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INTRODUCTION

Capacity control of a reciprocating compressor can be accomplished directly by varying the speed or by methods designed to allow direct discharge of vapor from a cylinder to the low pressure or suction side during the so-called compression stroke. The scope of this paper is limited to the latter methods in which the compressor design engineer is interested. The application of these methods is commonly limited to multi-cylinder compressors with displacements above 25 CFM.

The need for compressor capacity control arises from the dilemma of a constant displacement pump being coupled to a system with varying refrigerant flow rates. This system, of course, has the function of balancing the load of a heat influx at a selected temperature at the evaporator. This load will vary from none to a maximum for which the system was selected. This function is most commonly accomplished with a thermal expansion valve which regulates the flow of liquid phase refrigerant into the evaporator. Simply stated, if the heat load decreases the expansion valve will reduce the refrigerant flow rate and vice-versa. If the system has a compressor without means for capacity control the effect of a reduction in refrigerant flow rate will be a lowering suction pressure. The compressor capacity is somewhat reduced as it operates at a higher compression ratio and a lower volumetric efficiency. To some extent this can be tolerated and has been taken into consideration in the system design. However, there will be a minimum suction temperature or pressure below which either the compressor or the system should not operate. It may be the saturated suction temperature at which frost begins to form on an air cooling evaporator coil, or the rather obvious minimum temperature limitations of a water chiller. In some cases, particularly in low temperature refrigeration systems the compressor would overheat as the compression ratio became excessive with reduced flow. Such systems having fixed displacement compressors are controlled simply by starting and stopping the compressor with a thermostat or a suction pressure operated switch. From the household refrigerator through the fifteen ton air-conditioner this arrangement has been satisfactory or made tolerable with the addition of a hot gas bypass system.

An ideal system would have a compressor continually matching the expansion valve with a variable displacement. Ideal temperature control could be realized, the power input would never exceed the requirement and compressor starts and stops would be very infrequent. Presently developed capacity control devices on reciprocating compressors are a practical compromise with this ideal.

There are two functional components to a capacity control device: The sensor-controller and the unloader or unloaders. The sensor-controller may be a pressure actuated type or one which responds to temperature changes, and typically will control the unloader or unloaders electrically, hydraulically, pneumatically or a combination of these. The two common types of unloaders are the discharge bypass type and the suction valve opening mechanism. In order to appreciate the design considerations of the sensor-controller one should first become familiar with the types of unloader and their applicability to certain aspects of compressor design.

DISCHARGE BYPASS UNLOADER

We turn our attention first to the more recently developed of the two types, the discharge bypass unloader. As implied by the name, this unloader in its action isolates a cylinder head from the high pressure side of the compressor and vents it directly to the suction side. The effect is to virtually eliminate the work of compression in the unloaded cylinder or cylinders. There remains only the friction of the moving parts plus the slight compression required to overcome the spring loaded discharge valve(s). It is interesting to note the typical valve used for this type of unloading is an adaptation of the heat pump reversing valve. Both are pilot operated sliding spool type valves capable of handling large gas flowrates relative to their size and cost. In both the moving force is discharge gas pressure controlled by an electric three port solenoid valve. One arrangement is such that the compressor will be loaded so long as the solenoid valve is deenergized. This is felt to be more nearly a fail-safe design with respect to electrical failure of the solenoid valve in a hermetically sealed compressor. The unloader valve is designed with an unbalance so that the cylinder discharge will drive it to the loaded position when the pressure actuating port is vented to the crankcase or low side. Energizing the solenoid pilot valve supplies high pressure to the actuating port taking advantage of the unbalance and moving the spool piston to the unloaded position. In this position the high side of the of the compressor is sealed from the high side of the unloaded cylinder or cylinders while a vent has been opened from this unloaded cylinder(s) high side to the low side.

The particular advantage of this type of unloader is its adaptability to compressors with suction valves of a type which cannot be held open mechanically as for example the reed which opens toward the piston and must be closed to clear the piston at top dead center. It is by no means limited to this type of compressor since in some cases it is found to be adequate and less costly than the suction valve opening mechanism. It will normally be found on compressors in the 25 cfm to 60 cfm range (10 to 40 tons air-conditioning capacity with R-22). In the lower end of this range one unloader to reduce the capacity by 50% is sufficient. Such a compressor will be a four cylinder Vee type with one bank of two cylinders served by the unloader. Larger compressors with more than two cylinder banks can have two or more unloaders controlled in stages to match the system load more closely.

