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HIGH-EFFICIENCY DUAL-STROKE COMPRESSOR FOR HEAT PUMPS *

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ABSTRACT

The development of a high efficiency, two-capacity, three cylinder, reciprocating piston, hermetic compressor for the advanced electric heat-pump is described. The patented capacity modulation concept employs an eccentrically bored circular cylinder cam between crankpin and connecting rod which acts to change the stroke of the compressor upon a change in the direction of rotation of the crankshaft. Complementary design features that result in high efficiency include a circulating oil system that cools the drive motor and the compressor and rejects heat external to the compressor shell, a suction piping system that routes gas into the cylinders in a manner to impede heat transfer, a discharge piping scheme that impedes heat transfer and reduces pressure loss, an improved valve and valve plate system that reduces flow impedance and in conjunction with a crowned piston reduces clearance volume thereby improving both capacity and efficiency, and a high efficiency, dual strength drive motor (not detailed here).

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INTRODUCTION

The objectives of the compressor development program were: (a) to identify the detailed changes necessary to make a significantly better compressor than currently available, and (b) to incorporate these changes into a reliable, cost effective dual capacity compressor for the advanced electric heat pump.

The key to a commercially viable high annual performance advanced electric heat pump is a high efficiency hermetic compressor with low cost capacity modulation means (Kirschbaum and Veyo 1977, and Veyo 1982). The performance advantage due to capacity modulation depends upon the range over which the compressor capacity can be modulated ,i.e., the ratio of minimum to maximum capacity at a fixed set of conditions, and upon the number of capacity steps available in that range. Although continuous modulation would appear to be most advantageous in terms of performance,capacity modulation by means of two steps can yield performance nearly as good, provided the capacity ratio is judiciously chosen. The known means to provide stepwise variable compressor capacity include two-speed drive motors, cylinder suction cut-off, valve unloaders, and parallel compressors. To complement these, the development program summarized here has reduced to practice a dual-stroke mechanism that provides a step-wise capacity control of almost any desired capacity ratio at a lower cost than any other system currently available.

The overall efficiency for an electrically driven hermetic compressor is defined, on a steady basis, as the isentropic or ideal work required to compress the gas throughput from its suction state ,i.e., the steady evaporator exit pressure and temperature, to the time averaged discharge pressure, divided by the electric power consumed by the compressor drive motor. Overall efficiency is often expressed as the product of the process efficiency times the compressor mechanical efficiency times the motor electrical efficiency as in Equation (1), where Eta represents efficiency:

$$Eta(overall)=Eta(process)*Eta(mechanical)*Eta(electrical) \quad (1)$$

Dissipative processes such as Joulean heating of motor windings, motor iron losses, and motor mechanical friction absorb electrical power without contributing to the work of vapor compression and are obviously deleterious to efficiency. Similarly, mechanical friction and hydraulic windage detract from compressor mechanical efficiency. Other dissipative processes such as throttling and heat transfer from high pressure side to low pressure side alter the compression process from one of least work and therefore reduce process efficiency.

The volumetric efficiency of a positive displacement compressor may be defined as the mass of gas induced, hence compressed and throughput, per cycle divided by the mass of gas that could occupy the compressor displacement at suction pressure and temperature,per cycle. It is a measure of how large and therefore how expensive a compressor must be in order to compress a

unit mass of gas. Volumetric efficiency can be defined in terms of the clearance volume and other pertinent parameters as follows:

$$Eta(volumetric) = (1 + C*(1 -((Phi/Plo)*((1 + DPhi/Phi)/(1 - DPlo/Plo)))^{(1/n)}))*(1 - DPlo/Plo) - Qin*(1 - 1/gamma)/((PD)*Plo) \qquad (2)$$

<u>Where</u> C = clearance volume to displacement ratio

Phi = condensing pressure (at shell boundary)

DPhi = pressure drop from cylinder discharge pressure through discharge valve and internal discharge line

Plo = evaporating pressure (at shell boundary)

DPlo = pressure drop from hermetic shell entrance through suction valve to cylinder suction pressure

n = polytropic exponent

Qin = heat deposited in suction gas within the shell

PD = compressor displacement

gamma = isentropic exponent

In hermetic compressors, electric drive motor dissipation as well as mechanical and hydraulic friction losses must be removed as heat by some cooling means. In suction gas cooled compressors, this heat detracts from compressor volumetric efficiency directly as can be seen by observing the effect of "Qin" in equation (2). It is also obvious from equation (2) that compressor mass flow hence heat pump heating capacity will decrease as suction pressure declines due to cold ambients. The means then for cooling the

drive motor in a suction gas cooled machine also decreases. The result can be excessively high motor temperature and high suction gas superheat if the gas entering the hermetic shell is dry. If the gas must be wet for adequate motor cooling, especially in cold ambients, then suction gas motor cooling imposes expansion device control problems, and the danger of liquid "slugging". A scheme to cool the motor with externally cooled oil and pipe the suction gas directly to the compressor was conceived and implemented during this development program as a significant step toward improving the performance and durability of heat pump hermetic compressors, especially for heating during low ambient temperature conditions.

Also obvious from equation (2) is the negative impact of clearance volume and pressure drop upon volumetric efficiency and the fact that a given pressure drop value is much more deleterious in the suction path. Compressor efficiency is heavily dependent upon good valves to minimize pressure drop. Further, both good valves and low clearance volume are means to accomplish high volumetric efficiency. Clearance volume is determined by the geometry of the valve plate, valves and piston. A comparative testing program conducted during this development effort identified a valve and valve plate configuration that in combination with a crowned piston has considerable merit as a means to improve volumetric efficiency.

DUAL STROKE MECHANISM DEVELOPMENT

Concept Description

The patented stroke changing mechanism, (Sisk 1980 and 1981), consists of a circular cross section, eccentrically bored, cylindrical cam that mounts on the crankpin of the compressor and changes in angular position relative to the crankshaft as a function of crankshaft rotational direction, thereby changing the effective throw of the crankshaft. The action of the cam is such that it is driven against one of two rotational stops depending upon the direction of rotation of the crankshaft. A sketch of the eccentric cam in its two positions is shown in figure 1.

In one cam position, the connecting rod reciprocates at its maximum stroke, displacing the piston to its minimum clearance volume position at top dead center. At the other cam position, corresponding to crankshaft rotation in the opposite sense, the connecting rod reciprocates with a much shorter stroke and thus with a much greater clearance volume at top dead center. In the shorter stroke position, compressor flow rate ,i.e., capacity, is decreased by both the reduction of piston displacement and the reduction of compressor volumetric efficiency because of the increased clearance volume. As a consequence, compressor flow rate is reduced more at high pressure ratios than at low pressure ratios. However, since the low capacity mode of the machine is generally required only during low pressure ratio conditions on the system, cooling mode and heating mode above 30 degrees F (-1

degree C) ambient, it is possible to select short stroke cam eccentricities that provide the optimum capacity reduction without appreciably affecting system performance. The ratio of minimum to maximum compressor capacity is determined by the cam eccentricity. The magnitude of the mimimum stroke is bounded between zero and the maximum stroke; its precise value is a subject for system optimization.

Initial Embodiment--Mod 0

Project resources would not permit the procurement of any major compressor components of non standard design such as crankcases or crankshafts. The mechanism was installed therefore in a host compressor, a standard compressor of nominally 4.0 tons (14.0 kW) capacity, somewhat near the correct displacement size anticipated for the advanced heat pump system (3.5 tons [12.3 kW].

A three cylinder radial compressor was used so as to ensure a positive torque to hold the cam against its stop during operation.

Three alternative cam designs were investigated. All had the same cam geometry, including the internal key pocket to provide a rotational stop. The first design, identified as "A" in figure 2, used a two piece circumferentially split cam, which slipped over the motor shaft and upper bearing journals before being installed on the crankpin. The other designs used a two piece axially split cam assembly. The longitudinal pins of cam "B" guaranteed that the cam would not separate during operation, whereas cam "C" used no locking device. Geometric details for the cam and crankpin are

given in table 1.

In order to accomodate the stroke changing mechanism, several modifications to the host compressor were necessary. First, a new crankshaft was machined from a standard casting with a crankpin eccentricity of 0.25 in (6.35 mm) , instead of the standard 0.35 in (8.89 mm) and a crankpin journal diameter of 1.00 in (25.4 mm) instead of the standard 1.312 in (33.32 mm). The crankpin journal diameter was reduced because the standard crankcase could not accomodate the cam and connecting rod that otherwise would have been required. In addition a key slot was machined in the motor journal to facilitate easy removal of the motor rotor. Second, new connecting rods with larger center distance and larger crankpin bores (to accomodate the cam) were machined from castings. Third, reversible single phase drive motors were procured from the supplier of the standard motor. The only modification consisted of having the electrical leads from both the main and auxiliary windings available for connection.

The first Mod 0 dual stroke compressor used cam "B" of figure 2, an axially split cam with longitudinal locking pins. It was exercised on a gas loop test stand at various loading conditions with gradually increasing intervals between teardown and inspection. This initial test successfully demonstrated that the the cam mechanism would change the stroke reliably upon motor reversal. Several concerns were raised however.

The first concern was the light fretting corrosion discovered between cam and crankpin after forty hours of operation. It was concluded that fretting of the cam bore and the crankpin was to be expected since the journal was cast iron and the cam mild steel, both relatively soft materials that are not particularly resistant to fretting wear. The second was the accumulation in the key pocket of black sludge-like wear debris (equal quantities of iron, copper, and aluminum), since this might eventually restrict cam motion if the key pocket acted as a collection trap. The third was the potential for inadequate wrist pin bearing lubrication during short stroke operation at high pressure ratio since there would be no force reversal on the wrist pin because of the large clearance volume. The fourth was the potential for wearing a step in the cylinder bore during extended periods of short stroke operation.

