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Feasibility Study and Hardware Design of A Pivoting-Tip Rotary-Vane Compressor and Expander Applied to a Solar-Driven Heat Pump

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FEASIBILITY STUDY AND HARDWARE DESIGN OF A PIVOTING-TIP ROTARY-VANE COMPRESSOR AND EXPANDER APPLIED TO A SOLAR-DRIVEN HEAT PUMP

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INTRODUCTION

Battelle's Columbus Laboratories has developed an advanced design concept for a rotary-vane compressor and expander. Since its conception in 1965, a number of in-house and sponsored programs have been conducted to explore the feasibility of the design and application in a Rankine-cycle-powered heat-pump system and to design experimental-prototype hardware. Past sponsors have included the American Gas Association (1972), the National Science Foundation (1974), and the Solar Research and Development Branch of the U.S. Energy Research and Development Administration (1976). Experimentation has been limited to measurements of vane-tip friction in a non-stroking laboratory apparatus; however, in the ERDA-sponsored program, layout-design drawings of experimental hardware were prepared.

The program with ERDA/DOE is currently inactive, but a Phase 3 program is being considered. Battelle holds a background patent on the basic concept and has demonstrated technical feasibility in both theoretical and experimental studies. Battelle has prepared layout designs; the next phase of the development will probably be conducted in conjunction with a manufacturer.

This paper provides an overview of the progress in development of the solar-driven heat-pump concept with an emphasis on compressor and expander design features.

THE SOLAR-DRIVEN HEAT-PUMP CONCEPT

Figure 1 shows components of an advanced concept for a Rankine-cycle-powered vapor-compression heat pump. Principal features of the concept are:

- Pivoting-tip rotary-vane compressor and expander directly coupled to a motor/generator
- Single refrigerant and common condenser in the power and refrigeration loops
- Auxiliary energy supplied principally by shaft power
- Electrical power-generation capability.

The rotary-vane type of compressor and expander provides a number of technical and marketing advantages which are:

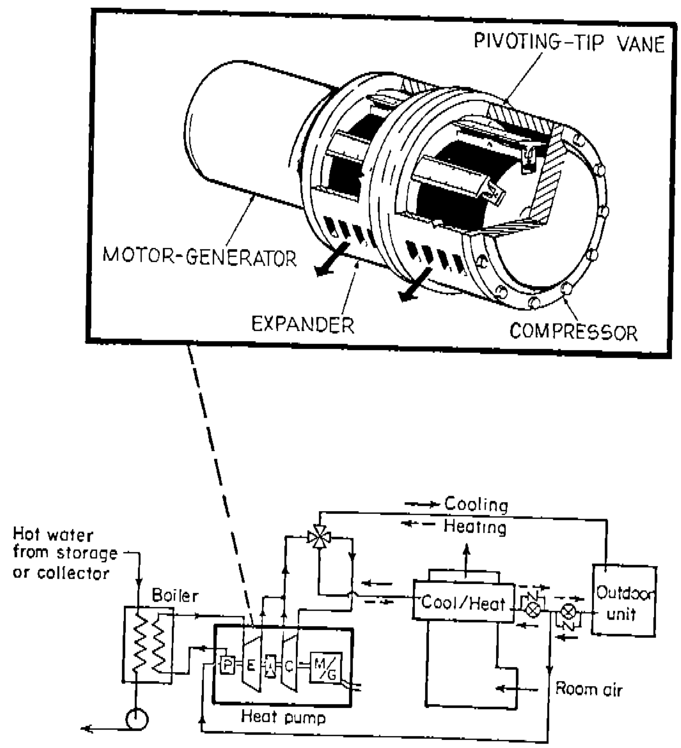


FIGURE 1. COMPONENTS OF THE BATTELLE SOLAR HEAT PUMP

1. Compact construction
2. Quiet, vibration-free operation
3. High volumetric efficiency
4. Low shaft speeds
5. Simplicity
6. Improved commonality of parts between compressor and expander
7. Potential for high efficiency and low cost in manufacturing.

Pivoting-pad hydrodynamic gas bearings are used on the tips of the vanes to reduce vane-tip friction, a major source of energy loss in rotary-vane machines.

If sealing at either end of the rotor can be accomplished as intended without oil lubrication, there is good potential for eliminating oil in the power and refrigeration loops. Eliminating oil from the working-fluid circuits provides these additional advantages:

- Connecting piping can be adequately sized for low flow losses without regard for return of oil to the compressor and expander.