SUCTION VALVE OPENING UNLOADER

Looking now at the suction valve opening type of unloader we must recognize a design prerequisite which is that the suction valve must not interfere with the piston when held open continuously and, of course, it must be physically possible to include a mechanism in or about the valve assembly which can force the suction valve open and hold it open. The arrangement we see in most modern refrigerant compressors has a non-flexing ring shaped suction valve plate located concentric to and outside of the cylinder bore near the top of the piston stroke. Typically, this valve opens by lifting upward against springs which are selected for efficient closing of the valve at the bottom of the stroke. Suction vapor in the chamber around the cylinder flows upward through ports between the concentric suction valve seats and into the cylinder. The discharge valve and ports are at the top of the cylinder. (We recognize here the added clearance volume due to the space provided for the lift of the suction valve). The means for mechanically opening the suction valve typically consists of six or eight rods evenly spaced around the cylinder and positioned so that the upper end can move up through guide holes between the valve seats to contact and lift the valve. At the lower end the rods rest on a movable ring positioned concentrically around the outside of the cylinder sleeve. This ring in turn is a part of or is moved by a hydraulic mechanism. We should note here that the valve lifter rods are either permanently affixed to the lifter ring or are headed at the lower end, and both types have small compression coil springs slipped over them and positioned between the heads and guide holes so as to hold the rod and ring away from the valve during loaded operation. Some type of retainer is often provided so that these parts can become a part of the cylinder sleeve assembly for convenience in handling.

The operating part of the unloader is a hydraulic mechanism working with lubricating oil pressure against a return spring or springs. Common to several designs in use is the "fail safe" feature wherein the loss of lubricating oil pressure results in the unloading of all cylinders having the mechanism. This likewise means that all such cylinders are unloaded at start up. Perhaps it is already evident from the preceding remarks that oil under pressure when supplied to the operating mechanism causes the cylinder to load which means in turn that in the absence of oil under pressure the return spring or springs provide(s) the force to lift the suction valve and hold it open. It is interesting and important to note here that this lifting force must be around 60 lbs with a cylinder bore of around 3-1/2 inch if the valve is to be restrained from any movement at high back pressures such as compressor start-up with R-22 at 100 psig suction pressure in an air conditioning system. Suction vapor density, height of suction valve lift, and other details affecting gas flow past the edge of the valve are significant in the determination of an adequate unloader lift force. At Airtemp we have relied successfully on past experience plus laboratory tests with operating compressors to work this out. Once the unloading force is established and a spring or springs are selected to provide this force the rest of the mechanism must be designed to overcome the unloader springs with a selected oil pressure. This will usually be the minimum oil pressure at which the compressor is expected to operate safely.

The most common mechanism is essentially a first class lever, yoke shaped, so that the open ends support the lifter ring at points 180° apart, and the closed end is either in direct contact with a piston or connected to a piston through a link. A pair of springs may be located under the open ends of the yoke or one spring may be positioned so that it directly opposes the piston. The unloader cylinder is a part of a casting which includes the pivot points and the mounting means. An older design uses the yoke as a sliding member with enlarged ends which ride on ramps affixed to the crankcase on either side of the cylinder sleeve. The upper portion of the enlarged ends are in contact with a lifter ring raising it or lowering it as they move up or down the ramps. The travel required to accomplish the loading and unloading does not take the points of contact between the yoke and the ring far enough away from center to cause the ring to tip and bind. The mechanical advantage of the ramps provides a strong lift force with a comparatively lighter spring, and a smaller piston area is possible for a given oil pressure.

One make of compressor has a distinctly different mechanism which is in effect an annular cylinder and piston around the outside of the cylinder sleeve. The stationary portion fits on the outside of the cylinder sleeve and is flanged outward at the top. The movable member is essentially a sleeve-shaped part clearing the stationary flange at the top and flanged inwardly at the bottom. Large O-rings seal at the flanges which serve to retain them. When oil under pressure is supplied to the annular space between the stationary inner and movable outer members the latter is driven downward and a lift ring resting on the upper edge of the movable part transmits the motion to the lifter rods as in other mechanisms. In such a design as this it is no simple matter to select O-ring materials which will neither shrink nor swell appreciably in the refrigerant and oil environment. The several refrigerants vary in their effect on elastomers, and Oil Brand "B" will swell a certain compound much more than Brand "A" although either brand is a good refrigeration grade of oil. Thus, it is not unusual for a refrigeration compressor manufacturer to be quite specific in naming the brand of oil which is acceptable in their equipment.