Mod 0 Durability Demonstration

A simple solution to the fretting wear problem appeared to be an improvement in the surface wear characteristics of the materials in contact. A variety of coating systems were considered. Since the number of candidates was rather large however, only two extreme coating alternatives were selected for experimental evaluation These were: (a) the simplest and lowest cost system that could be considered for production parts, and (b) the most wear resistant coating that was currently available regardless of cost. The first approach consisted of plating the crankpin with 0.0002 in (0.0051 mm) of hard chrome and using a tool steel cam, heat treated to about Rc = 40. The second approach consisted of flame spraying

both the crankpin OD and the cam ID with tungsten carbide and grinding the surfaces to leave about 0.003 in (0.076 mm) of carbide thickness. Three Mod 0 compressors were assembled with the chrome plated crankpin and hardened cam and one with the flame sprayed, tungsten carbide coated, crankpin and cam ID. The four Mod 0 compressors were then tested for various periods as part of the endurance demonstration program.

In order to ensure adequate wrist pin lubrication the force on the pin must reverse during reexpansion in order to admit oil to the side of the bearing loaded during the compression stroke. Calculations indicated that with 22.5% clearance volume the force on the wrist pin will reverse for pressure ratios less than 3.5 : 1 . This implies then that short stroke operation should be prohibited for pressure ratios greater than 3.5 :1. In order to verify that this pressure ratio limit was adequate, a single stroke version of the host compressor was modified to have a 22.5% clearance volume and operated at a pressure ratio o 3.5 : 1 for about 3900 hours to demonstrate endurance.

In order to verify that a step would not be worn in the cylinder bore as a result of prolonged short stroke operation, the test program included operation of two Mod 0 compressors in the short stroke mode continuously for abour 4000 hours each.

A matrix defining the Mod 0 endurance test program and indicating results is shown in table 2. Note that cam "C" of figure 2 has three locating pins in the mating flanges, but no pins to lock the cam halves together, whereas cams "A" and "B" have

longitudinal pins to lock the cam halves together. Testing indicated that it was not necessary to lock the cam halves together since an adhesive which was applied to a type "C" cam decomposed without ill effect.

Compressor performance was measured using a commercially available compressor calorimeter. Data were obtained for suction pressures corresponding to R22 saturation temperatures ranging between 40 and 0 degrees F (4.4 and -17.8 degrees C) and for discharge pressure corresponding to R22 saturation temperatures ranging between 120 and 100 degrees F (48.9 and 37.8 degrees C). Suction superheat was maintained at 20 degrees F (11.1 degrees C) and measured capacity was corrected to 15 degrees F (8.33 degrees C) condenser subcooling.

Characteristic performance curves for the Mod 0 dual stroke compressor are given in figure 3. It was not possible to operate below 30 degrees F (-1.1 degrees C) saturated suction temperature at 120 degrees F (48.9 degres C) saturated discharge temperature in the short stroke mode because of inadequate motor cooling. Note that in the long stroke mode the efficiency (EER or COP) of the Mod 0 compressor is better than the baseline host compressor whereas the capacity is less. This correlates with the fact that the dual stroke machine has the same bore but a shorter stroke. Geometric characteristics of the compressors used in this development program are listed in table 3.

Mod 0 Observations

Mod 0 compressor tests permitted the following observations: (1) The durability of the dual stroke mechanism using the axially split cam design made of tool steel with a hardness of Rc = 38 - 42 and a crankpin plated with 0.0002 in (0.0051 mm) of hard chrome appears to be satisfactory for production. (2) The accumulation of a sludge of wear debris in the cam key pocket is a nuissance type of problem and if a copious supply of oil were to be provided, such as from a positive displacement oil pump, then a flushing flow of oil would keep the pocket clear. (3) Compressor performance in long stroke mode is satisfactory. (4) Short stroke performance at low pressure ratios is also satisfactory. (5) Short stroke performance would benefit significantly from an increase in motor efficiency. (6) Short stroke performance would also benefit significantly from a more positive means for cooling the drive motor. (7) Continuous operation in the short stroke mode at pressure ratios less than 3.5 : 1 should not adversely affect wrist pin bearing lubrication. (8) Cylinder wear in the form of a step is not a problem.

EFFICIENCY IMPROVEMENT PROGRAM

Approach

The efficiency improvement effort involved three parallel development programs:

(1) valve improvement and clearance volume reduction ; (2)

exploration of means to eliminate heat transfer to the suction gas; and (3) the improvement of drive motor efficiency, especially for low capacity operation.

Valve Improvement

Constraints. The goals of the valve improvement program were to conceive valve designs that would decrease pressure drop and facilitate reduced clearance volume, to embody the most promising concepts in hardware, and to compare the resulting compressor performance with the performance of a reference compressor through test. All design alternatives were constrained to be compatable with the family of compressors to which the dual stroke concept was being applied. This dictated that the basic ring valve design be maintained and that the port size of the existing valve system also be maintained. Decreased pressure drop therefore could be accomplished only by providing a wider opening valve.

Because of the ring valve configuration, clearance volume reduction means were limited to either decreasing the thickness of the valve plate in the vicinity of the discharge valve ports, or eliminating the valve support from the center of the valve plate and incorporating it into a support structure that mounts on top of the valve plate. This latter configuration would require an open center suction valve and could therefore accomodate a crowned piston as a means to reduce clearance volume.

Suction Valve. The standard ring type suction valve incorporates a diametral web that spans the central void. The valve is supported by pinning the web at its center to the valve plate structure. When opening it deflects in a butterfly mode with the web along the node. Three alternative open center, ring type suction valves were investigated. The first consists of a relatively stiff ring valve that is held against the suction valve seat by four small helical springs mounted in recesses at the top of the cylinder wall. These springs act on four equally spaced tabs that are integral with the ring valve. The second open center suction valve configuration resembles the first, but is not rigid, and does not use the helical return springs. Instead, it deflects upon opening into a dished trough configuration. In order to permit this, the cylinder cutouts accepting two support tabs on the ring valve were relieved at a 20 degree angle to allow the dished deflection. The valve also included two stop tabs 90 degrees away from the support tabs. Three valve thicknesses were investigated: 0.018,0.015, and 0.012 in (0.457,0.381 and 0.305 mm). The third open center suction valve configuration is a dished deflection type like the second except that the ring is of non uniform width and utilizes only two support tabs and no stop tabs. Valve thicknesses of 0.015 and 0.012 in (0.381 and 0.305 mm) were investigated.

Discharge Valve. Three discharge valve configurations were investigated. The first used the standard ring type discharge valve with diametral web supported at its center by a superstructure that mounted on top of the valve plate. The second discharge valve is a rigid, direct lift, open center ring type.

Three return spring systems were investigated: six helical springs, three helical springs, and a single curved spring washer. The springs are contained in a support structure that mounts on top of the valve plate. The third discharge valve configuration consists of a shaped cantilevered flapper that has the center portion configured into a ring that seats on the standard discharge port. The flapper is rigidly clamped at one end, but permitted to slide under the valve stop at the opposite end. The valve stop is a double curved arch with equal radii of curvature. Stop curvature limits the maximum stress in the valve as well as the maximum lift. Curvature radii of 3.00, 2.75, and 2.50 in (76.2, 69.9, and 63.5 mm) were investigated with valve thicknesses of 0.012, 0.010, and 0.008 in (0.305,0.254, and 0.203 mm).

Valve Plate. Three valve plate designs were investigated. The first was designed to be used with a flat top piston, the direct lift or standard discharge valve and either of the open center dished deflection suction valves. The valve plate was thinned from the top and yielded a calculated clearance volume of 3%. The second valve plate was designed to be used with a crowned piston, any type of discharge valve, and the dished deflection, open center, uniform ring width suction valve. The valve plate was thinned from the bottom and yielded a calculated clearance volume of 3.2%. The third valve plate was designed to provide minimum clearance volume by eliminating entirely the central valve support section thereby creating an open center valve plate . The direct lift or standard discharge valve can be used with a special top mounted support structure. Any open center suction valve can be

used with this configuration and with a crowned piston the calculated clearance volume is 2.0%.

Valve Selection and Performance Impact. All valve testing was done on a two cylinder radial compressor in order to minimize the number of parts that had to be fabricated. The family of compressors considered included no single cylinder compressor. Tests were executed using a commercially available compressor calorimeter at various suction pressures and a discharge pressure corresponding to a condensing temperature of 120 degrees F (48.9 degrees C) . Based upon the results of this comparative testing program, the valve configuration for the prototype dual stroke compressor was selected. This configuration uses the 0.012 in (0.305 mm) thick, uniform width, open center suction valve, crowned piston, and valve plate thinned from the bottom shown in figure 4; plus the cantilevered flapper discharge valve of 0.010 in (0.254 mm) thickness with the machined valve stop of 2.75 in (69.9 mm) curvature radius shown in figure 5. The calculated clearance volume of this configuration is 3.2%. At an evaporator pressure corresponding to a saturation temperature of 40 degrees F (4.4 degrees C) the improvement in compressor capacity is 14.4% and in efficiency is 8.5%; whereas at an evaporator pressure corresponding to a saturation temperature of 10 degrees F (-12.2 degrees C) the improvement in compressor capacity is 7.1% and in efficiency is 9.0%. These improvement figures are relative to the baseline two cylinder compressor which served as the valve testbed. Note that further development and optimization may yield still better performance.

Elimination of Heat Transfer to Suction Gas

Motor Cooling. The second program of the efficiency improvement effort had as its objective the elimination of heat transfer to suction gas. For thermodynamic efficiency, it was desired to reject dissipation originating inside the compressor shell to the condenser side of the system. Several alternatives were considered but rejected. The concept adopted relies on pumped oil to cool the drive motor and compressor with heat rejected to indoor air through an oil to air plate fin on tube heat exchanger. A positive displacement, reversible direction gear type oil pump was provided at the bottom of the vertically oriented crankshaft in order to generate more oil flow and higher oil pressure than was possible with the eccentric oil passage in the standard compresser crankshaft. As illustrated in figure 6, the oil after being cooled in an external heat exchanger, is piped into the hermetic compressor shell and showered on the upper end of the motor rotor. The rotating rotor slings a portion of the oil over the upper end turns. From there it drains by action of gravity over the motor stator and the compressor crankcase into the bottom of the shell. The remainder of the oil drains through the holes in the rotor and any space in the winding slots and through the rotor-stator gap. From the bottom of the rotor the oil is slung against the lower end turns. Oil draining from the bottom of the motor drains to the diaphragm plate surface and then flows out through three radial holes in the crankcase and into the lower shell. The oil pump then picks up the oil and raises its pressure. Part of the oil is

circulated to the bearings while the remainder is routed to the external heat exchanger.