- Thermal decomposition of the working fluid in the boiler will be minimized without an oil catalyst.

- Thermal decomposition of the oil in the boiler will be eliminated.

Rotary-vane machines inherently have high volumetric efficiency because "trapped" volume or clearance volumes for carry-over of discharge gas can be relatively small and flow areas for suction or admission flows are relatively large with porting provided by vane action. With the proper working fluid in the Rankine-cycle-powered heat pump, rotary-vane machines for the compressor and expander can be designed with many common parts. Displacement differences required in the compressor and expander can be provided simply by a difference in axial length of the rotating members. Additionally, since a rotary-vane machine is of the positive-displacement class of machines, cost-effective construction and good performance can be realized with low shaft speeds. As a result, conventional low-friction ball bearings can be used to support the shaft, high-speed shaft seals are eliminated, and auxiliary energy can be supplied as shaft power by a conventional electric motor.

The use of a common working fluid in the power and refrigeration loops simplifies condenser construction and design of change-over controls for switching from cooling to heating operation. In addition, rotating components can be enclosed in a hermetic enclosure and simpler internal seals can be used to separate zones of different pressure.

A direct-coupled electric motor can provide auxiliary shaft power input as well as electric power output. In many applications it is expected that electrical power can be generated during the spring and fall when the heating and cooling demands are low.

SOLAR HEATING AND COOLING PERFORMANCE SIMULATION

The Battelle solar heat pump was designed for operation on solar-heated water available from high-quality flat-plate solar collectors. All of the development work to date has been directed toward the residential application, although the unit can be scaled up to commercial sizes of under 100 tons cooling. Also, the unit can be adapted to higher-temperature evacuated-tube or focusing solar collectors with some design modifications or change in working fluid.

Design conditions for the solar heat pump are:

- 3-ton cooling capacity
- 202 F turbine-inlet temperature
- 108 F condensing temperature
- 50 F evaporator temperature.

A performance simulation of the unit was prepared in order to assess overall performance of the solar system with real weather data in several locations. In this simulation, the heat-exchanger areas and compressor and expander displacements were sized to give a power balance at design conditions with no auxiliary shaft-power input.

The simulation was set up to utilize working-fluid properties in the ASHRAE format[1] so that operation with a variety of fluids could be explored. The final choice of fluids was narrowed to R-11 and R-12. On the basis of thermodynamic properties alone, cycle efficiency with R-11 was up to 10 percent higher than with R-12; however, the size of the compressor using R-11 would be over 5 times larger than one designed with R-12. Compression and expansion characteristics of the fluid must be considered in order to avoid compression or expansion into the two-phase region. With R-12 the inlet to the expander must be superheated somewhat to avoid expansion into the two-phase region, but "wet" expansion or compression is easily avoided with R-11. Operating pressures are strongly affected by the working-fluid selection. For example, condenser pressure with R-11 is very low and boiler pressures with R-22 or R-114 are very high. Boiler design is affected by the working fluid. In this study, fluids with sub-critical boiler pressures were preferred to simplify boiler construction.

From this study, R-12 was selected as the preferred fluid primarily because the required compressor and expander displacements are of reasonable size for good efficiency and cost competitiveness. Cycle efficiency with R-12, although not as good as with R-11, is superior to that with other candidate fluids. Condenser pressure is positive with R-12 and boiler pressure at design condition is reasonable.

Allen[2] has shown that higher efficiencies can be achieved with the use of separate fluids in the power and refrigeration cycles. Advantages cited previously for a single-fluid system would be lost in such a two-fluid system, however,

Cooling coefficient of performance (COP) at design conditions is 0.56. This value compares favorably with equivalent values for absorption cooling equipment and is defined as the ratio of the cooling effect to heat energy and thermal equivalent of electrical energy supplied to the unit (not including the power for the outdoor fan or controls). In the simulation, no pressure losses in piping and heat exchangers were considered because larger piping can be used without regard for oil return. Increased cost of larger piping and ample heat-exchanger areas were felt to be justifiable in respect to the total solar system cost and long-term performance improvement. Compression and expansion efficiencies of 85 percent were used to reflect improvement expected over values for similar components without pivoting tips.

