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DESIGN AND EXPERIMENTAL STUDY OF HIGH-SPEED LOW-FLOW-RATE CENTRIFUGAL COMPRESSORS

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ABSTRACT

This paper describes a design and experimental effort to develop small centrifugal compressors for aircraft air cycle cooling systems and small vapor compression refrigeration systems (20-100 tons). Efficiency improvements at 25% are desired over current designs. Although centrifugal compressors possess excellent performance at high flow rates, low-flow-rate compressors do not have acceptable performance when designed using current approaches. The new compressors must be designed to operate at a high rotating speed to retain efficiency. The emergence of the magnetic bearing provides the possibility of developing such compressors that run at speeds several times higher than current dominating speeds.

Several low-flow-rate centrifugal compressors, featured with three-dimensional blades, have been designed, manufactured and tested in this study. An experimental investigation of compressor flow characteristics and efficiency has been conducted to explore a theory for mini-centrifugal compressors. The effects of the overall impeller configuration, number of blades, and the rotational speed on compressor flow curve and efficiency have been studied. Efficiencies as high as 84% were obtained. The experimental results indicate that the current theory can still be used as a guide, but further development for the design of mini-centrifugal compressors is required.

INTRODUCTION

Centrifugal compressors have historically only been used in large commercial systems, such as air conditioning and refrigeration systems with cooling capacities larger than 200 tons. These compressors have shown numerous inherent advantages over screw, scroll, and reciprocating compressors, being smaller, lightweight, and up to 30% more efficient. Having only two or three moving parts results in higher reliability. The centrifugal compressor can operate at variable

speeds for partial-load operation, and can effectively use the latest HFC refrigerants.

However, when the centrifugal compressor size is reduced to accommodate the low-flow-rate (smaller) applications, the flow passage becomes too narrow and the fluid specific losses (loss per unit mass) increase significantly. The compressor efficiency drops sharply as the flow rate is reduced below its lower operating limit. The compressor is typically operating out of the performance map if the flow rate is too small. Casey and Marty (1986) indicated that the overwhelming causes of losses at relatively low flow coefficients are friction and leakage loss. It was these effects that have limited centrifugal compressors to large flow rate applications.

In many applications, such as relatively small vapor compression systems (under 100 tons) and aircraft air-cycle air-conditioning systems, a high power to weight ratio of the centrifugal compressor is desired (Rahman and Scaringe, 1992). Further, the new environmentally safe alternate refrigerants (HFC-134a, HFC-227, HFC-236, HFC-245) will perform better in a cycle without lubricants (Gottschlich *et al.* 1994). These challenges set a goal to develop a lightweight, lubricant free centrifugal compressor with an extended operating range while retaining satisfactory efficiency.

Although the efficiency of centrifugal compressors decreases at low flow rate, Gui *et al.* (1994a) indicated that the efficiency would be recovered as the compressor rotating speed increases. The high-efficiency small compressor is possible if the flow passages can be widened, and an increased rotating speed utilized. Unfortunately, a quantum leap in development is required to demonstrate this technology due to the miniaturization required. High-speed operation (50,000 RPM and above) is critical for successful use. This will require the development and integration of aerodynamically-designed impellers and diffusers and high-speed bearings.

The high rotational speed requirement has led to the use of the newly emerged active magnetic bearings. The first magnetic bearing came into use in 1972 in France. Active magnetic bearings exhibit excellent dynamic characteristics when used in centrifugal compressors (Hustak 1986,1992, Kirk 1987, Pinckney and Keesee 1991). One obvious advantage of the magnetic bearing is the ability to levitate the shaft assembly (shaft, motor rotor, and impeller) before, during, and after the rotor rotation. The shaft is rotating in the air (or working fluid), and nothing but air fills the small gap between the magnetic bearings and the shaft, so that the drag in the magnetic bearing is negligible compared with contact type bearings. The rotor can virtually run at any speed (within the limits of rotor strength and fluid choke), and can always remain centered. This differs with shaft position in journal bearings that vary slightly with rotational speed.

