURE PULSATIONS

The effects of pressure pulsations can be grouped into two broad categories. The first is its effect on compressor performance and operation and the second is mathematical simulation model accuracy if pulsation effects are not modeled. The latter one is a by-product of the first.

Performance and Operation

Generally speaking, the pressure oscillations affect the valve behavior and the thermodynamic performance, produce piping vibrations & structural problems and radiate noise. All of these are discussed below in detail.

1. Valve Behavior: If a suction pipe is fitted in an air compressor (always true for refrigerating compressor), pulsation amplitudes are increased and are particularly significant during valve closure. If pressure amplitudes are greater then the pressure loss in the valve, flow inversion results. Although pressure difference across the discharge valve (few tens of psi) exceeds greatly the pressure difference across suction valves (few psi), pulsations effects are more pronounced in the discharge lines. Due to the pulsations and possible flow inversion, valve will behave in the following way,

i) Contacts between valve plate and the seat increase.

ii) Impact velocity of valve plate on seat increases.

iii) Since pressure difference across the valve is fluctuating, valve may flutter.

In short pulsations may cause high valve impacts against seats and stop, valve bounce and flutter, thus resulting in shortening of the service life and failures of the valves. The remedy of this particular problem is to reduce the valve lift and strengthen springs. These would increase the flow resistance and decrease the main pulsation effect viz. the flow inversion but it will be accompanied by the thermodynamic losses. Brablik's [1,2] analytic models and experimental results support the theory outlined above. MacLaren and Tramschek [3,4,5] have confirmed it.

2. Thermodynamic Performance: Pulsations and flow inversion affect the capacity of the compressor. The volumetric efficiency depends upon suction and discharge pressures

and since these pressures are fluctuating, the volumetric efficiency is affected adversely and so is the capacity of the working fluid. Also, the cylinder pressure tries to follow the lines pressure because the flow into and out of the compressor is a function of the pressure difference across the valve.

Flow inversion around suction valve decreases the volumetric efficiency as reversed flow across the discharge valve mixes up with clearance volume gas and is re-expanded. It delays the suction valve opening and decreases the compressor capacity. The most unfavorable case of pulsation effect arises in multistage compressors; if the timing of the suction and the delivery coincide in two successive stages i.e. if the pistons are in phase.

Pulsations also have favorable effects on performance. According to Bannister [6], the fitting of a plain inlet pipe to a compressor may increase or decrease the throughput to up to 18%, depending upon the delay angle ϕ, defined as $\phi = \frac{12NL}{c}$ where N is the rpm, c is the sonic velocity, L is the length of the inlet pipe. Throughput is a maximum, zero, a minimum and zero at $\phi = 80^{o}$, 155, 190 and 300^{o} respectively. With moderate pipe lengths, induction ramming (effect of wave action) causes an increased flow. The effect of supercharging in suction line can increase the volumetric efficiency up to 15% and increase in mass flow rate by up to 30%. These are based on the investigations carried out by Czaplinski [7], Wallace [8], and Jasper [9]. The supercharging is achieved by tuning the system i.e. making the natural frequency of the intake system equal to the fundamental frequency of compressor exitations. The effect of discharge line tuning has been reported by Stein and Eibling [10]. Although the thermodynamic performance may be improved by tuning but it is not advisable to tune the system merely for the sake of improving the capacity because tuning can cause troubles in the systems, as discussed below.

3. Piping Vibrations: Pulsating flows cause the vibrations of the piping system. The magnitude of the vibration depends upon the amplitude and nature of the pressure pulses, thickness, length and the material properties of the pipe. This problem can be analyzed analytically by considering it to be a forced vibration case. The most serious trouble arises when the system is tuned or having resonance, then the large amplitudes are built up in the system and may cause pipe failure, damage pipe support and produce other structural problems. The structural significance of the problem is as follows.

i) frequency dictates the number of cycles and hence fatigue problems.

ii) and amplitudes indicate maximum stress and range of stress, thus a measure of yield stress and fatigue life.

Piping vibrations might be transmitted to the condenser, evaporator, expansion valve and other system components of a refrigerating compressor and can induce serious vibration problems in these components at resonance conditions.

4. Standing Waves & Noise: The two acoustic effects of the gas oscillations are the formation of the standing waves and the noise radiation. If the piping dimensions are of the order of the sonic wavelength, then due to the interaction of the incident and reflected waves, standing waves are formed. It would amplify pressures at some points (antinodes) and cancel pressure effects at other points (nodes), thus giving rise to harmful unbalanced forces.

