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AN OPTIMIZED TWO-CAPACITY ADVANCED ELECTRIC HEAT PUMP*

S. E. Veyo
Manager, Heat Exchange Systems Research
Westinghouse Electric Corporation, R&D Center
1310 Beulah Road
Pittsburgh, PA 15235

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S. E. Veyo

S. E. Veyo, Manager, Heat Exchange Systems Research

Westinghouse Electric Corporation, R&D Center

1310 Beulah Road

Pittsburgh, PA 15235

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ABSTRACT

A two-capacity residential electric heat pump utilizing a unique dual-stroke compressor has been developed, designed, constructed in preprototypic form and laboratory tested. The estimated annual energy efficiency of this system is twenty percent better than a presently available two-speed electric heat pump. A computer based constrained optimization procedure was used to select heat exchanger proportions, air flow rates and compressor stroke ratio in order to minimize the annual ownership cost subject to comfort constraints upon delivered air temperature and dehumidification. A high-efficiency dual-stroke compressor, indoor blower and outdoor fan plus a microprocessor based control system were specifically developed for this system. This paper highlights the hardware development and summarizes system and component performance.

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* S. E. Veyo, Mgr., Heat Exchange Systems, Westinghouse R&D Center, Pittsburgh, PA 15235

INTRODUCTION

The residential electric heat pump is an embodiment of the well known vapor compression refrigeration machine. As such it represents a mature technology and higher efficiency therefore inevitably is achieved only with higher product cost. The market for residential heat pumps is extremely cost competitive and has been dominated historically by the new home builder in our rapidly growing southern states where comfort cooling is in demand. It is not surprising, therefore, that until very recently heat pump manufacturers have concentrated on large volume, low cost, near state-of-the-art cooling machines of modest efficiency that can also provide limited heating.

The demonstrated fragility of our imported oil supply has caused market forces to rapidly escalate the price of energy over the past decade. Energy conservation, particularly a reduction in imported oil, has become an objective of all oil consuming nations. As a direct result of escalating energy prices, fear of fuel nonavailability, and the threat of minimum efficiency regulations by government, industry has responded by introducing higher efficiency residential heat pumps. Perusal of the latest ARI Directory[1], however, reveals that few heat pump models are offered with a coefficient of performance (COP) even modestly exceeding 3.0 at 8.3°C (47°F) or 2.0 at -8.3°C (17°F).

In order to accelerate the availability of heat pumps with the highest economically viable annual efficiency, the U. S. Department of

Energy initiated, in 1979, the Advanced Electric Heat Pump Research and Development Project. The objective of the resulting cost-shared contract[3] was the development of a preproduction prototype, commercially and economically viable, air source, advanced electric heat pump that can provide performance and comfort with an annual energy efficiency at least twenty percent better than that provided by the highest efficiency heat pumps available in early 1979. A concommitant objective was that the degree of efficiency improvement realized combined with the projected units sold result in a significant national energy savings within fifteen years of product introduction.

Because of a higher population density and a more severe winter climate, ninety percent of the energy consumed for space conditioning in the United States is consumed in our northern climates. This makes the development of an economically viable northern climate heat pump an obvious goal if we are to effect a significant national energy savings. Previous work[3] has shown that the key to an effective northern climate heat pump is a low cost means to achieve capacity modulation.

In early 1979 a newly introduced two-speed, therefore, modulatable capacity, heat pump was identified as the unit whose annual performance was to be surpassed by twenty percent. We have achieved this objective with a system that uses the well known vapor compression refrigeration cycle with R22 as the working fluid. The development process and results achieved are detailed in the following.

TECHNICAL APPROACH

In order to achieve our contract objective we proposed to improve system reliability through demonstrated component durability, to improve component efficiency, to reduce to practice a unique stroke changing concept for compressor capacity modulation, and lastly, to optimize the configuration of the heat pump by minimizing the annual cost of ownership. As a means for providing satisfactory system performance a constraint was placed upon the minimum acceptable delivered air temperature during heating and upon the maximum acceptable sensible-to-total capacity ratio during cooling.

Project time constraints forced the component hardware development phase to be concurrent with the development of the annual optimization procedure and the subsequent analytical system optimization process. In order to carry out the development process, targets for component characteristics and efficiency were specified based upon available information and component development or selection initiated. As achievable component performance and geometric characteristics were made available, they were incorporated into the system analysis, optimization, and design process. The optimization process determined the size influencing design parameters such as heat exchanger surface area and air flow rates. These size parameters were then studied by a

package design team that formulated the various package constraints. These constraints were then considered in an additional phase of system analysis and optimization. Selected components such as reversing valves and model compressors were subjected to continuous testing in order to establish both performance and durability. A breadboard system was built using developmental models of advanced components. This breadboard system was thoroughly instrumented and tested to verify design assumptions and computer codes used for system analysis. A design for the prototype heat pump was completed during the breadboard stage and subsequently two preprototype heat pump systems were built using prototypic materials and components. One preprototype was thoroughly instrumented and tested. The second preprototype will be tested to verify performance.

COMPRESSOR DEVELOPMENT

The major performance fault of the simple air-to-air heat pump during heating is the decline in capacity as the outdoor ambient temperature declines. Since this is the result of basic thermodynamic considerations, it cannot be eliminated or significantly improved. The low temperature capacity can be increased by utilizing a higher capacity unit, but this increases cycling at mild ambient temperatures with a concomittant reduction in efficiency and perceived comfort. It is therefore unattractive. The obvious answer is a heat pump with variable capacity. Since the compressor is the heart of the system, this means that the effective displacement (the product of piston displacement and volumetric efficiency) of the compressor must be variable. Several means are available to provide this function: two-speed compressor motor, parallel compressors, independent refrigeration machines, variable frequency electric drive, and cylinder suction cutoff. All of these suffer from either economically unjustifiable high cost, inflexibility relative to capacity ratio, or both.

In lieu of these, we have reduced to practice a unique low cost means to achieve compressor capacity modulation via stroke change. The concept is disclosed in References 4 and 5. The concept employs the circular, eccentrically bored cam, shown in Figure 1, between crankpin and connecting rod. The action of the cam is such that it is driven against one of two rotational stops attached to the crankpin, depending upon the direction of rotation of the crankshaft.

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 position. The ratio of minimum to maximum compressor stroke is determined by the cam eccentricity. 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 ratio than at low pressure ratio. However, since the low capacity mode of the machine is generally required only during low pressure ratio conditions for the system (i.e., cooling mode and heating mode above 0°C (32°F) ambient),, it is possible to select the cam eccentricity that provides optimum capacity reduction while maintaining efficient system performance. The magnitude of the minimum stroke is bounded between zero and the maximum stroke with its precise value a subject for system optimization.

The adopted cam design is bisectioned with the plane of separation parallel to the crankpin axis. Because of this split arrangement, the cam can be installed without passing over the main bearing journals. The design includes a key pocket in one section to provide end stop positions for a loading key that is inserted in the crankpin.

The preprototype cam is fabricated from machined, heat treated and ground 4140 tool steel. It is anticipated that this component would be manufactured in quantity from pressed powder, sintered and ground tool steel.

This cam was incorporated into a slightly modified standard 14.1 kW (4.0 ton) three cylinder hermetic compressor of radial construction. It was necessary to machine a new crankshaft of lesser crankpin eccentricity with a key slot and new connecting rods with larger center bore. This resulted in a slight reduction in compressor displacement resulting in an equivalent 12.3 kW (3.50 ton) compressor. It was also necessary to procure a drive motor reversible in rotation direction through external reconnection of the windings.

This stroke changing mechanism has been incorporated into nine compressors. Based upon periodic tear down and inspection there is no evidence to suggest abnormal wear or any form of mechanical deterioration in the adopted design. Thermodynamic performance has remained stable. The most tested unit has endured 89,500 cycles of mode reversal and 10,000 hours of operation. The pressure ratio at high capacity was approximately 3.8:1 and at low capacity approximately 3.4:1. Of the nine compressors built, four are preprototypic and intended for system use.