SENSOR-CONTROLLERS

To control the compressor capacity we must arrange to sense the need for greater or lesser refrigerant flow and as directly as possible cause the unloading mechanism to actuate. Looking first at the discharge bypass unloader you will recall that an electric 3-port solenoid valve when energized supplies discharge gas pressure to cause unloading or when deenergized vents the larger area piston to the crankcase to permit loading. Control can therefore be quite straightforward. A pressure switch sensing suction pressure can be set to close an electric circuit when the pressure drops to a selected level. A practical setting for a 1-step unloading compressor might be around 55 pisg for an R22 system with an air cooling evaporator coil. The compressor will unload just above a temperature which would cause frost to form on the coil. The pressure setting for loading the compressor will probably be about ten to fifteen psi higher so that loading will occur when truly needed, and too frequent cycling of the unloader will be avoided. A six cylinder compressor (with three banks of two cylinders) having two steps of unloading will be controlled with two pressure switches staged so that unloading to 2/3 capacity occurs when the load drops just below the design conditions. Such a pressure control may be in fact a single unit having two sets of electrical contacts arranged to effect the staging.

Direct response to temperature changes simply requires use of a thermostat to control power to the unloader solenoid valve. This is particularly desirable in liquid chilling systems where closer control is usually required. Again staging with a two-step thermostat is possible with compressors having two steps of unloading. Incidentally the electrical contacts in the controls discussed here are subject only to the load of less than 10-watt solenoid coils. Also worth mentioning is the feasibility of temperature control through a pneumatic system to pressure-electric switches.

A fascinating degree of mechanical ingenuity has gone into designs of controllers for the hydraulically operated suction valve opening unloaders. Control by response to changes in suction has been with us for at least thirty-five years, and one such design, basically unchanged, is still in use today. The basic components are a spring balanced metal bellows about three inches long and one inch inside diameter, a detenting piston-in-sleeve type valve and the connecting lever. This mechanism is built into one of the cover plates closing the compressor crankcase. The bellows is soldered to the inner edge of a flange at one end such that the bellows can be inserted inwardly through a hole in the cover and the flange bolted to the cover with a gasket. A rod about three times the length of the bellows with a washer shaped plug brazed about one-fourth of the way from the inner end is assembled to the bellows by soldering the plug to the inner end of the bellows. The outer fourth of the rod is threaded. A tubular piece is also attached to the inside of the flange such that the inner end is a support and guide for the bellows as well as a positive stop for the inner end to prevent overcompression. The outer end of the tubular piece is threaded on the outside and has two length wise slots 180° apart running from the flange to the outer end. To complete this portion of the assembly a compression coil spring is inserted over the rod into the bellows followed by a winged washer fitted in the slots and a nut over the threaded tube engaging the tabs of the washer driving it to compress the spring. A special nut is assembled to the rod so that a pilot diameter centers the rod in the tube, and a shoulder bears against the end of the tube to limit inward travel. The nut can be adjusted to permit full travel or no travel at all. The inner end of the rod is connected to a lever which is pivoted to multiply the bellows travel about three to one. Since the bellows travel is limited to about 1/4 inch to avoid fatigue failure the valve piston may have up to 3/4 inch travel.

The valve assembly consists of a body, sleeve, spool type piston and detenting parts. They are so designed that oil under pressure supplied to the valve either can be directed to the several unloading stages in sequence or can be cut off, and the individual unloader oil circuits vented to the crankcase. As the piston is pulled out of the valve sleeve oil is fed to the unloaders. This will occur when rising suction pressure (which is vented to the crankcase) compresses the bellows. Proper positioning of the piston with respect to the valve ports is very important and is achieved by having a series of Vee notches in the lever end of the piston into which diametrically opposed spring loaded closely guided steel balls tend to nest. It is critical that, when the bellows is moving, the piston not move unless and until the suction pressure has changed sufficiently to require an increase or decrease in compressor capacity. Dragging of the piston through from one position to another slowly closeing or opening ports causes partial or slow lifting of the suction valves resulting in a severe pounding of the cylinder unloading mechanism by the valve hammering on the lifter rods. To avoid this the lever is in fact a straight piece of spring wire which deflects until enough movement of the bellows causes it to overcome the detent mechanism and move the piston quickly to the next position. In practice a loose fitting stamped steel lever is used along with the spring wire as a precaution against binding.