Pressure pulsations are also a source of noise radiation. It might radiate the noise to surroundings either through the compressor shell [29] or through the pipings. Generally the high frequency content of the oscillations is responsible for the noise radiation.

In pneumatic and refrigeration industries, a lot of attention has been given to the noise & vibration problems but emphasis has been mainly on control rather than on correct prediction of the effects on the system [11]. In this connection, the efforts of Grover [12], Chilton and Handley [13], Miller and Hatten [14] and Nimitz [21] are significant.

Simulation

Analysis of the pressure pulsations and their incorporation in the mathematical model has become an integral part of the simulation of reciprocating compressors. Exclusion of modeling of the suction and discharge lines from the computer simulation program might lead to performance prediction which may not be realistic as the program would predict,

i) incorrect mass flow rate through the valves

ii) incorrect valve response

iii) and incorrect pressure distribution.

Not only the modeling of lines but their interaction with the valves is also important (discussed in Part II). Brunner [15], Brablik [1,2], Benson [16], Soedel [17] et al, Elson & Soedel [18,19] and Schwerzler [20] are some of the various investigators who have included these effects in their mathematical simulation models.

MUFFLER APPLICATIONS

From the above considerations, it is obvious that flow smoothening devices are needed to reduce the pressure pulsations, to shift or avoid resonances, to reduce piping and structural vibrations and finally to attenuate sound energy which might radiate noise. These flow smoothening devices are generally referred to as mufflers or filters or dampers, or flow smootheners or silencers or as snubbers. In connection with filteration of pulsations, the most important question is whether and when is a muffler necessary? The answer to this question is that even if it is not very necessary, it might prove to be beneficial. Since pulsation flows are inherent to the reciprocating compressor installations, it is inconceivable that the compressor might be free from its ill-effects, which may differ from one system to another, thus some form of a muffler element is required in compressor installations. Proper selection of muffler requires an analytical or experimental investigation of the existing conditions in the suction and discharge lines (which is discussed in Part II of the paper), system requirements and the characteristics of the filter elements.

MUFFLER DESIGN CRITERIA

Before choosing any muffler element, it is desirable to select the criteria for design. Muffler design criteria can be classified broadly into acoustic criterion and general criteria. Proper design requires that both should be satisfied.

General Criteria

1. Pressure Drop. It is an important criterion because it limits the length and geometry of the muffler elements. For each particular system, maximum pressure drop allowable should always be calculated beforehand. For example in the refrigeration system, after the muffler element, pressure of the gas should be equal to the condenser pressure. Pressure drop (Δp) depends upon the length L, total instantaneous velocity, u_t (sum of mean and fluctuating parts) and fanning friction factor f as shown below

   $$\Delta p = 2\, f \frac{L}{d} (\rho_o u_t^2) \qquad (8)$$

   where d is the diameter of tube and ρ_o is the gas density. All flow direction changes, e.g. bends, sharp corners etc., should be kept to minimum for low pressure drop.

2. Space Requirement. A designer has to keep in mind the geometric and space requirements in compressor installations before picking up a filter element.

3. Material Selection. Material of the muffler devices should not react with the gases and also should be able to endure high temperatures (if existing) or satisfy any other particular requirement.

4. Cost Criterion. This criterion sometimes is the override factor as far as commercial competition is concerned. The manufacturing, installation and the operating costs of mufflers should be kept to the minimum possible.

Acoustic Criteria

Acoustic criteria of the muffler design is specified by the frequency response of the filter element and sound energy transmission and attenuation characteristics. Mufflers in general, are classified by either as dissipative or as reactive types. A dissipative muffler has usually the flow resistent characteristics i.e. in electrical analogous term, resistive element. The mode of filteration is the absorption of sound energy. In the air compressor installations, control valves are also used to smoothen the flow but they absorb energy and create an undesirable pressure drop. Thus, reactive type muffler elements are generally preferred and frequently employed. Its performance is determined mainly by the geometrical shape of the element and it varies with the frequency just like an electrical filter. The reactive muffler does not absorb energy but rather reflects part of the sound energy back by offering an impedance mismatch. The electrical analogous of a reactive muffler is a circuit composed of inductances and capacitances only. Chilton & Handley [13], Wallace [2], and Miller & Hatten [14] have investigated and developed some simple reactive mufflers for the compressors. The general theory of acoustic transmission line filters is described in references [23,24,25]. The important factor to be considered in designing the elements is to select the proper frequency dependent sound transmission or attenuation characteristics. Also, it is important that either the resonances be avoided or shifted. Therefore, the elements must be chosen so as to attenuate the sound energy at that particular frequency or must shift natural frequency of the system to avoid the resonance conditions. From the noise reduction point of view, elements with high frequency attenuation characteristics are generally selected. The muffler should be located as close to the cylinder as possible so as to eliminate pulsations in pipings. Also, care should be taken to investigate, either theoretically or experimentally, the effect of muffler elements on the value behavior and thermodynamic performance because the muffler may have favorable or unfavorable effect on these.