A detailed study of the effect of values selected for design parameters and operating modes was conducted[3]. From this study it was concluded that the design condensing temperature should be 108 F and the evaporator temperature set at 50 F to maximize COP. Conventional systems operate with 125 F condensing and 40 F evaporating temperatures. Additional heat exchange area from that used in conventional systems is required to achieve 108 F condensing temperature with air cooling and the 10 F increase in evaporator temperature will reduce dehumidification capability. However, those departures from accepted design practices are felt to be justifiable in the solar application in order to significantly improve system COP.

In a study of the effect of speed control, it was found that overall system performance can be improved with constant-speed operation provided by auxiliary shaft power input/output by an electric motor/generator. With variable-speed operation, shaft rotation would cease below about 175 F expander-inlet temperature in the cooling mode. Cooling capacity with constant speed operation is nearly independent of expander-inlet temperature. Heating capacity falls off with decreasing expander-inlet temperature but not nearly so much as with variable-speed operation.

Table 1 shows the results of transient thermal performance of the solar heating and cooling system in a residence with 1600 sq ft of floor area located in a representative midwestern climate (Columbus, Ohio) and in a dry southern climate (Albuquerque, New Mexico) simulated with the Battelle SOLTRM* code in response to real hour-by-hour weather data. Results for Columbus are for the entire season; whereas, the results for Albuquerque were for the months of January and August.

TABLE 1. SOLAR SYSTEM PERFORMANCE DATA

	Columbus, Ohio	Albuquerque, New Mexico
Collector area, sq ft	800	450
Percent of energy supplied by solar for:		
Heating	63	87
Cooling	78	90
Hot water	∿90	∿90
Average coefficient of performance for:		
Heating	1.4	1.5
Cooling	0.72	0.60

Negligible electrical power was generated in the spring and fall in the Columbus, Ohio, installation and power generation was not calculated for the Albuquerque installation. The unit was run in the fixed-speed operational mode in both locations. Expander-inlet temperature was fixed at 202 F and 140 F for cooling and heating in Albuquerque; whereas the boiler-water temperature was allowed to float for cooling operation and a minimum value of 140 F was maintained for heating operation in Columbus by electrical-resistance heat input to the boiler.

* Developed from the University of Wisconsin TRNSYS Code.

DESIGN OF EXPERIMENTAL PROTOTYPE COMPONENTS

A design study was conducted to prepare layout drawings of experimental prototype hardware based on available experimental data and theoretical optimization. It is recognized that experimentation in the laboratory will be needed to finalize some design values or material selections affected by thermal expansion or relative motion at interfaces with no liquid lubrication.

Figures 2 and 3 show cross-sectional views of the compressor and expander. The compressor is designed with two vanes and a discharge valve, whereas the expander has six vanes and no discharge valve. Vanes are made of high-quality graphite, tips are made of nickle-plated aluminum and the cam ring and rotor heads (not shown) are faced with carbon graphite.

