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A COMPARISON OF CAPACITY MODULATION MEANS FOR HERMETIC COMPRESSORS

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INTRODUCTION

Increased energy costs and Government regulations have caused considerable attention to be directed toward capacity modulation to achieve energy conservation. The major advantage claimed for modulation over conventional on-off systems is that the dynamic losses during start up and shutdown are minimized because the modulated systems experience fewer on-off cycles. Also, for heat pump systems, compressor capacity can be better matched to the heating load, thereby reducing the amount of auxiliary heat required.

Capacity modulation systems with separate compressors and separate refrigeration circuits have been in use for many years. In some systems, common air handling equipment with fan speed controls are used for the evaporator and condenser. In some of these designs, interlaced evaporator tubes are used to take advantage of the total fin area required for maximum capacity when the system operates at reduced capacity.

This paper discusses compressor capacity modulation techniques that can be applied to single refrigerant circuit systems to achieve energy conservation. Three concepts of compressor capacity modulation are considered to be within the "state of the art" to meet the objectives of high efficiency, satisfactory reliability, reasonable cost, and early availability. These concepts are: twin compressors, compressors with blocked-suction cylinder unloading, and two-speed compressors. Although these modulation concepts are considered within the "state of the art", production compressors are not generally available. The data and conclusions in this paper are based on laboratory data, cost projections, and reliability estimates for prototype designs. Prototypes of all three concepts have been constructed and tested.

DESIGN FEATURES

Twin-Compressors - Single Shell

Figure 1 shows the twin-compressor in a single shell concept. The major advantage of this concept is that standard compressors, with their standard protectors, and standard starting contactors can be used.

The space between the compressors is used for a "built-in" suction-line accumulator which protects the compressors from liquid refrigerant. The refrigerant from the evaporator enters the accumulator through a single line. Liquid refrigerant and oil separate from the suction gas in the accumulator and drop to the bottom. Liquid-free refrigerant flows from the accumulator to one or both compressors through the top openings. Oil and liquid refrigerant are metered through the small hole in the bottom of the accumulator into the oil. Venting of the space around each compressor is accomplished through the area between the top of the accumulator and the top of the shell.

The ends of the accumulator are welded to the upper shell sides, which strengthens and stiffens the shell and improves its hydrostatic and acoustic characteristics. Because of the large shell required to enclose the twin compressors, it is believed that this concept will be limited to the 1 to 5 horsepower size range. This range is adequate to serve the room and unitary residential air conditioning market.

The oil level in this twin compressor concept is at the same level as a standard compressor. Therefore, the total quantity of oil, because of the added volume underneath the accumulator, is slightly more than twice the quantity of a standard compressor. With the large vent area available across the top of the accumulator, the oil level is the same for both compressors during operation of one or both compressors.

The entire oil sump is kept warm even when only one compressor is operating. This warm oil heats the cylinder head of the idle compressor and prevents liquid refrigerant from migrating into this volume. Liquid migration into the cylinder head of the idle compressor has been the cause of slugging and early compressor failures on some twin compressor installations.

Each compressor has its own discharge line, which is joined to the other outside the shell. Separate lines are desirable in order to minimize stress during compressor starting and stopping.

Twin compressors with single phase motors may require auxiliary starting devices in some applications. If it is desired to modulate capacity without stopping the compressor, capacitor or PTC start assist is required by one of the compressors. Alternatively, if the compressors are turned off and the system is allowed to equalize pressure in the process of changing capacity, then standard PSC motors can be used. Of course, twin compressors driven by three-phase motors can be modulated in either manner without any auxiliary starting or protective devices.

One of the intriguing characteristics of the twin compressor is the ability to modulate capacity in more than one step. For example, a three ton unit could combine a two ton and a one ton compressor, and with the proper control circuits could have 33%, 67%, and full load capacities.

Twin-Compressors - Separate Shells

Figure 2 shows the twin-compressor in separate shells. As with the single-shell concept, the major advantages are that standard compressors, protectors, and contactors can be utilized. In addition, standard shells - with the exception of the provision for the oil and pressure equalizing lines - are used.