The piston, sleeve and body are close fitting to minimize oil leakage, and the latter two parts combine to provide precisely located ports and connecting passages to the cover plate where drilled holes are arranged to connect the valve to tubing in the crankcase.

To summarize the description of this mechanism, it is mounted on the inside of a cover plate in such a way that increasing suction pressure resulting from increased refrigerant flow at the evaporator compresses the bellows and spring causing motion of the spring wire lever which tends to pull the valve piston out of the valve. Sufficient bending of the lever will snap the piston out one step. Oil under pressure will then be fed to the hydraulic mechanism of a particular cylinder which moves to drop the lifter ring and rods letting the suction valve open and close normally. Conversely a reduction in refrigerant flow at the evaporator will cause a drop in suction pressure until the balancing spring moves the bellows inward enough to move the piston one step back. This vents the port of a particular unloader to the crankcase, and the lifter spring or springs in the hydraulic mechanism force out the oil and the ring and lifter rods open the suction valve and firmly hold it wide open. A selection of pressure balancing springs permits control from suction pressures below one atmosphere up to about 85 psig covering a temperature range from -80°F to +50°F with the commonly used halocarbon refrigerants. The mechanism is also adaptable to external drives connected to the outer end of the movable rod.

Another make of suction pressure sensor-controller utilizes the detenting spool piston and the spring balanced bellows, but in place of the lever is an interesting oil valving, hydraulic action design. Briefly, movement of the bellows opens or closes a needle valve, and the spool piston having a similar detenting design to that described above is moved in one direction as a hydraulic piston and returned by the force of a compressed spring. There is a small bleed in the oil supply circuit to the piston such that the bellows operated needle valve may balance the outflow of oil to hold the position of the piston unchanged, overcome the bleed to move the piston against the spring, or close and let the spring return the piston. The piston, serving as a spool valve, is controlling the unloader oil supply as described in the preceding paragraph. A design of this type includes ingenious means for damping of bellows and detenting piston movement and balancing of small port valves and bleed orifices.

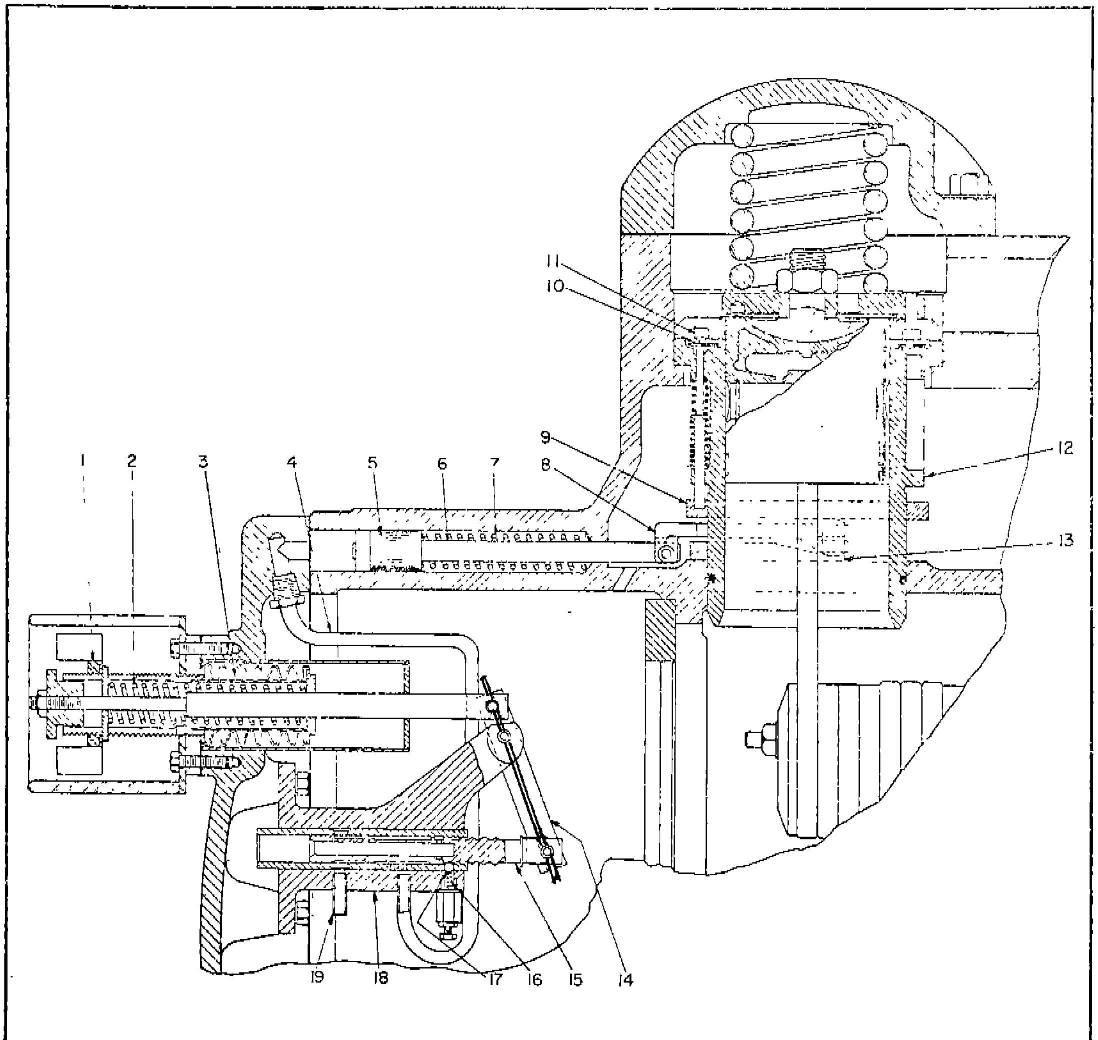
Finally, we will look at the electrically controlled systems which generally are of more recent development. Basically these differ from the suction pressure controlled type in that only the oil valving part of the system is built into the compressor as a cover assembly. The sensing part of the system may be elsewhere and connected electrically.