Direct Suction. The suction gas enters the compressor shell through the top cover and is piped directly to one end of a semicircular chamber of rectangular cross section that serves as a liquid separator. Drain holes in the separator permit liquid to drain to the bottom of the compressor shell. A suction pipe connects the other end of the separator to a suction plenum formed by sealing off the inlet ports from the bottom of the motor by a diaphragm plate that spans the crankcase. The performance of an oil cooled two cylinder compressor is compared with that of the standard suction gas cooled compressor in figure 7. Incorporation of this scheme into the three cylinder dual stroke compressor as shown in figure 6 presented no problem other than a more complex suction plenum and diaphragm plate.

Reduced Heat Transfer. In addition to eliminating suction gas cooling of the motor, steps were taken to reduce the transfer of heat from the discharge gas to either the cooling oil or the vapor in the hermetic shell. The internally mounted muffler used in the baseline compressor was eliminated in favor of an external discharge line muffler. This greatly reduced the hot surface area exposed to both the gas in the shell and the oil. In addition, the internal discharge lines from each cylinder were judiciously cut to length, gracefully curved, and joined together to form a colinear flow quadripod. Further, a phenolic fiber spacer was placed between the valve plate and the cylinder head as a heat break so as

to reduce the temperature of the cylinder head exposed to cooling oil and shell gas.

Impact of Cool Suction Gas. The improvement in compressor capacity is 12.5% at 10 degrees F (-12.2 degrees C) evaporating and 3.0% at 40 degrees F (4.4 degrees C) , with condensing pressure corresponding to a saturation temperature of 120 degrees F (48.9 degrees C) . The improvement in compressor efficiency is 11.3% at 10 degress F (-12.2 degrees C) evaporating and 6.1% at 40 degrees F (4.4 degrees C) .

Drive Motor Efficiency Improvement

The third program of the efficiency improvement effort sought to improve the efficiency of the drive motor. Reference was made earlier to the difficulty experienced with the standard motor when the Mod 0 compressor was tested in low capacity mode. The compressor would operate satisfactorily at an evaporating temperature of 30 degrees F (-1.1 degrees C) , but would cut-out on thermal overload at an evaporating temperature of 25 degrees F (-3.9 degrees C) . The dual strength motor concept evolved from the solution to this problem. Figure 8 shows the efficiency of the standard drive motor as a function of delivered shaft power and the efficiency of the improved motor in both high and low strength modes. As configured for the preprotype dual stroke compressor, drive motor efficiency exceeds 88 percent in both high and low capacity operation at the respective design points of 1.5 and 4.5 hp (1.1 and 3.4 kW).

PREPROTYPE (Mod II) COMPRESSORS

Embodiment Description

The prototype compressor is shown in figure 9 and includes the following advanced features: (a) A capacity modulation system that changes compressor stroke upon reversal of drive shaft rotation. See figures 1 and 2. (b) An internal suction piping system that bypasses the drive motor and directs the incoming gas to a liquid separator and then to a section of the crankcase common to all cylinder suction ports. See figures 6 and 10. (c) A system for motor and compressor cooling that uses externally cooled oil supplied by a reversible positive displacement oil pump driven from a bottom extension of the crankshaft. See figure 6. Oil pump flow is approximately 1.3 gpm (0.082 L/s). (d) A drive motor of high efficiency in both capacity modes. (e) A valve and valve plate system that allows a low clearance volume (3.2%) in the long stroke mode while providing improved valves of increased open area and rapid response. See figures 4 and 5. (f) A discharge piping system that minimizes heat exchange between discharge gas and the interior shell environment while providing an aspirating arrangement for low pressure drop. See figure 10. In addition, a phenolic fiber spacer between valve plate and cylinder head reduces the temperature of the cylinder head which is also part of the discharge gas containment. (g) An oil to forced air heat exchanger located near the compressor to minimize the inventory of oil required in the system and to reject heat to the ambient air near

the compressor. An exploded view of the prototype compressor is shown in figure 10. The impact of the individual advanced features upon compressor performance is estimated in table 4. Measured performance is given in figure 11 and in tables 5 and 6.

Disposition of Compressors

All three cylinder development compressors built for this project are identified in table 7. The testing of all suction gas cooled compressors (items 2 through 6 of table 7) was terminated on December 31, 1981. Compressor #2 had completed 89,500 cycles of mode reversal and 10,000 hours of operation. Compressor #11 is part of preprototype heat pump system #2, and has been operating in the field since November 24,1982. A more detailed account of the compressor development effort may be found in a funding agency topical report (Young,Kastovich, and Veyo 1983).

Discussion

The Mod II oil cooled dual-stroke hermetic compressor is clearly superior to compressors presently available in the market place in terms of efficiency across the entire range of saturated evaporator temperatures of interest. This superiority is especially pronounced at low evaporator temperature where the Mod II compressor has a substantial capacity advantage. The incremental compressor cost relative to the baseline, host compressor, as seen by the manufacturer, will be reflected in the market place as a system cost premium of approximately $110 (1981

U.S. dollars). Although good durability has been demonstrated in the limited testing permitted by project resources, significantly more testing would be required before this compressor could be considered ready for the market place.

CONCLUSIONS

1. The dual stroke compressor is a low cost, high efficiency, flexible capacity ratio machine.

2. The dual stroke compressor has significantly better performance than any other compressor available in the market today.

3. The dual stroke compressor concept is generally applicable to any residential advanced electric heat pump or air conditioning system.

4. The incremental cost associated with upgrading the baseline compressor to an advanced dual stroke unit can be justified through energy savings in any suitably designed system.

5. In quantity manufacture (100,000 units annually) the dual stroke capacity modulation means is less expensive than presently known alternatives such as two pole/four pole two speed motors, electronic variable speed drives, geared two speed drives, parallel compressors, and probably even cylinder suction cut-off schemes.

6. The ratio of low capacity to high capacity is not fixed but is a parameter for system optimization.

7. The dual stroke concept as reduced to practice appears to be applicable only to multicylinder radial compressors.

8. Low capacity operation should be limited so as to impose pressure ratios less than 3.5 : 1 in order to ensure wrist pin lubrication. (Most residential heat pump applications with saturated suction temperatures greater than 30 degrees F [-1.1 degrees C] meet this requirement.)

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able 1. Geometric Details of Cam and Crankpin

	inches	millimeters
rankpin eccentricity	0.250	6.35
rankpin OD	1.000(+0.003,-0)	25.40(+0.076,-0)
am ID	1.0006(+0.003,-0)	25.42(+0.076,-0)
am OD	1.4375(+0,-0.003)	36.51(+0,-0.076)
am eccentricity	0.100	2.54
am rotation (degrees)	90.0	90.0
ax throw	0.350	8.89
ax stroke	0.700	17.78
in throw	0.269	6.83
in stroke	0.538	13.67
in clearance vol (%)	6.0	6.0
ax clearance vol (%)	22.8	22.8
iston diameter	1.750	44.45
on Rod C/C	2.160	54.86

able 2. Mod 0 Dual Stroke Compressor Endurance Test Program

		--------Cam-------			----------Test (R22)----------			
omp.	Pin	Style	Coat	Rc	Type	Pres	Hours	Cond
	Rc=10	B	none	12	cycles 100	hi=P1/P7 lo=P6/P11	40	fret sludge
	CrPlt	B	none	40	cycles 89500	hi=P2/P10 lo=P3/P9	10000	ok
	CrPlt	C	none	40	cycles 100	P5/P11	2899	ok
					steady	P5/P10	4375	ok
	WC	B	WC	32	cycles 5	hi=P1/P7 lo=P5/P11	1359	ok
	CrPlt	C	none	40	cycles 6	P4/P11	2886	sludge
					steady	P5/P10	3838	ok
	Rc=10	none-single stroke			steady	P2/P8	3901	ok

ote: P(n=1 to 11)=50,55,60,65,70,75,225,230,235,260,300 psi
=345,379,413,448,482,517,1550,1585,1619,1791,2067 kPa
ote: For cam type see figure 2.