All of the other characteristics and requirements are the same as the twin-compressors in a single shell, except:

1. Specific applications will have to be evaluated to ensure that the idle compressor will not slug when cycling on.

2. This concept does not have the size limitations of the single shell and is, therefore, better suited for large hermetic compressors.

Blocked-Suction Cylinder Unloading

Figure 3 is a schematic diagram of the blocked-suction cylinder unloading system. Figure 3A shows the unloaded mode of operation where the unloader valve is actuated into a position to prevent flow of suction gas to one or more cylinders of a multi-cylinder compressor. In the unloaded mode, the discharge valve is exposed to condenser pressure. Therefore, because of minor leakage between piston and rings, the cylinder pressure will be referenced to the condenser pressure at top-dead-center. The pressure between the suction valve and the unloader valve - within a few cycles - will reach a pressure that is a function of the clearance volume, the displacement of the cylinder, and the compression exponent of the refrigerant. When this minimum pressure is reached, there is theoretically no flow through the unloaded cylinder(s), and all of the energy delivered to the gas on the compression stroke is returned on the expansion stroke. Losses do occur, however, because of minor leakage across the piston and rings, back-flow through the discharge valve, and the irreversibility of the compression and expansion processes. Figure 3B shows the loaded mode of operation where the unloader valve is actuated into a position to allow suction gas flow to all of the cylinders and normal full-capacity operation.

Figure 4 shows an installation of a blocked-suction cylinder unloader on a two-cylinder hermetic compressor. Figure 4A shows this compressor under full load operation where the suction gas flows through the suction tubes into the suction plenum of the top cylinder and then through the unloader port into the suction plenum of the lower cylinder. Figure 4B shows the compressor under the unloaded mode where the unloader valve is actuated to close the unloader port and prevent flow of suction gas to the lower cylinder.

Actuation of the unloader valve to the unloaded position is by discharge gas acting on the unloader valve piston; introduction of the discharge gas to the piston is through a three-way solenoid valve. For full load operation, the solenoid valve vents the unloader piston to suction pressure and a spring moves the unloader valve from the unloader port.

Figure 5 shows an outline view of a hermetic compressor with the solenoid valve mounting. This compressor modulation system does not require any increase in the size of the compressor.

The advantage of this concept is that standard compressor components - except the cylinder head - are used, and only the addition of the modulation valve and the three-way solenoid valve is required. This system can be modulated from high to low capacity without stopping the compressor and does not require the addition of a starting assist.

Two-Speed Compressor

Figure 6 shows the connections and features of three different types of three-phase consequent-pole motors for two and four-pole operation: variable torque, constant torque, and constant horsepower. For three-phase power, the consequent-pole, two-speed motor is a single winding two-pole motor that has the current flow through one pole reversed and connected for the same magnetic polarity to create a four-pole equivalent motor. Since the compressor requires essentially a constant torque, the constant horsepower technique is not considered practical. The variable torque technique produces a full load torque at four-pole operation that is only half of the two-pole torque. Therefore, the motor is weak for four-pole operation and could result in rundown and starting problems. The constant torque motor has the same full-load torque at four-pole operation and two-pole operation. This motor type, therefore, should not have rundown and starting difficulties; however, the power factor is low. Also, a longer stack than the variable torque approach is required to keep the four-pole flux density low.

Figure 7 shows typical winding and connection diagrams for a single phase two-speed motor with a 24 slot stator lamination. The single phase motor has two sets of main windings. One is two pole winding which can be connected either in parallel for two-pole operation, or in series as consequent pole connection for four-pole operation, while the other single coil winding is connected in conjunction with the consequent pole winding for two-pole operation only, and is a dummy for four-pole operation. Separate phase windings are used for two-pole and four-pole with only one phase winding energized at one time.

Because of tooling cost and availability problems, only two-pole laminations have been investigated. The use of two-pole laminations results in lower motor efficiency at four-pole operation. It is possible, however, to design special laminations and windings for two-speed motors such that the high and low speed motor efficiencies can be essentially the same. However, the efficiency at two-pole operation will not be as high as the efficiency of a single speed, two-pole motor. Compared to a single speed motor for the equivalent horsepower rating, the consequent-pole, two-speed motor requires a longer stack and a greater stator slot area, whether it is a single-phase or a three-phase motor.