One such unloader control consists simply of three-port solenoid valves which control the supply of oil to individual steps of unloading. Energized the solenoid valve supplies oil under pressure to the cylinder unloader mechanism, deenergized it relieves the oil to the crankcase unloading the cylinder or cylinders.

Another unloader control utilizes much smaller less expensive solenoid valves as pilot valves which supply oil under pressure to spring loaded spool pistons. Movement of the latter control oil to the unloader mechanisms. A bleed is used in conjunction with each solenoid valve so that oil supplied by the solenoid valve can be bled off when the valve closes permitting the spool piston to return. The orifice is very small and is protected by a sintered metal filter from small foreign particles which might obstruct the opening. The advantage of this pilot operated system is that larger porting for oil supply and drain is possible economically, and more rapid response by the cylinder unloading mechanism is assured. Control of the solenoid valves by staged pressure or temperature sensors is straightforward with temperature control being the more common. Pneumatic control through staged pressure electric switches is also possible.

Important to much of the international market where star-delta type of electric motor starting is commonly used is the availability of compressors which can be completely unloaded so that the starting motor is not overloaded. If the compressor has unloaders in all cylinders it is no problem with electric unloader control to hold all cylinders unloaded until after starter transition where the motor has come up to speed. Then the power to the capacity control system can be closed and normal control assumed. Except when starting, the compressor should never be operated completely unloaded as friction will eventually cause serious internal overheating and damage to the compressor will result.

There are distinct advantages and disadvantages to both the suction pressure sensing and the temperature sensing type of unloader control. The former is very satisfactory in systems having an air cooling evaporator coil usually some distance away from the compressor. Normally the compressor capacity follows the dictates of the thermal expansion valve, or valves if there is more than one circuit and/or coil. Thermostats may control solenoid valves in the liquid refrigerant lines to the expansion valves so that flow is stopped entirely when the thermostats are satisfied. If all flow is stopped the compressor operating at minimum capacity will pump the boiling refrigerant out of the evaporators reducing the suction pressure until a pressure control is tripped and opens the compressor control circuit stopping the compressor. No matter what the circumstances when



SLIDING YOKE AND RAMP UNLOADER WITH SUCTION PRESSURE SENSOR-CONTROLLER

1. Unloader spring adjusting nut
2. Unloader spring
3. Unloader bellows
4. Unloader oil tube
5. Piston
6. Plunger
7. Spring
8. Unloader lifter arm assembly
9. Unloader lifter ring assembly
10. Suction valve plate
11. Suction valve plate spring
12. Cylinder liner
13. Unloader ramp
14. Linkage
15. Indexing piston
16. Indexing springs
17. Indexing ball
18. Unloader body
19. Oil supply tube

the compressor is started, the suction pressure sensing controller will quickly regulate the compressor to balance the conditions of the system.