The advantages of high speed, smaller size, and adequate pressure ratio (at small flow rates) are obtainable with such contact-free bearings. The magnetic bearings do not need lubrication, completely eliminating oil contamination and the auxiliary lubrication system. The power needed for magnetic bearings is quite small compared to contact type bearings. So far, over 5,000 compressors and turbopumps have used magnetic bearings. The magnetic bearings have proven to be excellent and reliable at rotational speeds from 10,000 to 800,000 RPM. The total accumulated times for these machines exceed 7 million hours.

Compressor aerodynamic performance can be further improved with three dimensional blades, increasing power transmission, smoothing the air flow, and reducing frictional losses. These efforts extend the lower limit of the centrifugal compressor operational range to a significantly lower flow rate. Centrifugal compressors made with this technology will demonstrate superior performance in terms of pressure ratio, flow range, size, weight, and efficiency. With magnetic bearings and three dimensional design, the compressor efficiency can be raised to an acceptable level.

A program to develop high-speed magnetic-bearing mini-centrifugal compressors is currently being pursued by Mainstream Engineering. As part of this program, several aerodynamically-designed compressors with three dimensional blades have been fabricated and tested for improved performance. During the first stage, the rotating speed was set at 24,000 RPM. An experimental analysis has been conducted on the compressors, using different impeller and blade geometries. Efficiencies as high as 84% have been obtained for these small compressors.

BACKGROUND

For decades, a general rule was that centrifugal compressors were limited to large flow rate applications. The compressor efficiency is closely linked to the compressor flow patterns and operating conditions. The centrifugal compressor relies on high fluid velocity to produce pressure. Such high flow velocities inevitably result in flow losses caused by the frictional and aerodynamic losses, which dominates the compressor efficiency. The specific frictional loss is typically unacceptably high and efficiency very low for small compressors when applied without a significant increase in rotating speed. For large, low pressure ratio air compressors, the efficiency reaches 90% (Casey and Marty 1986), while for large, high pressure ratio air compressor the efficiency is nearer to 80%.

Some small 2-D compressors, which are essentially unchanged since the 1950's, have a lower efficiency, around 60%. Efficiency concerns have often excluded a centrifugal compressor from low flow rate applications. A few small 3-D compressors have been developed, but little, if any, technical information is available in the literature.

The value of the compressor's specific speed, to some extent, implies the compressor geometry and efficiency. It accounts for air flow rate $\dot{m}$, specific compression work H_{ad} for a given pressure ratio, and compressor rotational speed ω , and can be used as a guide in compressor design.

$$n_s = \frac{\omega \sqrt{\dot{m}/\rho_i}}{(H_{ad})^{3/4}} \quad (1)$$

Balje (1981), basing his analysis on a group of large compressors, indicated that centrifugal compressors can achieve high efficiency when the specific speeds fall between 0.62 and 1.08 and the specific diameters are at their optimal values. The compressor efficiency drops sharply as the specific speed falls below 0.62. It may be seen from Eq.(1) that n_s reduces with the flow rate, but is proportional to the compressor rotating speed. It is possible to design a low-flow-rate centrifugal compressor for smaller applications if the rotational speed can be increased significantly (by a factor of two, three, or four). Variation of the specific speed and efficiency, as a function of the air flow rate and rotating speed, has been analyzed before by Gui *et al.* (1994a), using the theory obtained for large, low speed (around 10,000 RPM) centrifugal compressors. The results showed that, at a rotational speed of 24,000 RPM, the efficiency increases monotonically with the flow rate up to 1 kg/s of air (at a constant pressure ratio). At a moderate pressure ratio, the point of maximum efficiency shifts in the direction of high rotating speed as the flow rate decreases. Thus for a small flow rate compressor, the rotating speed must increase to retain efficiency. While it was believed that efficiency could indeed be improved with increased rotational speed, it was questionable whether the results from the existing theory would still be valid at the changed operating conditions. It was necessary to conduct an experimental study to verify the applicability of the existing theory.

DESIGN AND FABRICATION

Having established the potential of high rotational speed, the impeller blades must be designed aerodynamically for high performance. Due to the complexity of the three dimensional turbulent flow and the complex compression process, the blade configuration on many current (large) compressors is not optimized in terms of aerodynamics and fluid mechanics. Many blade designs are in fact optimized in terms of manufacturing, casting, and other considerations. The compressor design must consider the flow passage width, length, tip clearance, blade inlet angle, blade curvature, and other mechanical concerns such as machining feasibility and thermal expansion occurring during compressor operation. Proper blade geometry is critical to compressor performance.