In addition to the dual-stroke mechanism, the preprototype compressor embodies complementary efficiency-improving design features. These include: a. Means for keeping compressor losses out of the suction gas and available elsewhere as useful heat; b) means for increasing the

volumetric efficiency of the compressor; c) improved suction and discharge valving; d) means for maximizing drive motor efficiency over the complete range of operating conditions.

Figure 2 shows efficiency curves for the motors used in our compressor development effort. Curve 1 is the efficiency curve for a new motor designed specifically to match the requirements of the dual-stroke compressor when operating in the high capacity mode. Curve 3 shows the efficiency of the standard production motor. As can be seen, the efficiency of both motors suffers greatly when used to drive the compressor in its low capacity mode. Curve 2 shows the efficiency of the new motor designed so as to maintain high efficiency when lightly loaded. At 1.5 kW (2.0 hp) the efficiency of the new motor is 87.8 percent compared with 77.7 percent for the standard motor. At 3.0 kW (4.0 hp), a typical operating point in high capacity mode, the efficiencies of the new and old motors are 89.0 and 85.7 percent respectively.

The performance of a preprototype compressor is shown in Figure 3. In the high capacity mode at 4.4°C (40°F) evaporating and 48.9°C (120°F) condensing with 11.1°C (20°F) suction superheat and 8.3°C (15°F) condenser subcooling the compressor capacity is 13.5 kW (46,000 Btu/h) with a COP = 3.28 (EER = 11.2 Btu/Wh).

The efficiency of this compressor at these conditions exceeds that of presently available compressors by at least ten percent. More significant is the performance at low evaporator temperatures. At an evaporator temperature of -17.8°C (0°F) and a condensing temperature of

37.8°C (100°F) the measured compressor capacity is 5.6 kW (19,000 Btu/h) and the COP = 2.1 (EER = 7.2 Btu/Wh). At 0°C (32°F) evaporating temperature and 37.8°C (100°F) condensing temperature in low capacity, 43.3°C (110°F) condensing temperature in high capacity, the capacity ratio is approximately 2:1 with the mean COP between high and low capacity approximately 3.3. As installed in a system the compressor will not be operated in low capacity heating mode when the suction saturation temperature is less than approximately 0°C (32°F). Similarly, it will not be operated in the low capacity cooling mode when the pressure ratio exceeds 3.5.

FAN DEVELOPMENT

A 12.3 kW (3.5 ton) state-of-the-art heat pump outdoor air mover is required to move relatively large volumes of air, (typically 1416 l/s (3000 scfm) through the relatively low flow resistance (typically 37.4 Pa (0.15 in of water)) due to heat exchanger and protective grilles.

The outdoor air mover is typically a two-to-four bladed propellor fan turning at six-pole (1020 rpm) or eight-pole (825 rpm) speed. The propellor blades are of large chord, more or less shovel-shaped and riveted to a simple steel spider hub driven by the motor. The blades are commonly of constant thickness, perhaps ribbed for stiffness, and of constant camber and incidence and with high solidity, overlapping in some designs. The fan hub is open and air can recirculate in some applications. The propellor rotates in a running ring or shroud, many times without curved inlet flare, with a generous clearance of a quarter-inch or more between blade tip and shroud. A restricted fan inlet and short or nonexistent diffuser ring are common place. Consequently, outdoor air mover static efficiencies as applied are generally 20 to 35 percent rather than the 45 percent peak efficiency indicated in some vendor literature.

The fan motor for residential units is typically a permanent split capacitor type of 124 W to 250 W (1/6 to 1/3 hp) rating and of weatherproof design yielding about 55 percent efficiency. With 35 percent

fan static efficiency and a 55 percent efficient motor, the overall air moving efficiency is only 19 percent and power consumption is of the order of 250 W. Our goal was to halve the outdoor air moving power.

The preliminary specification for our outdoor air mover called for an air delivery of 1369 l/s (2900 scfm) against an upstream flow resistance of 49.8 Pa (0.20 in of water) with a static efficiency of 60 percent. The motor was required to be at least 65 percent efficient. Air mover size was constrained to 610 mm (24 in) in diameter and 300 mm (12 in) in height, exclusive of mounting flanges. The fan noise[6] was not to exceed a noise rating (SRN) of 20 and 18 or less was desired.

Since a quiet fan with good static efficiency was required a multivane axial flow design was proposed. A fan of this type has five or more blades of relatively narrow chord and low solidity. A large solid hub is used to prevent recirculation.

The fan design procedure used had three levels. First, since both low noise and high static efficiency were desired, the interaction of fan performance and size constraints was examined with the objective of producing a fan with minimal blade surface velocities in order to minimize fan noise. Second, since high static efficiency was the primary design criterion, the impact of the fan design parameters developed in the first level upon static efficiency was examined in greater detail through systematic variation of parameter values. Third, the best candidate designs were studied in detail using more rigorous procedures embodied in proprietary computer programs used for turbomachinery design.

Our design procedure specified the largest allowable diameter and the lowest allowable rpm. The propellor was thus fixed at 610 mm (24 in) tip diameter and 850 rpm, consistent with a high-efficiency eight-pole motor.

The breadboard fan employed eight cambered plate blades with variable pitch, a 0.5 hub-to-tip diameter ratio, a thirty-one bladed stator vane row, a radiused inlet flare, a conical after body over the motor, and a 152 mm (6 in) constant diameter diffuser stack. This fan was tested over a range of values for blade pitch and rpm and for several options concerning diffuser design. Configured to meet the preliminary specification for the system, the breadboard fan delivered 1463 l/s (3100 scfm) with a pressure rise of 52.3 Pa (0.21 in of water) with a static efficiency of 52 percent and absorbed approximately 138 W (0.185 hp) of shaft power.

Subsequent system design efforts resulted in modifications to the breadboard fan design. The inlet flare radius was reduced, the stator vane row eliminated, and the diffuser stack redesigned upon a relaxation of the diameter constraint and the fan blade pitch altered slightly. The outdoor air mover as employed in the preprototype system is shown in Figure 4. It delivers 1449 l/s (3070 scfm) through the outdoor unit with an upstream (suction side) static pressure drop of 22 ~~67~~ Pa (0.09 ~~0.27~~ in of water) at 850 rpm. The fan motor consumes 150 W of electrical power. In low speed operation the air mover rotates at 609 rpm and the motor consumes 78 W of electrical power.

BLOWER DEVELOPMENT

The duty for a heat pump indoor air handler differs considerably from that of the outdoor fan. The indoor air flow is typically half that of the outdoor unit while the static pressure rise requirement is much higher. The indoor air handler must provide sufficient static pressure rise to overcome the air flow resistance of the house duct system, the electrical resistance heating elements, the air filter and the indoor heat exchanger. For a typical 12.3 kW (3.5 ton) heat pump, the indoor design air flow is 743 l/s (1575 scfm) with an air mover static pressure rise requirement of 212 Pa (0.85 in of water).

In a state-of-the-art heat pump the indoor air mover typically uses a squirrel cage (forwardly curved vane) blower wheel 230 to 250 mm (9 to 10 in) in diameter and 200 to 300 mm (8 to 12 in) wide with double air entry. The blower wheel is mounted in a simple volute scroll. The motor is of the permanent split capacitor type and of open design with a rated capacity of 250 W to 375 W (1/3 to 1/2 hp). It is typically mounted in the eye of the blower wheel to take advantage of the excellent air cooling available and it is not uncommon for the delivered power to be fifty percent more than the motor's nameplate rating.

Although the efficiency of a forwardly curved vane centrifugal blower can approach 60 percent, the measured efficiency as typically used is at most 35 percent. This lower achieved efficiency is due in part to the flow restricting placement of the direct drive motor in the

eye of the blower wheel. This motor placement saves space, reduces cost and improves reliability relative to a less flow restricting belt drive arrangement. The principal trade off however is between cost and efficiency. The typical six-pole single phase ac drive motor, well cooled and operating at high power output, runs with high slip and is typically between 50 and 60 percent efficient. Overall indoor air moving efficiency is thus between 17 and 20 percent with a power consumption between 800 and 900 watts.