The principal disadvantage of suction pressure control is evident when the volume of suction vapor is relatively small compared to the system capacity, as in a liquid chiller with compressor and evaporator mounted together connected with a short suction pipe. The effect of loading or unloading a cylinder or bank of cylinders will be a change in suction pressure so rapid and so great that the reverse action will quickly follow and rapid cycling will persist. Widening the pressure difference between the loading and unloading of a particular step results in poor control of chilled liquid temperature.

Since most chilled liquid systems whether chilled water for air-conditioning, chilled liquid for process control or chilled brine for ice rinks or controlled cold rooms, are design based on a closely controlled liquid temperature, only a temperature sensing capacity control system will be satisfactory. Thus, the staged thermostat is directly controlling the refrigerant flowrate by regulating the pumping rate of the compressor while the thermal expansion valve follows in controlling the feed of liquid refrigerant into the evaporator as needed. Typically the thermostat senses the temperature of liquid returning to the chiller.

The most serious disadvantage of the temperature sensing system is that it will not recognize an unusual deviation of suction pressure from normal. Too low a suction pressure in a water chiller might result in a quick freezing condition which the thermostat would not sense soon enough especially if the water flow had slowed or stopped. Protection is obtained by adding a pressure switch to the system set to open at least half of the capacity control stages immediately and ultimately to stop the compressor if the partial unloading does not relieve the condition.

Before closing let us look at the performance of the two types of cylinder unloaders. This is not so much a comparison since basic design differences in the two compressors contribute to the differences apparent here.

POWER AND CAPACITY VERSUS STAGE OF LOADING
(Suction Valve Opening Type)

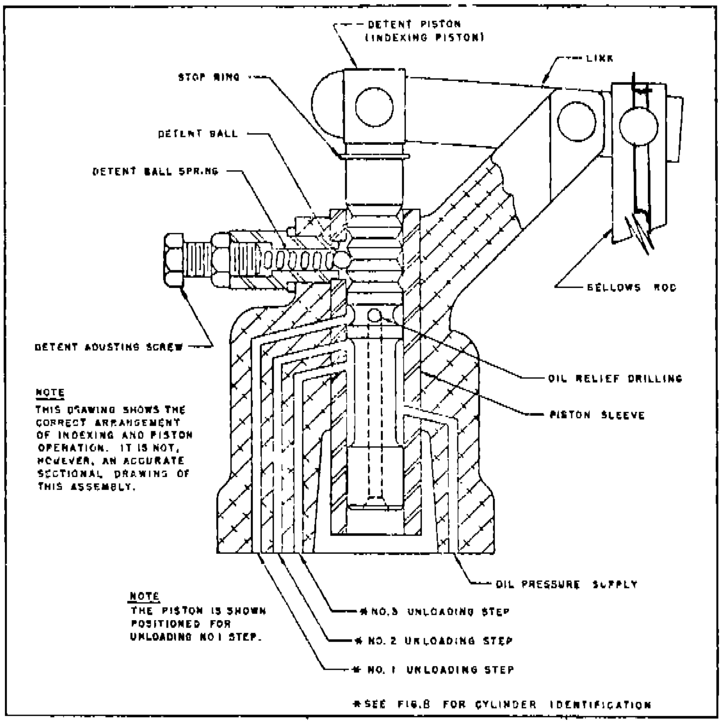
Stage	KW	Capacity (Tons)
100%	37.0 (100%)	38.0 (100%)
75%	28.9 (78.1%)	28.6 (75.3%)
59%	20.9 (56.5%)	18.8 (49.5%)
25%	12.5 (33.8%)	8.8 (23.2%)

(Discharge Bypass Type)