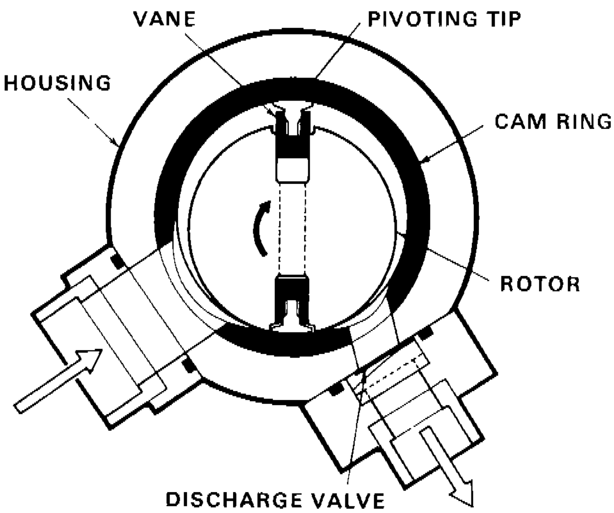


FIGURE 2. PIVOTING-TIP COMPRESSOR

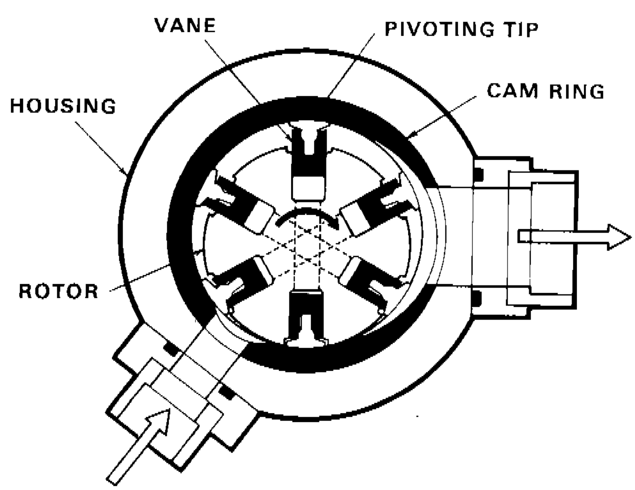


FIGURE 3. PIVOTING-TIP EXPANDER

Figure 4 shows the power train of the heat pump enclosed in a hermetic enclosure. A detailed layout of the motor/generator, expander, and compressor has been prepared showing all shafts, bearings, and seals. The shaft bearings are oil lubricated. Relatively inexpensive spring-loaded Teflon lip seals are used to keep oil from the expansion and compression spaces and to seal between zones of different pressures.

Cool vapor from the evaporator is discharged into the enclosure above the motor to provide cooling for the motor and to pressurize the enclosure at evaporator pressure to reduce the pressure differential on the shaft seals.

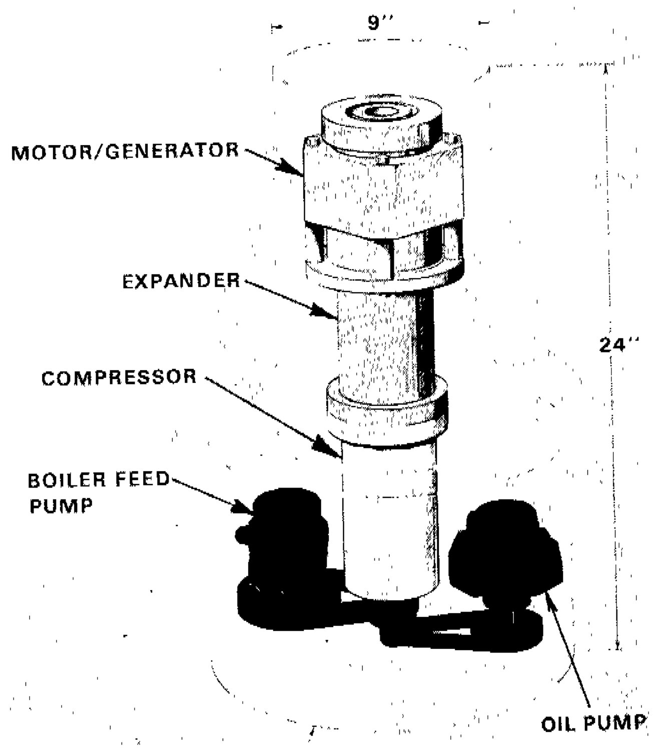


FIGURE 4. POWER TRAIN OF HEAT PUMP

Table 2 shows values of design parameters for the compressor and expander. A value of 0.90 was selected for the ratio of rotor diameter to cam-ring diameter to minimize stroking acceleration forces and side loading on the vanes in the extended position[4]. A cam ring diameter of 2.5 inches was selected based on tip-speed requirements at 3600-rpm shaft speed for good performance of the pivoting-tip gas bearings (from experimental data), rotor length/diameter values for adequate rotor strength, and length/width requirements in the gas bearing pads for minimum end losses. The vane width of 0.35 inches was selected based on a tradeoff analysis of the minimum width needed to support the tip and the maximum width limited by strength of the 2.25-inch-diameter rotor with six vanes.

TABLE 2. VALUES FOR COMPRESSOR AND EXPANDER DESIGN PARAMETERS

	Characteristics Common to Both the Compressor and Expander	
Cam-ring diameter, in.	2.50	
Rotor diameter, in.	2.25	
Vane width, in.	0.35	
Pad width of the pivoting tip, in.	0.44	
	Compressor	Expander
Rotor length, in.	3.08	1.92
Number of vanes	2	6
Intake/admission valving	by vanes	by vanes
Discharge valving	Cantilevered strip	by vanes
Port slots: Number	4	3
Width of end slots	1/2 X*	1/2 X
Width of uninterrupted bearing surface on cam ring between slots	3 X	4 X

* X is width of center slot(s).

Pad width dimension of the pivoting tip (0.44 in.) was selected from an analysis of the minimum required for acceptable performance of the pivoting tip. The rotor lengths were calculated from displacement requirements in the heat pump application assuming appropriate volumetric efficiency values. The number and width of port slots were optimized to minimize end losses in the gas bearings at the pivoting tips and to minimize flow losses in the ports.