In order to achieve a doubling in overall air mover efficiency we proposed a high-efficiency single-entry blower wheel design with multiple backwardly curved air foil shaped blades driven by a high-efficiency motor. Our design goal was a 65 percent efficient blower driven by a 65 percent efficient motor. The preliminary performance specification for this blower called for a delivered air flow of 743 l/s (1575 scfm) with a static pressure rise of 212 Pa (0.85 in of water). Dimensionally, the air mover was constrained to fit within a cabinet 572 mm (22.5 in) wide. As a result of the heat pump system optimization and design effort the performance specification for the air mover was later revised to require a 660 l/s (1400 scfm) delivery rate with a static pressure rise of 237 Pa (0.95 in of water).

The indoor blower wheel as produced for our preprototypic heat pump system is shown in Figure 5. The air mover assembly fits inside the indoor air handler which measures 572 mm (22.5 in) in width. The 406 mm (16.0 in) effective tip-diameter blower wheel uses a conical backplate, a conical inlet plate, and ten backwardly curved

cambered plate blades. Except for the steel mounting hub it is constructed entirely of aluminum. The blower wheel is mated to its own shaft at the mounting hub. The shaft is carried within a simple volute sheet steel scroll by two sets of bearings, one set mounted to the back plane of the scroll, the other set centered over the inlet opening by a wire form bearing carrier. A rounded entry section seals the inlet side of the scroll to the conical inlet plate of the blower wheel. The 224 W (0.30 hp) blower motor is axially aligned with the blower shaft and supported by a wire form belly band carrier. The motor shaft and blower shaft are joined by a flexible coupling. The air mover assembly slides into the air handler package and is captured therein using the flanges at the discharge plane of the scroll. A lower support bracket secures the bottom of the scroll to the air handler package.

The performance of the preprototypic blower wheel mounted in a mock-up version of the volute scroll and tested in a blower test cell is shown in Figure 6. At the design point, the blower requires less than 224 W (0.30 hp) shaft power to deliver 660 l/s (1400 scfm) with a pressure rise of 237 Pa (0.95 in of water) at 1070 rpm. The design point static efficiency is 63 percent. At the peak efficiency of 70 percent, the blower will deliver 520 l/s (1100 scfm) with a static pressure rise of 286 Pa (1.15 in of water) and requires 209 W (0.28 hp) shaft power. The blower will deliver 660 l/s (1575 scfm) with a static pressure rise of 199 Pa (0.80 in of water) at an efficiency of 58 percent and will require 246 W (0.33 hp) shaft power.

As installed and tested in our preprototype heat pump, the indoor air mover delivers 663 l/s (1405 scfm) of 16.7°C (62°F) air at 1092 rpm into an external downstream flow resistance of 87 Pa (0.35 in of water). A 224 W (0.30 hp) high-efficiency motor was designed to mate with this blower. Measured electrical power is 390 W at full air flow and 190 W at reduced air flow corresponding to 770 rpm.

ANNUAL OPTIMIZATION

In order to select the compressor capacity ratio, the proportions of the heat exchangers and the indoor and outdoor air flow rates we have employed a computer-based optimization procedure. An optimum system is defined as that which minimizes the annual ownership cost. Annual ownership cost is defined as the sum of the annual charge for consumed electrical energy plus an annual charge for the capital invested in the system plus an annual allowance for maintenance. We have used a constant dollar analysis wherein $0.0497/kWh is the cost of electricity, 15 years is the expected useful life of the system, 3 percent is the true cost of invested capital and 2.5 percent of first cost is the levelized annual maintenance cost. A hierarchy of computer programs is used to carry out the optimization procedure.

A proprietary system simulation program is the major program at the first level. Given a performance map for the compressor, design point data for the air movers and the geometry of the heat exchangers this program will yield the capacity and efficiency of an arbitrarily configured system as well as loop state point information as a function of the specified outdoor ambient temperature and the indoor return air temperature (and humidity when cooling). Systems employing expansion valves (either condenser subcooling or evaporator superheat control), capillary tubes, orifices, or fixed parameter values for subcooling or superheat can be accommodated. Charge distribution and inventory calculation are part of the procedure.

Heat exchangers are modeled in simplified form within this program. When exercised as a subprogram within the optimization procedure certain simplifications are made regarding refrigerant side pressure drops in order to minimize code execution time.

A program to estimate seasonal performance of the heat pump system as applied to a dwelling is at the second level. The procedure promulgated by DOE[7] is used with certain modifications. The DOE procedure synthesizes a system performance map based upon system performance at five outdoor dry bulb temperatures and two sets of indoor conditions. Our seasonal performance estimation procedure does likewise. Capacity degradation for frosting conditions is based upon empirical data.

In order to obtain the load specificity required for system optimization we use well specified load lines rather than the ranges used in the DOE procedure. The heating load line is anchored at 18.3°C (65°F) and the cooling load line at 23.9°C (75°F). Offsets from these anchor points can be used if desired. The lowest ambient temperature system balance point at maximum system capacity mode is a parameter in the optimization procedure. For a fixed value, the heating load line is then defined as connecting the lowest ambient temperature balance point with the anchor point. The magnitude of this slope is retained for the sensible cooling load but opposite in sign. The true cooling load line has the slope of this sensible load line multiplied by a factor (1.4) in order to account for solar and latent loading. The value of this multiplicative factor can be made a function of the climate zone considered.

In our optimization procedure we use the climate data prescribed by DOE[7]. We prefer, however, to use site specific weather data within each of the DOE climatic zones both heating and cooling with site specific heating and cooling hours when estimating the seasonal performance of fixed hardware. The well known bin analysis is used to calculate the annual energy usage given specification of the system performance, the load line, and the climate. In estimating seasonal performance we use $C_D = 0.1$ with separate calculation of that parasitic power devoted to the crankcase heater and control system.

It is clear that given a complete specification of the heat pump system hardware, repeated execution of the system simulation code will yield the points needed to estimate seasonal and annual performance using the modified DOE procedure.

The computer program at the third level is the optimizer itself. A performance function is formulated and the gradient of this function is determined by numerically evaluating the derivatives of the performance function with respect to each of the independent variables defining the system. The independent variables considered are the face areas of the heat exchangers and the indoor and outdoor air flow rates.

The performance function used is the thermal energy beneficially delivered, both heating and cooling, per unit of annual ownership cost. Using the gradient of beneficially delivered thermal energy per unit of annual ownership cost, the proportions of the system (heat exchanger sizes and air flow rates) are adjusted so as to maximize the performance function.

In order to accommodate comfort constraints the performance function is modulated by a penalty function so as to avoid systems that deliver air colder than 32°C (90°F) when heating or a sensible to total capacity ratio greater than 0.7 when cooling. Only one set of data for heat exchanger tube diameter, tube pitch, fin thickness and fin spacing was considered.

As used for this project, the system low temperature heating balance point, the compressor capacity ratio and the number of heat exchanger tube rows was externally controlled. That is, the procedure is carried out for several fixed values of each and the best balance point and best capacity ratio are selected on the basis of the maximum value of the previously determined "quasi-optimum" performance function.

This optimization procedure has been carried out for a system using an early version of the dual-stroke compressor in DOE climatic regions II, III, IV and V. See Reference 8. Based upon this study, only optimization for region V was considered in subsequent applications of the procedure.

