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Historically in reciprocating refrigeration compressors there have been three basic types of capacity modulation. These are: 1) suction valve lifter mechanisms; 2) hot gas by-pass means; and 3) high to low side equalization or discharge to suction equalization of cylinders.

Suction valve lifter mechanisms, such as shown in figure 1, have been historically used in reciprocating refrigeration compressors approximately 30 tons and up. Basically, the mechanism consists of an annular piston that, when supplied with pressurized lube oil or discharge gas, overcomes the spring force and moves downward as shown in the loaded position. When forcing pressure is removed, the cylinder is to be unloaded and the spring forces the piston upward and the lifter pins hold the valve open. Gas is drawn into the cylinder on the intake stroke and expelled back into the suction manifold on the compression stroke. These mechanisms have the advantages of: 1) reducing the torque required to start the compressor; 2) providing as many steps of modulation as there are unloaded cylinders available; 3) relatively efficient power conversion when unloaded; and 4) the actuating mechanism can be signalled by either suction pressure or conditioned medium temperature. The disadvantages of these mechanisms are: 1) complication and consequently expense; 2) the mechanism is not applicable to flapper valve machines; 3) a limited number of steps of capacity modulation; and 4) generally these mechanisms cannot be field installed.

The hot gas by-pass method shown in Figure 2 is often used where very close humidity or temperature control is required. There are many variations of these systems; however, their function is basically the same. The system shown operates with a valve that senses downstream pressure and opens to maintain a preset minimum pressure. The hot gas is bled into the normal refrigerant circuit upstream of the evaporator and downstream of the expansion valve. The temperature of the suction gas returning to the compressor is controlled by the expansion valve as in fully loaded operation. The advantages of the hot gas by-pass system are; 1) the capacity modulation is infinitely variable from 0% to 100% modulation; 2) the system is easily field installed; and 3) the systems are applicable to all types of compressors. The disadvantages are: 1) the power required to operate the system is not reduced when the system is operating at less than 100% of load; 2) the system has no reduction in required starting torque; and 3) the system can be signalled to operate only by suction pressure.

One method of achieving high to low side equalization is as shown in Figure 3. The valve as shown is fitted to a cylinder head that contains one or more cylinders. In the normal position, gas is discharged into a manifold leading to the condenser. In the unloaded position, the cylinder discharge is directed to the suction and the connection to the discharge manifold is blocked so that no back flow can occur. The cylinder(s) then operate with only the pressure difference caused by fluid flow losses across it. The advantages of this system are: 1) reduce load start; 2) generally the system can be field installed to the compressor; 3) the system may be used with most compressor types; i.e., it is not limited to ring valve type machines; and 4) the system can be signalled to operate by either suction pressure or conditioned medium temperature.

The disadvantages are: 1) the unloaded efficiency suffers somewhat when compared with the suction lifter mechanism; and 2) the systems may be somewhat limited to the operating range when a large part of the compressor capacity has been unloaded. These operation limitations are generally caused by temperatures of the suction gas going into the remaining operable cylinders.

Reciprocating compressors with flapper valves have grown in size to approximately 400,000 BTU. Their size and the growing awareness of the necessity for improved humidity and comfort control has increased the need for capacity modulation of flapper valve machines down to approximately 120,000 BTU machines. In searching for methods to modulate the capacity of flapper valve machines, the suction lifter mechanisms were ruled out immediately because it is not applicable leaving only the hot gas by-pass and the high to low side equalization system. Even before the present energy crisis, the hot gas by-pass systems inefficiency was ruling itself out of many applications.

Large reciprocating air compressors have been using a method of reducing their capacity to 0% of full load by blocking off the suction inlet to the compressor. Patent searches revealed that the initial patents applying to this method of unloading air compressors was patented in 1932. The compressors that these systems are used on generally have some external means of cooling the compressor other than the flow of gas through the compressor.

Theoretically the work done in a cylinder or a compressor which has had the suction inlet blocked is zero if friction is neglected and irreversibility is neglected. However, some means of removing heat caused from cylinder friction must be provided. In a refrigeration compressor, this heat can either be removed by suction gas flowing to the operable cylinders or removed to the surrounding ambient.