Design approaches, including computer-aided-design and aerodynamic analysis, have been established. The air through the

compressor will experience property changes and encounter acceleration, diffusion, friction, turbulent loss, and flow separation. A verified Navier-Stokes computer code, which takes into account the effects of viscosity, compressibility, turbulence, and friction, was used to analyze the fluid velocity, pressure, and temperature fields.

Small centrifugal compressors were designed for high rotating speed, but as a first step a conservative 24,000 RPM rotational speed has been utilized. In order to expedite the fabrication and test cycle (thereby allowing a greater number of tests to be performed), compressors were designed smaller and vaneless diffusers were used. The design flow rates are therefore even lower than that required in new applications, such as 20-100 ton refrigeration compressors. The motivation for this approach is that if the fundamental information can be obtained on these very small compressors, then this information, combined with the classical theory derived from large compressors, can provide guidance for compressors of any size between them. An unshrouded impeller configuration was selected for the compressors developed. This type of impeller can minimize the friction loss between the impeller and its mating cover, and can be more easily and quickly machined.

There were many conflicts in determining the ideal compressor design parameters for small compressors. For example, a small impeller inlet diameter is required for an appropriate inlet velocity, but this would make the ratio of the inlet-to-outer diameters extremely small, well off the recommended value (for large compressors). If the blade angles on the shroud and hub sides are designed aerodynamically, they may not be physically practical. The absolute and relative Mach numbers at the inlet are very small due to relatively small impeller sizes.

Three dimensional blades were used for the impeller. The inlet of the impeller is perpendicular to its axis rotation. Part of the impeller blade near the inlet serves as an inducer with air entering the impeller axially.

Gui *et al.* (1994b) have developed a set of dimensionless equations for preliminary determination of design parameters such as impeller diameter and Mach number. One such equation, Eq.(2), is for Mach number determination at the impeller inlet. This equation allows direct calculation of the inlet Mach number with accuracy and almost no iteration. No static properties (initially unknown) of the fluid at the inlet are required.

$$M_1 \left(1 + \frac{k-1}{2} M_1^2 \right)^{\frac{k+1}{2(k-1)}} = \frac{4\dot{m}}{\pi (D_{1s}^2 - D_{1h}^2) B_1 p_0} \sqrt{\frac{R_a T_0}{k}} \quad (2)$$

The fluid properties and blade angles at the inlet can then be determined from their known stagnation values. The relative Mach number can then be obtained with the blade angles and absolute Mach number.

The impeller tip (outer) diameter is basically determined by the work it has to transmit. The specific energy imparted to the fluid is usually evaluated with the Euler equation

$$E = U_2 V_{\theta 2} - U_1 V_{\theta 1} \quad (3)$$

The specific energy E , after deducting losses in the diffuser, should be sufficient to provide the adiabatic energy H_{ad} for the required compression. The adiabatic energy for air at moderate pressure rise can be obtained with Eq.(4).

$$H_{ad} = \frac{k}{k-1} R_a T_{01} \left[\left(\frac{p_{03}}{p_{01}} \right)^{\frac{k-1}{k}} - 1 \right] \quad (4)$$

There are several features in the compressor geometry which require discussion. The ratio of inlet shroud-to-hub diameter is large, resulting in a large variation of blade angle from shroud to hub. The Mach number is small, on the order of 0.1. Blade thickness takes a significant percentage of the total inlet area. The ratio of the impeller outer diameter to the inlet shroud diameter is much larger than those of large compressors. Tip clearance to blade height ratios could be 2-4 times as large as those found in large compressors, leading to leakage loss concerns. The blade angle β_1 decreases greatly from the shroud to the hub to reflect the radius change.

