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A HIGH-EFFICIENCY INDOOR AIR MOVER*

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ABSTRACT

A high-efficiency indoor air mover has been developed for the advanced electric heat pump. Preprototype air mover overall efficiency is approximately 40 percent, twice that of the conventionally applied squirrel-cage blower. The air mover consists of a 411 mm (16.2 inch) diameter single entry blower wheel carried in a volute sheet steel scroll and driven by a 250 W (1/3 horsepower) high-efficiency motor. The blower wheel uses ten backward curved, uniformly spaced, cambered plate blades. As installed in the advanced heat pump the air mover consumes 390 W of electrical power and delivers 662 ℓ/s (1405 scfm) at 1092 rpm into an external static flow resistance of 87 Pa (0.35 inches of water). Although this air mover will cost about twice as much as the conventional squirrel-cage blower and motor of comparable flow performance the incremental premium cost at the retail level can be recovered in less than one year through energy savings assuming 6000 hours of operation per year with electricity at $0.05/kWh.

INTRODUCTION

The duty for a heat pump indoor air handler differs considerably from that of the outdoor fan. The indoor airflow 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 airflow 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 airflow is 743 ℓ/s (1575 scfm) with an air-mover static pressure rise requirement of 212 Pa (0.85 inches of water). Consideration of specific speed[1] clearly shows that a centrifugal type fan is appropriate for this flow and pressure rise with rpm as available from direct connected 4, 6, and 8 pole motors.

In presently available heat pumps the indoor air mover typically uses a squirrel-cage (forwardly curved vane) blower wheel 228 to 254 mm (9 to 10 inches) in diameter and 203 to 305 mm (8 to 12 inches) 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 to 375 W (1/3 to 1/2 horsepower). 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 50 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 static 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 and reduces cost relative to a less flow restricting belt-drive arrangement. 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.

Static efficiencies ranging from 60 to 80 percent have been achieved commonly in industrial practice using centrifugal blowers with backward inclined airfoil-shaped blades.

A comparison of the performance characteristics for the two types of blowers, both scaled for size and speed to meet a common design point, and employing a volute scroll reveals a continuous slope in the pressure rise versus flow curve to the right of the peak efficiency point for the backward curved airfoil blower and a flat pressure-flow relationship for the forward curved blading in the vicinity of the peak efficiency point. This is indicative of a more limited stable operating regime for the forwardly curved vane blower when both are operated near peak efficiency. Note that at the same specific operating point the efficiency of the backward curved airfoil blower mounted in a volute scroll with external motor mounting is roughly twice the value of the forward curved blower similarly mounted but with the motor in the eye of the blower wheel.

After accounting for scale or size difference, there is a significantly higher cost to be associated with this higher efficiency due to critical clearance dimensions. As partial cost compensation however, note that if blower efficiency is improved then motor size, and presumably cost is reduced. There exists also the prospect for an increase in motor efficiency. It is well known that because of geometric effects, the achievable efficiency of small fractional horsepower single phase motors is at most 70 percent. The practical horizon for motor efficiency is about 65 percent, which represents a significant improvement relative to current practice. The prospect for a high-efficiency indoor air mover for residential heat pumps that would roughly halve the required electrical power with economic and commercial viability is therefore very good.

In order to achieve a doubling in overall air-mover efficiency, a high-efficiency single-entry blower wheel design was proposed, with multiple backwardly curved airfoil-shaped blades driven by a high-efficiency motor. The 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 airflow of 743 ℓ/s (1575 scfm) with a static pressure rise of 212 Pa (0.85 inches of water). Dimensionally, the air mover was constrained to fit within a cabinet 572 mm (22.5 inches) 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 inches of water).

PRELIMINARY REPORT

Preliminary calculations indicated that a 330 mm (13 inch) diameter backward curved airfoil blower wheel would be approximately correct for the initial specification at 4 pole motor speed. We had on hand an untested 343 mm (13.5 inch) wheel from some previous work.[2] It was deemed prudent to test this wheel for baseline data and scale a specific design if that proved necessary. The tubular shroud rather than the more conventional volute shroud was selected for this first test in order to determine if the potential advantages of compactness, lower cost and more uniform exit flow were available without penalizing performance. Test data for a similar blower wheel in a volute shroud were available from previous work.[2]

The baseline blower is defined as follows:

Wheel diameter	343 mm (13.5 inches)
Tip width	133 mm (5.25 inches)
Blades	10 airfoil
Casing	590.6 mm (23.25 inch) ID cylindrical
Inlet cone	30 degrees
Inlet vanes	none
Outlet vanes	4
Speed	1725 rpm

For the tubular shroud configuration, it is necessary to recover the swirl velocity at the periphery of the wheel in order to reduce losses and power requirement. This may be accomplished by properly designed outlet guide vanes.

For testing purposes the blower wheel was supported on an arbor and shrouded by a tubular duct. An inlet cone was mounted to the face of the duct and nested into the wheel. The rear flange of the duct was attached to a nozzle plenum chamber. All test procedures and data reduction methods are in accordance with AMCA Standards, Bulletin 210-74.[3]

The characteristic performance curve as developed by test at 1725 rpm is shown in Figure 1. A parabolic system curve is shown passing through the specified design point of 1575 cfm and 0.85 inches of water static pressure rise. The baseline blower exceeds the required pressure rise and flow at 1725 rpm.

The effect of casing-to-wheel diameter ratio was investigated. The baseline wheel was maintained and the casing size reduced in increments. There is a marked reduction in performance for smaller casing diameters. At a casing diameter of 559 mm (22 inches) the static efficiency is reduced by about 3 percentage points and at 457 mm (18 inches) by 13 points.

The effect of outlet guide vanes on both performance and efficiency was also investigated. Four circular arc cambered outlet guide vanes yield a 5 percentage point improvement in static efficiency over a clear duct while the use of straight radial vanes yields a reduction in static efficiency of about 3 percentage points. Outlet guide vanes in conjunction with the variation of casing sizes was not investigated. There is a good probability that casing size could be reduced beyond that required to meet geometric constraints without severe efficiency penalty by use of outlet guide vanes for swirl velocity recovery.