The contactors for two-speed motors are more complicated than single speed motors. By comparison, the contactors for a single-speed, single-phase motor have only two poles, while the contactors used for two-speed, single-phase motors have five poles. The contactors used for single-speed, three-phase motors have only three poles, while the contactors used for two-speed, three-phase motors have eight poles.

SIZE, WEIGHT, AND COST COMPARISONS

Table I shows a comparison of the size and weight of the various concepts for different capacities. In all cases, cylinder unloading is the smallest and lightest, and twin compressors are the largest and heaviest.

Table II shows a comparison of the relative cost premium over standard non-modulated compressors. The rating numbers are not absolute and are rankings only, with the value 1 being the lowest cost premium and the value 3 being the highest cost premium.

PERFORMANCE

Four performance characteristics of the prototypes can be compared: capacity, efficiency, noise, and reliability. Final production designs will have improvements over the prototypes; however, the relative comparisons of the prototype data is considered reasonable.

Capacity-Efficiency Performance Comparison

Table III shows the calorimeter performance comparison for the three different compressor modulation methods at full capacity and half capacity with single- and three-phase motors at ASRE/T condition. Table IV provides the same comparisons at heat pump conditions. Table V shows the performance of a twenty-two ton hermetic compressor with blocked-suction cylinder unloading.

The twin compressor is the most efficient with no loss at half capacity. With an improved motor, the loss at half capacity for blocked suction unloading can be improved. The loss at half capacity for the two-speed compressor can also be reduced by utilizing a better, more expensive motor. It is believed that an EER loss at half capacity of about 9% can be achieved, while maintaining the high EER at full capacity, for both the blocked suction and two-speed compressors.

RELIABILITY

Laboratory and field tests to establish reliability have not yet been conducted. However, a failure mode and effect analysis shows that the reliabilities of all the modulation approaches are lower than a conventional single-capacity compressor, but probably adequate if properly applied. In relative terms, cylinder unloading appears to be the most reliable, and two-speed motors the least reliable.

NOISE

Table VI shows the measured noise level for three types of modulated compressors in the size range of four tons. As can be seen, the twin compressors are the least noisy.

SUMMARY AND CONCLUSIONS

The characteristics of three approaches for modulating capacity of hermetic compressors have been reviewed in this paper. All three compressor modulation systems are considered feasible and practical from the standpoint of performance, reliability, and cost. No one system is superior in all three characteristics. Therefore, depending upon the system requirements, all three concepts could have their place in the market. In general, the modulation systems are rated as follows:

1. Cylinder unloading is the most reliable and least expensive with acceptable unloaded efficiency.

2. Twin compressors are the most efficient in the unloaded mode with acceptable reliability and cost.

3. Two-speed motors are acceptable in performance, cost, and reliability.

TABLE I

SIZE & WEIGHT COMPARISON

Type	4 Ton	10 Ton	22 Ton
Twin Single Shell	15.0 X 13.3 X 11.0 120 lbs.	NA	NA
Twin Separate Shell	14.5 X 15.3 X 9.4 122 lbs.	17.8 X 20.0 X 11.7 225 lbs.	18.6 X 27.2 X 14.2 320 lbs.
Cylinder Unloading	17.8 X 9.0 X 11.7 104 lbs.	18.6 X 12.6 X 14.2 155 lbs.	23.3 X 14.7 X 17.4 255 lbs.
Two-Speed Motors	18.8 X 9.0 X 11.7 114 lbs.	19.6 X 12.6 X 14.2 170 lbs.	24.3 X 14.7 X 17.4* 275 lbs.