SYSTEM PERFORMANCE

Based upon the heat exchanger sizes and air flow rates resulting from the optimization procedure a preprototypic system was designed and built. Certain compromises were necessary, however, because the optimization procedure does not consider the detailed mechanical design of the hardware. In addition, the preprototypic air flows are larger than the indicated optimimum because the air movers had to be specified prior to the last optimization pass in order to ensure that hardware would be available to meet contract deadlines. Similarly, the last optimization pass used compressor characteristics one model generation prior to those shown in Figure 3. The preprototype system uses the compressor whose characteristics are shown in Figure 3, the fan shown in Figure 4, and the blower shown in Figure 5. System performance was measured in the laboratory in accordance with applicable codes and procedures[7]. Measured heating performance is shown in Figure 7. Measured cooling performance is shown in Figure 8. These performance curves represent data from the system as first tested. There are several adjustments available by which we can tune the system to achieve better performance. The estimated seasonal performance of this system based upon the data shown in Figures 7 and 8 is compared with the estimated performance of currently available high-efficiency systems, based upon published manufacturer's data, as installed in a 149 m^2 (1600 ft^2) dwelling with level II insulation[3] in Table 1.

DISCUSSION

The advanced electric heat pump is a high efficiency design with two discrete capacity modes in both heating and cooling. Optimally sized to a dwelling in a northern climate (i.e., DOE[7] regions IV and V) the low temperature, therefore high capacity, heating balance point is approximately -8.3°C (17°F). The low capacity balance temperature is then approximately 1.7°C (35°F). At the low capacity balance temperature the ratio of high to low capacity is 1.8 for the system with low capacity COP = 3.26 and high capacity COP = 3.20. The two-speed heat pump that served as our system of comparison is also well sized to the same dwelling. Compared on the same basis with clear coils at 1.7°C (35°F) the two-speed unit has slightly higher capacity, a capacity ratio of about 1.7, a low capacity COP = 2.70 and a high capacity COP = 2.55. The advanced heat pump with dual-stroke compressor is therefore 20 percent more energy efficient in either capacity mode at the ambient temperature corresponding to the mode in the distribution of hourly temperature occurrence in the northern United States. At an ambient temperature of -8.3°C (17°F) the dual-stroke heat pump has an efficiency advantage of 35 percent with very nearly the identical heating capacity. At the high ambient temperature rating condition of 8.3°C (47°F) the advanced heat pump has a heating capacity of 8.41 kW (28,700 Btu/h) in low capacity mode and a COP = 3.83. It is approximately 25 percent more efficient than the two-speed heat pump of comparison. Note should be taken of the fact that the delivered

air temperature of the advanced heat pump exceeds 32.2°C (90°F) for all outdoor ambient temperatures greater than -8.3°C (17°F). Since the advanced dual-stroke heat pump has an efficiency advantage exceeding 20 percent across the operating ambient temperature range, it will clearly have a similar heating season performance advantage. Inspection of Table 1 reveals this to be the case in the northern climates. Seasonal performance is estimated using a calculational procedure based upon that promulgated by DOE[7] but differing as outlined previously.

In the cooling mode, low capacity operation is adequate up to an outdoor ambient temperature of 35°C (95°F). At 27.8°C (82°F) the cooling capacity of both the dual-stroke and two-speed heat pumps in low capacity mode are equivalent. The dual-stroke unit enjoys an efficiency advantage of better than 20 percent over the low capacity mode operating range. In high capacity cooling mode the advanced heat pump is 20 percent more efficient than the two speed unit at 35°C (95°F).

Table 1 indicates that the advanced dual-stroke heat pump will exceed the energy efficiency of an existing two-speed high-efficiency heat pump during the heating season by 20 percent or more in the northern climatic regions of the U.S. (DOE regions III, IV, V). Annually the two-speed heat pump consumes more than 20 percent more electricity than the advanced dual-stroke unit in DOE climatic regions IV and V and 20 percent more in DOE region III.

Also shown in Table 1 are the estimated performance figures for current high-efficiency single-speed heat pumps of nominally 8.8 kW (2.5 ton) and 14.1 kW (4 ton) capacity. The smaller capacity unit would be sized appropriately to the same dwelling considered for the two capacity units.

It can be seen that in a northern climate (Pittsburgh) the larger single-speed heat pump provides a 6 percent higher heating season efficiency but a 12 percent poorer cooling season efficiency than the smaller unit. Note that the cost of thermal energy beneficially delivered per unit of energy is 2 percent higher with the larger, over-sized unit. Obviously over sizing of single capacity units does not pay if the same level electrical rates prevail both summer and winter. Consideration of comfort requirements would provide additional reasons for rejecting the over-sized single capacity heat pump.

Comparing the two-speed unit and the smaller single-speed unit we see that in Pittsburgh, the two-speed unit has a 13 percent better heating season efficiency and a modest 1.5 percent advantage in cooling season efficiency. Since the instantaneous heating efficiency of the single-speed unit of comparison is comparable to the instantaneous low capacity efficiency of the two-speed unit, the heating season performance advantage is attributable to the ability to modulate capacity.

Comparing the advanced dual-stroke heat pump with the two-speed heat pump we can see the heating season energy efficiency advantage. A small portion of this advantage is due to the slightly larger capacity ratio available to the dual-stroke unit at capacity switch-over conditions. Of greater significance is the fact that the heating capacity of the advanced unit does not decline as rapidly with declining temperature as does that of the two-speed unit. Note that at the low temperature rating condition -8.3°C (17°F), the heating capacities are nearly identical. This advantage stems primarily from the superior low evaporating temperature performance of the compressor (see Figure 3). We are aware of no

hermetic, residential service compressor with equivalent performance. Further, the dissipation from this compressor is available to heat the indoor air stream in the heating mode of operation. The compressor is mounted indoors as we recommended in Ref. 3 but in a manner different from that practiced by others.

An advantage in both heating and cooling stems from our high-efficiency air movers. In the high capacity mode of operation the indoor air mover delivers roughly 660 l/s (1400 cfm) of air while consuming 390 W of electrical power. In the low capacity mode of operation the indoor air mover delivers roughly 470 l/s (990 cfm) of air and consumes 190 W of electrical power. The outdoor air mover delivers 1450 l/s (3070 scfm) through a suction side flow resistance of 67 Pa (0.27 in of water) static pressure and consumes 150 watts of electrical power. Operation of the outdoor air mover in low speed is restricted to cooling mode for ambient temperature less than 27°C (80°F). The electrical power consumed by the air moving equipment has been reduced to about half that consumed by state-of-the-art equipment.

In the study of Ref. 3 we advocated two parallel compressors as the most cost effective means then known to achieve satisfactory capacity modulation for a northern climate heat pump. Gratifyingly such a system is now available in the market place. The advanced electric heat pump with dual-stroke compressor has two significant advantages over the parallel compressor concept however. The first is smaller compressor package size while the second and most important is significantly lower cost. We estimate that the high-efficiency dual-stroke compressor will represent a premium cost to the retail customer of $110 over the basic single capacity

compressor upon which it is based. The stroke changing mechanism, cam plus modified crankshaft, represents only 15 percent of this additional retail cost. The most significant limitation to the stroke changing concept is that it appears to be limited in application to multicylinder radial compressors.

While the system design procedure has forced the system cost to be economically justified on a life cycle basis, this is not sufficient to guarantee commercial success. Based upon market analyses[8], the selection of alternative higher efficiency components in place of existing state-of-the-art components was governed by the rule that the expected incremental cost at the retail level had to be recovered by energy savings in less than four years. Of the components described here, all can more than satisfy the requirement.

Two prototype sample advanced electric heat pumps have been built and tested in the laboratory. In appearance and construction they represent mass producible hardware. Considerably more testing, especially under field conditions, is required to verify system durability and the design concepts embodied before it can be said that the system, in its entirety, is ready for commercialization.

CONCLUSIONS

1. The advanced electric heat pump with dual-stroke compressor meets its performance objective by providing an estimated annual energy efficiency 20 percent better than the most efficient heat pump available at contract inception.

2. The dual-stroke compressor, indoor blower and outdoor fan can save sufficient energy relative to their state-of-the art alternatives so as to permit recovery of their premium first cost in less than four years of operation.

3. The dual-stroke mechanism for capacity modulation is low in cost and is in all likelihood the lowest cost alternative available.

4. A procedure for optimizing heat pump system proportions based upon estimated annual performance and life cycle cost has been developed.