The data for impeller blade configuration were produced according to basic parameters determined with the methods mentioned previously. Special consideration was given to the effects of friction and flow separation. For the first group of impellers, the flow passage was made short by increasing the blade angle in the middle and then reducing to the desired value at the exit. The data were then imported and converted into a three dimensional model using AutoCAD. Modification for a tapered blade tip was made for most impellers to improve the inlet flow. The blade configuration consisted of a set of ruled surfaces suitable for fast machining. CNC (computer numerical control) codes were developed thereafter and directly sent to a five-axis milling machine for manufacturing. Seven compressors of the first group were fabricated for the test. This group of compressors had the same meridional cross-section and essentially the same blade configuration, but had some differences. These differences included the number of blades, whether or not splitter blades were used, the lengths of the splitter blades if used, and whether or not slight blade exit modifications were used. Other groups of compressors are currently under design. The major parameters of the seven compressors in the first group are listed below.

Inlet shroud diameter	d_{1s}	43.64 mm
Inlet hub diameter	d_{1h}	14.00 mm
Inlet (shroud) angle	β_{1s}	30.5
Exit diameter	d_2	63.00 mm
Exit width	b_2	5.23 mm
Blade exit angle	β_2	64.0

Differences between compressors in the first group were made to investigate the effects on performance of the blade number, the blade inlet tip configuration, the length of the splitter blades, and the blade exit shape. The impeller blades of compressors M006 and M007 were trimmed to tapered edges from 1.02 mm to 0.40 mm at the exit in a length of 2.5 mm in the radial direction. This trimming was designed to reduce the impeller jet/wake effect. The differences between compressors in the first group are listed in Table 1.

Table 1. The differences of the compressors in the first group

Parameters	Symbol	M001	M002	M003	M004	M005	M006	M007
Total No. of blades	z	10	8	8	12	12	12	12
No. of splitter blades	z_s	-	-	-	6	6	6	6
Blade thickness	$t(\text{mm})$	1.02	1.02	.86	1.02	1.02	1.02	1.02
Blade inlet tip		flat	flat	round	round	round	round	round
Blade cut length (shroud)	$\Delta z_s(\text{mm})$	-	-	-	3.18	5.72	3.18	5.72
Blade cut length (hub)	$\Delta z_h(\text{mm})$	-	-	-	6.35	8.89	6.35	8.89
Blade exit modification		-	-	-	-	-	Yes	Yes

* The blade cut length refers to the projection length on the x axis of the cut part of the splitter blades.

EXPERIMENTAL TEST STAND

Figure 2 shows a schematic of the centrifugal compressor test stand equipped with instrumentation for overall compressor performance evaluation. Compressors are driven by a 400 Hz electrical motor controlled by a variable-frequency drive. Air is drawn in at ambient conditions by the compressor through a 52.5 mm (ID) tube with a radial opening. A 40.5 mm (ID) exhaust tube directs the flow back to ambient. A valve controls the compressor back pressure and flow rate. Both inlet and outlet fluid temperature and pressure can be measured. The compressor pressure rise is measured using a high resolution Setra differential pressure transducer, with a range of $0-32.47 \pm 0.045$ kPa ($0-5.000$ psid ± 0.007 psid). The compressor inlet volumetric flow rate is measured by a Diamond II Annubar Flow Meter, in tandem with a Setra differential pressure

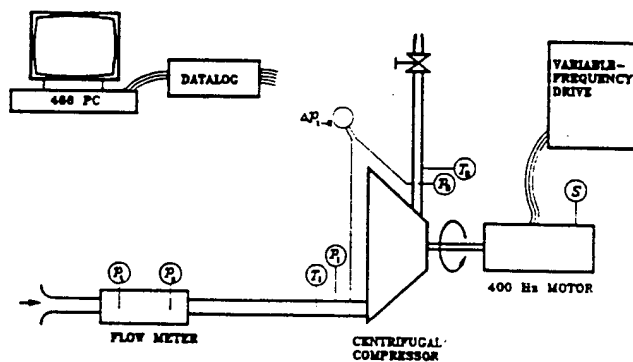


Figure 1 Air Compressor Testing Stand Schematic

transducer. This combination gives a range of $0-104 \pm 0.0038 \text{ m}^3/\text{s}$. The inlet and outlet temperatures are measured with two calibrated 100Ω platinum RTDs with a deviation of less than $\pm 0.015^\circ \text{C}$ from each other in the range of interest. The inlet pressure is measured by a Setra pressure transducer with a range of $0-162.3 \pm 0.2$ kPa ($0-25.00 \pm 0.03$ psia). Data is collected by a Fluke Hydra Data Logger and stored on a PC.