MUFFLER ELEMENTS

Numerous muffler elements are in existence as reviewed from the references [22,23,24,25,26,27,28]. Here, only some very basic and simple elements will be discussed to illustrate the point.

Maximum sound energy transmission takes place only when there is impedance matching and the amplitudes of the pulsations are maximum at the resonance conditions, thus the main objective in picking up the muffler elements is such that they should provide impedance mismatch to the system and shift the system resonance frequency. However, some muffler elements provide maximum sound attenuation at their resonance frequency i.e. offer complete impedance mismatch at their natural frequency. Such mufflers elements are referred to as the resonators. The sound transmission characteristics of muffler elements depend not only upon its own impedance but also upon the source and the load impedances. Acoustic impedance Z_a is given as

$$Z_a = \frac{P}{Q} = R_a + j\,X_a = R_a + j\,(\omega M_a - \frac{1}{\omega C_a}) \tag{9}$$

where

$$M_a = \frac{\rho_o L}{S}\,; \quad C_a = \frac{V}{\rho_o c^2} \tag{10}$$

where R_a is the real part of the acoustic impedance, the resistance and X_a is the imaginary part of the impedance. Thus impedance takes into account the inertia effects (M_a - inertance) and elastic properties (C_a - compliance). Refer table 1. S is the cross sectional area, L is the length, V is volume (V = SL), ρ_o is the mean density and c is the sonic velocity. The impedance electrical analogy suggests that R_a is analogous to electrical resistance, M_a is analogous to the electrical inductance L_e and C_a is analogous to the electrical capacitance C_e. The electrical analogy and its familiar results have been employed extensively in formulating silencers for the acoustic systems. Acoustic mufflers are broadly classified, by their frequency response characteristics, as low pass filters, high pass filters, band pass filters and band elimination filters. For instance, a simple expansion chamber is a low pass filter as it attenuates only the high frequency contents and its transmission loss, T.L., is given as (22)

$$\text{T.L.} = 10 \log\,[1 + \frac{1}{4}(m - \frac{1}{m})^2 \sin^2 \frac{\omega}{c} L]\ \text{dB} \tag{11}$$

where m = area ratio = S_2/S_1 and L is the length of the chamber (Refer Fig. 3). If several expansion chambers are connected to each other, then the transmitted (output) acoustic volume velocity, would be as given by Fig. 4, whose ω_c is the cutoff frequency which depends upon the inertance and the compliance distributions. Electrical analog of the acoustic system has also been shown. A high pass filter, as shown in Fig. 5 along with its electrical analog and transmission characteristics, is used to attenuate the low frequency contents of the sound energy. The band pass filter is a combination of a low pass and a high filter. Sound energy between two cutoff frequencies only is allowed to pass unattenuated. It is shown in Fig. 6. A band elimination filter is composed of resonators. The best example of a resonator is the Helmholtz resonator. If a Helmholtz resonator is put in the side branch, as shown in Fig. 7, its transmission loss at the resonant frequency ω_r will be maximum and is given by (22)

$$\text{T.L.} = 20 \log_{10} (\frac{\alpha + 0.5}{\alpha}) \tag{12}$$

where α = resonator dimensionless resistance $= S\,R_a/A\rho_o c$

where S and A are the areas of pipe and Helmholtz resonator neck respectively. R_a is the Helmholtz resonator resistance and is given by

$$R_a = \frac{\rho_o \omega^2}{2\pi c}$$

Resonance frequency of the resonator is

$$\omega_r = c\sqrt{\frac{A}{LV}} \tag{13}$$

where L is the neck length and V is the volume of cavity. Several Helmholtz resonators in series will constitute a band elimination filter i.e. it will attenuate sound completely between two cutoff frequencies, as shown in Fig. 8. Several books [23,24,25] and good papers [22,26] on acoustic filters provide a good coverage on the general theory of filters. However, while picking up a particular muffler for a particular frequency response, attention must be given to its effect on compressor efficiency in all frequency ranges.