5. Additional testing, especially under field conditions is necessary before commercialization can be expected.

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Table 1 - Comparison of Seasonal Performance[1]
(149 m^2 Level 2 House)[2]

Climate Region (DOE)		II[3]	III[4]	IV[5]	V[6]
Heating Load MWh		9.25	20.15	32.17	37.68
Cooling MWh		24.59	11.68	4.87	6.11
Dual-Stroke	Installed Cost[7]	1.49	1.49	1.49	1.52
Heat Pump	Ownership Cost[9]	0.77	0.74	0.84	1.02
	$/MWh	27.65	27.88	27.68	29.01
	SCOPH	2.99	2.74	2.60	2.26
	SCOPC	2.67	2.95	3.01	3.02
High-Efficiency	Installed Cost[8]	0.97	0.97	1.00	1.03
Single-Speed	Ownership Cost[9]	0.76	0.79	1.00	1.28
Heat Pump[10]	$/MWh	27.92	29.86	32.32	35.02
(8.8 kW)	SCOPH	2.47	2.08	1.86	1.61
	SCOPC	2.36	2.46	2.46	2.48
High-Efficiency	Installed Cost[8]	1.50	1.50	1.53	1.56
Two-Speed	Ownership Cost[9]	0.87	0.86	1.02	1.27
Heat Pump[11]	$/MWh	30.92	32.42	33.07	34.57
(13.5 kW)	SCOPH	2.54	2.25	2.10	1.83
	SCOPC	2.41	2.53	2.50	2.54
High-Efficiency	Installed Cost[8]	1.22	1.22	1.25	1.28
Single-Speed	Ownership Cost[9]	0.91	0.87	1.02	1.27
Heat Pump[12]	$/MWh	32.01	32.63	32.97	34.68
(14.1 kW)	SCOPH	2.27	2.08	1.97	1.74
	SCOPC	2.07	2.20	2.21	2.22

NOTE: Installed cost and ownership cost are multiples of those for a high-efficiency single-speed heat pump in region IV.

1. W modified DOE Procedure (Fed. Register 12-27-79)
2. EM319 EPRI
3. Phoenix, AZ
4. Nashville, TN
5. Pittsburgh, PA (Cleveland weather)
6. Minneapolis, MN (Madison weather)
7. Estimated
8. Estimated via market study
9. Energy (@ 4.97¢/kWh + Amortized) Installed cost (15 year life and 3% true interest) + Maintenance (2.5% of installed cost per year)
10. Mfg. A, single speed
11. Mfg. B, two speed
12. Mfg. A, single speed

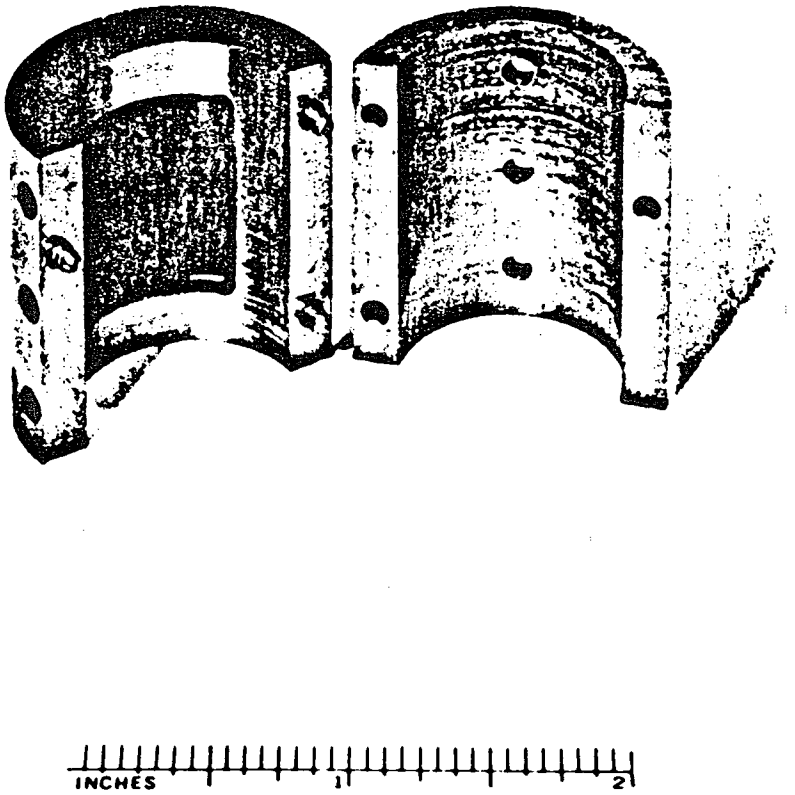

Figure 1 - Dual-stroke cam

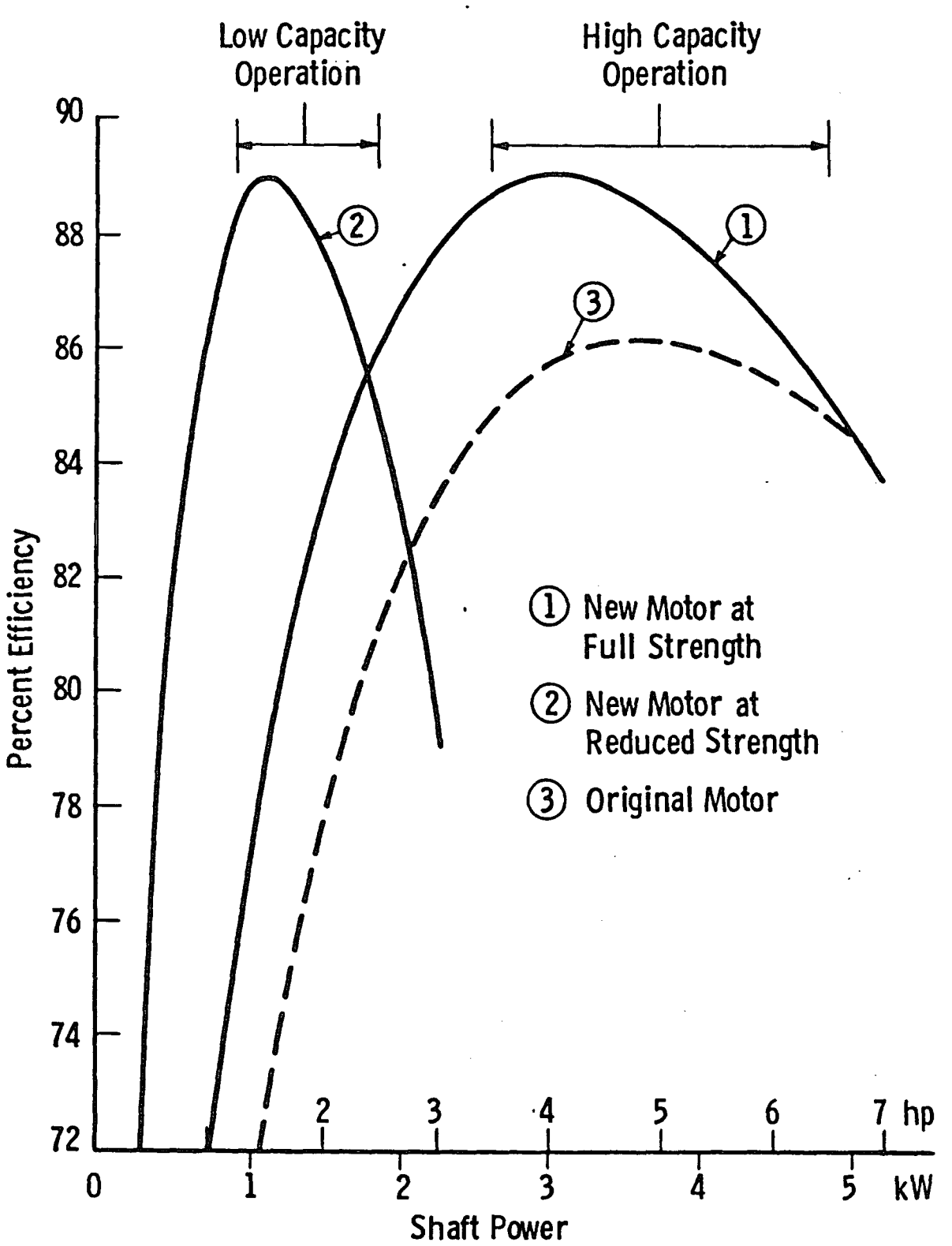

Fig. 2—Compressor motor efficiency vs shaft power (25°C winding temp.)