PERFORMANCE

Compressor performance includes the compression capability, described as the flow curve of pressure ratio versus flow rate, and compressor efficiency.

In pressure ratio calculation, the outlet pressure was evaluated by adding the pressure difference between the inlet and outlet to the inlet pressure to avoid error accumulation. The differential pressure transducer had an accuracy and resolution five times higher than the inlet and outlet pressure transducers. This arrangement ensures a high accuracy. The static pressure ratio of outlet to inlet was taken as the pressure ratio.

$$PR = \frac{p_1 + \Delta p_{1-3}}{p_1} = 1 + \frac{\Delta p_{1-3}}{p_1} \quad (5)$$

The overall efficiency was determined thermodynamically. Overall efficiency includes the compressor (impeller and diffuser) aerodynamic efficiency as well as mechanical efficiency. The latter accounts for mechanical losses due to friction between mechanical parts. Since the mechanical loss of the compressor is very small compared with the aerodynamic loss, the aerodynamic efficiency could be approximately considered as a measure of the compressor overall efficiency. In calculations, adiabatic reversible (isentropic) compression processes were used as a reference to determine the compressor overall efficiency. The energy consumed in a real compression process was the summation of the fluid enthalpy increase and the dynamic energy gain from the compressor inlet to the outlet. By neglecting the slight variation of the specific heat at constant pressure, the isentropic efficiency or adiabatic efficiency of a compression process of air at a low pressure rise can be obtained as follows:

$$\eta_c = \frac{(h_{3s} - h_1) + E_{d1-3}}{(h_3 - h_1) + E_{d1-3}} = \frac{C_p T_1 \left(\left(\frac{p_3}{p_1} \right)^{\frac{k-1}{k}} - 1 \right) + \frac{1}{2} (V_3^2 - V_1^2)}{C_p T \left(\left(\frac{T_3}{T_1} \right) - 1 \right) + \frac{1}{2} (V_3^2 - V_1^2)} \quad (6)$$

Another direct method to evaluate compressor efficiency is to replace the denominator in Eq.6 with the compressor input shaft power. A new test stand capable of measure the shaft input power is being developed for this approach.

EXPERIMENTAL RESULTS AND ANALYSES

The performance charts were developed from experimental data. Both pressure rise and efficiency versus volumetric flow rate (at 24,000 RPM) are plotted in Figure 2. (Since the initial interest of this experiment was focused on the compression capability, only steady state pressure readings were recorded for compressors M001 and M002.) There are many characteristics common among these compressors, but these data also show differences from large industrial compressors. The range of flow rate corresponding to a range of high efficiency is between 0.015 to 0.035 m³/s. No choking was observed, although the compressors were running at a high rotating speed (twice that of existing industrial compressors).

The flow rates continue to increase as the back pressure is reduced, or, described in another way, the pressure rises decreases gradually as the flow rate increases. This is due to a low Mach number effect. This soft characteristic is different from that of large compressors where the pressure versus flow rate curve is quite flat (Kenny, 1984) in the designed operation range, dropping suddenly as the flow rate reaches the choking limit. Surges were observed as the pressures rose beyond their peak limits. Efficiency points for flow rates below 0.015 m³/s were not plotted in the figure because the temperature readings, at low flow rate, were affected by flow leakage.

There are slight differences among the curves of the seven compressors. The 12-blade compressors have a higher compression capacity than the 10-blade and 8-blade compressors throughout the entire flow rate range. Among the 12-blade compressors, the compressor with the longest splitter blades has the highest compression capability. From the curves of compressors M001 and M002 (both have flat blade tips), the 10-blade compressor (M001) has a higher compression capability than the 8-blade compressor (M002). The curves of compressors M002 (flat blade tip) and M003 (tapered blade tip) show the effect of the blade tip configuration. Significant differences in compression capability can be seen between these two curves.