MUFFLER DESIGN PROCEDURE

A compressor designer should proceed as follows for the proper muffler selection for any installation.

1. Investigate the level of pressure pulsations. Sound pressure spectrum of the existing suction and discharge systems can be measured. However, for a proposed design, computer simulation should incorporate the pulsation analysis.

2. Estimate the piping natural frequencies either with the aid of analytic study of piping geometry or by observing the pressure spectrum. This knowledge would provide the designer with the idea of the frequency or frequency bands to be avoided. Sound spectrum level would also indicate the noise level.

3. Select the general criteria for the flow smoothing devices, as discussed earlier. Muffler designer should assess the degree of importance of each criterion like pressure drop, geometric and space limitations, material selection and the cost. Also their relative weightage with the acoustic criteria should be studied.

4. Choose the acoustic criteria. For muffler selection, transmission loss, frequency band elimination, shifting of a particular frequency, as discussed earlier, should be specified. An upper limit of 2% pulsations after muffler element seems safe for the average installation. This figure should be decreased somewhat for the high pressure lines and may be increased for the low pressure lines.

5. Select the muffler element according to the requirement of the installation. If transmission characteristics of a particular element are not known, then these can be investigated experimentally or analytically before and after muffler inclusion in the system.

6. Locate muffler at the appropriate place, preferable as close to the inlet or exhaust as possible so that in the piping no dangerous pulses remain to cause the noise and vibrations.

7. Lastly, determine pressure spectrum after inclusion of muffler devices for the confirmation of satisfactory results.

CONCLUSION

No matter how close the muffler is located to the suction or discharge, there shall be substancial pulsations in the immediate vicinity of the valves, thus affecting the valve response and service and also the capacity of the working fluid. Muffler elements may cause favorable or adverse effects on the valve chamber pulsations. A complete theoretical or (and) experimental information regarding pulsating flows are required so that the performance of the pulsation effects can be predicted with a fair degree of accuracy. Finally optimization of the compressor lines is advocated [28]. It should take into account all the constraints and design criteria. Although the problem is complex, the answer is not elusive.

NOMENCLATURE

A	area
c	speed of sound
C	capacitance; compliance
L	length
M	mass
n	integar
N	rpm
p	pressure
Q	volume velocity
R	resistance
S	area
t	time
u	velocity
V	volume
ρ	density
ω	circular frequency

Subscripts

a	acoustic	c = cut off
e	electrical	r = resonance
o	mean	
t	total, instantaneous	

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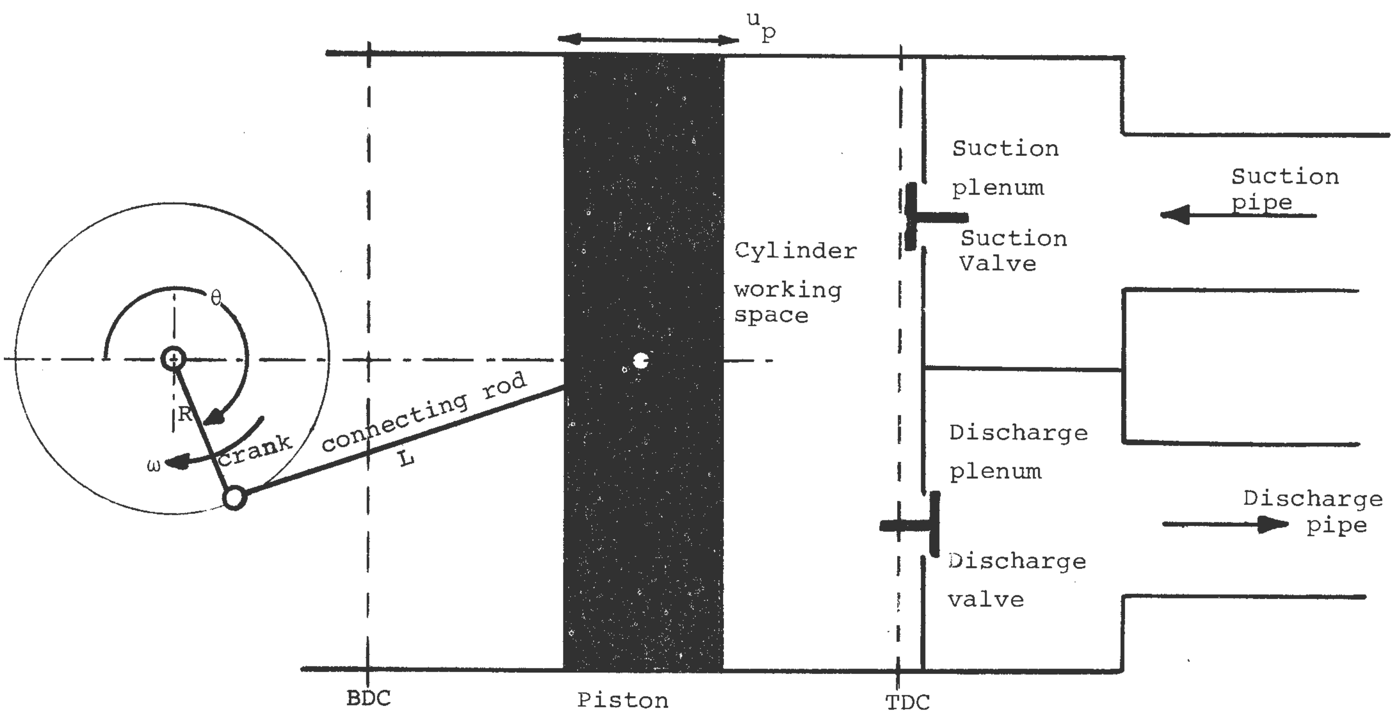