Figure 5 shows porting events for the compressor and expander. Suction porting events for the compressor and discharge porting events for the expander are fixed by the number and width of vanes in each unit, since suction on the compressor must be cut off and discharge on the expander must begin when the volume between vanes begins to decrease. Rotor angle at the opening of the discharge port was selected to minimize overcompression to less than 15 percent of the compressor discharge pressure[5]. In the expander, the admission port must be cut off by the vane at a rotor angle called the arc of admission, which is determined from the number of vanes and the required design-point expansion ratio.

Figures 6 to 8 show final dimensions for the pivoting-tip, vane, and rotor slot from the optimization study. Holes and slots are included in the tip and vane to provide hydrostatic lift to the face of the gas bearing from undervane pressure. The vane and tip slots in the rotor are shaped to minimize clearance volumes in which vapor will be carried over from discharge to suction or admission ports. Teflon strip seals on the sides of the vane are loaded by undervane pressure.

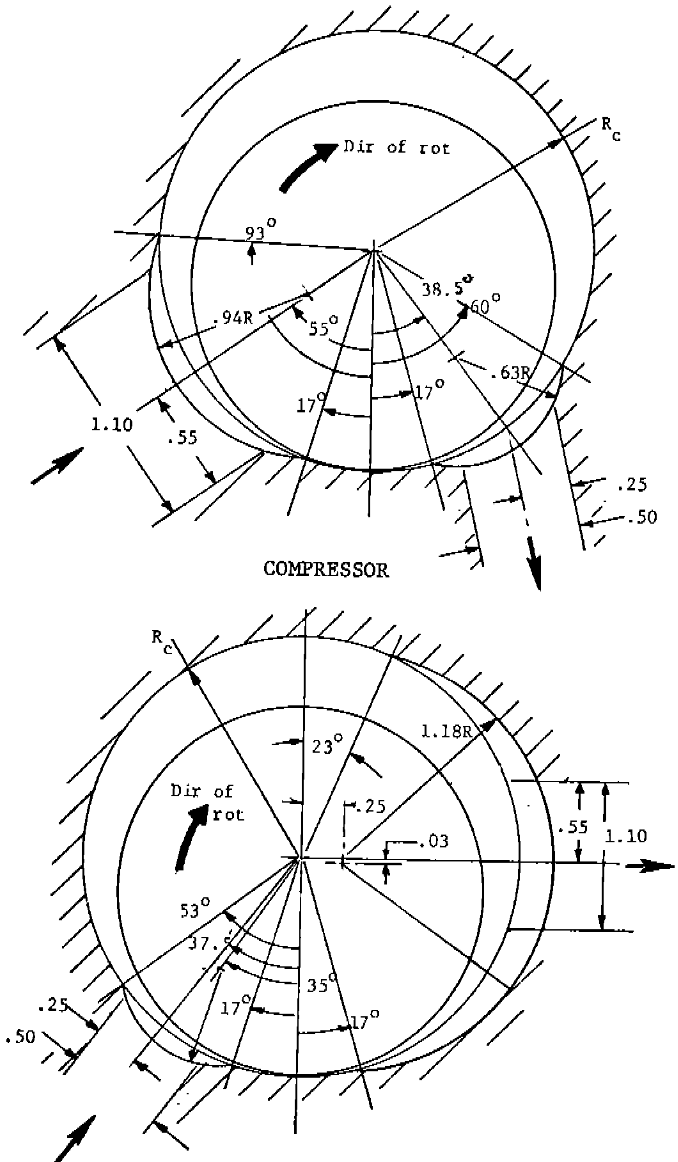


FIGURE 5. PORTING EVENTS AND PORT GEOMETRY

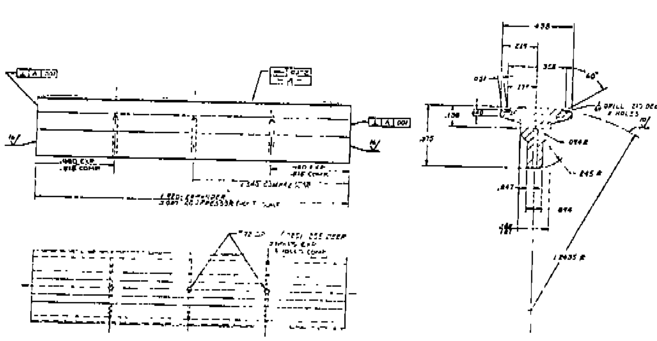


FIGURE 6. PIVOTING-TIP

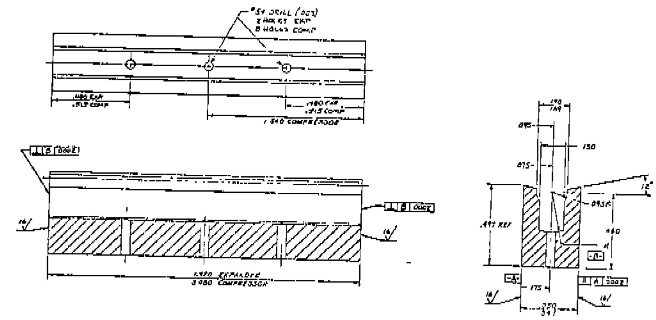
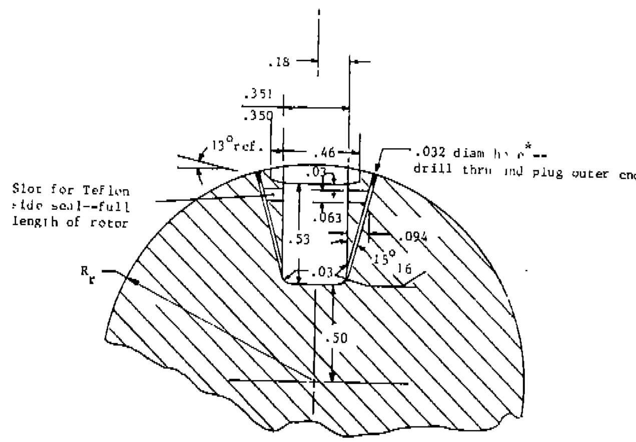


FIGURE 7. VANE



* 3 holes equally spaced on both sides of vane for compressor

2 holes for expander

FIGURE 8. ROTOR SLOT

PERFORMANCE OF THE PIVOTING-TIP GAS BEARINGS