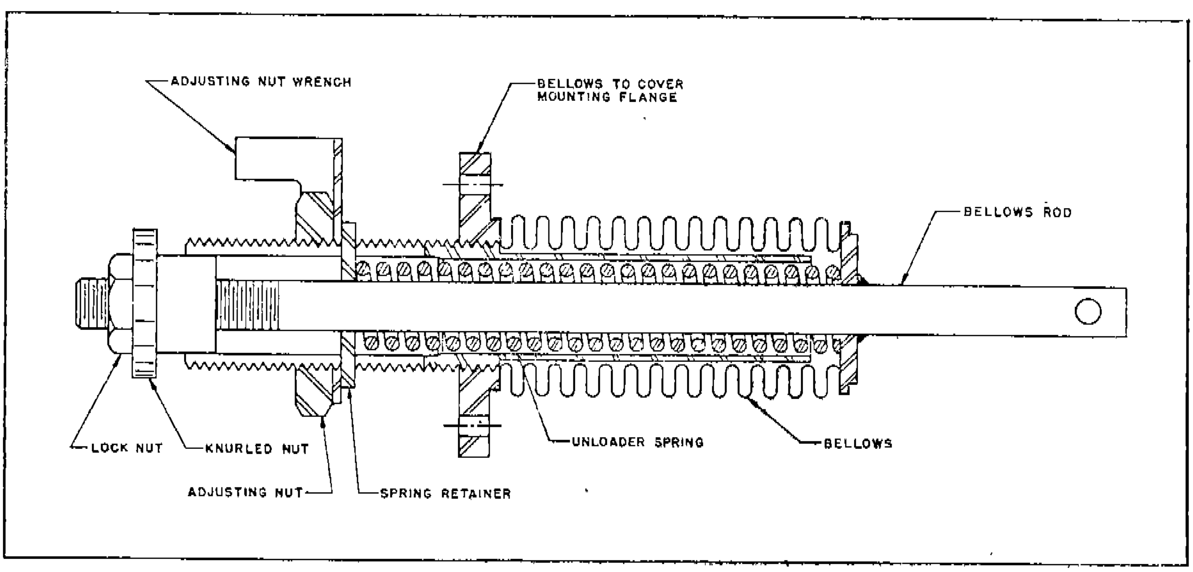
Stage	KW	Capacity (Tons)
100%	17.9 (100%)	14.3 (100%)
50%	12.2 (71.8%)	7.0 (49%)

The difference in per-cent of power reduction at 50% capacity cannot escape attention although it is exaggerated as an indication of the difference between the two types of unloaders. To some extent it reflects the added power required to push the cylinder gas through the discharge valves in the discharge bypass system. It also reflects a characteristic difference between higher and lower speed compressors, the latter tending to be more efficient. In both cases as the load is reduced friction of moving parts and characteristics decrease in motor efficiency become more dominant.

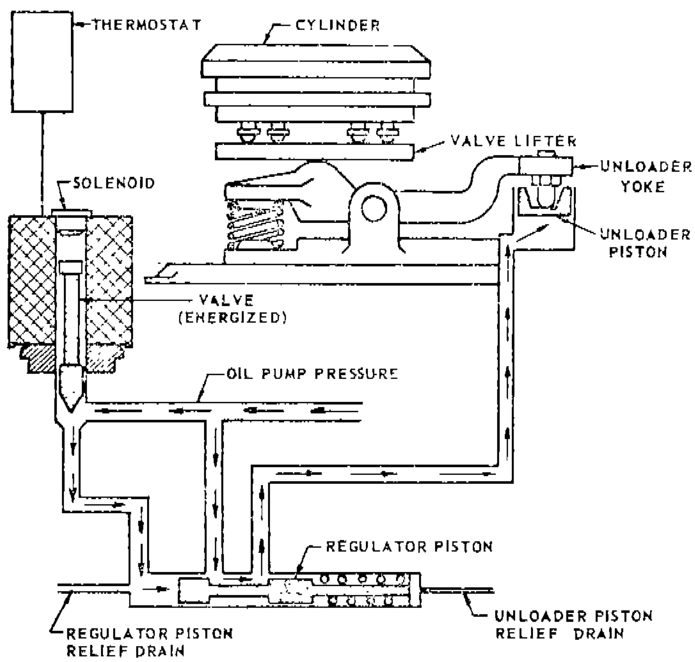
In conclusion it appears to this writer that the state of the art is quite well developed and that most of what we will see henceforth will be variations and refinements of the mechanisms and controls described herein. The principle shortcoming of any of these is that reciprocating compressor capacity is controlled in finite steps so that hunting between two steps can be annoying. This probably indicates the major capacity control design challenge today.



UNLOADER CONTROL VALVE - SUCTION PRESSURE SENSING TYPE



UNLOADER BELLOWS AND SPRING ASSEMBLY



ELECTRIC UNLOADER MECHANISM