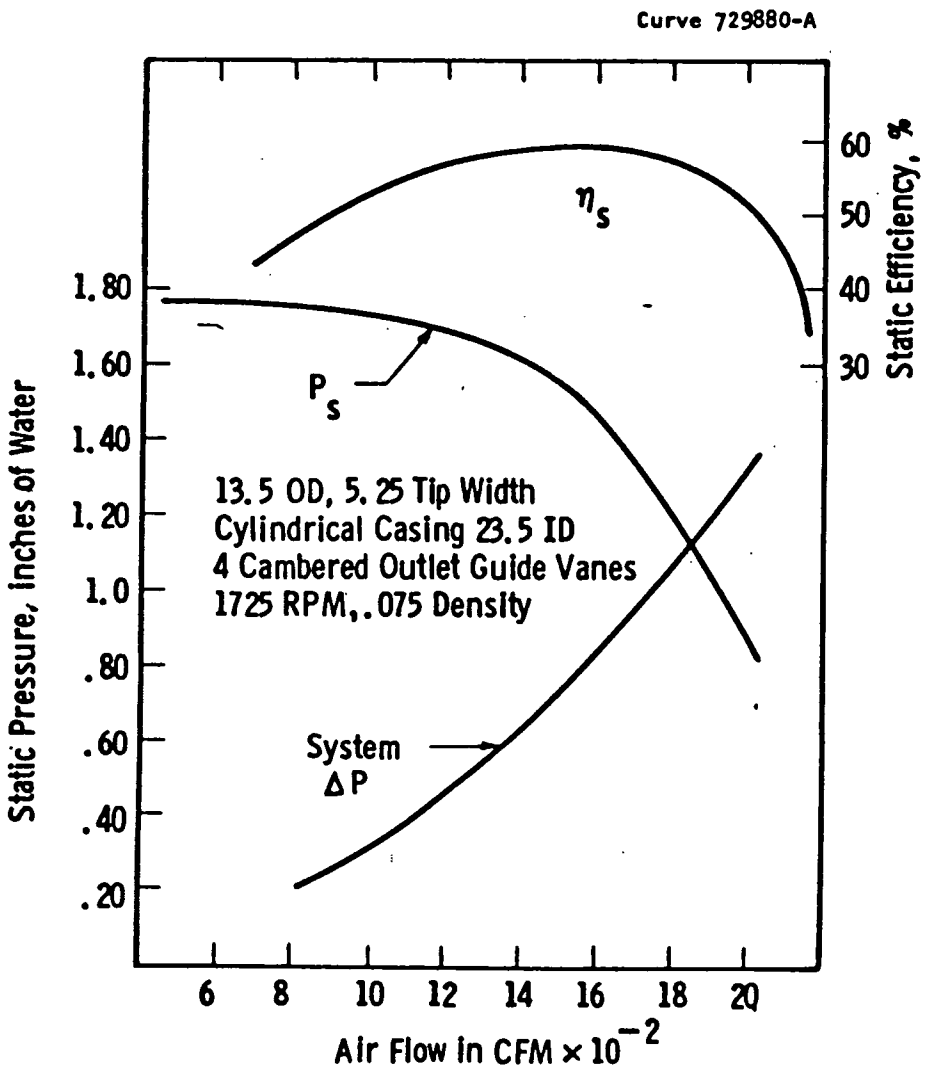


Figure 1 - Baseline Airfoil Blower Performance Characteristics (dimensions in inches)

Inlet guide vanes were investigated as a means for flow control only during initial set-up rather than as a means for continuous flow modulation. Three uncambered airfoil-shaped stub vanes were used for this purpose and manually set at 15 degree intervals. Tests were performed on the baseline blower without outlet guide vanes.

The turndown in pressure-flow performance is shown in Figure 2 with a system curve passing through the specified design point. The corresponding map of power requirements for the array of vane settings and flow rates is also shown. The dashed line represents the power requirements along the parabolic system curve.

INITIAL BLOWER DESIGN

Even though a 343 mm (13.5 inch) tip diameter blower was tested, the final specifications for the 12.3 kW (3.5 ton) heat pump indoor air mover may be different from the preliminary specifications and a variation in both fan size and/or speed must be expected.

Using the test results described, the blower design can be downsized to meet the specified performance using the fan scaling laws[4]:

$$\Delta P_s \propto D^2 N^2 \rho_a \tag{1}$$

$$Q \propto D^3 N \tag{2}$$

$$W_s \propto D^5 N^3 \rho_a \tag{3}$$

where D = wheel diameter

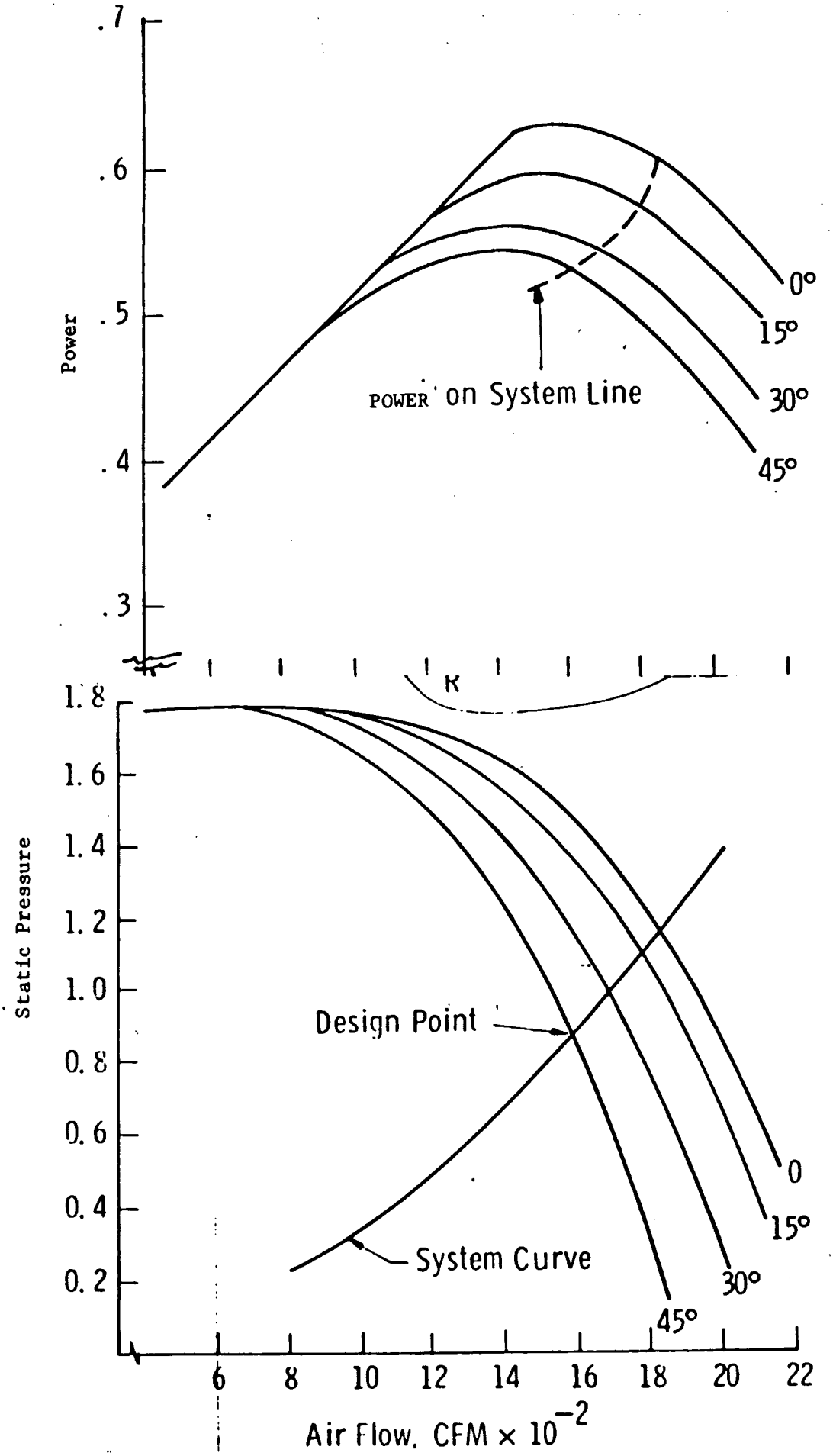
N = rotational speed

ρ_a = air density

ΔP_s = static pressure rise

Q = volume flow rate

W_s = fan shaft power



Curve 740914-A

Figure 2 Effect of Inlet Vane Angle on Performance

The effect of Reynolds number on blower efficiency may be approximated by

$$\frac{\eta_m}{1 - \eta_m} \frac{1 - \eta_p}{\eta_p} = \frac{D_m}{D_p} \left(\frac{\eta_m}{\eta_p}\right)^{1/2} \tag{4}$$

where η = efficiency

m = model

p = prototype

The predicted performance of a 324 mm (12.75 inch) tip diameter, 1650 rpm wheel geometrically similar to the 343 (13.5 inch) wheel tested is shown by Figure 3. The design goal can be met with 300 W (0.40 horsepower) shaft power. Note that the baseline blower was tested at 1725 rpm because it was assumed that this was appropriate for high-efficiency operation at maximum flow. Multispeed blower operation was later determined to be necessary and 1650 rpm is judged appropriate for a multispeed four-pole blower motor.

TECHNICAL ANALYSIS

A convenient method of presenting blower characteristics for analysis purposes is in a nondimensional format[4]:

$$\psi = \Delta P/D^2N^2 \quad \text{pressure coefficient} \tag{5}$$

$$\phi = Q/D^3N \quad \text{flow coefficient} \tag{6}$$

$$\nu = \phi^{1/2}/\psi^{3/4} \tag{7}$$

where in mutually consistent units:

D = wheel diameter

Q = volume flow rate

ΔP = pressure rise

N = wheel speed

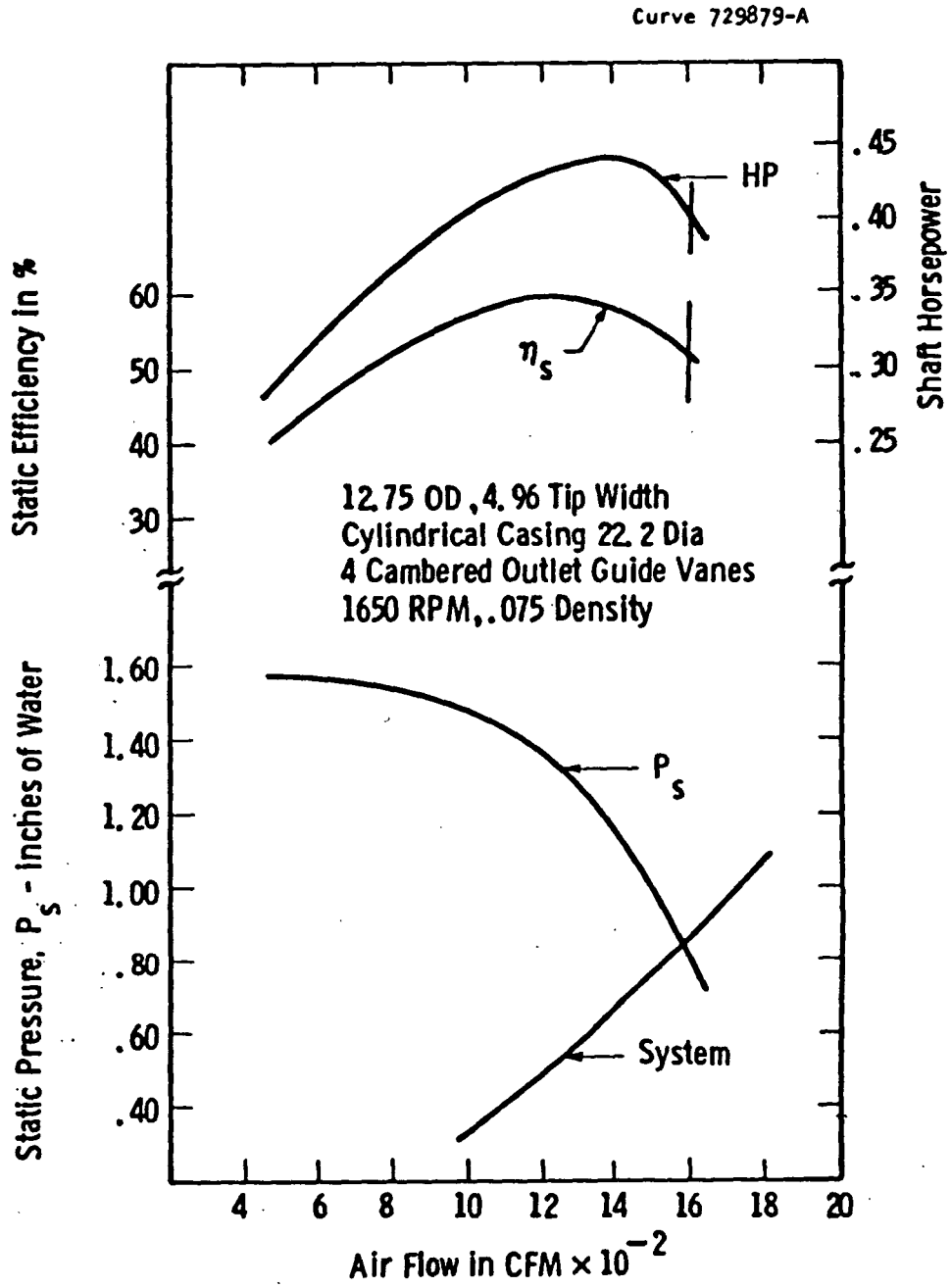


Figure 3 - Centrifugal Blower Scaled to Performance Specifications (dimensions in inches)

Note that the centrifugal blowers considered here are applied in closed ducted systems and under these circumstances the static pressure rise at design point is much greater than the change in dynamic pressure across the air mover. For the purposes of this comparative analysis a good assumption therefore is $\Delta P \cong \Delta P_s$.

The design point for the advanced heat pump was initially specified as 743 ℓ/s (1575 cfm) at 212 Pa (0.85 inches of water) static pressure rise. The dimensionless specific speed at 1650 rpm is $\nu = 1.04$.

Performance data for a backward curved airfoil bladed blower wheel in a volute housing was available from previous work[2]. These data are presented in dimensionless form along with data from our baseline blower mounted in a tubular casing in Figure 4.

Performance and efficiency of the 1650 rpm design in either mounting are nearly identical at the design flow coefficient, $\phi \cong 0.32$. It is necessary to select the blower with volute casing at a much lower specific speed in order to obtain a significant reduction in power or increase in efficiency.

A backward curved airfoil bladed blower matched to a volute housing selected at the peak efficiency of 78 percent (find ϕ at peak efficiency from Figure 4) and meeting the preliminary specifications would have the following characteristics:

$$\nu \cong 0.64$$

$$\phi \cong 0.25$$

$$\psi \cong 0.75$$

where ψ is found from Figure 4 and $\nu = \phi^{1/2}/\psi^{3/4}$. Therefore

$$N = 1019 \text{ rpm}$$

$$D = 415 \text{ mm (16.32 inches)}$$

$$W_s = 200 \text{ W (0.27 hp)}$$

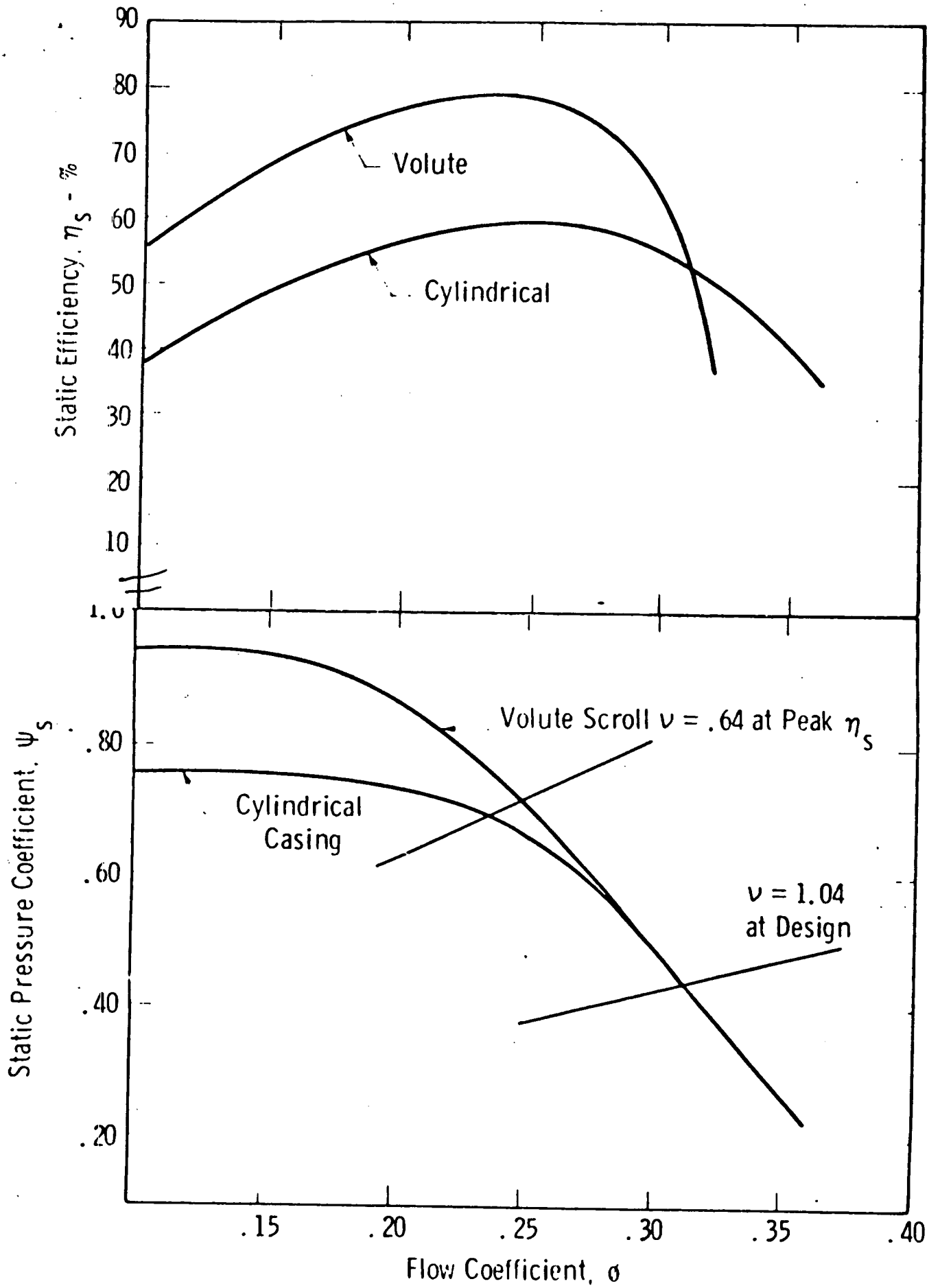


Figure 4 - Effect of Casing on Blower Performance (airfoil-shaped, backward-inclined blades)