The theory of self-acting pivoting-pad gas bearings is well established(6) but application to tips of vanes in rotary-vane machines is not as well established. Battelle has successfully applied this concept to high-speed fuel and oil pumps(7) and encouraging results have been obtained from theoretical studies aimed at applying the concept to compressible-flow machines. The specific challenges faced in applying the pivoting-pad gas bearing to reduce vane-tip friction in a compressor or expander are unique. The gas bearing must support (1) cyclic forces due to gas pressures and frictional forces at the sides of the vanes, (2) acceleration forces of the stroking vanes, and (3) constant forces due to centrifugal load of the vane and under-vane pressure.

Figure 9 shows the general configuration of the pivoting-tip vane for a compressible-flow rotary-vane machine which was conceived and patented(8) by Battelle as a result of in-house research. A design methodology for computing the performance of the pivoting-pad gas bearings in the rotary-vane application has been developed in theoretical studies of gas-bearing performance. This methodology applied in a computer program is capable of

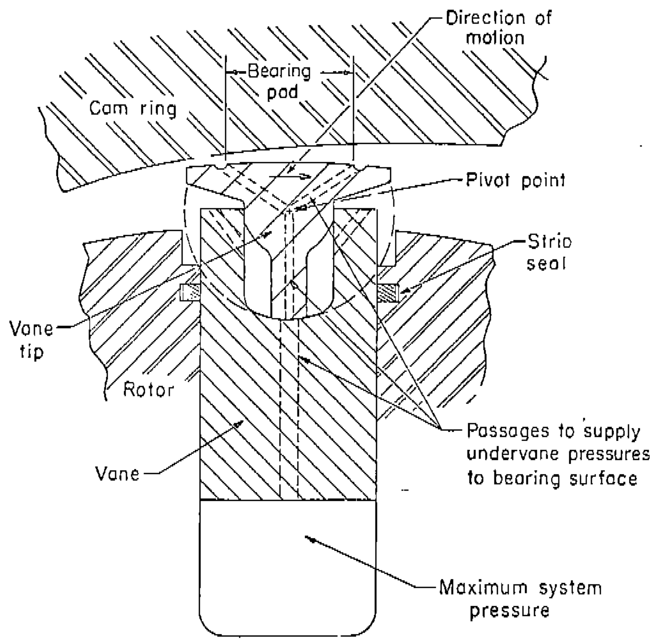


FIGURE 9. PIVOTING-TIP VANE CONFIGURATION

performing design point analyses for a specified-pad width, peripheral speed, and imposed leading- and trailing-edge pressures. Gas-bearing performance parameters can be computed as a function of rotational angle and shaft speed for either a compressor or expander, such as:

- Minimum film thickness from a radial-force balance
- Location of the point of minimum film thickness with respect to the leading edge of the bearing pad from a moment balance.