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Curve 731248-B

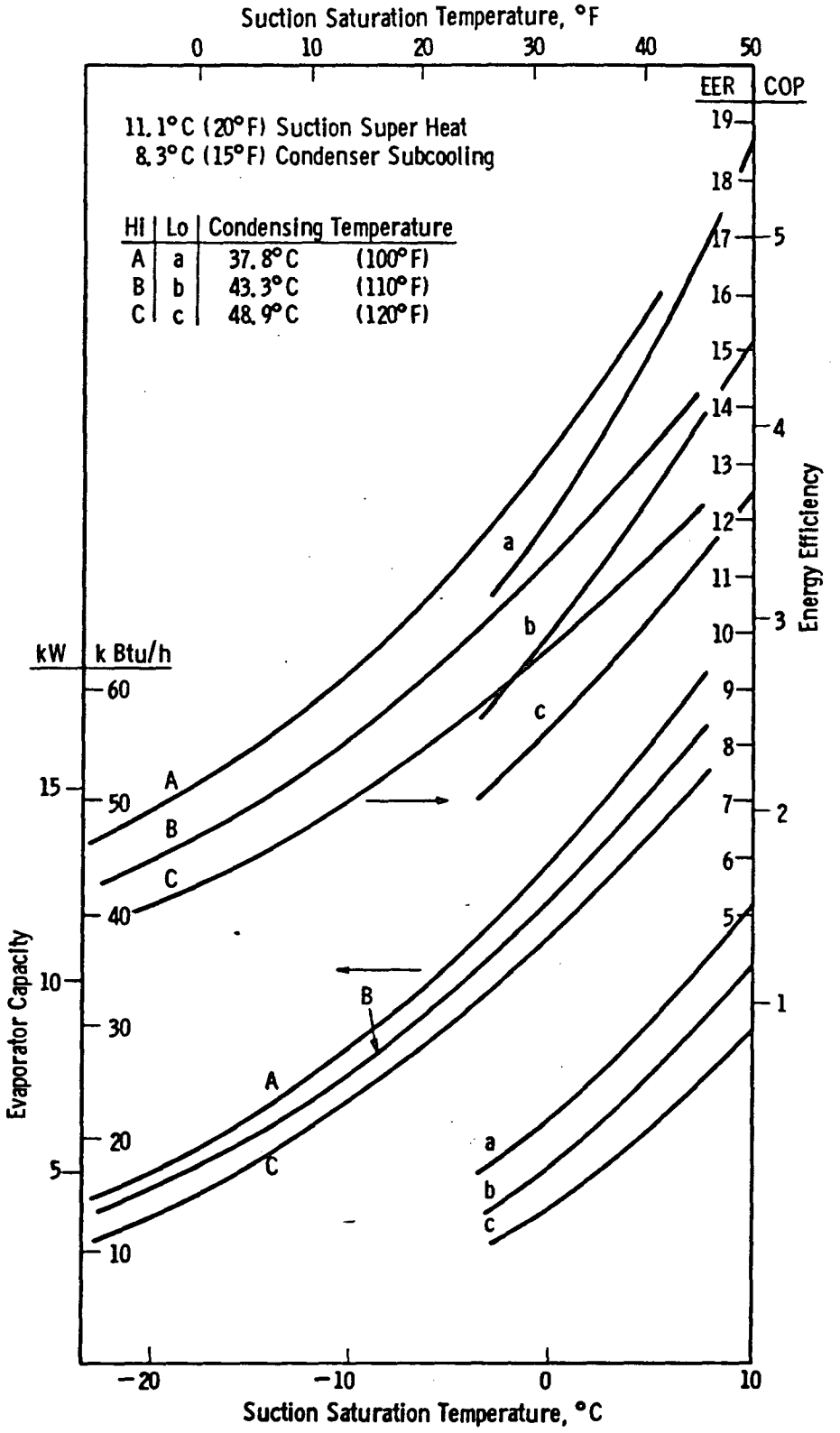

Fig. 3—Prototype dual stroke compressor performance

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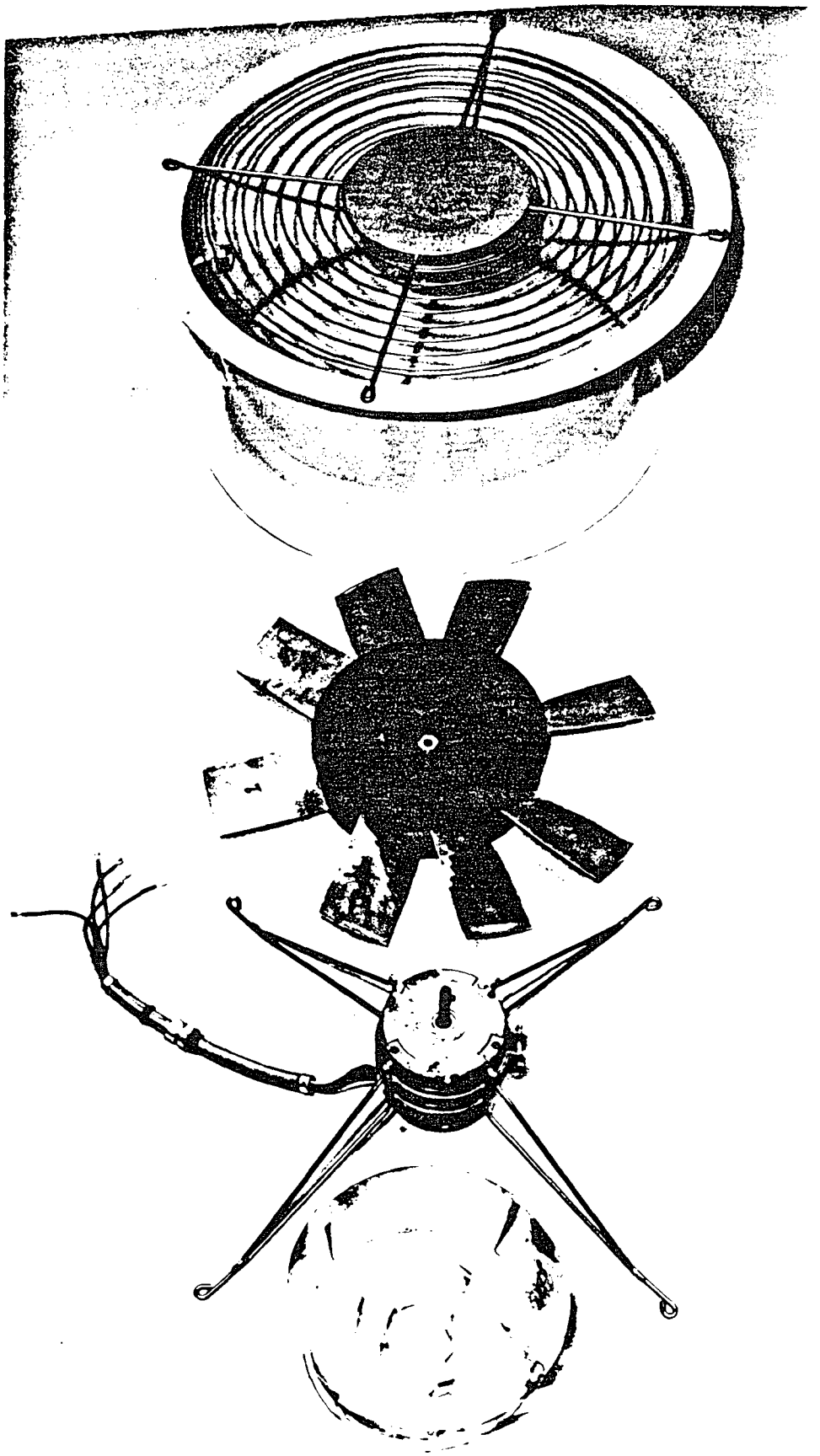