Table 3 - Compressor Geometric Characteristics

	Host Compressors		Dual Stroke Compressors		
	2 Cyl.	3 Cyl.	Mod 0	Mod I	Mod II
Bore (mm)	44.45	44.45	44.45	44.45	44.45
Stroke [Max] (mm)	20.09	21.79	17.78	17.78	17.78
Stroke [Min] (mm)	20.09	21.79	13.67	13.67	13.67
Discharge Valve					
Port OD (mm)	27.61	27.61	27.61	27.61	27.61
Port ID (mm)	19.99	19.99	19.99	19.99	19.99
Port Area (sqmm)	284.4	284.4	284.4	284.4	284.4
Lift Stop (mm)	1.778	1.778	1.778	1.778	2.159
thickness (mm)	.203	.203	.203	.203	.254
Suction Valve					
Port OD (mm)	39.70	39.70	39.70	39.70	39.70
Port ID (mm)	32.23	32.23	32.23	32.23	32.23
Port Area (sqmm)	1008.1	1008.1	1008.1	1008.1	1008.1
Lift Stop (mm)	1.651	1.651	1.651	1.651	1.651
thickness (mm)	.305	.305	.305	.305	.305
Con Rod Length (mm)	53.67	52.68	54.84	54.84	54.84
Main Journal Dia (mm)	28.58	28.58	28.58	28.58	28.58
Crankpin Dia (mm)	33.32	33.32	25.40	25.40	25.40
Piston Pin Dia (mm)	17.45	17.45	17.45	17.45	17.45
Clearance Vol (%)	6.0	6.0	6.0	6.0	3.2
Displacement (mL)	62.40	101.5	82.84	82.84	82.84

meas

Table 4 - Impact of Advanced Features Upon Compressor Performance

	40°F (4.4°C) Evaporator[4]			10°F (-12.2°C) Evaporator[4]		
	Capacity (kW)	Power (kW)	COP	Capacity (kW)	Power (kW)	COP
Baseline Host[1]	15.15	5.622	2.69	6.930	4.002	1.73
Mod O[2] (High Cap)	11.88	4.260	2.79	4.754	2.982	1.59
Mod Ia[3] (High Cap)	12.58	4.336	2.90	5.327	3.081	1.73
Oil Cooled						
Direct Suction						
Discharge Piping						
Mod Ib[3] (High Cap)	12.58	4.175	3.01	5.327	2.940	1.81
Oil Cooled						
Direct Suction						
Discharge Piping						
2 Strength Motor						
Mod II (High Cap)	13.51	4.115	3.28	6.065	3.175	1.91
Oil Cooled						
Direct Suction						
Discharge Piping						
2 Strength Motor						
Better Valves						
Reduced Clearance						

Notes:
1 Displacement = 6.191 cubic inches (101.5 mL)
2 Displacement = 5.051 cubic inches (82.84 mL)
3 Estimated
4 120°F (48.9°C) Condensing, 20°F (11.1°C) Evaporator Superheat, 15°F (8.3°C) Condenser Subcooling, Refrigerant R22

Table 5 - Preprototype Dual-Stroke Compressor Performance - High Capacity
(20°F superheat, 15°F subcooling, 230 V, 1 phase, 60 Hz)

CAPACITY - BTU/HR

EVAP TEMP DEG F	CONDENSING TEMPERATURE DEG F 90.0	95.0	100.0	105.0	110.0	115.0	120.0
0.0	21336.4	19603.8	18257.5	17203.0	16345.9	15591.5	14845.5
10.0	28326.2	26798.1	25394.8	24099.7	22896.4	21768.4	20699.4
20.0	35633.4	34432.6	33153.4	31807.6	30407.1	28963.5	27488.7
30.0	44649.1	43612.1	42376.6	40958.6	39374.1	37639.1	35769.5
40.0	56764.8	55441.5	53907.4	52184.2	50293.7	48257.7	46097.7
50.0	73371.6	71025.9	68589.2	66116.3	63662.2	61281.6	59029.5

POWER INPUT - WATTS

EVAP TEMP DEG F	CONDENSING TEMPERATURE DEG F 90.0	95.0	100.0	105.0	110.0	115.0	120.0
0.0	2572.8	2530.8	2541.1	2586.7	2650.9	2716.8	2767.7
10.0	2828.9	2860.7	2905.5	2961.3	3026.2	3098.1	3175.1
20.0	2942.6	3055.1	3158.5	3253.8	3342.2	3424.8	3502.7
30.0	2989.8	3171.0	3336.4	3484.1	3611.9	3717.7	3799.5
40.0	3046.2	3265.2	3475.7	3671.9	3847.8	3997.5	4114.8
50.0	3187.6	3394.5	3612.7	3837.2	4062.9	4284.9	4497.9

ENERGY EFFICIENCY RATIO - BTU/W-HR

EVAP TEMP DEG F	CONDENSING TEMPERATURE DEG F 90.0	95.0	100.0	105.0	110.0	115.0	120.0
0.0	8.29	7.75	7.18	6.65	6.17	5.74	5.36
10.0	10.01	9.37	8.74	8.14	7.57	7.03	6.52
20.0	12.11	11.27	10.50	9.78	9.10	8.46	7.85
30.0	14.93	13.75	12.70	11.76	10.90	10.12	9.41
40.0	18.63	16.98	15.51	14.21	13.07	12.07	11.20
50.0	23.02	20.92	18.99	17.23	15.67	14.30	13.12

Table 6 - Preprototype Dual-Stroke Compressor Performance - Low Capacity
(20°F superheat, 15°F subcooling, 230 V, 1 phase, 60 Hz)

CAPACITY - BTU/HR

EVAP TEMP DEG F	CONDENSING TEMPERATURE DEG F 90.0	95.0	100.0	105.0	110.0	115.0	120.0
30.0	24418.8	22410.1	20337.1	18253.9	16215.1	14275.0	12488.1
35.0	28454.1	26466.2	24414.8	22322.6	20212.2	18106.2	16027.1
40.0	33591.9	31481.4	29311.3	27085.2	24806.8	22479.5	20106.9
45.0	39510.3	37234.5	34862.1	32405.9	29878.7	27293.2	24662.1
50.0	45887.3	43504.4	40903.0	38148.9	35307.8	32445.6	29627.8

POWER INPUT - WATTS

EVAP TEMP DEG F	CONDENSING TEMPERATURE DEG F 90.0	95.0	100.0	105.0	110.0	115.0	120.0
30.0	1820.2	1792.4	1759.2	1720.9	1677.9	1630.2	1578.3
35.0	1880.5	1884.6	1883.3	1874.9	1857.8	1830.4	1790.9
40.0	1940.8	1976.2	2005.5	2025.8	2033.9	2026.9	2001.8
45.0	1993.7	2056.9	2113.6	2160.1	2192.6	2207.2	2200.4
50.0	2031.8	2116.5	2195.3	2264.5	2320.2	2358.7	2376.3

ENERGY EFFICIENCY RATIO - BTU/W-HR

EVAP TEMP DEG F	CONDENSING TEMPERATURE DEG F 90.0	95.0	100.0	105.0	110.0	115.0	120.0
30.0	13.42	12.50	11.56	10.61	9.66	8.76	7.91
35.0	15.13	14.04	12.96	11.91	10.88	9.89	8.95
40.0	17.31	15.93	14.62	13.37	12.20	11.09	10.04
45.0	19.82	18.10	16.49	15.00	13.63	12.37	11.21
50.0	22.58	20.55	18.63	16.85	15.22	13.76	12.47

Table 7 - Summary of 3 Cylinder Development Compressors

Comp.#	Cooling	Oil	Shell	Motor	Cam	Cycles	Hours	P-ratio	Term	Sludge	Cond.	Status
1	gas	wax	bolt	std	pin	100	40		1980	Heavy	scrap	scrap
2	gas	wax	bolt	std	pin	89,500	10,000	hi=3.90 lo=3.33	12/31/81	none	ok	apart
3	gas	wax	bolt	std	free	100 0	2899 4375	3.74 lo=3.26	12/31/81	lite	ok ok	apart
4	gas	wax	bolt	std	pin	5	1359	hi=3.69 lo=3.74	12/31/81	lite	ok	
5	gas	syn	bolt	std	free	6	2886 3838	3.97 lo=3.26	12/31/81	lite	ok ok	apart
6	gas	syn	bolt	std	none	0	3901	lo=3.54	12/31/81	N/A	ok	apart
7	oil	syn	bolt	2str	free	0	2083	lo=3.40	3/5/82	none	ok	apart
8	oil	syn	weld	2str	free	transferred to Unit-1			active	-	ok	Unit-1
9	oil	syn	bolt	2str	free	45,150	3733	3.50	3/5/82	none	ok	apart
10	oil	syn	weld	2str	free	failed under test			2/16/82	none	valve	apart
11	oil	syn	weld	2str	free	in field use			active	-	ok	Unit-2
12	oil	syn	bolt	2str	free	available			active	-	ok	spare