The minimum film thickness in the bearing pad (HMIN) is a useful parameter to be evaluated with respect to the average surface roughness to determine onset of surface contact. This thickness would be very difficult to measure in an operating machine. The stability of the bearing pad is related to the location of the minimum film thickness (XMIN). For example, the bearing should operate stably with values for XMIN greater than 0.5.

In addition, other parameters (for either a compressor or expander) that are a function of rotational angle are also computed, such as:

- Pressures at the leading and trailing edge of the pad
- Tilt angle of the tip as the tip oscillates within the vane to conform to the cam ring
- Axial length of the pad as affected by ports in the cam ring.

An unknown coefficient in the differential equation[6] for pressure variation across the pad is determined by iteration for each operating condition. Other bearing parameters can then be computed from the known pressure profile across the pad. The detailed analysis includes calculation of 22 forces and 10 torques acting on the tip or vane.

Figure 10 shows representative pressures in the gas bearing at three vane angles for the experimental prototype compressor and expander, showing substantial hydrodynamic pressures generated in the gas bearing at most vane angles.

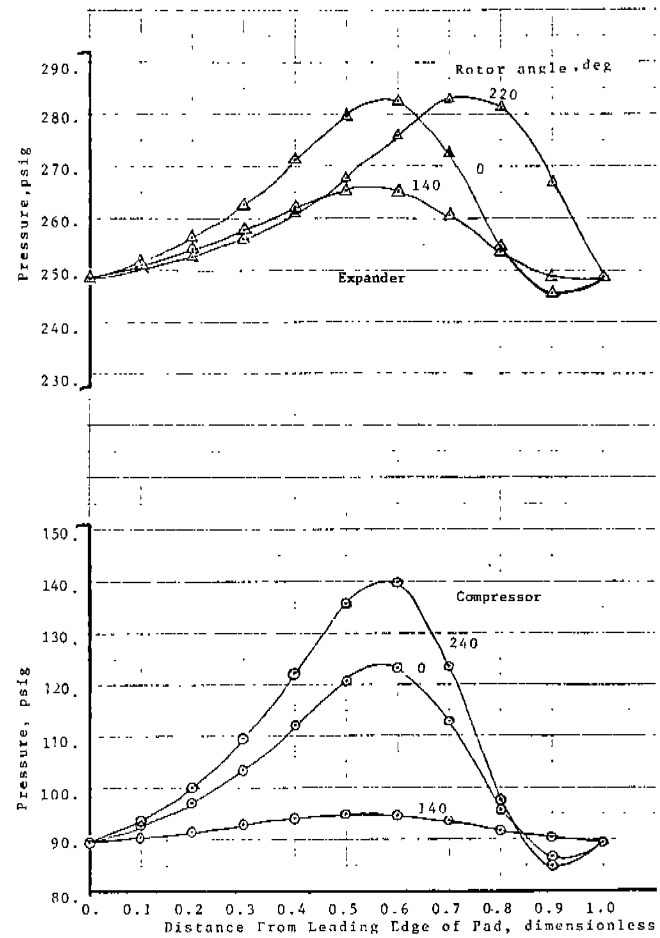


FIGURE 10. REPRESENTATIVE PRESSURES IN THE GAS BEARING

Figure 11 shows representative values for minimum film thickness and location of minimum film thickness for the experimental prototype compressor and expander with no frictional forces in the vane slots. The XMIN data show good potential for stable operation. Values for minimum film thickness averaged about 40 microinches, indicating that a tip with a surface finish of 10 microinches, which is not difficult to achieve, would function satisfactorily. With frictional forces in the vane slots, somewhat lower values for minimum film thickness were computed at some vane angles. For example, at a vane angle of 320 degrees, with vane-slot friction, HMIN was about 20 and XMIN was about 0.61.