Figure 4 - Outdoor fan preprototype

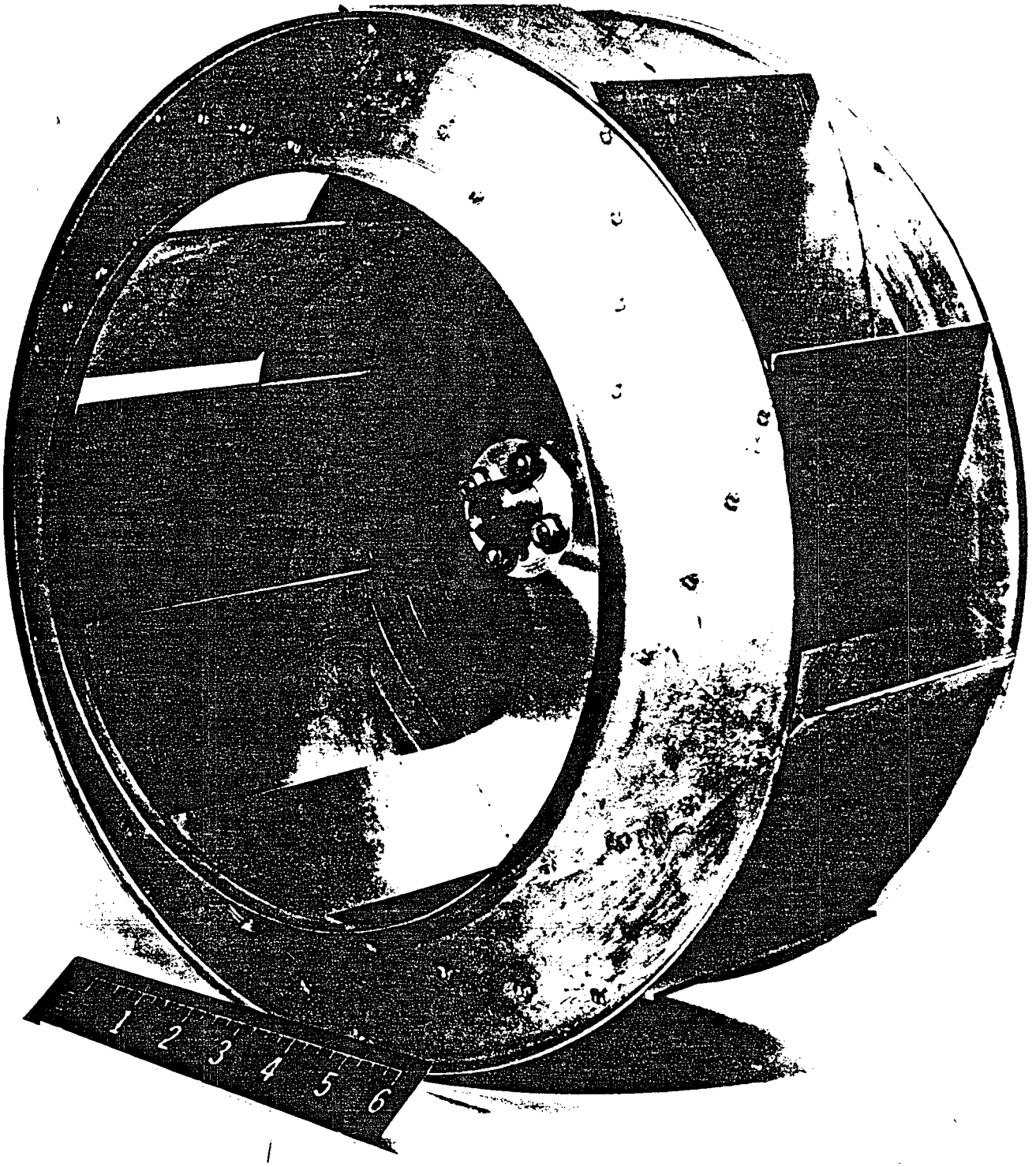

Figure 5 - Indoor blower preprototype

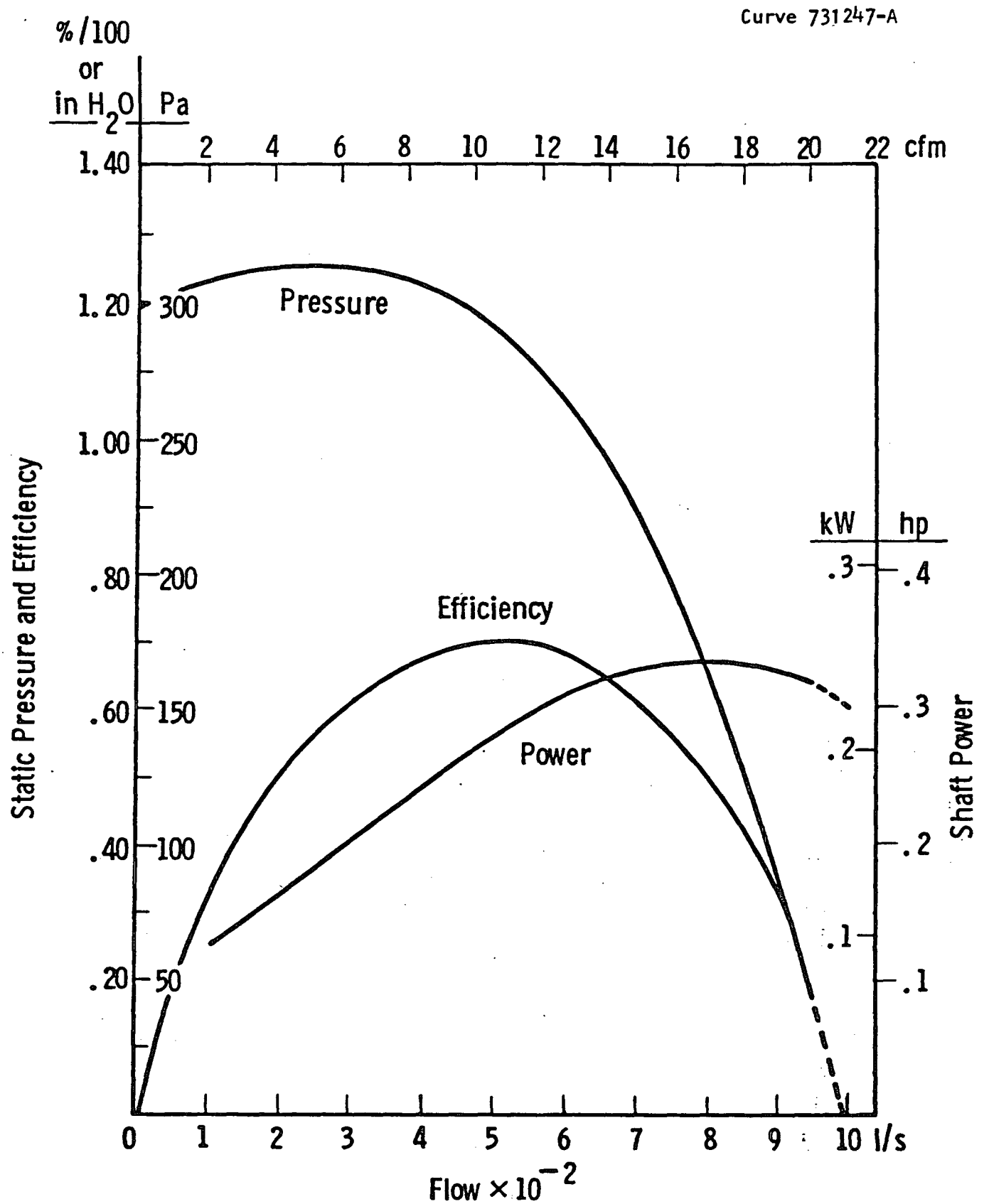

Fig. 6 — Blower performance characteristics

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Curve 731186-B

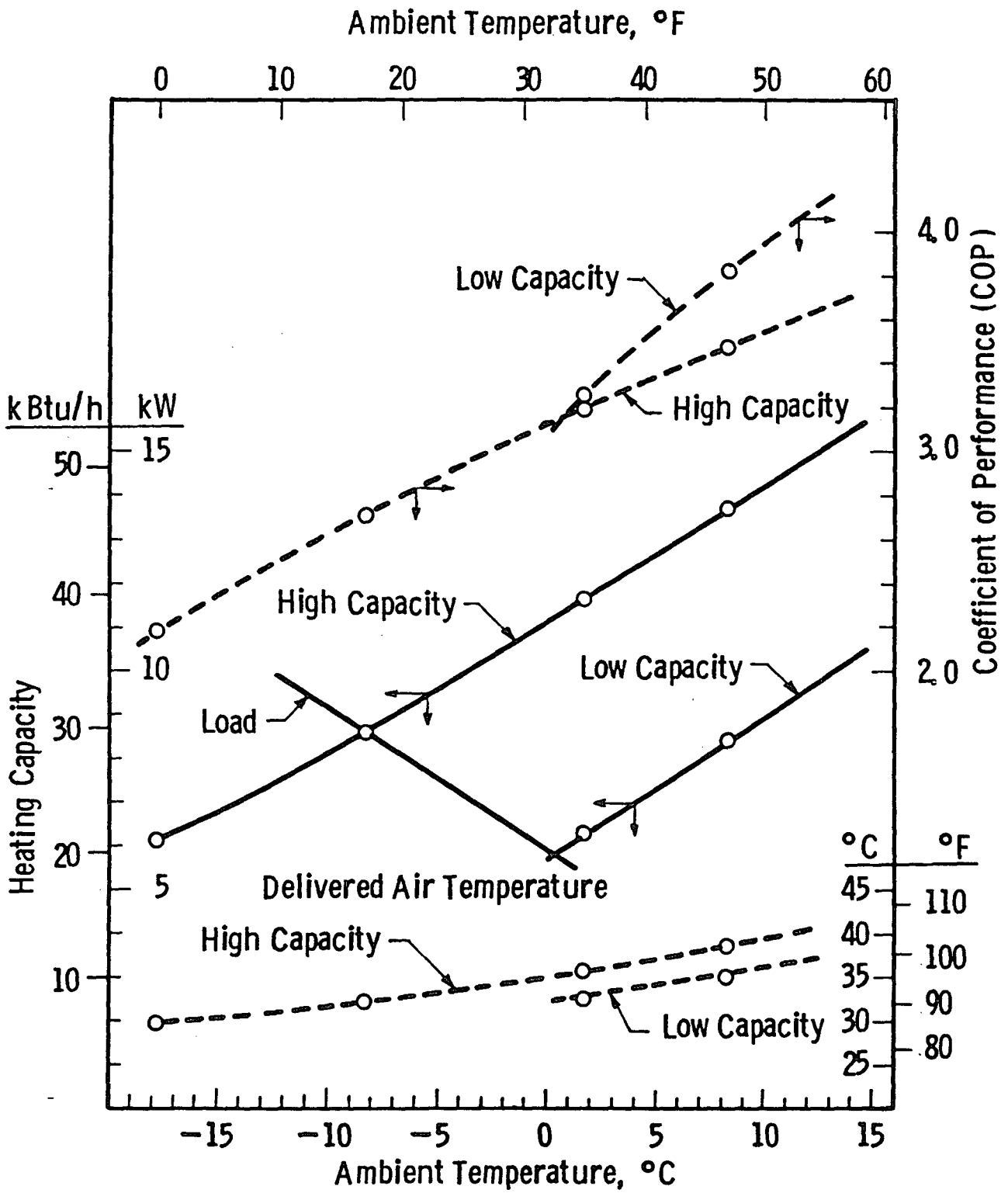