All the efficiency curves have the same tendency. Peak efficiencies, obtained at a flow rate around 0.027 m³/s, are between 79% and 84%. The efficiency drops in both directions of the flow rate. When the flow rate is reduced, there is insufficient dynamic momentum to push against the back (leaking) flow through the clearance between the impeller and its shroud (cover). It was observed in these tests that the temperature at the compressor inlet was affected by the back flow when the flow rate was reduced significantly. When the flow rate increases, this phenomenon disappears, however, the large flow losses due to a high flow velocity and a high Reynolds number result in a low efficiency as well. Furthermore, as the flow rate increases in the compressor, the tangential velocity of air flowing out of the impeller drops,

leading to a smaller loss-free pressure gain, a gain through centrifugal effect (Cumpsty 1989). At a moderate flow rate, the pressure rise obtained from the loss-free centrifugal effect takes a higher percentage, so the compressor has a low frictional loss, a low separation loss, and a moderate leaking loss, and therefore exhibits a higher efficiency. As the flow rate increases, the turbulence becomes severe and frictional loss increases.

The efficiency curves show large differences at low and moderate flow rates (<0.035 m³/s). Compressors M003, M004, and M006 have a higher efficiency than the others. It is interesting that the 8-blade compressor with elliptical tips (M003) has a high efficiency, even though its compression capability is less than that of 12-blade compressors. However, if the compression capability is considered, the 12-blade compressors (M004 and M006) have a better performance. A 10-blade compressor with elliptical tips may have better performance. But it is too early to make this conclusion because 10-blade compressors with round tips have not yet been tested. Further investigation will be conducted.

Among the four 12-blade compressor, compressor M004 still has the highest efficiency. From the difference between the two curves of compressors M004 and M005, it appears that the longer splitter blades are preferred.

By analyzing the curve of the compressor with a modified blade exit (M006), it is worth noting that this compressor

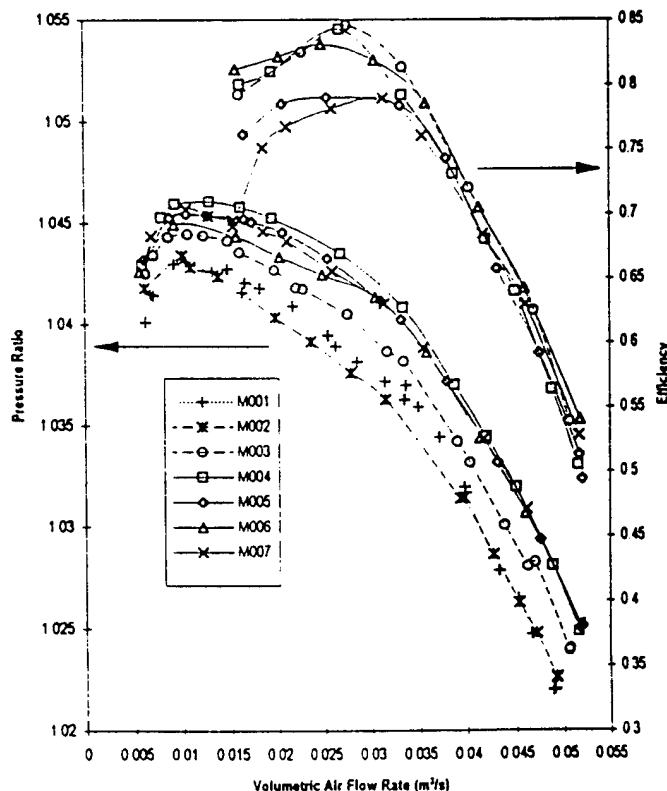


Figure 2 The Pressure ratio and Efficiencies of Compressors (at 24,000 RPM)