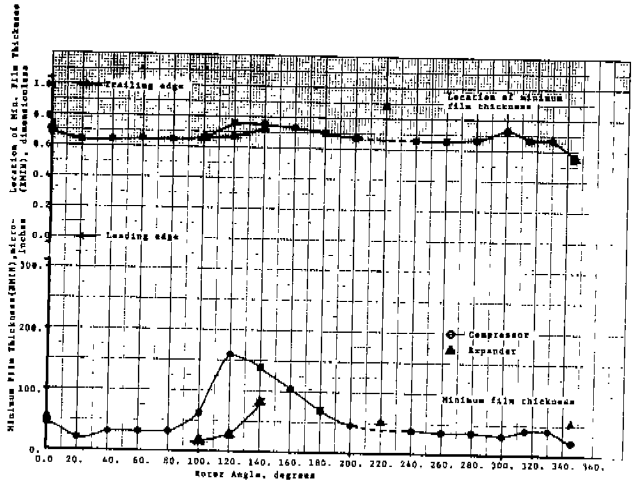


FIGURE 11. GAS-BEARING PERFORMANCE PARAMETERS

Experimental Studies of Gas Bearing Performance

Theoretical studies of gas-bearing performance was supplemented with experimental measurement of vane-tip friction in a non-stroking test rig. In this apparatus, the cam-ring was supported in trunion bearings so that the torque on the cam ring from the pivoting-tips could be measured by a strain-gaged cantilever stop.

Figure 12 shows measured tip friction expressed as a dimensionless friction coefficient with a conventional nonlubricated carbon-graphite vane and a pivoting-tip vane as a function of shaft speed. The value of 0.25 for the friction coefficient with the solid vanes is in good agreement with published data. Friction with the hydrodynamic pivoting-tip vanes was lower by an order of magnitude. Additional tests with hydrostatic support from under-vane pressure supplied to the bearing face further reduced the friction by an appreciable amount.

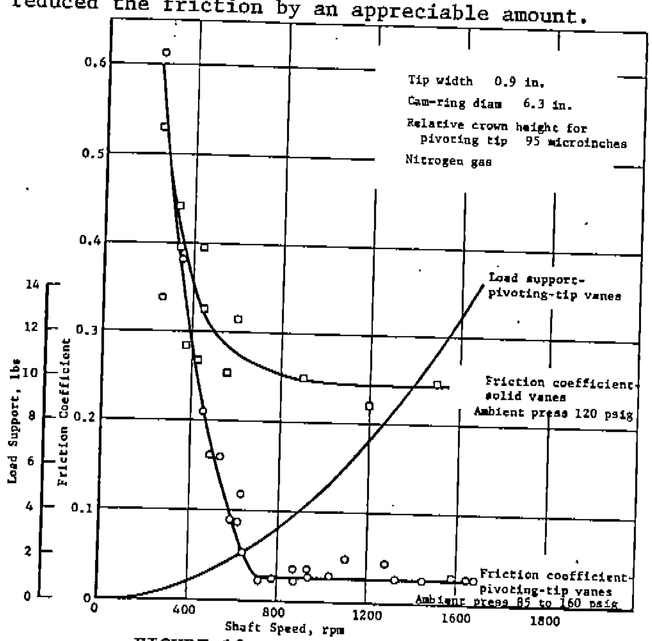


FIGURE 12. EXPERIMENTAL DATA

A minimum value for a key bearing design variable was established in these studies. It was found that the relative curvature of the tip with respect to the cam-ring radius had to be increased from a theoretical optimum value of 50 microinches to 100 microinches in order to achieve stable operation. This value was used in the theoretical design of the tips for the experimental prototype.

Load supported by the gas bearing is also shown in Figure 12 to illustrate speed dependency.

CONCLUSIONS

Conclusions arising from these studies are:

1. The development of the advanced compressor and expander design concept presented in this paper has progressed satisfactorily to a point where it is appropriate in the next phase to give considerable attention to commercial manufacturing. Performance simulation results have been positive, layout-design information is relatively complete, and analytical tools are available to support further development.

2. In a residential application of the Battelle solar heat pump, it is predicted that a substantial fraction of heating, cooling, and hot-water energy demands can be met with a reasonably-sized solar collector system.

3. Stable tip operation and adequate film thickness in the gas bearings are expected based upon theoretical and experimental analyses.

4. The preferred operational mode of the solar heat pump is to maintain speed constant with a directly-coupled electric motor and to let the expander-inlet temperature float above some minimum level.

5. For minimum electrical consumption, with the solar heat pump, auxiliary energy should be input as shaft power at the electric motor; if needed, additional heating capacity should be provided by electric resistance heating to the boiler to maintain temperatures above the level for net power output.

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