CURVE 729873-A

REVISED BLOWER DESIGN

The indoor airflow rate was a parameter of optimization during the course of our heat pump system development effort. The optimum airflow rate depended, therefore, upon the design of the entire system. The preliminary specification was later revised to 660 ℓ/s (1400 cfm) at 212 Pa (0.85 inches of water) static pressure rise. A tip diameter of 400 mm (15.75 inches) was indicated and the back plate and inlet cone for the blower wheel were so ordered.

A blower wheel was fabricated at 400 mm (15.75 inches) diameter with airfoil-shaped blades and tested in a mock-up version of a volute scroll. At this diameter pressure rise was inadequate but the efficiency exceeded that of subsequently tested and scaled blowers with cambered plate blades by 5 to 10 percentage points.

Based upon subsequent measurements taken with a mock-up of the indoor unit, the pressure rise specification was increased to 240 Pa (0.95 inches of water). Modifications to the blower design to accommodate the final specification (660 ℓ/s at 240 Pa [1400 cfm at 0.95 inches of water]) were therefore somewhat constrained. In order to achieve the higher pressure rise, the effective blower diameter was increased to 406 mm (16 inches) by projecting the blades outward beyond the diameter of the back plate and inlet cone.

The performance of the 406 mm (16 inches) blower wheel with cambered plate blades was reported in Ref. 5. Although the performance of this blower is adequate, a version with the blades protruding 6.4 mm (0.25 inches) beyond the back plate was also built and tested.

The performance of the 413 mm (16.2 inch) diameter preprototypic blower wheel mounted in a mock-up version of the volute scroll and tested in a blower test cell is shown in Figure 5. Into the design resistance (660 ℓ/s at 240 Pa [1400 cfm at 0.95 iches of water]) the blower requires 260 W (0.35 hp) of shaft power to deliver 667 ℓ/s (1415 scfm) with a pressure rise of 248 Pa (0.98 inches of water) at 1070 rpm. The static

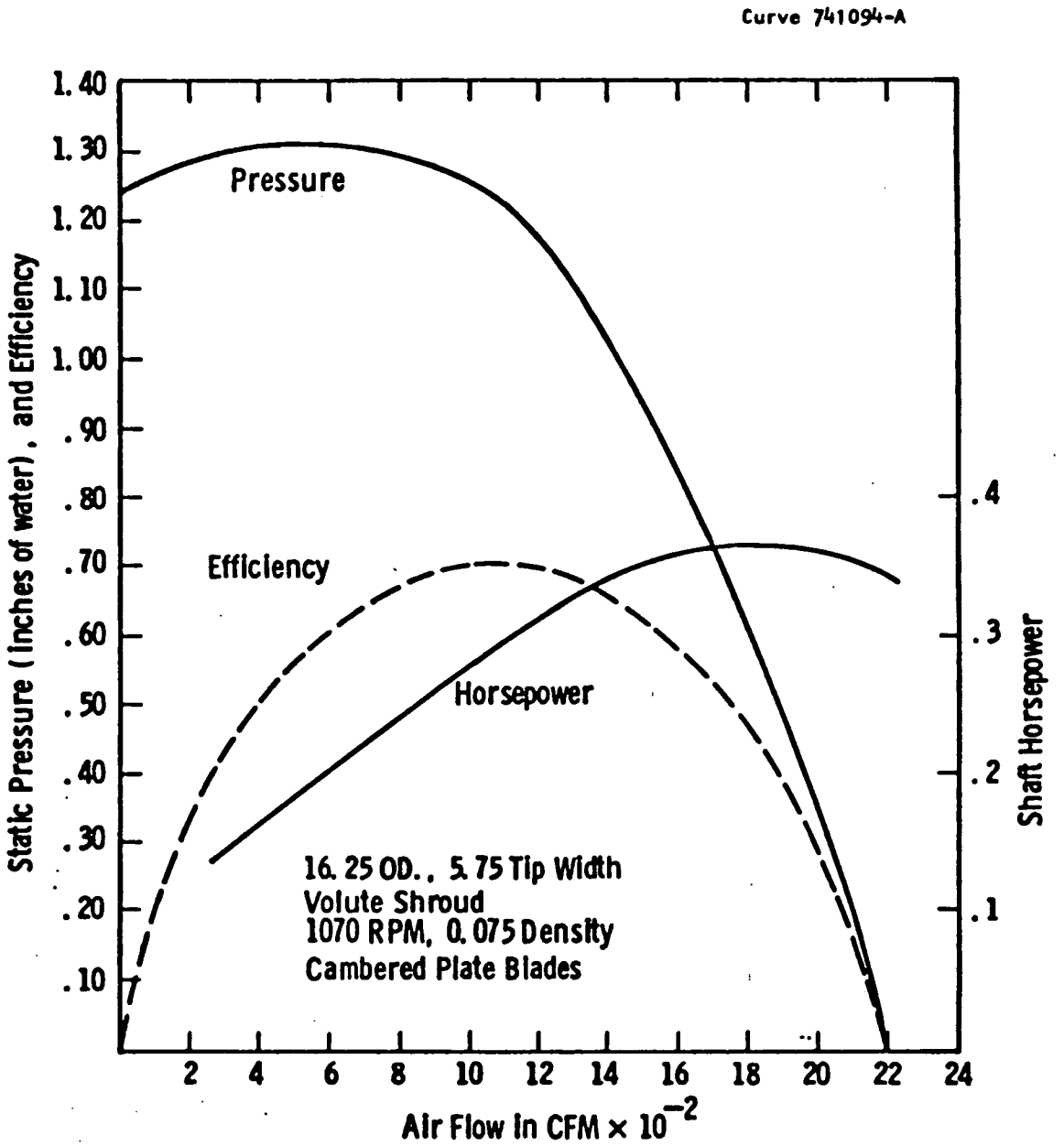


Figure 5 - Performance Characteristics, Preprototype Blower (dimensions in inches)

efficiency is 64 percent. At the peak efficiency of 70 percent, the blower will deliver 500 ℓ/s (1060 scfm) with a static pressure rise of 311 Pa (1.23 inches of water) and requires 216 W (0.29 hp) shaft power. The blower will deliver 743 ℓ/s (1575 scfm) with a static pressure rise of 220 Pa (0.87 inches of water) at a static efficiency of 58 percent and will require 270 W (0.36 hp shaft power.