A method of accomplishing blocking the suction inlet to a cylinder head is shown on Figure 4. A servo valve with a common connection that connects either the suction or the discharge to an unloader cylinder that actuates the unloader piston. When discharge pressure is applied to the cylinder, it drives a piston down to close off the suction inlet to the cylinder head. This occurs when the solenoid valve is energized and is shown in Figure 4A. When the servo solenoid is de-energized, the unloader cylinder is connected with the suction of this cylinder head. The discharge pressure bleeds down to the cylinder head pressure and the suction pressure and spring drive the unloader piston to the upmost position (shown in 4B).

In order to do as little work as possible in the cylinders, it is necessary that leakage into the cylinder during unloaded operation be limited to a practical minimum. The seal around the unloader piston is shown in Figure 4C and consists of an elastomeric "O" ring with a TFE shoe or slipper. This mechanism has proven to be very wear resistant and on a production basis a dry leak rate maximum of 50 cubic centimeters per minute has been obtainable.

Figure 5 is a computed torque effort diagram of a cylinder that is unloaded by blocking the suction inlet. Because the inlet to the cylinder is blocked, the pressure in the cylinder at bottom dead center is a function of the discharge pressure, minimum volume, maximum volume, and the compression exponent. Mathematically it is expressed as $P_2 = P_1 \left(\frac{V_1}{V_2}\right)^n$.

Therefore, from 0° crank angle (bottom dead center) to approximately 158°, the cylinder pressure is below suction pressure; from 158° to 202° the cylinder pressure is above suction pressure; and from 202° to 360° the cylinder pressure is below suction pressure again. The cylinder shown has 2.2% clearance volume and is operating at 300 psig discharge. Changing either of these parameters would change the place where cylinder pressure exceeds suction and the general shape of the curve but would not change the summation of the torque.

The piston imparts torque to the crankshaft when the suction pressure is higher than cylinder pressure and the cylinder volume is decreasing. Also, when the cylinder pressure is higher than suction and the cylinder volume is increasing. Conversely, torque is required from the crankshaft when the suction pressure is higher than cylinder pressure and the cylinder volume is increasing. Also, when the cylinder pressure is higher than suction pressure and the cylinder volume is decreasing. Examination of this torque effort diagram reveals that the summation of torque in any one revolution equals zero. However, instantaneously energy is either being put into or taken out of the crankshaft by the unloaded cylinder. If this cylinder's torque effort diagram is added to other loaded cylinders, the variation in torque required in one revolution could be considerable. Consequently, modulation of this type is generally limited to modulating two cylinders simultaneously that are 180° apart in relation to each other. The modulated cylinder torque effort diagrams then counteract each other so that their torque at any one instant is zero and the only variation in torque effort of a complete machine would then be caused by variations in the pumping cylinders.

In any theoretical comparison of the efficiency of the modulated performance of the suction valve lifter mechanism, the high to low side equalization method or the blocked suction inlet, the friction required to operate the modulated cylinders is a constant and will be neglected from the following comparisons.

Both the suction valve lifter mechanism and the high to low side equalization methods losses are caused by fluid flow. Comparatively, the high to low side equalization method probably has more losses than the suction lifter mechanism because gas is pulled into the cylinder through the suction valves, exhausted through the discharge valves and has to flow through some mechanism back to the suction manifold. Gas flowing through the suction lifter mechanism has only to flow through the suction valve and exit through the suction valve. The suction valve's area generally exceeds that of the discharge valve's area; consequently, its exit losses should be less than the high to low side equalization method. This compares to the blocked suction method whose losses are irreversibility and compression efficiency.

Chart number 1 shows the comparative BTU/watt of a flapper valve machine on a percentage basis. One hundred percent being fully loaded BTU/watt. The improvement in the efficiency between the blocked suction and high to low side equalization is approximately 30% in the normally expected operating range for an air conditioning compressor. This comparison would vary somewhat depending upon flow area of the unloader mechanism of the high to low side equalization system. However, it is thought that they are generally representative. These comparisons were made on a flapper valve machine that could not be fitted with a suction valve lifter mechanism. However, based upon some catalog data, the suction valve lifter mechanism shows an efficiency approximately equal to the blocked suction.