Fig. 1 Physical Model of Reciprocating Compressor

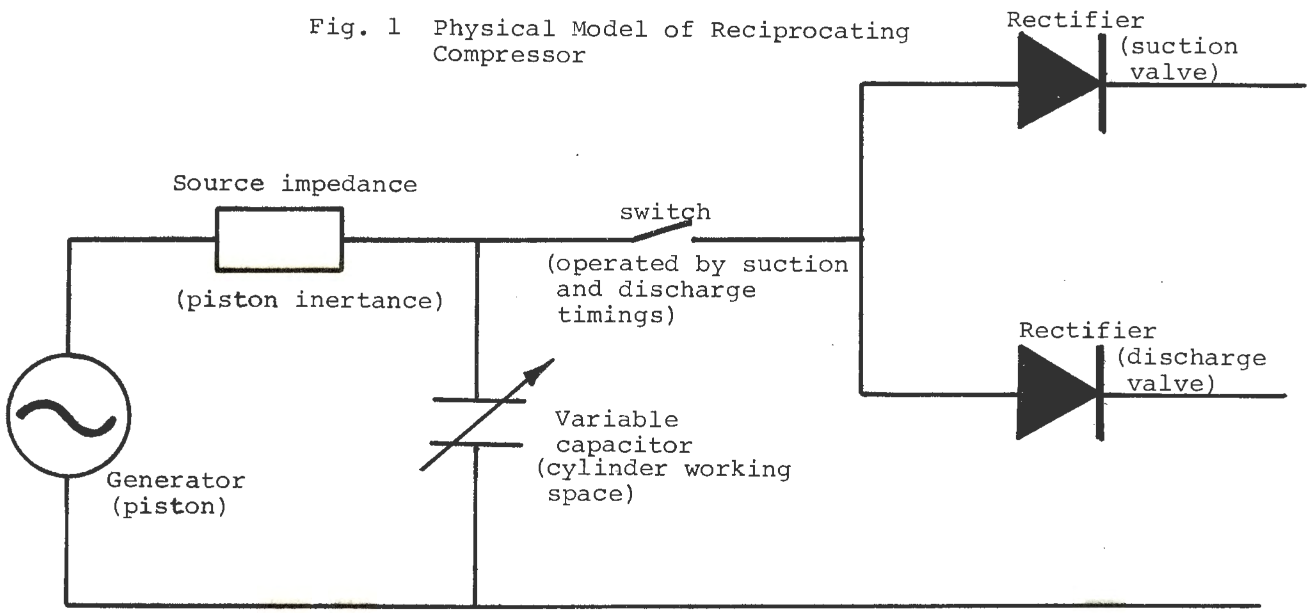


Fig. 2 Electrical Analog of Compressor Cylinder Piston, Working Space and Valves. Switch is Controlled by Thermodynamic Processes.