Fig. 7 — Two capacity advanced electric heat pump measured heating performance

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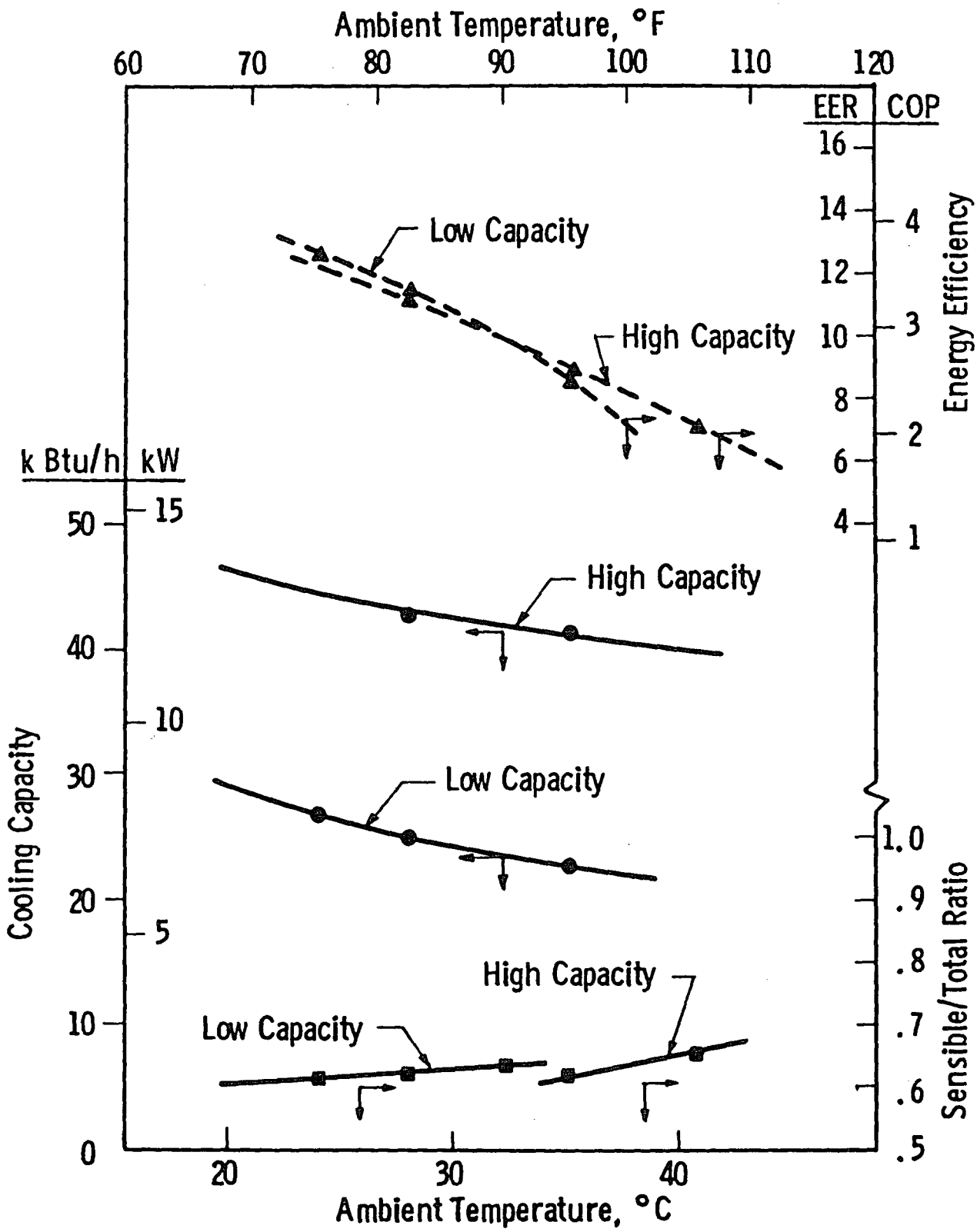

Fig. 8—Two capacity advanced electric heat pump measured cooling performance

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