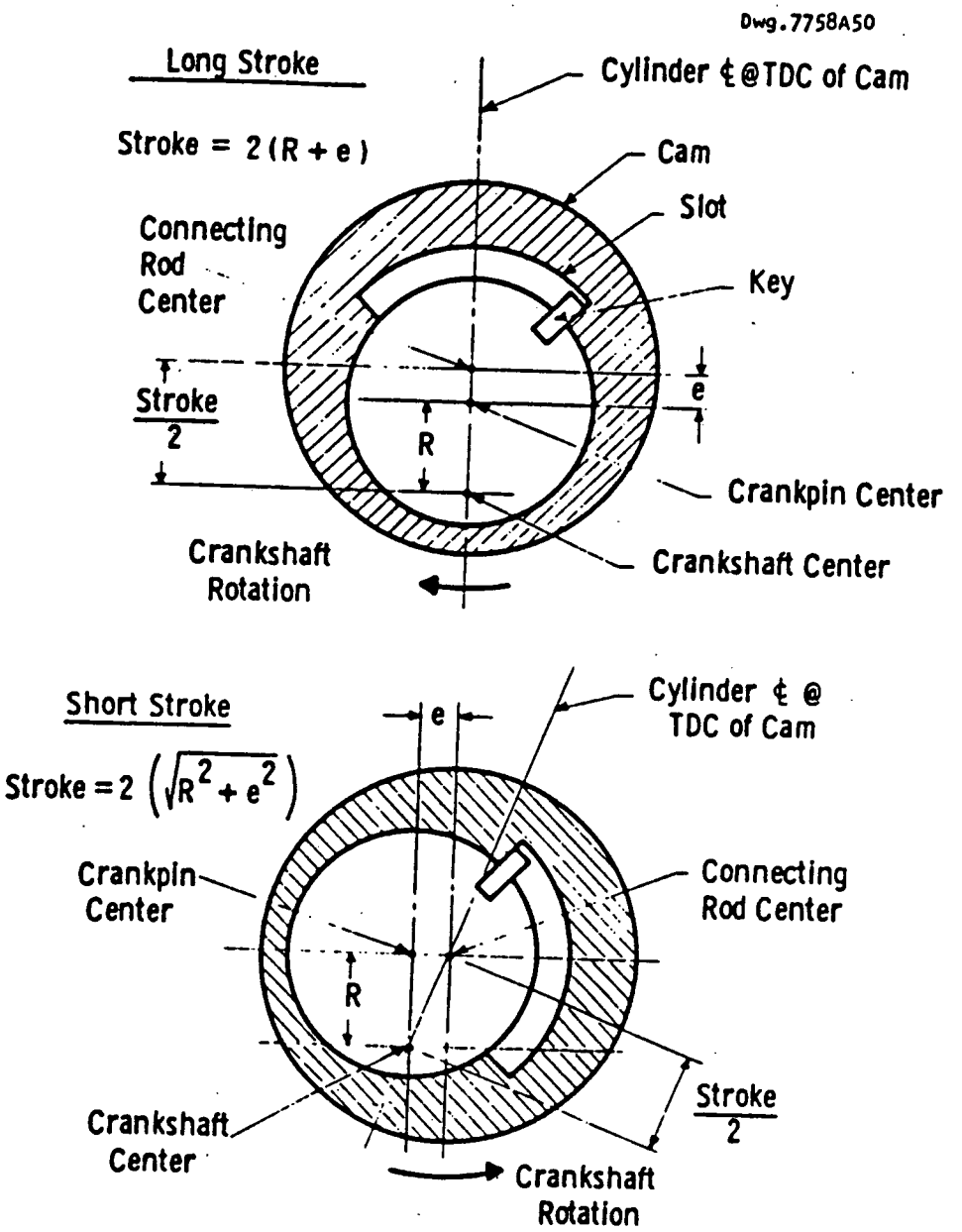

Fig. 1 - Dual-stroke cam configuration

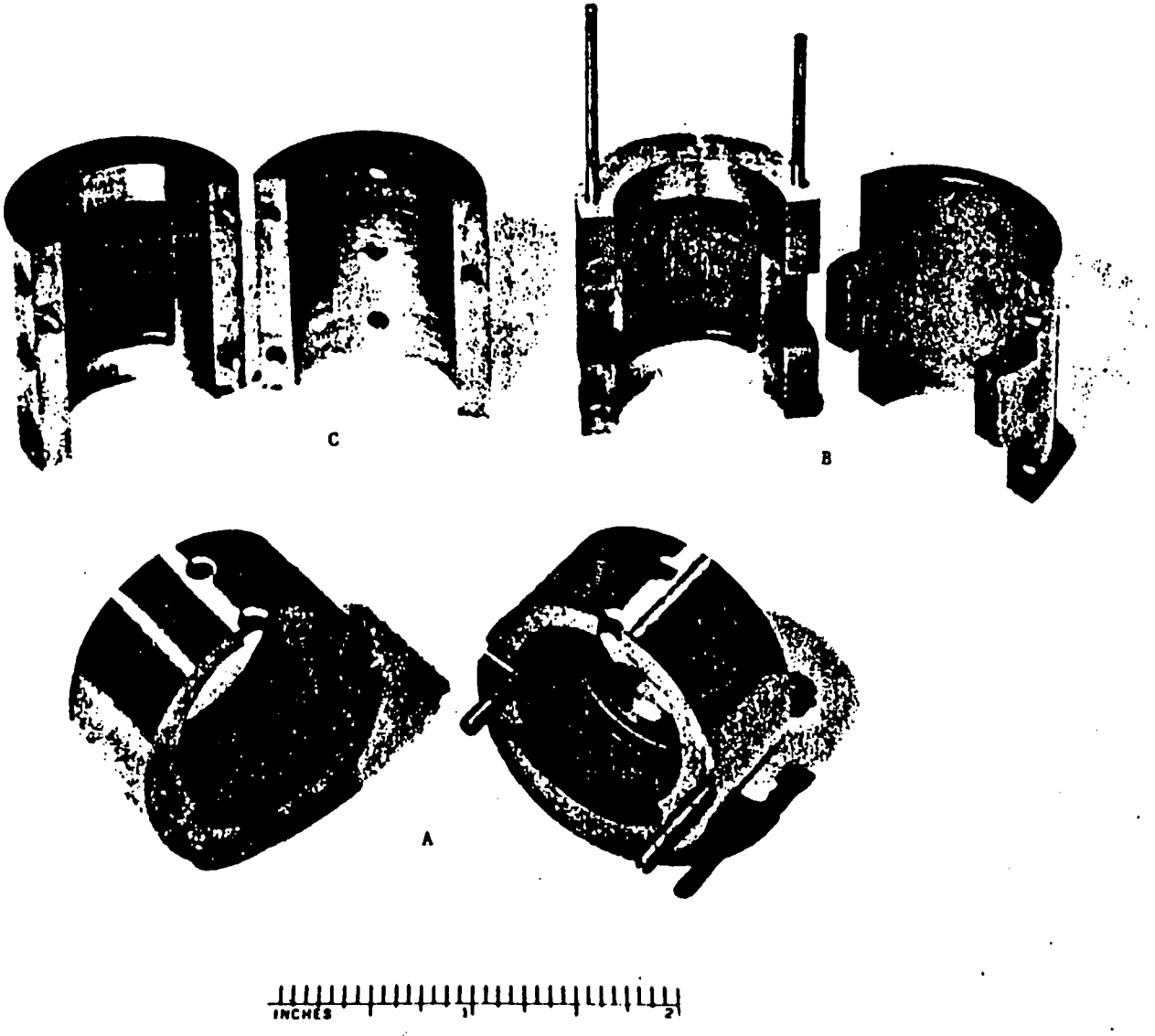

Fig. 2 - Dual-stroke cam designs - disassembled

Curve731394-B

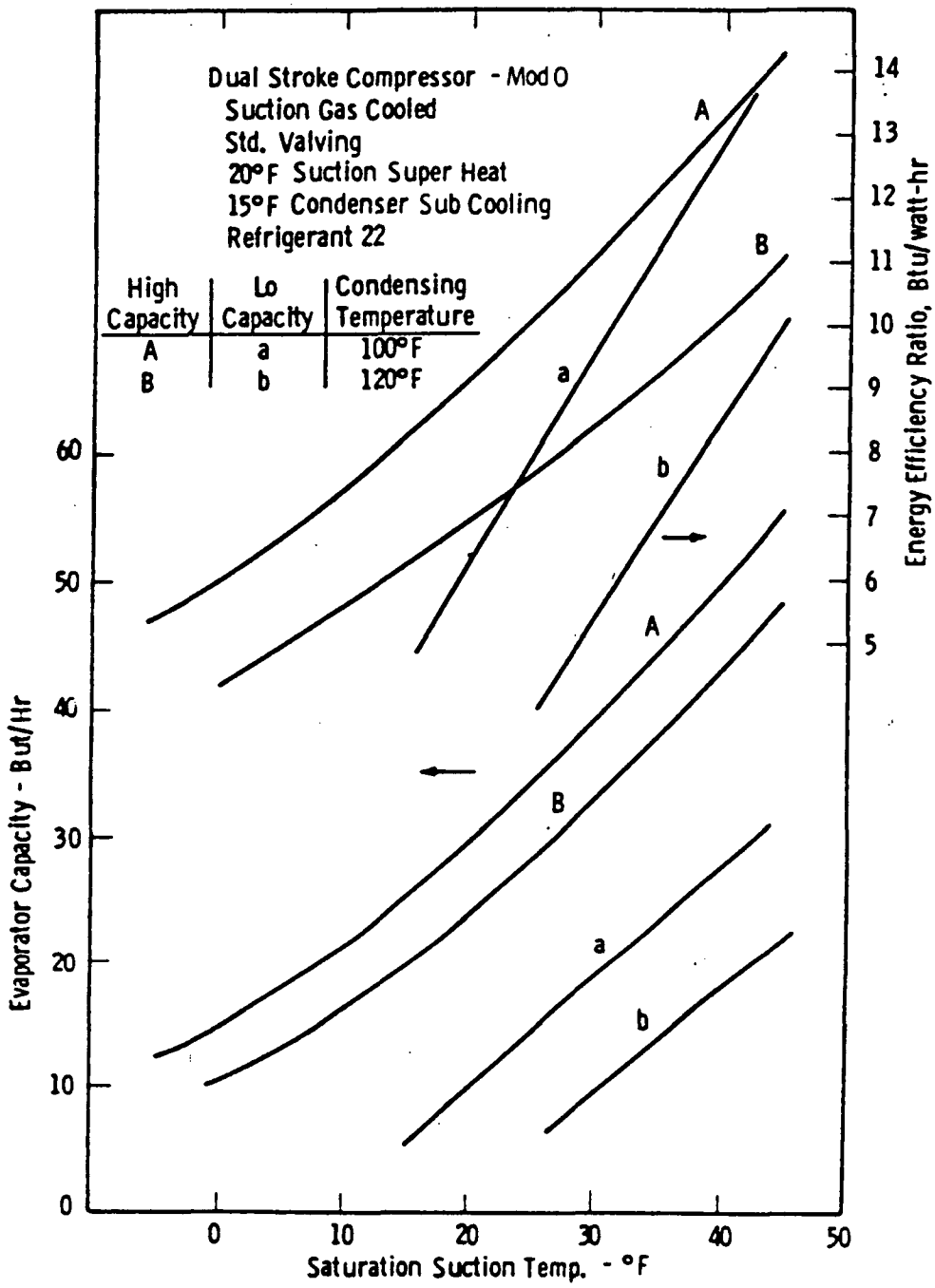

Fig. 3 - Performance of suction gas cooled dual-stroke hermetic compressor (Mod O)