Generally manufacturers have established minimum suction pressures beyond which compressors can not have their capacity modulated. These limits are the result of low mass flow through the compressor and the resultant high temperatures somewhere in the compressor. Limits for discharge valve back stop temperatures have been established. These temperature limits often establish the operating limits of modulated compressors at higher suction pressure than fully loaded. The modulated compressor's discharge back stop temperature is higher because it has lower mass flow to conduct essentially the same energy (neglecting motor losses) from the compressor. In hermetic compressors, the reduction in mass flow when modulated reduces the cooling to the motor and may cause motor overheating at high voltages. Thus motor overheating is another source of modulated operation limits. The limits of the blocked suction and high to low side equalization slightly favor the blocked suction when established on an equal basis.

In conclusion, we would submit that to the traditional three methods of capacity modulation, a new method should be added that is really an adaptation of an old method from another industry that compares favorably with methods available for flapper valve type compressors.

POWER CONVERSION EFFICIENCY

50% Loaded

Fully loaded equals 100

	Condensing Temperature			
	110°F		120°F	
Saturated Suction	High To Low	Blocked Suction	High To Low	Blocked Suction
45	65.5	87.6	67.6	86.9
40	64.2	83.2	67.4	86.3
35	65.2	82.8	67.3	85.2
30	60.9	81.6	67.9	84.2

Chart 1

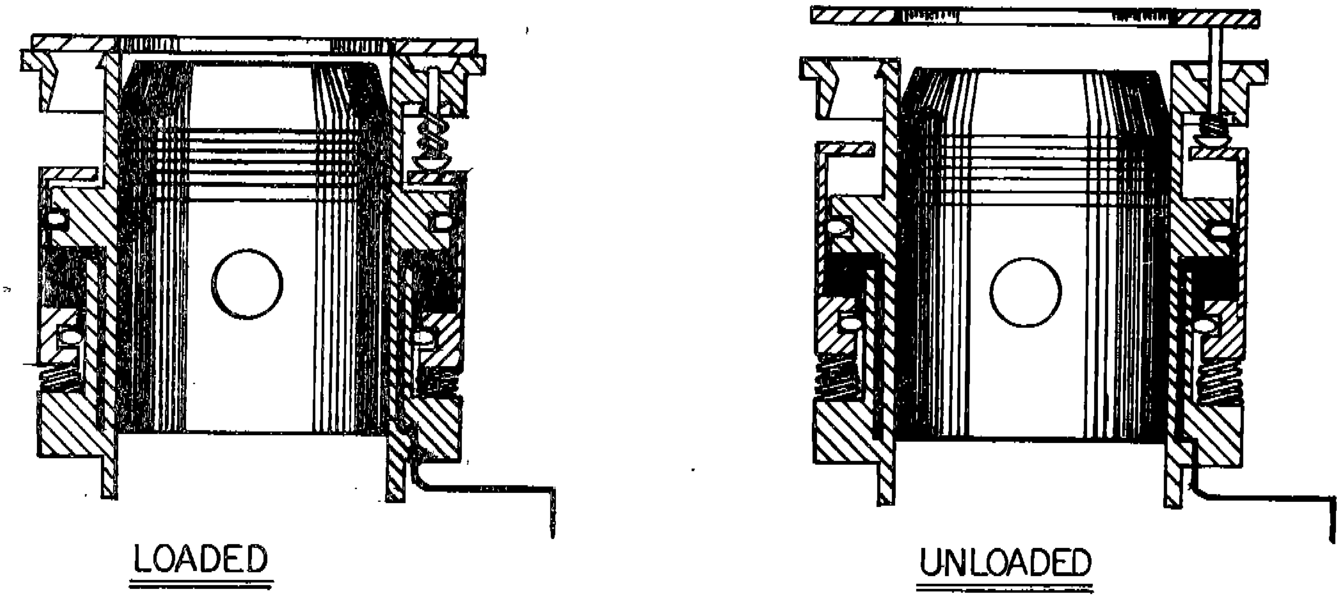
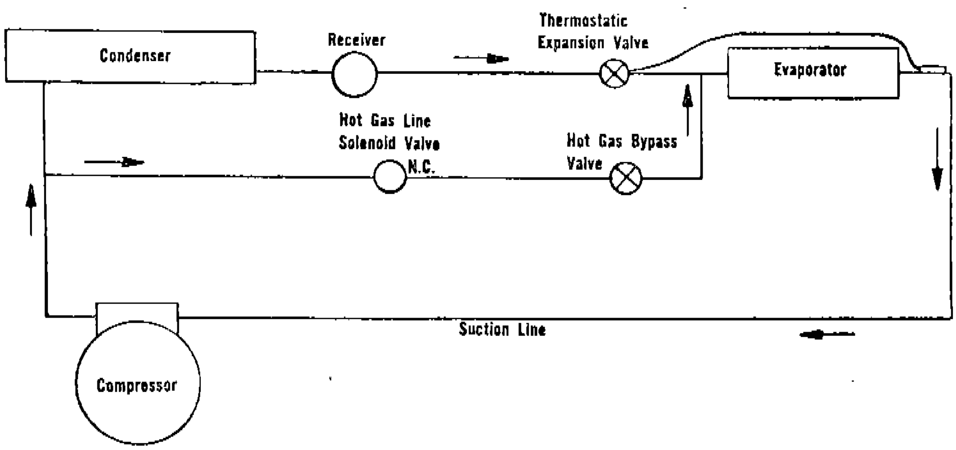