BLOWER MOTOR

The blower motor needed for the preprototype heat pump evolved from a 300 W (4/10 hp), four-pole motor with "deep" speed reduction capabilities to a 250 W (1/3 hp), six-pole design with somewhat less speed reduction potential than in the case of the four-pole motor. The motor finally selected does not represent a precise optimum. This is partially due to the nonstationary targets for the design, necessitated by the overall system optimization study, and by the fact that the effort required to perform a complete suboptimization of the motor did not seem warranted since the contribution of the blower motor to overall system efficiency and system cost is small. Further, motor optimization was specifically excluded from the scope of our contract effort.

The analytically estimated torque speed curve for this six-pole motor is shown in Figure 6 along with the 406 mm (16 inch) diameter blower characteristics. The estimated efficiency at its design point 224 W (0.3 hp), is 67.0 percent.

SUMMARY

Preliminary specifications for the indoor blower called for a delivered airflow of 743 ℓ/s (1575 cfm) with a pressure rise of 212 Pa (0.85 inches of water). The use of backward curved blade blower technology was known to be necessary if a significant increase in operating efficiency relative to the conventionally applied squirrel-cage blower was to be achieved. Originally, a four-pole blower motor was considered because the motor would be less expensive, and the space occupied by the air mover less because the higher speed blower wheel is smaller. In an additional effort to contain cost and minimize size a tubular blower shroud was used during preliminary testing.

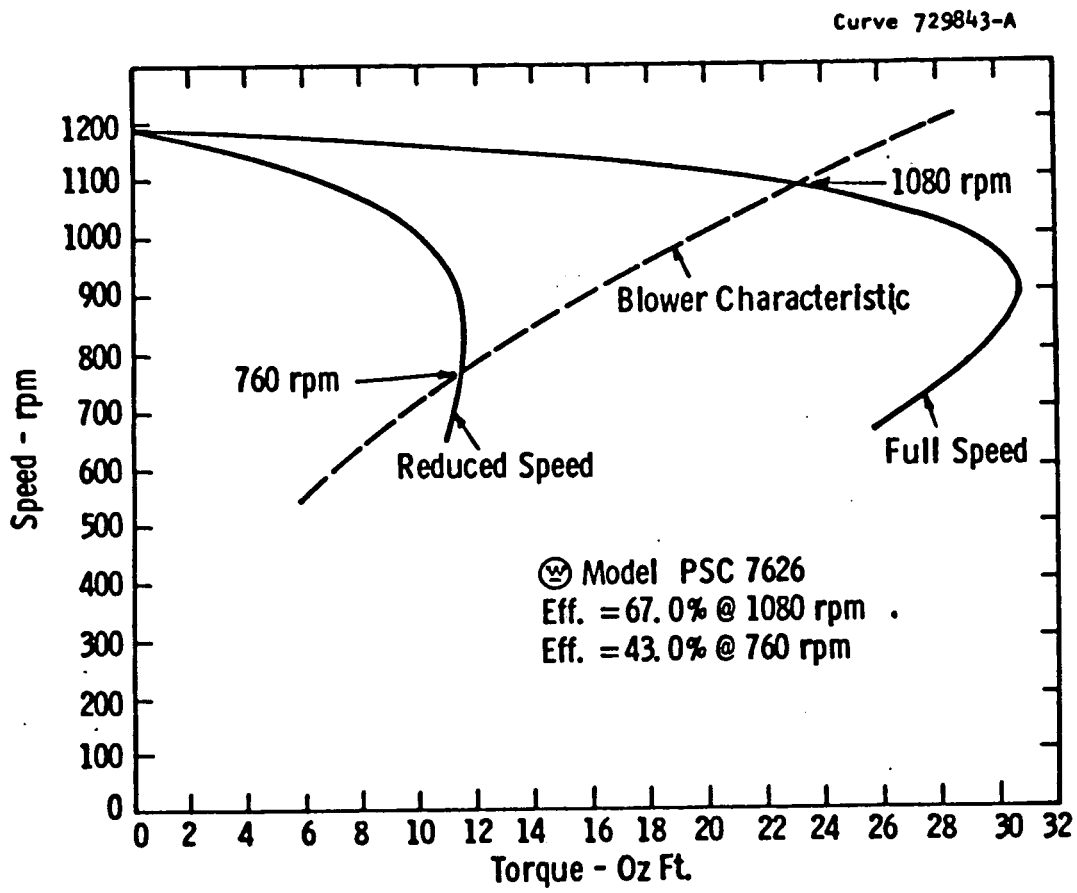


Figure 6 - Blower Motor Speed-Torque Characteristics (analytically estimated)

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Testing of the breadboard blower wheel designed to operate at four-pole speed in a tubular shroud revealed a characteristic howling noise that some found objectionable. Since both lower noise and better blower efficiency were achievable at six-pole motor speed, our design effort was redirected to this area. At this reduced rpm, significant efficiency benefits accompanied adoption of a volute shroud.

Prior to the manufacture of our prototypical units, a system design change required a modification in the blower specifications from 743 ℓ/s (1575 cfm) at 212 Pa (0.85 inches of water) static pressure rise to 660 ℓ/s (1400 cfm) at 240 Pa (0.95 inches of water) pressure rise. Since the backplate and inlet cone for the blower wheel had been procured prior to receipt of the final specification, modification of the design to meet the final design point was constrained slightly. Given the opportunity to redesign the blower again, the blower side plates would be increased slightly in diameter so as to be equal to the blade tip diameter and an increase of approximately 3 to 4 percent in efficiency could be achieved over the unit as built.