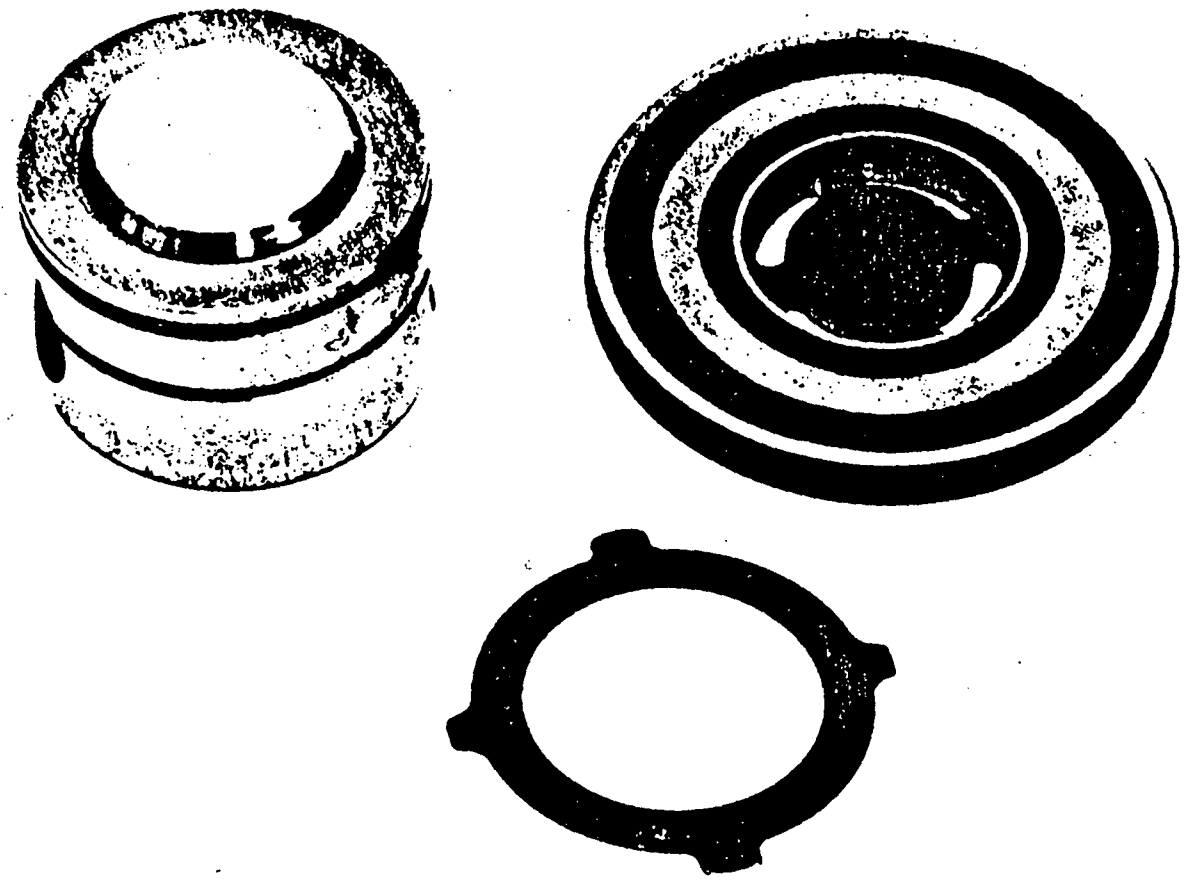

Fig. 4 - Valve plate thinned from bottom with open center suction valve and crowned piston