FIGURE 1



HOT GAS BY PASS SYSTEM

FIGURE 2

HIGH TO LOW SIDE EQUALLIZATION

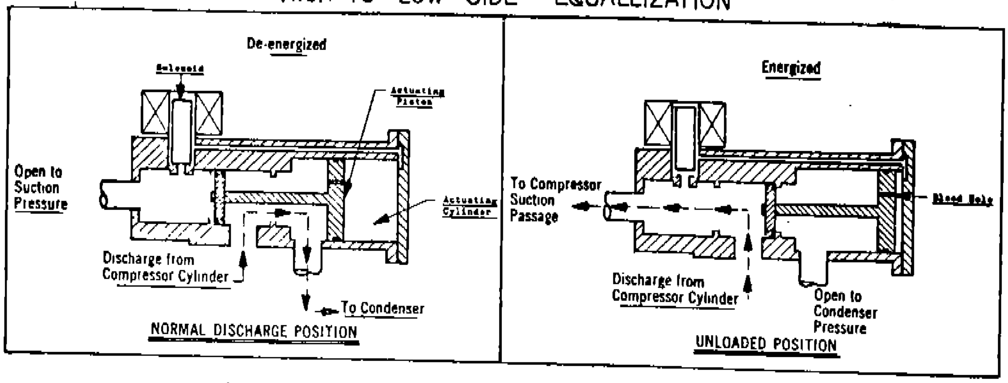


FIGURE 3

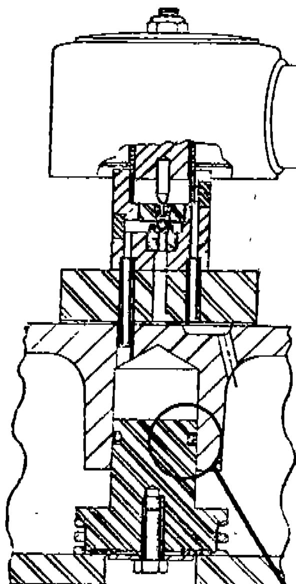


FIGURE 4A

COMPRESSOR UNLOADED

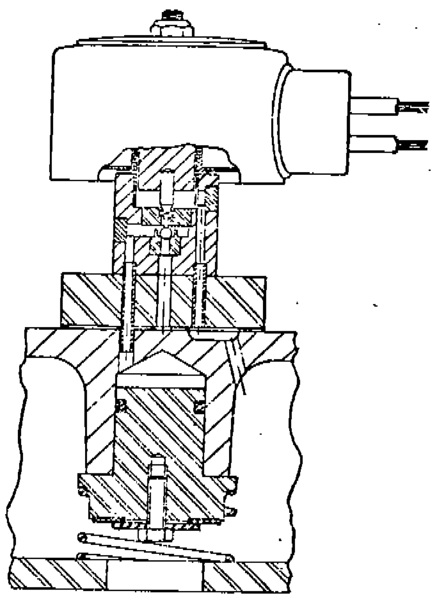


FIGURE 4B

COMPRESSOR LOADED

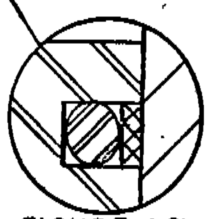