Two blower wheel variants were designed, built and tested, the first using airfoil-shaped blades and the second using cambered-plate blades. The blower with airfoil blading was 5 to 8 percent more efficient. Since the use of cambered plate blading allowed us to meet our performance objective while promising lower ultimate cost, cambered plate blading was adopted for the prototype indoor air movers.

The indoor air mover as developed for the preprototype advanced electric heat pump is shown in Figure 7. This assembly slips into an air mover cabinet with external dimensions 572 mm (22.5 inches) wide by 762 mm (30 inches) high by 762 mm (30 inches) deep. It is captured in the cabinet using the flanges at the discharge plane of the scroll. A support bracket (not shown) secures the end of the scroll opposite the discharge to the cabinet.

The aluminum (3003 H14) blower wheel shown in Figure 8 has an effective blade tip diameter of 413 mm (16.2 inches) and uses ten backward curved, uniformly spaced, cambered plate blades 2.29 mm (0.090 inches) thick spanning between a conical backplate 1.60 mm (0.063 inches) thick and an open center conical side plate 1.02 mm (0.040 inches) thick.

The blower wheel is mated to its 12.7 mm (0.5 inch) diameter shaft by a mounting hub. The blower shaft is carried within the scroll by two sets of permanently lubricated, sealed, ball 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 conical inlet cone seals the intake side of the scroll to the conical side plate of the blower wheel. The 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. A complete description of the air mover is available in Ref. 6.

As installed and tested in the 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 inches of water).

Measured electrical power is 390 W at full airflow and 190 W at reduced airflow corresponding to 770 rpm. At high speed, electrical consumption is half that of a conventional blower applied to a comparable capabity heat pump.

The estimated factory cost of this air mover (Figure 7) is $55, approximately twice that of the conventional air mover. Assuming 6000 hours of high-speed operation per year with an electricity cost of $0.05/kWh, this blower can save $117 per year. If we assume that the

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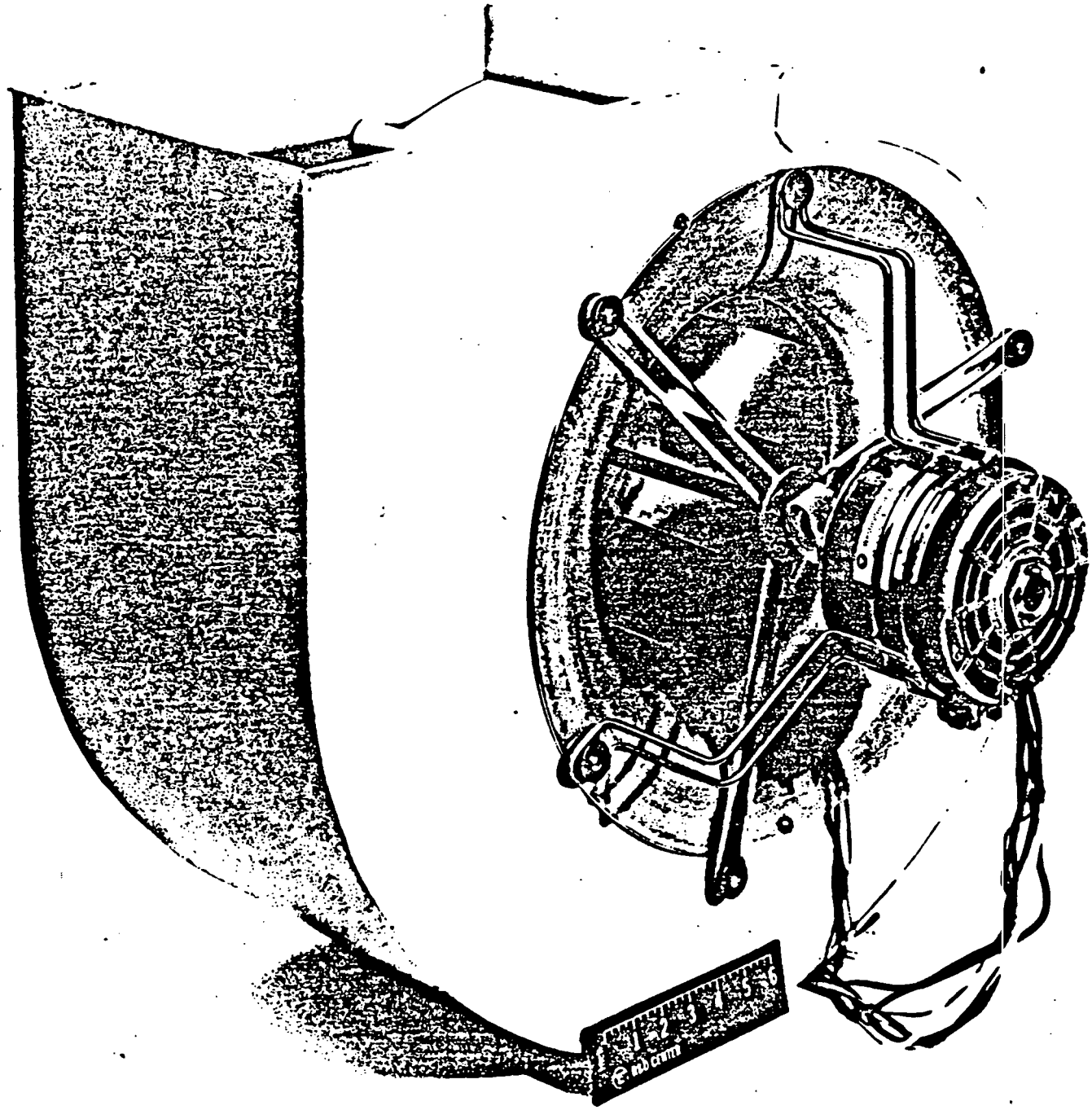

