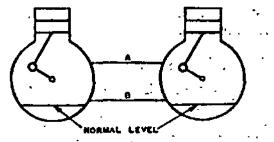

Fig. 4 .... Modified Oil-Equalizing System

Motor cooling and pressure ratio usually determine the pressure setting.

b. Oil pressure protectors are used extensively with forced feed lubrication systems to prevent the compressor from operating with insufficient oil pressure.

5. Time delay or lockouts with manual resets to prevent damage to compressor and contactors from repetitive rapid cycling.

Noise

Achievement of an acceptable noise level is a basic requirement of good design and application. The criterion for acceptance of a noise level must be based on the human ear as well as instrumentation. A discussion on the particulars and the limits of design criterion for air conditioning can be found in Chapter 14. The quality of the noise is extremely important in that those with pure tones or many high peaked discrete frequencies are particularly annoying. Whenever possible, final acceptance should be based on performance in the unit application, this is especially true of small sized equipment. Generally, a satisfactory compressor noise level will result in a satisfactory unit level, all other things being equal.

Sound becomes evident to the senses when a vibrating member (or structure) causes waves of compression and rarefaction to propagate through the air. In compressors, noise comes from the mechanical components, generation by valves and rubbing surfaces and by response of other members to generated frequencies, from the electric motor by generation and excitation and from the gas because of pulses, windage and turbulence.

A program to decrease noise involves the following:

1. Modification and refinement of the members generating the energy to reduce the driving forces.
2. Minimizing of response to the driving forces by designing components so that their natural frequencies do not coincide with the running frequency, twice line frequency or the lower harmonics of these. This is particularly true of hermetics where no isolation of the motor to the running gear and housing of the compressor can be made.
3. Internal and external isolation wherever possible.
4. Muffling of suction and discharge gases.
5. Use of materials that dampen or absorb sound energy.
6. Balance of running gear.

For additional information, see Chapter 14, Sound Control.

Vibration

Vibrations in compressors result from gas pressure pulses and inertia forces associated with the moving parts. In multi-cylinder designs (greater than 2 cylinders) the inertia forces and couples can be diminished to a great degree by balancing.

The problems of vibration can be handled in a number of ways:

1. *Isolation.* This is the most common method and the best results are obtained with spring mounting of the compressor and in the case of the welded shell hermetic spring mounting of the crankcase. Transmission by this means can be readily reduced to 5 percent. The degree of isolation dictates the flexibility of the system and hence the motion. To limit movement during starting, stopping and shipment spring stability in terms of lateral stiffness and/or snubbers and stops are commonly used. The effectiveness of the springs can be determined from the following formulas:

$$t = \frac{100}{1 - (f/f_n)^2} \tag{9}$$

$$f_n = 188/D \tag{10}$$

where

t = transmission, percent.
f = frequency of the impulses (compressor speed or cylinder impulse), cycles per minute.
f_n = natural frequency of the compressor spring combination in cpm.
D = static spring deflection for weight of compressor, inches.

Dampeners or dampen isolators are used to prevent excessive excursion of the system when going through resonance or when the isolator has extremely low rigidity and motion has to be limited.

Nonmetallic isolators of molded rubber and synthetic rubber-like compounds are available. While they possess inherent dampening and are superior in sound isolation, their application is limited for the following reasons:

a. Characteristics are not completely predictable, selection is not an exact science.

b. For equivalent flexibilty (low unit stress) they are bulky.

c. They are affected by changes in environment, deteriorate under high temperatures, and become ineffective under low temperatures.

d. Drifting, taking set.

2. *Reduction of Amplitude.* The amount of movement can be reduced by adding mass to the compressor. This is done by rigidly attaching it to a base, condenser, chiller or providing a solid foundation. Where structural transmission is a problem, the entire assembly is then resiliently mounted. This practice is followed when the machines are large and have shaking forces of appreciable magnitude.

3. *Elimination.* By following good design principles, maintaining close balancing tolerances and using properly selected isolators, vibrations can be reduced to acceptable levels. On small units, package type 5 hp and less, structural members and lines are checked for resonant frequencies and designed away from the fundamental and low harmonies.

On large units, generally field type installations, the problems must be resolved as they crop up. Often vibration will be found on installations where the compressor is well balanced and runs well at light loads or when independent of the system. The problem then is one of resonance of the adjoining members or the result of gas pulsations in the low or high side circuits.

Resonance of a member can be detected by the effect of clamping added mass, or by determining natural frequencies. Changing the stiffness of the member, changing its shape, and/or changing the mass will correct it. A word of caution, a line has many modes of vibration and many natural frequencies.

Vibrations from gas pulses cannot be readily dampened out and are more pronounced at bends in the lines, the frequency is equal to the rotational speed multiplied by the number of cylinders. Pulsations are best handled by a good muffler, the addition of one will do away with the necessity of matching impedances of lines and coils to the compressor. The muffler should be placed as close to the compressor as possible. Within the compressor, at the time of design and testing, port size and length, manifold volumes and internal line sizes and configuration should be carefully selected so as to smoothen gas pulses, reflect waves and eliminate conditions of resonance.

For additional information, see Chapter 14, Sound Control.

Shock

In designing for shock, three types of dynamic loads are recognized:

1. Suddenly applied loads of short duration.
2. Suddenly applied loads of long duration.
3. Sustained periodic varying loads.