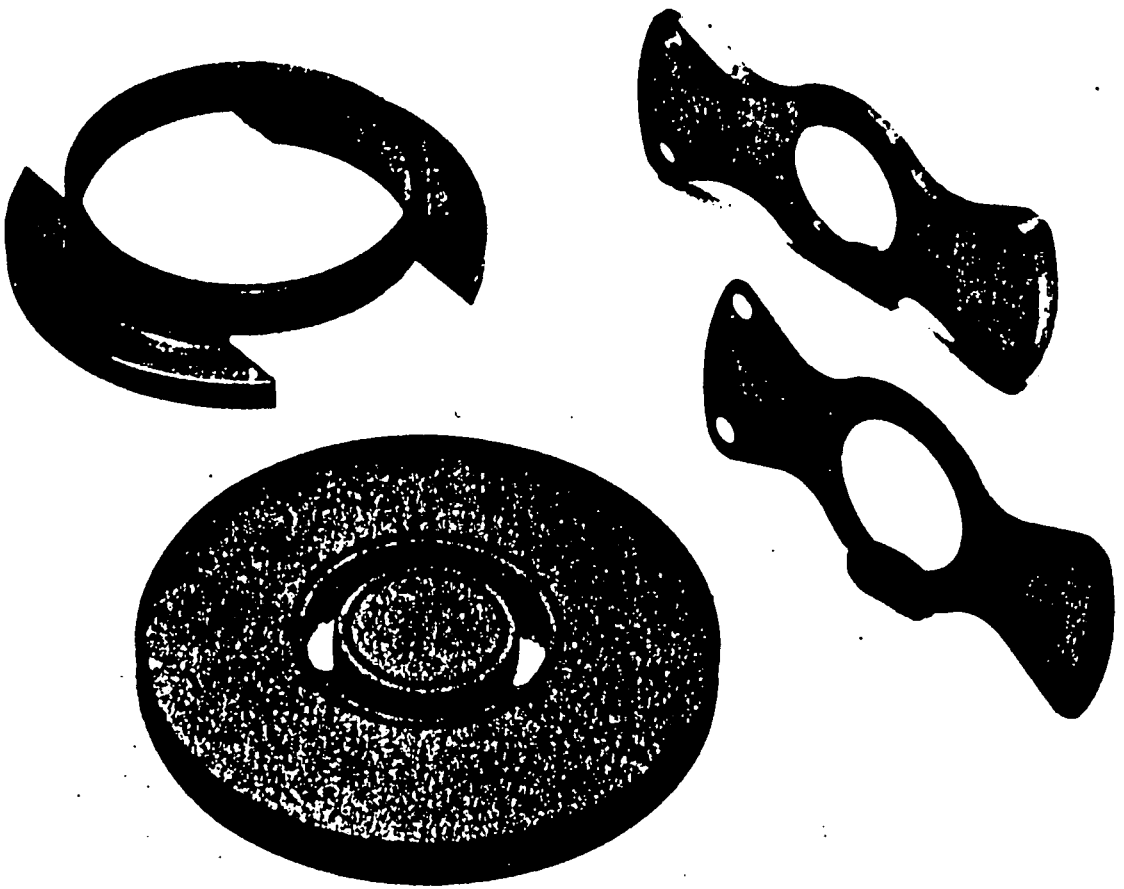

Fig. 5 - Cantilevered discharge flapper with end restraint

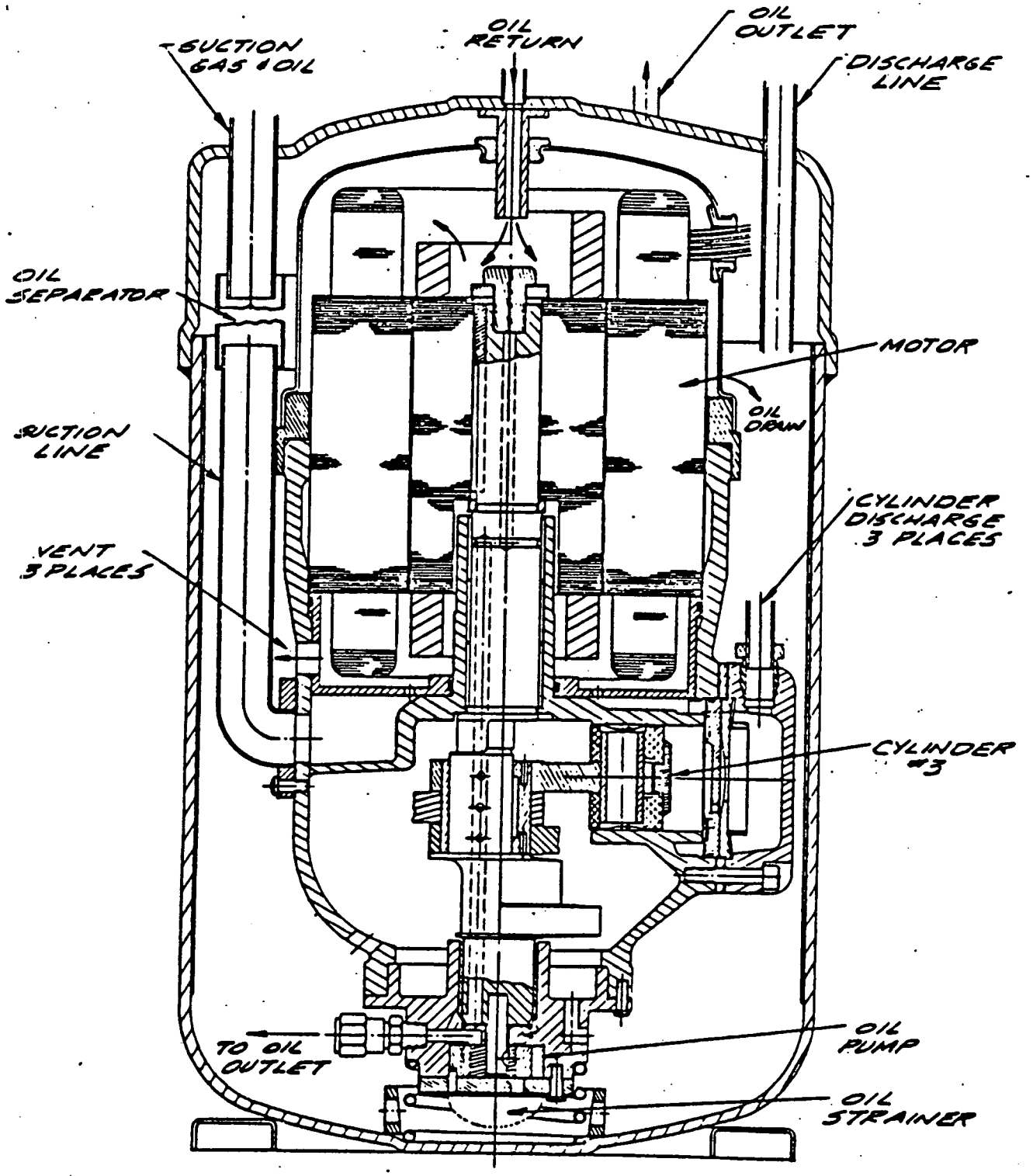

Fig. 6 - Cross section of oil cooled compressor with direct suction

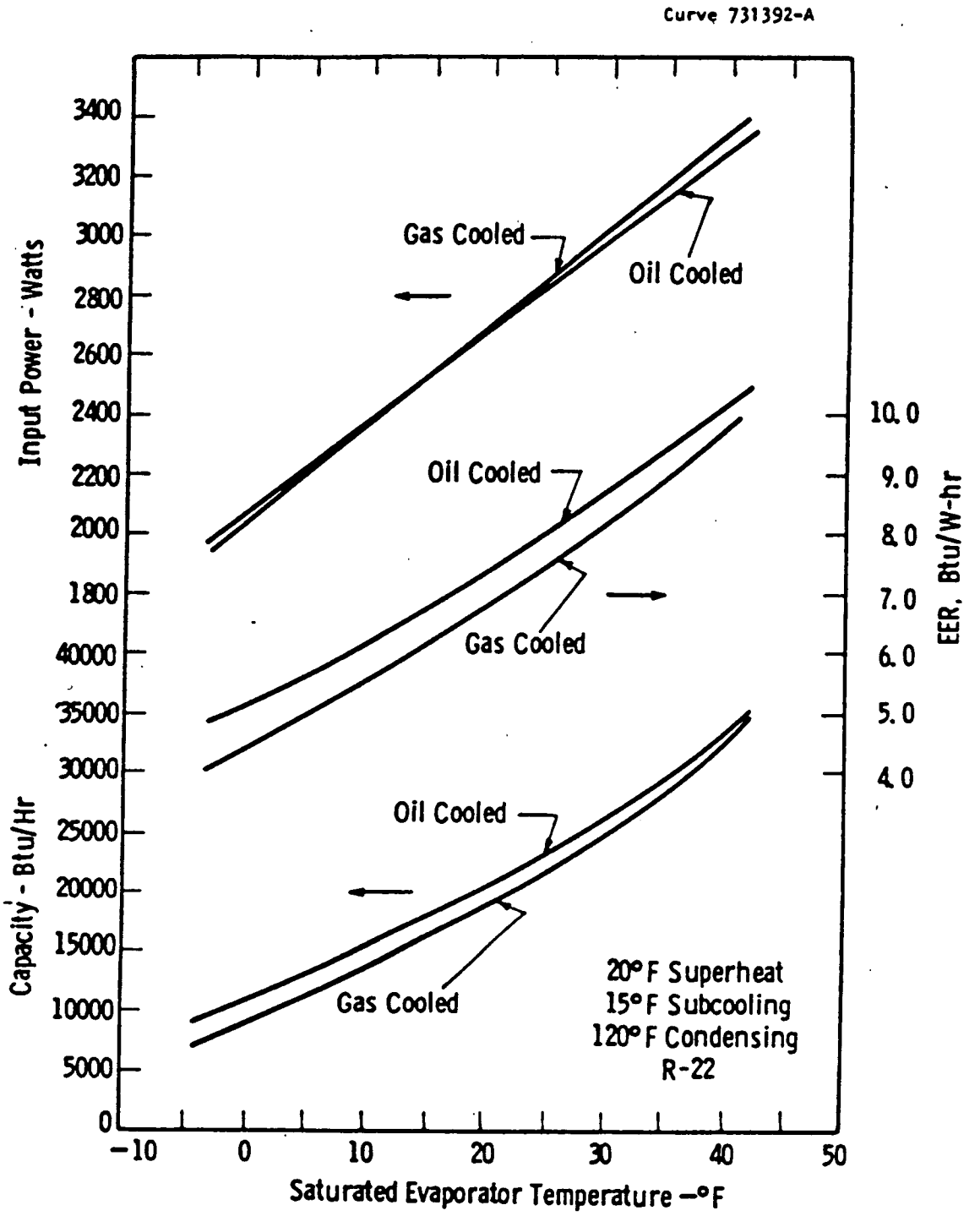

Fig. 7 - Performance comparison between oil cooled and suction gas cooled CD030 compressors

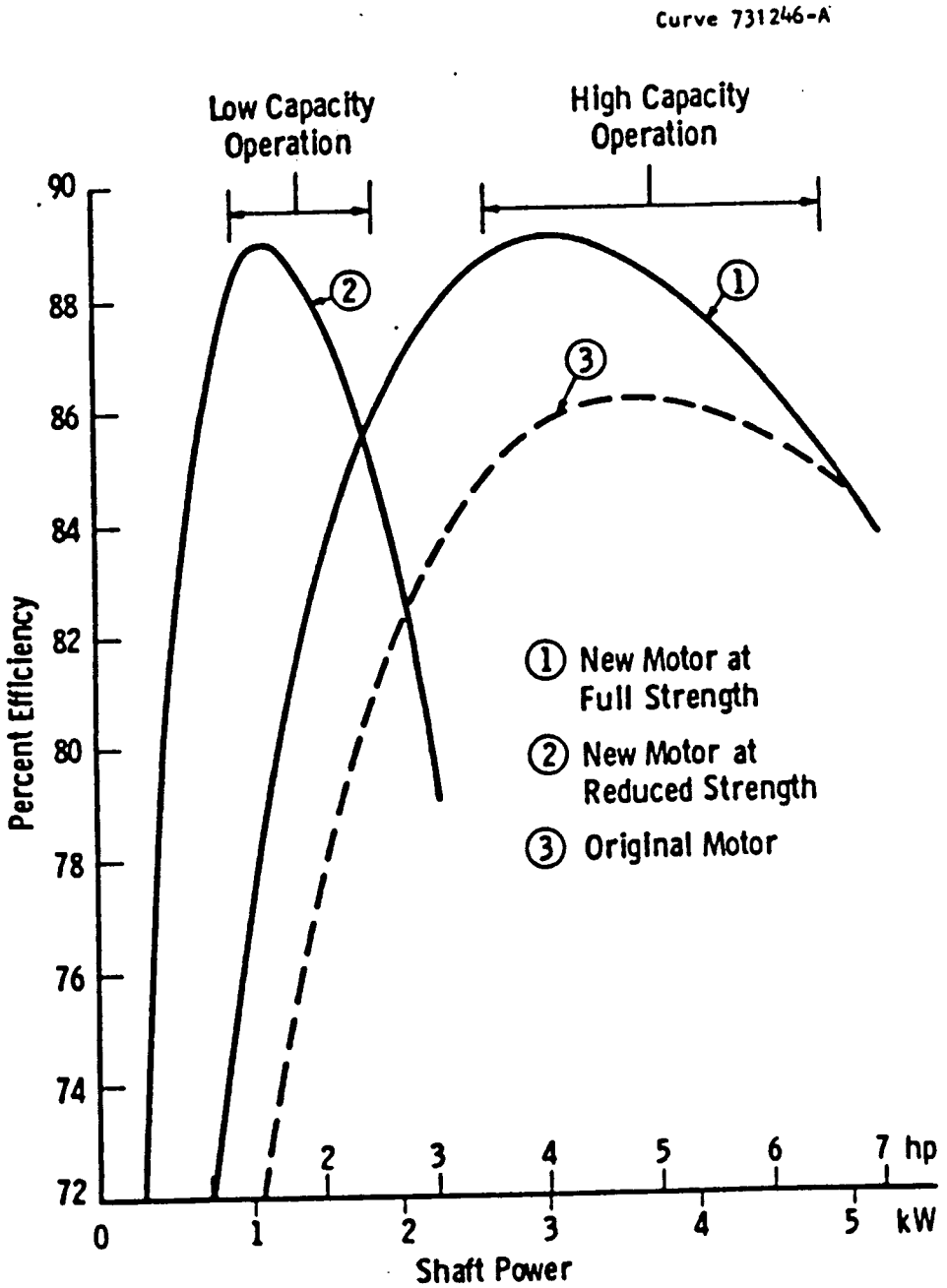

Fig. 8 - Compressor motor efficiency vs shaft power (25°C winding temp.)