Figure 7 - Indoor Air Mover Assembly

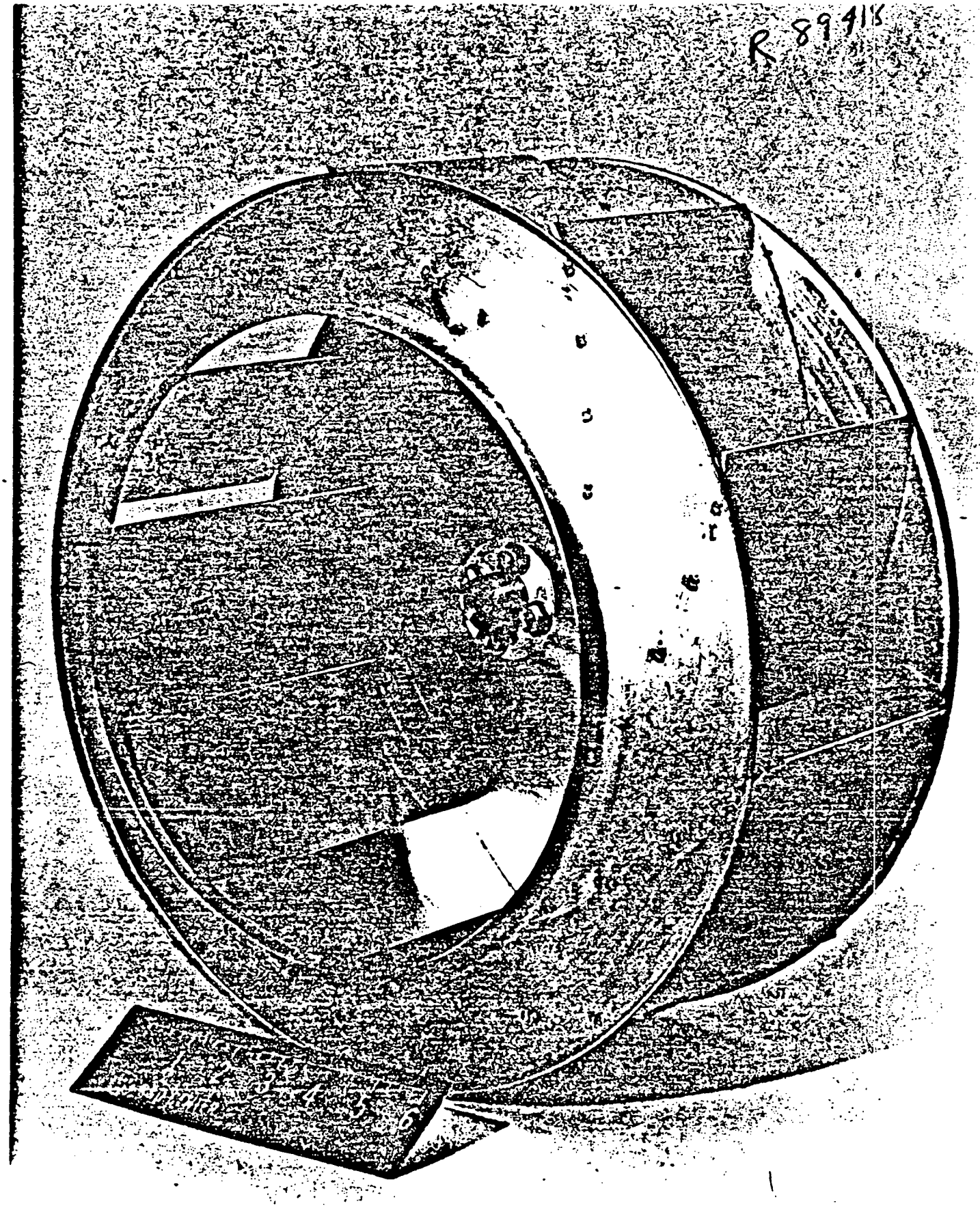

Figure 8 - Blower Wheel

premium incremental cost of this blower at the retail level is three times the factory cost, then this cost is recovered in less than one year through energy savings. Additional energy savings will be realized since a considerable number of hours of operation will be at low speed.

DISCUSSION

The predicted overall air moving efficiency at the estimated operating point (668 ℓ/s [1415 scfm], 248 Pa [0.98 inches of water] pressure rise, 1070 rpm) is 42 percent (64 percent blower static efficiency, 67 percent efficient motor) with a projected electrical consumption of 389 watts. As installed in the preprototype heat pump system, the air mover consumes 390 watts when delivering 663 ℓ/s (1405 scfm) at 1092 rpm. Since the observed flow at 1092 rpm is slightly lower than expected (681 ℓ/s at 258 Pa [1444 cfm at 1.02 inches of water]) it must be concluded that the internal flow resistance of the preprototype heat pump is larger than the design value of 152 Pa (0.6 inches of water). Assuming 65 percent blower static efficiency for the observed flow and a scaled pressure rise of 273 Pa (1.08 inches of water) then the required shaft power is 274 W (0.368 hp) and implies a motor efficiency of 70 percent and of higher than predicted power output in order to reconcile the lower than expected motor slip. Note that the flexible coupling represents a power loss that was not present in the laboratory test cell. The measured electrical consumption of the preprototype air mover is half that of the air mover in a conventional heat pump of the same nominal capacity (12.3 kW [3.5 tons]) working into the same external flow resistance (87 Pa [0.35 inches of water]) at slightly higher flow (743 ℓ/s [1575 scfm]).

Our preprototype indoor air mover with cambered plate blades has achieved its design goals, however, by yielding an air moving static efficiency approximately twice that of current practice with commensurate energy savings.

CONCLUSIONS

1. The achieved static efficiency of the preprototype indoor air mover working into a flow resistance curve passing through the design point and at tested rpm is approximately 64 percent (1070 rpm, 668 ℓ/s (1415 scfm), 248 Pa (0.98 inches of water) pressure rise).

2. The preprototype air mover at 1092 rpm consumes 390 W of electrical power to deliver 663 ℓ/s (1405 scfm) of 16.7°C (62°F) air through the preprototype heat pump indoor unit into an external flow resistance with 87 Pa (0.35 inches of water) static pressure drop.

3. The electrical power saved relative to a replacement air mover of conventional construction is approximately 400 watts or 50 percent.

4. The premium first cost of the air mover should be recoverable from energy savings in less than one year assuming 6000 hours of operation per year, the cost of electricity at $0.05/kWh, and the effective retail cost at three times the factory cost.

5. The use of airfoil-shaped blades instead of cambered plate blades would improve static efficiency by about five percentage points.

6. Replacing the inlet and back plates so as to eliminate the overhung blades would increase static efficiency by about three percentage points.

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