* Estimated

TABLE II

APPLIED COST COMPARISON

	Three Phase		Single Phase
	25 HP	5 HP	4 HP
Cylinder Unloading	1	1	1
Twin Compressor	3	2	2
Two-Speed Motors	2	2	3

TABLE III

AIR CONDITIONING PERFORMANCE COMPARISON

(Test Conditions: 45°E/130°C; 95°S/115°L; 95°A)

	Full Capacity		Reduced Capacity		Capacity Reduction	EER Loss(%)	Expected EER Loss(%)
	BTU/HR	BTU/W-HR	BTU/HR	BTU/W-HR			
Single Phase							
Twin Compressor	44,000	9.6	22,400	9.6	50%	0	0
Cylinder Unloading	53,800	9.6	28,200	8.4	48%	12	9
Two-Speed Motor	52,200	9.2	24,000	8.2	54%	11	9
Three Phase							
Cylinder Unloading	61,400	9.7	32,500	8.8	47%	9	9
Two-Speed Motor	53,000	10.0	24,400	8.3	54%	17	9
Two-Speed Motor	61,900	9.8	27,700	7.9	55%	19	9

TABLE IV

HEAT PUMP PERFORMANCE COMPARISON

(Test Conditions: 30°E/110°C; 40°S/95°L; 47°A)

	Full Capacity		Reduced Capacity				
	BTU/HR	BTU/W-HR	BTU/HR	BTU/W-HR	Capacity Reduction	EER Loss(%)	Expected EER Loss (%)
Single Phase							
Twin Compressor	34,800	9.7	17,400	9.7	50%	0	0
Cylinder Unloading	42,600	9.6	22,200	8.1	48%	16	9
Two-Speed Motor	42,700	9.3	19,900	8.4	53%	10	9
Three Phase							
Cylinder Unloading	49,500	9.8	26,300	8.6	47%	12	9
Two-Speed Motor	44,000	10.3	20,100	8.7	54%	15	9

TABLE V

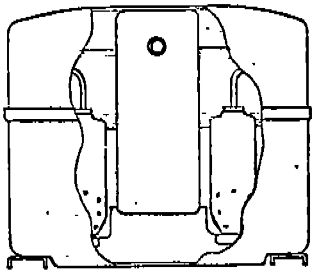
(Test Conditions: 45°E/130°C; 65°S/115°L; 95°A)

	Full Capacity		Reduced Capacity			
	BTU/HR	BTU/W-HR	BTU/HR	BTU/W-HR	Capacity Loss	EER Loss
Blocked Suction Cylinder Unloading	251,600	9.0	125,700	8.1	50%	10%

TABLE VI

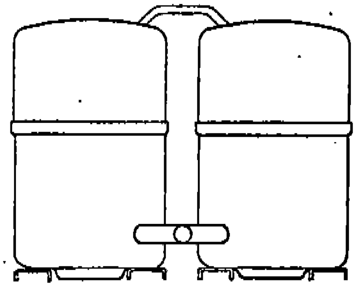
ACOUSTIC PERFORMANCE

	"A" Weighted Sound Power Level in dBA	
	Full Capacity	Reduced Capacity
Twin Compressor	75.7	72.6
Cylinder Unloading	82.6	84.4
Two-Speed Compressor	83.5	75.6



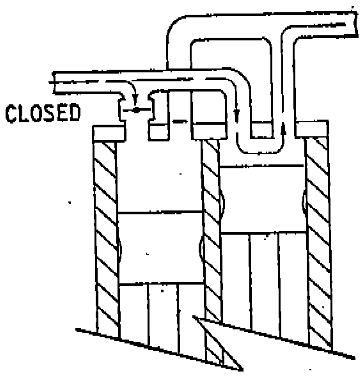
TWIN COMPRESSORS
NEW SINGLE SHELL

Fig. 1



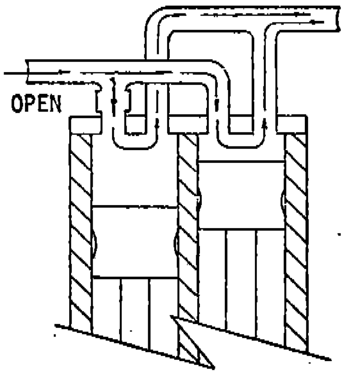
TWIN COMPRESSORS
EXTERNAL CONNECTIONS

Fig. 2



CYLINDER
UNLOADED

Fig. 3(A)