FIGURE 4C

FIGURE 4

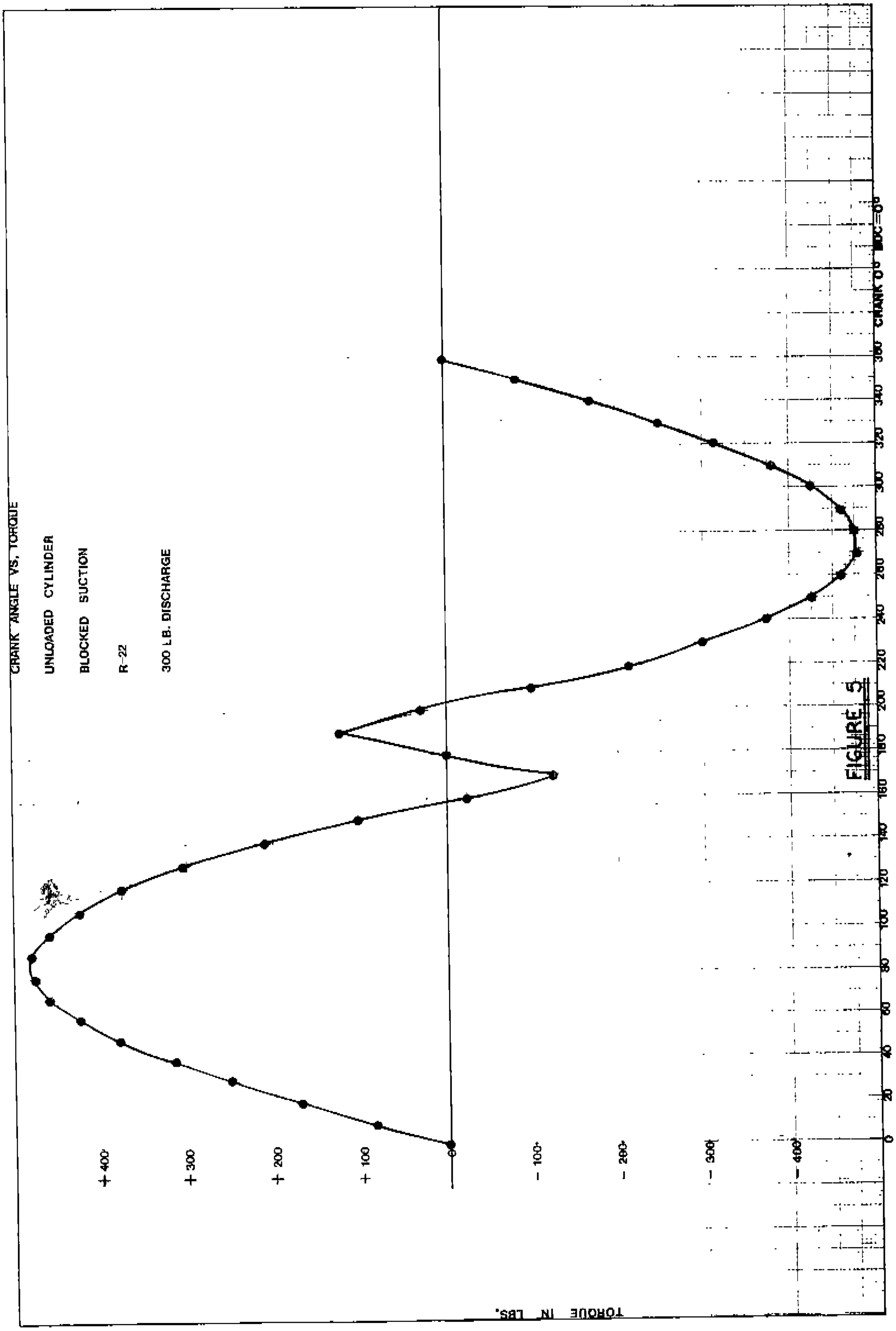

CRANK ANGLE VS. TORQUE
UNLOADED CYLINDER
BLOCKED SUCTION
R-22
300 LB. DISCHARGE
+ 400
+ 300
+ 200
+ 100
0
- 100
- 200
- 300
- 400
TORQUE IN LBS.
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280
300
320
340
360
CRANK 0° BDC=0°

FIGURE 5