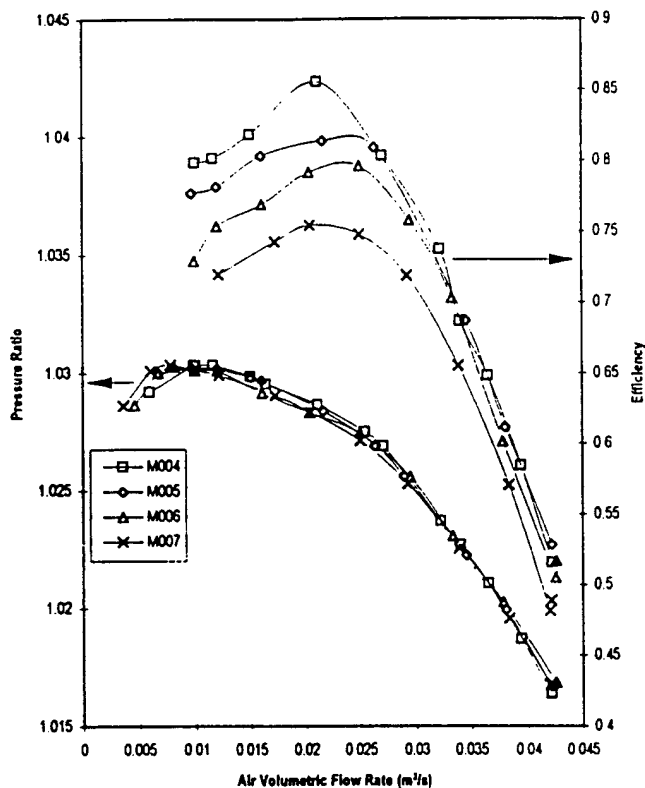


Figure 3 The Pressure ratio and Efficiencies of Compressors (at 20,000 RPM)

has an efficiency 1.5% higher than the unmodified one (M004) at a high flow rate, but has a peak efficiency 1% lower.

The same kind of curves have been plotted in Figure 3, for tests running at 20,000 RPM. The differences of pressure ratio among the compressors is reduced to an insignificant value; however, the differences in efficiency were retained. The peak values are between 78% and 85%.

Figure 4 demonstrates the performance of compressor M004 and M005 at rotating speeds of 20,000 RPM and 24,000 RPM. There is a significant difference in the pressure ratio. The peak pressure rise increased about 45% as the compressor rotating speed increased by 20%. This result is consistent with the analytical assertion that the pressure rise is proportional to the square of the rotating speed. The efficiency curves maintained the same shape as the rotating speed changes, but shifted to the high flow rate side as the rotating speed increased.

A curve which provides more insight for the compressor design and evaluation is a semi-dimensionless curve, plotting efficiency versus specific speed (Figure 5). Figure 5 shows that the specific speed corresponding to the peak efficiency is about 0.9. This result coincides with the prediction obtained from the curve of the maximum efficiency versus the specific speed from Balje (1981). However, further analysis shows that the specific diameter corresponding to a specific speed of 0.9 differs from the "optimum" for the maximum efficiency according to existing criterion. This phenomena means that the existing