Since the forces are primarily inertia, the basic approach is to maintain low equipment mass and make the strength of the carrying structure as great as possible. The degree to which this is carried out is a function of the shock loading.

(1) *Commercial Units.* The major concern here is to ship units or have them operate on commercial carriers without having them suffer any damage whatsoever.

Train service provides the severest test because of low forcing frequencies and high shock load. Shock loads as high as 10 g have been recorded with many lesser ones from 4 to 7 g.

Trucking service results in higher forcing frequencies, shock loads to 5 g can be expected.

Aircraft service forcing frequencies generally fall in the range of 20 to 60 cps with shocks to 3 g.

(2) *Military Units.* The requirements are spelled out in the specifications and greatly exceed anything that is expected of the commercial unit. In severe applications deformation of the supporting members and shock isolators will be tolerated providing the unit is able to perform its function.

Basically, as far as the compressor is concerned, it must be made of components rigid enough to avoid misalignment or deformation during the shock load. Therefore, it is important to avoid structures with low natural frequencies.

PART II: ROTARY COMPRESSORS

The term *rotary* means that compression is performed by a piston and cylinder arrangement that employs circular or rotary motion instead of reciprocating motion. The characteristic form of the machine is a direct-driven positive displacement mechanism. In these machines, performance of the thermodynamic process may be continuous or cyclic depending upon the choice of mechanism employed.

Figs. 5 and 6 show two common types of rotary machines, the *rolling piston* type (Fig. 5) and the sliding vane type (Fig. 6). These two machines have many similarities in respect to size, weight, thermodynamic performance, field of greatest application, range of Btu sizes, durability, noise level, etc. The principle difference is found in the mechanical parts. The rolling piston machine employs a roller on a shaft having an eccentric. In the vane type machine the rotor and shaft are one piece, eccentricity is provided by locating the shaft *off center* with respect to the cylinder.

A well designed rotary machine is relatively free from vibration and is therefore well suited to high speed operation such as direct drive with two-pole motors.

PERFORMANCE

The rotary compressor performance is characterized by high volumetric efficiency due to the small clearance volume and correspondingly low re-expansion losses inherent in their design. Fig. 7 and Table 2 show performance typical of compressors now in production. A careful analysis of performance data discloses potential for improvement which indicates that further progress can be expected in the future.

Improved techniques and new instruments which have become available in the past decade has made possible detailed study and analyses of the performance of these small machines. Table 3 gives a reasonable estimate for the *break down* of losses that can be accounted for in machines of known performance. The values given do not represent the best performance that may be expected. Different designs will exhibit individual variations depending on the factors involved. However, in well designed machines the relative values for most of the small losses would not be expected to change appreciably. Machines of significantly better performance would differ mainly in the major characteristics, such as motor efficiency, mechanical friction, heat transfer to the suction gas, overcompression, and suction throttling. Two-pole machines usually can be expected to achieve a somewhat better coefficient of performance than similar designs at 4-pole speed (in the order of 5 to 8 percent better for the 2-pole machine) if the machine is designed for 2-pole speed. This is so because 2-pole machines exhibit lower heat transfer losses and benefit from higher motor efficiencies. Increasing speed has the adverse tendency of increasing friction and over-compression losses. Therefore, it is important to include design features that will minimize these losses.

The prevalent use of the high-side crankcase is due to the simplicity of the lubrication system and the success of such a design with regard to oiling problems and compressor cooling. Low-pressure crankcase machines differ in respect to the oiling system in that an oil pump is required and also that suction gas is used to obtain compressor cooling.

The preferred arrangement on the suction side of the machine is one which reduces suction throttling to a minimum. This condition is more easily attained if the suction gas is brought into the cylinder from both sides. The suction entrance chamber is made large enough to minimize the possible effect of surging in the flow of suction gas. Heat transfer to the suction gas before actual compression begins, accounts for most of the volumetric loss on the inlet side of the machine. Therefore, heat transfer surface on the suction side is made small. The heating area is reduced most effectively if the important dimensions of rotor diameter and cylinder height are as compact as possible. This leads to smaller values of radius ratio and therefore to improved performance on the exit side of the machine as well.

Internal leakage is controlled through hydrodynamic sealing. The major design requirement is precision fits and optimum clearances. Particular attention is given to secure accurate fits for the blade with the corners where the cylinder

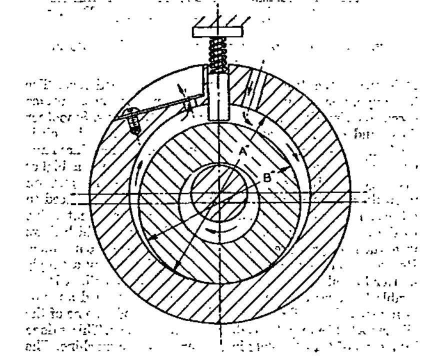

Fig. 5 .... Rolling Piston Type Rotary Compressor

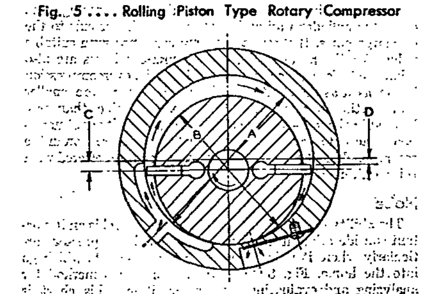

Fig. 6 .... Sliding Vane Type Rotary Compressor