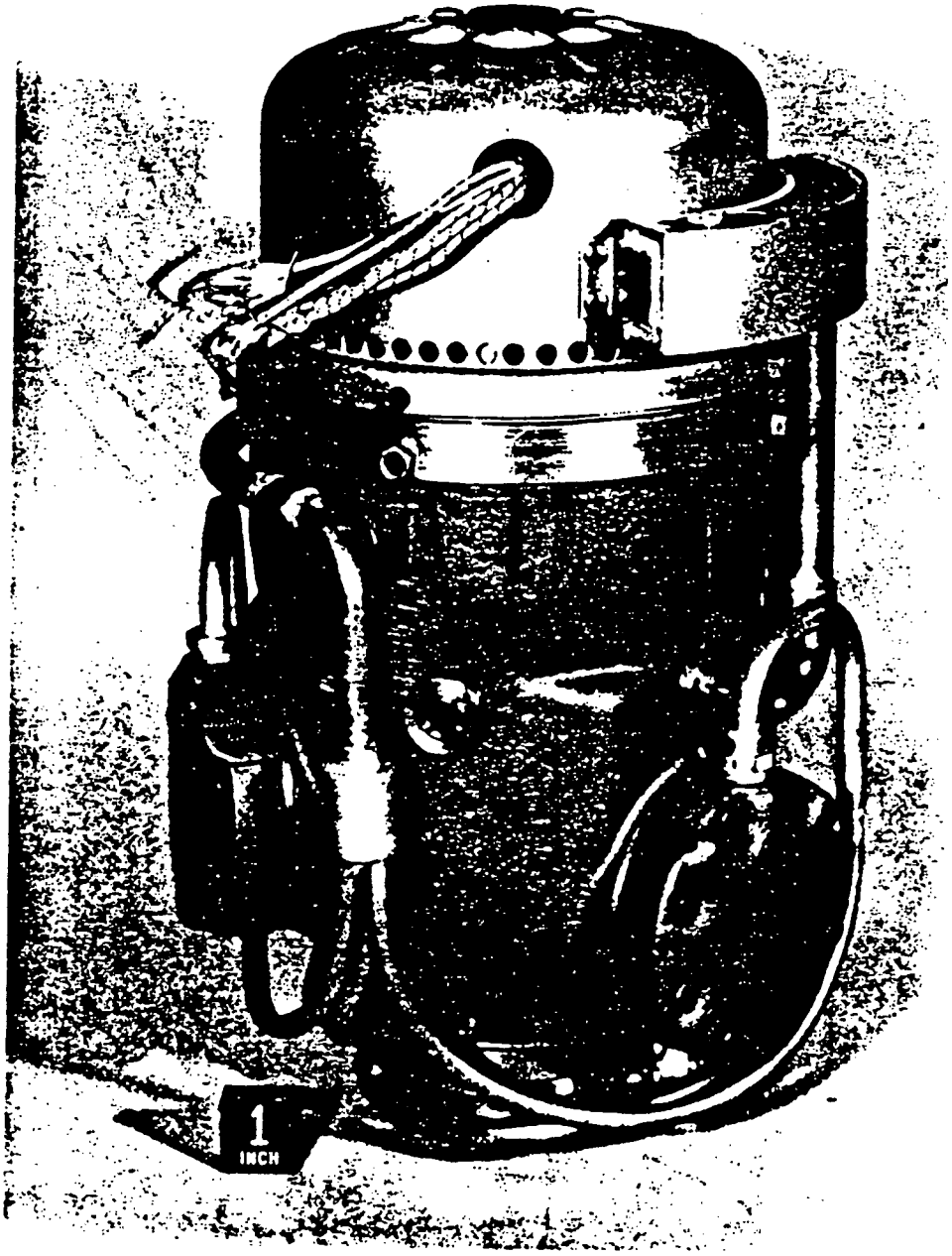

Fig. 9 - Preprototype oil cooled dual-stroke compressor - discharge line view

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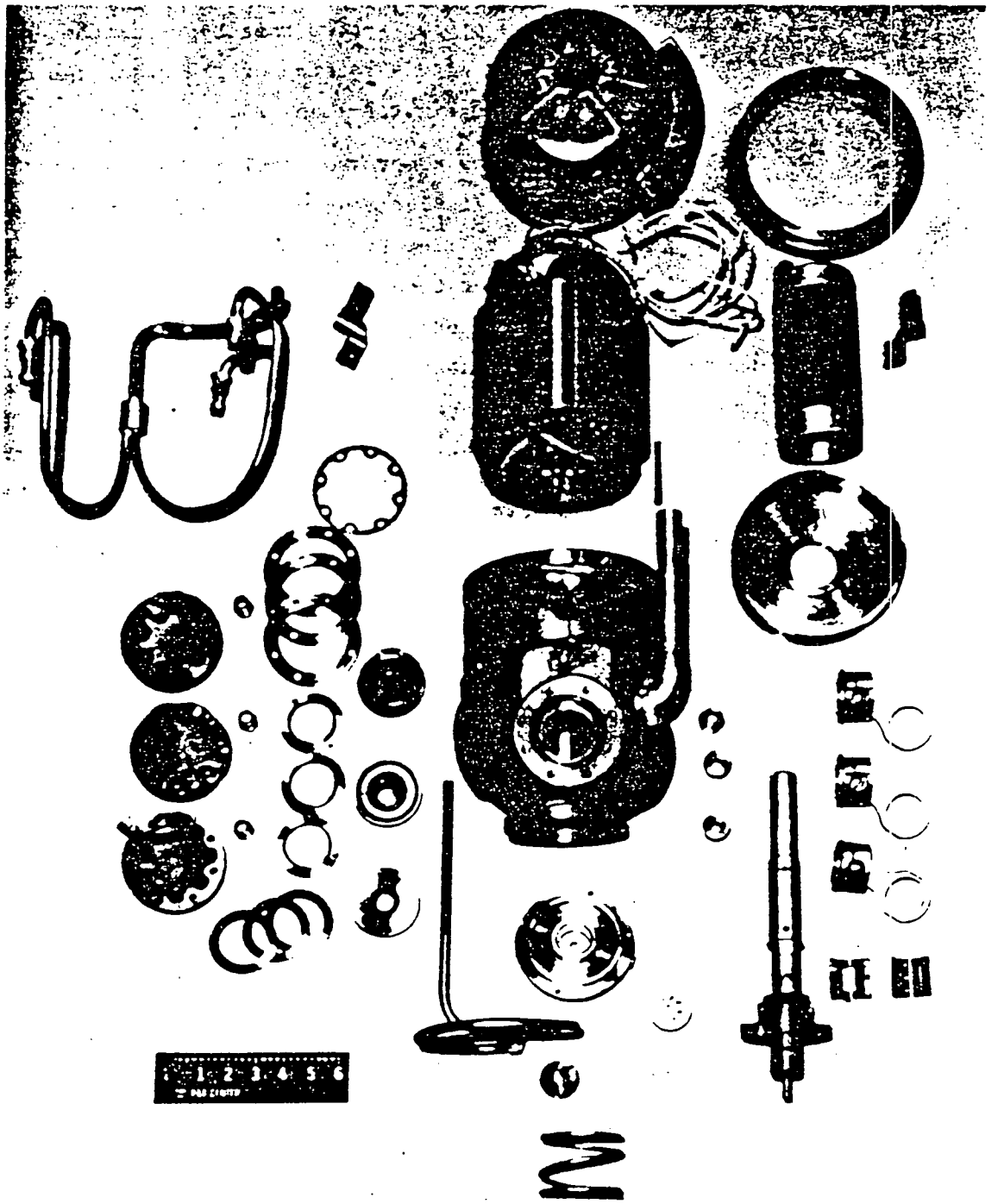

Fig. 10 - Exploded view of preprototype oil cooled dual-stroke compressor

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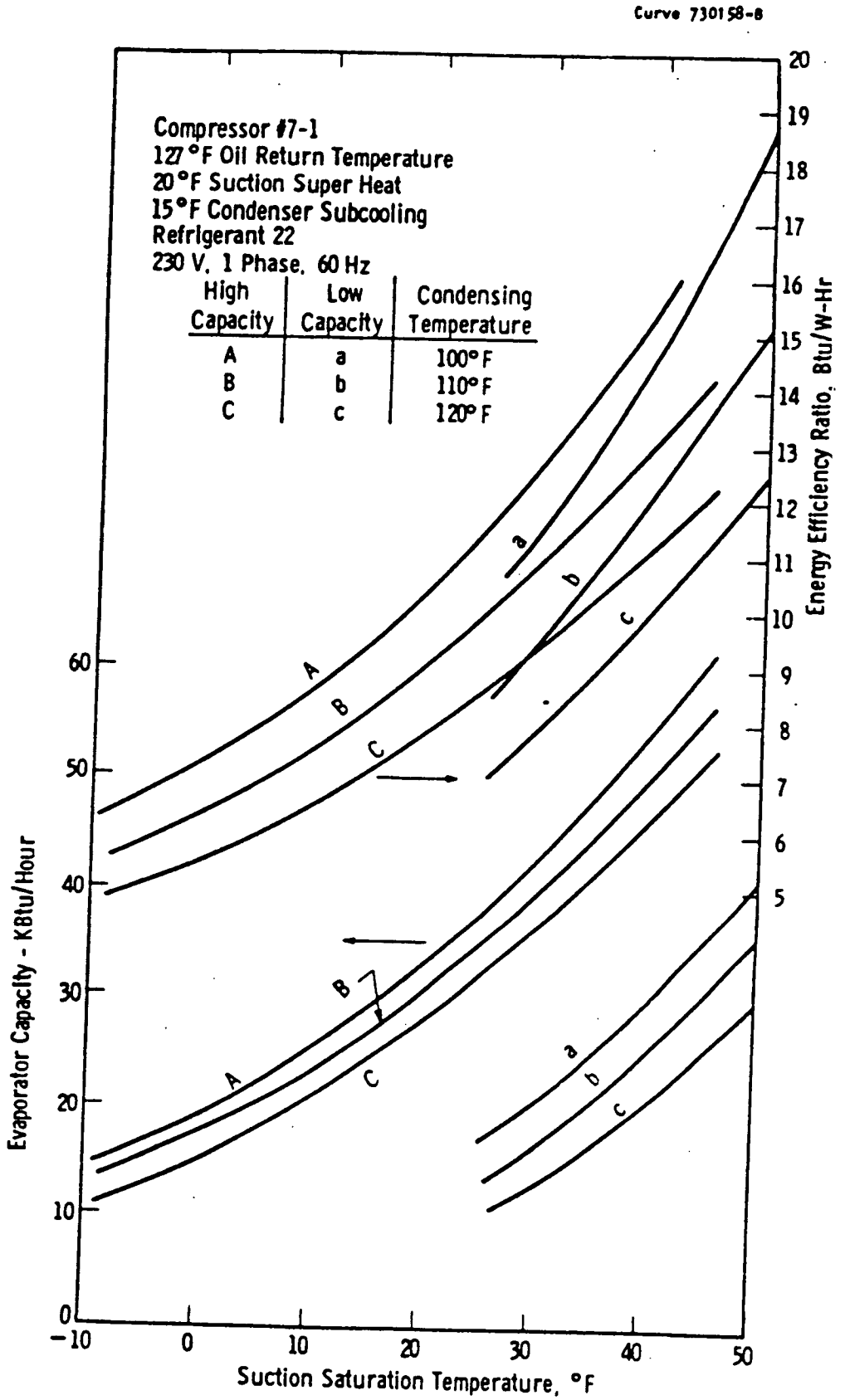

Fig. 11 Performance of prototype dual-stroke compressor (Mod II)