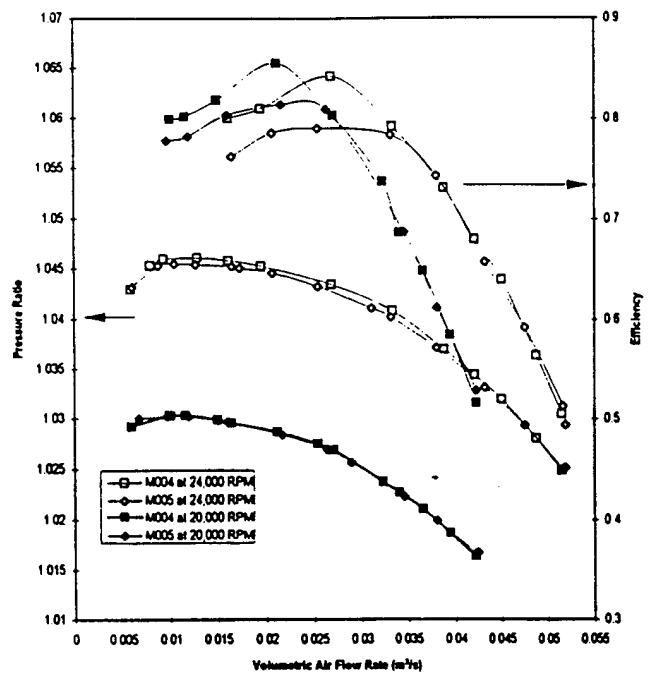


Figure 4 The Performance of Compressors M004 and M005

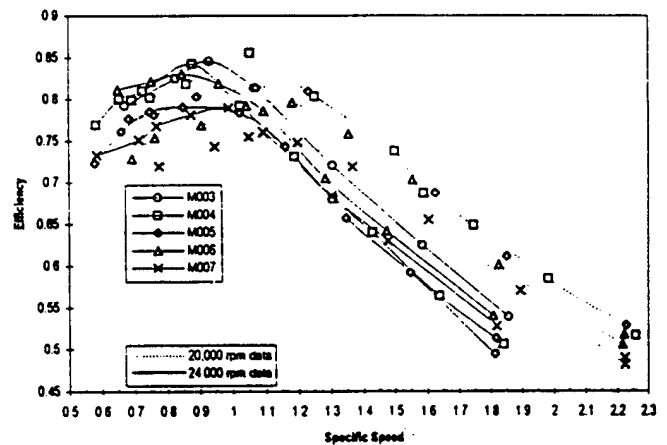


Figure 5 Effect of Specific Speed n_s on Efficiency η

design rules, to some extent, are still a usable reference. But it also indicates that there are differences between small compressor designs and large compressor designs. The empirical method is based on the results of large compressors, which may not be entirely applicable to the design of a small compressor. For example, the pair of n_s and d_s of a point at low flow rate (about $0.015 \text{ m}^3/\text{s}$) in Figure 2, would be close to the optimum pair specified by existing empirical results, 0.62 and 4.5 respectively, but the corresponding efficiency is low, as shown in Figure 2. The previously mentioned geometric features of a small compressor must be considered in the small compressor design. A new diagram of optimum specific

diameter and specific speed should be developed to reflect these features.

ERROR ANALYSIS

Stagnation effect and frictional heating on the RTD shield would influence the measured temperature, causing the RTD sensor temperature to be slightly higher than that of the flowing air. On the other hand, the heat loss through the RTD shield to the outlet pipe causes the RTD sensor temperature to be lower than the air temperature. This error has been determined to be less than 0.05 °C, and the total systematic error in the reported efficiency values is estimated less than $\pm 2\%$. The error in the flow rate measurement would not affect the calculated efficiency, but affected the specific speed, causing an error of $\pm 6\%$. An accurate flow meter will be used for a new test stand.

CONCLUSION

It is feasible to design low-flow-rate centrifugal compressors for aerospace applications and small vapor compression refrigeration systems. The inherent advantages of centrifugal compressors can be retained for small compressors when the compressor rotating speed can be increased by a factor of two or more. Efficiencies as high as 84% have been reached for a small compressor with impeller diameter of only 63.0 mm. Construction of high efficiency small, 100 mm (4 inch)-200 mm (8 inch), 3-D impellers appears practical. The experimental results are, to some extent, consistent with the similarity equation of efficiency and specific speed derived by Balje (1981). Existing rules and methods can still be a fairly good reference for small compressor design, but special features must be considered in design. New criterion for optimum design of small compressor should be developed with the state of the art technology.

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NOMENCLATURE

B	blockage factor $\beta_1 = 1 - \frac{2z\bar{t}_1}{\pi(D_s + D_h)\sin\beta_1}$
c_p	specific heat at constant pressure [kJ kg ⁻¹ K ⁻¹]
d	diameter [m]
d_s	specific diameter, $(D_2(H_{ad}))^{1/4} / (\dot{m}\rho_1)^{1/2}$
h	enthalpy [kJ kg ⁻¹]
H_{ad}	adiabatic head [kJ kg ⁻¹]
k	ratio of specific heat, c_p/c_v
M	Mach number, U/a
$\dot{m}$	mass flow rate [kg s ⁻¹]
n_s	specific speed, $\omega\sqrt{\dot{m}\rho_1} / (H_{ad})^{3/4}$
p	pressure [Pa]
R_a	air constant [kJ kg ⁻¹ K ⁻¹]
T	temperature [°C or K]
U	blade velocity [m s ⁻¹]
V	fluid absolute velocity [m s ⁻¹]
β	blade angle

η	efficiency
ρ	density [kg m ⁻³]
ω	impeller angular velocity [s ⁻¹]
Subscript	
1,2	impeller inlet, outlet
3	compressor discharge
h	hub
o	stagnation state
θ	tangential

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