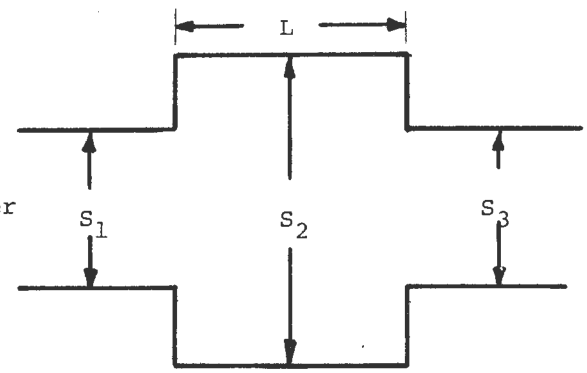
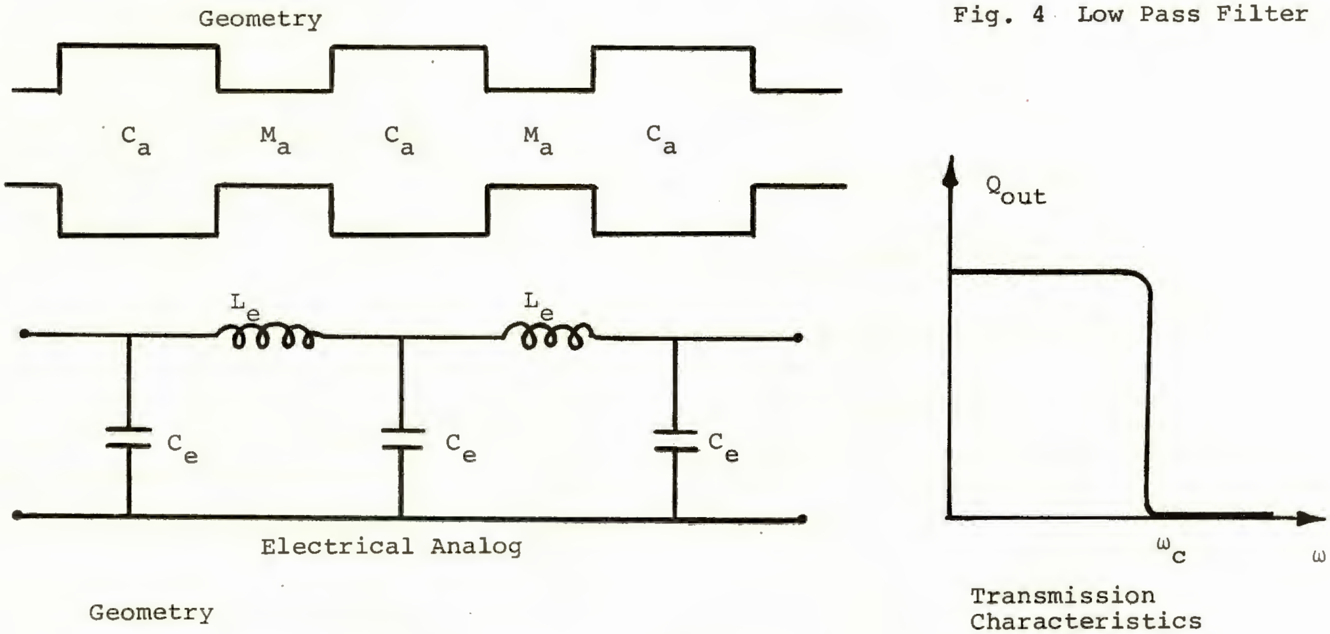


Fig. 3 Expansion Chamber

Table 1 Electrical Analogy of Acoustic Elements

Acoustic Elements	Electrical Elements
S, L — Inertance $M_a = \rho_o L/S$	L_e — Inductance
V — Compliance $C_a = \frac{V}{\rho_o c^2}$	C_e — Capacitance
Resistance $R_a = \frac{\Delta p}{Q}$	R_e — Resistance
Acoustic Impedance $Z_a = p/Q = R_a + j\omega M_a + \frac{1}{j\omega C_a}$	Electrical Impedance $Z_e = \frac{V}{I} = R_e + j\omega L_e + \frac{1}{j\omega C_e}$