CYLINDER
LOADED

Fig. 3(B)

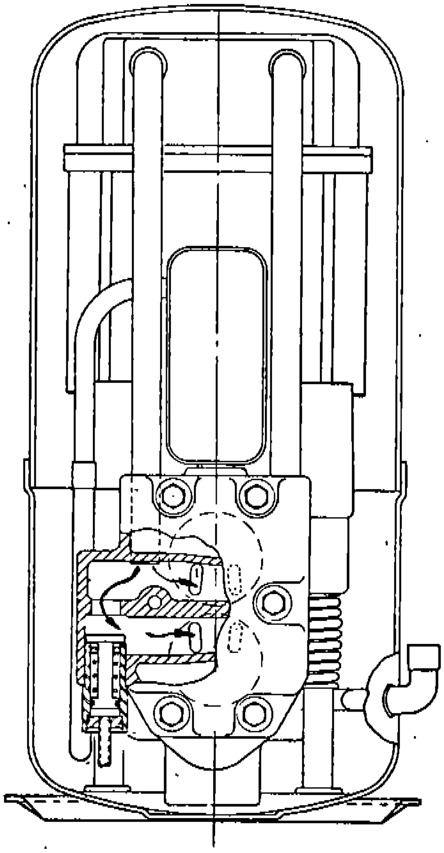


FIG.4 (A)

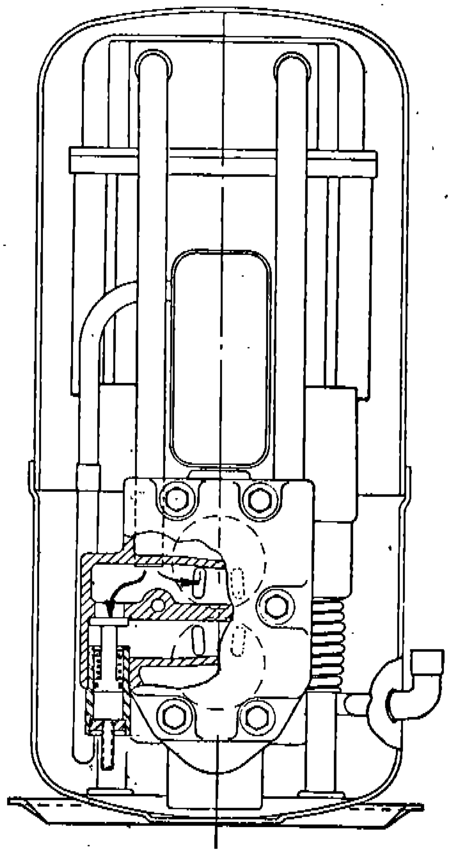
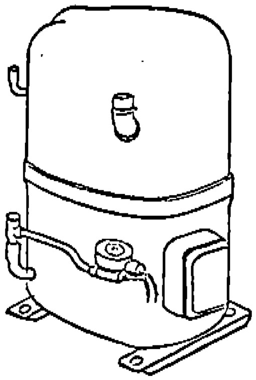


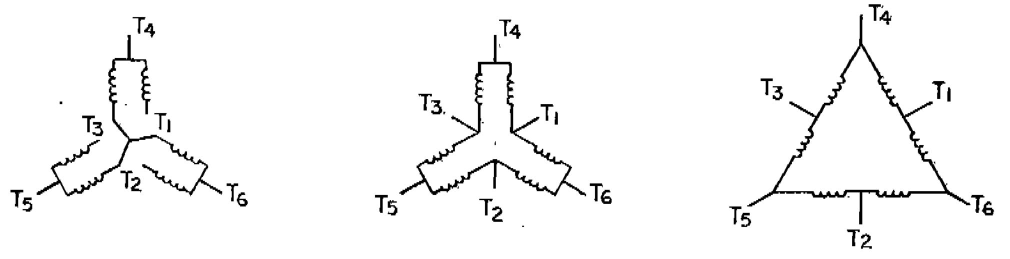
FIG.4 (B)