Fig. 4 Low Pass Filter



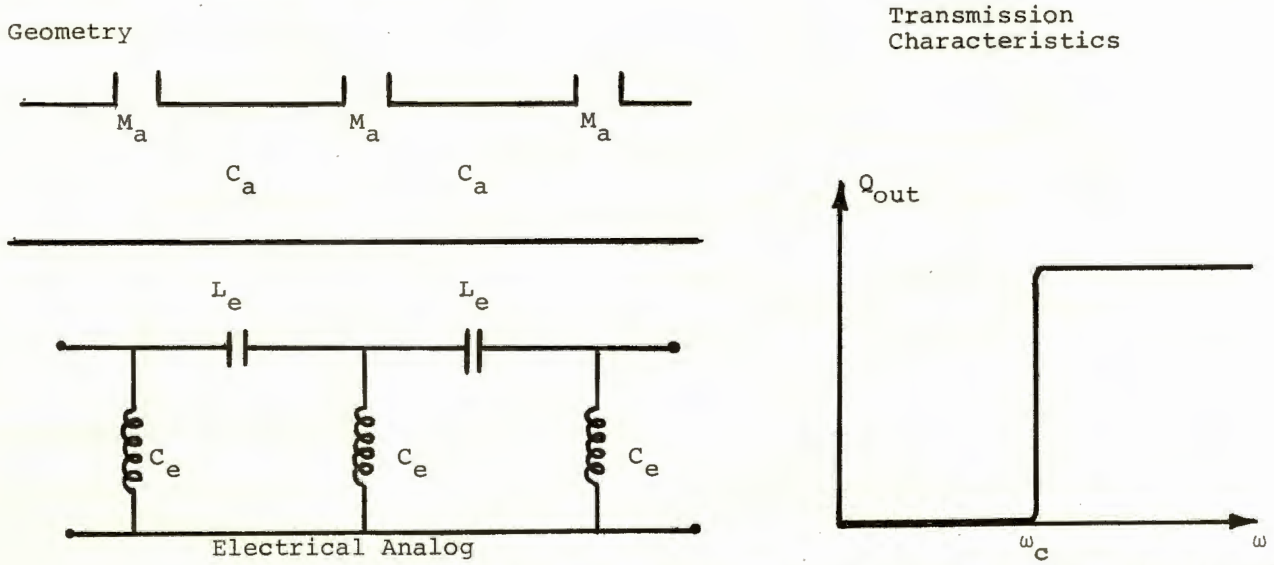
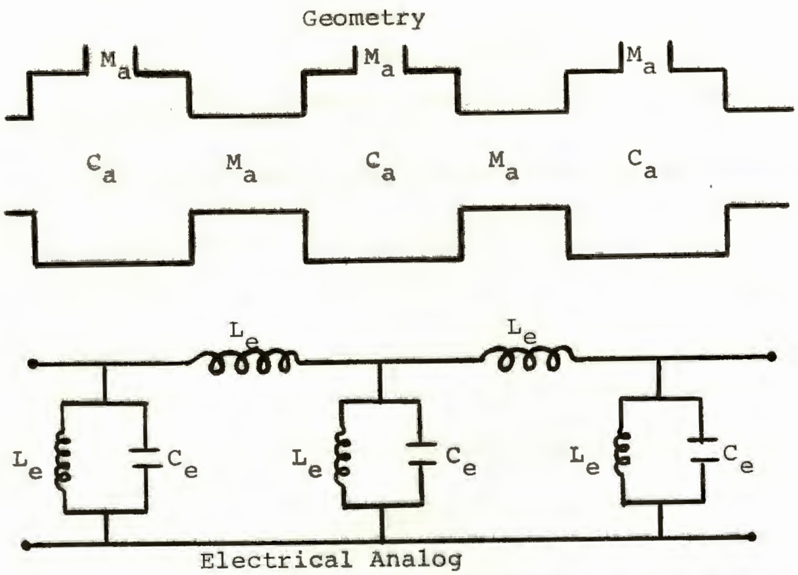