CYLINDER UNLOADED "YR"

Fig. 5

	LOW SPEED (% OF HI SPEED)	F.L. TQ. (LO SPEED) (% OF HI SPEED)	H.P. (LO SPEED) (% OF HI SPEED)	CONNECTION	
				HI SPEED	LO SPEED
VARIABLE TORQUE	50%	50%	25%	PAR. Y	SER. Y
CONSTANT TORQUE	50%	100%	50%	PAR. Y	SER. Δ
CONSTANT HP	50%	200%	100%	SER. Δ	PAR. Y



SPEED	L_1 L_2 L_3	OPEN	JOIN
LO	T_1 T_2 T_3	$T_4T_5T_6$	
HI	T_6 T_4 T_5		$T_1T_2T_3$

VARIABLE TQ

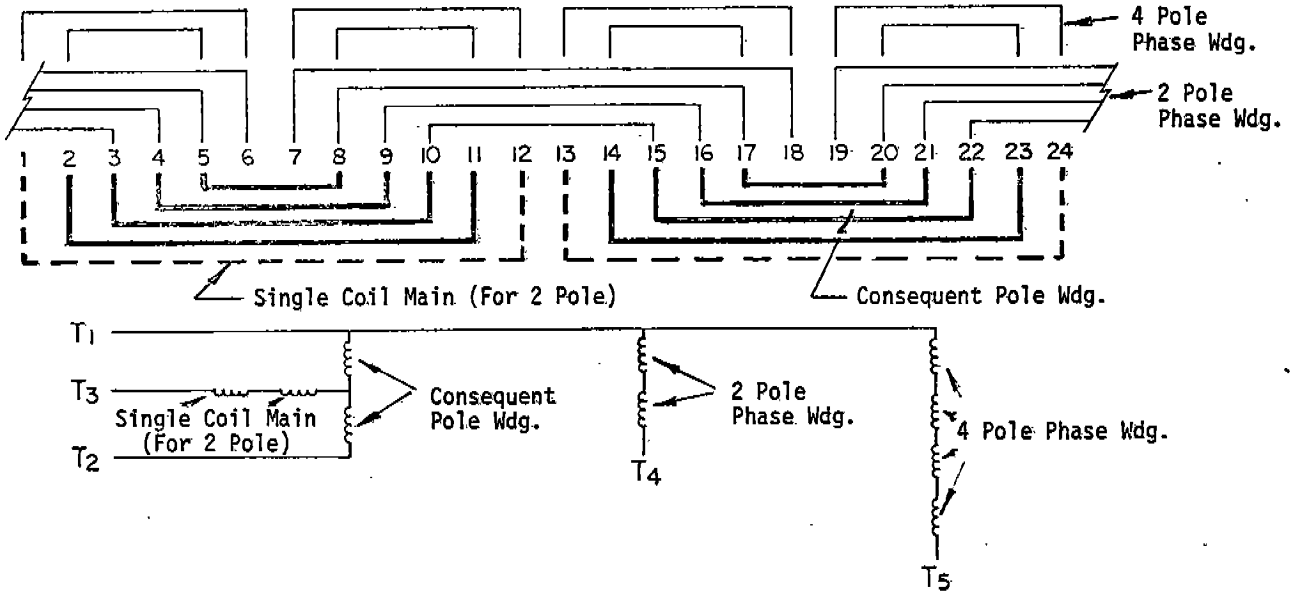
SPEED	L_1 L_2 L_3	OPEN	JOIN
LO	T_1 T_2 T_3	$T_4T_5T_6$	
HI	T_6 T_4 T_5		$T_1T_2T_3$

CONSTANT TQ

SPEED	L_1 L_2 L_3	OPEN	JOIN
LO	T_1 T_2 T_3		$T_4T_5T_6$
HI	T_6 T_4 T_5	$T_1T_2T_3$	

CONSTANT HP

FIG. 6



Speed	L_1	L_2	Open	Run Cap.
Low	T_1	T_2	T_3, T_4	Between T_2 & T_5
High	T_1, T_2	T_3	T_5	Between T_3 & T_4

Fig. 7