Fig. 5 High Pass Filter

Transmission Characteristics



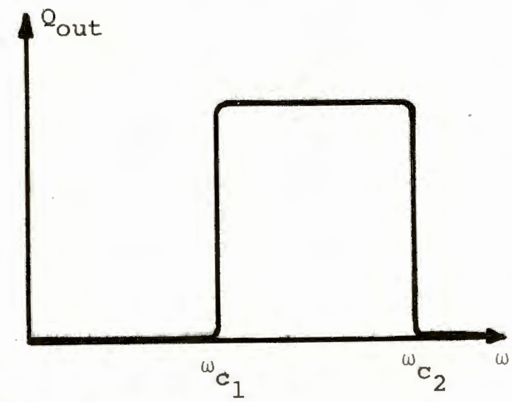
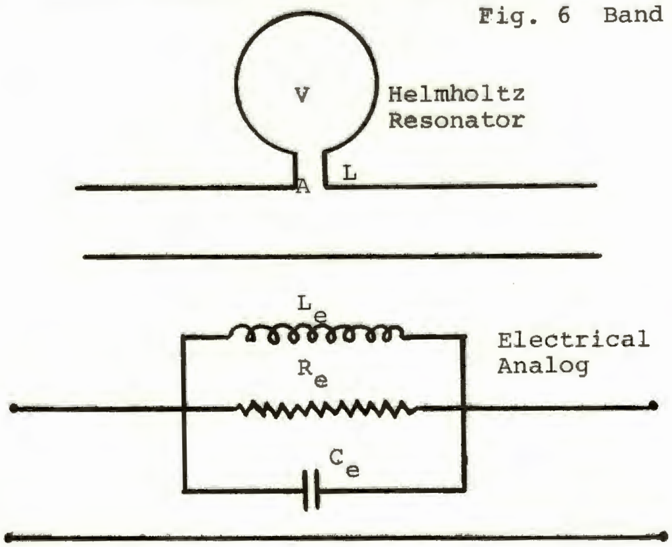


Fig. 6 Band Pass Filter

Transmission Characteristics



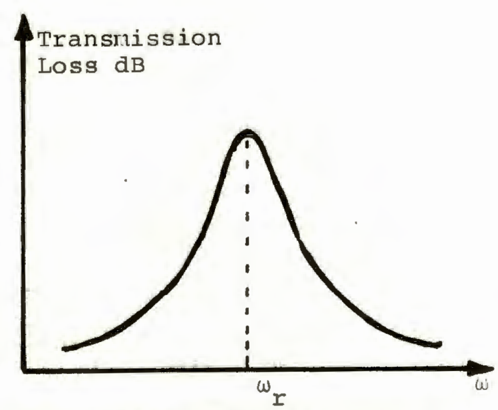
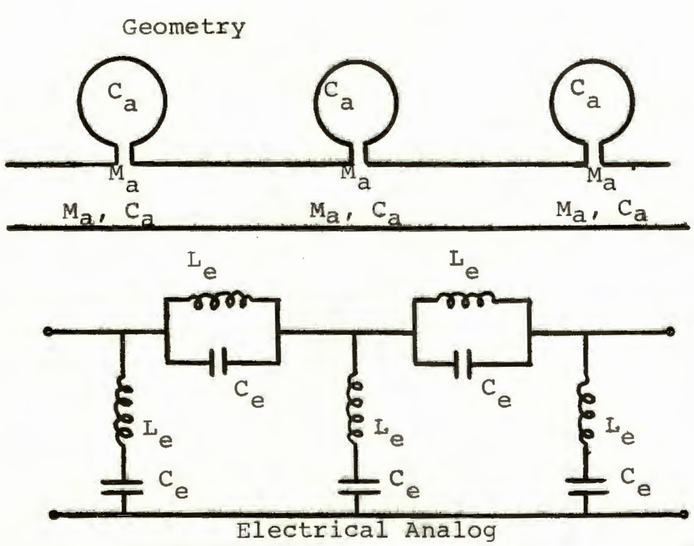
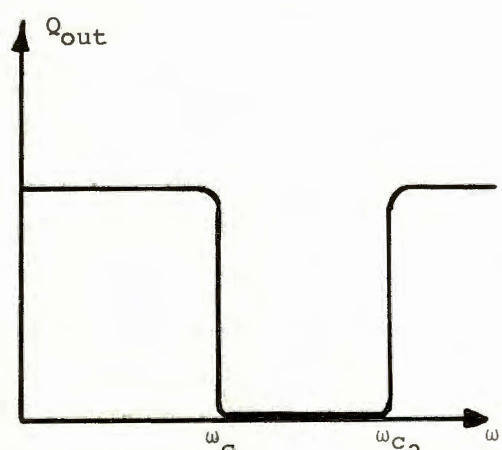


Fig. 7 Helmholtz Resonator

Attenuation Characteristics





Transmission Characteristics

Fig. 8 Band Elimination Filter

R. Singh and W. Soedel, "A Review of Compressor Lines Pulsation Analysis and Muffler Design Research. Part II: Analysis of Pulsating Flows," Purdue Compressor Technology Conference, July 10-12, 1974, Proc., pp 112-113.