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EXPERIMENTAL STUDIES OF NON-RADIAL VANE ROTARY SLIDING VANE AIR COMPRESSORS DURING STEADY STATE OPERATION

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ABSTRACT

A study of the effects of vane inclination upon the performance of a sliding vane air compressor was investigated experimentally. Measurements were made of free air delivery and compressor input power for a range of compressor speeds and operating pressure ratios. The data was used to determine values of an optimisation parameter FOPT defined as the free air delivery per unit of power input (litres/kWs). Experiments were performed on a sliding vane air compressor having a circular rotor mounted eccentrically within a circular stator and the vane inclination was varied to cover both forward and backward inclined vanes. Tests were conducted for machines having vane inclinations of -10^0, -5^0, $+5^0$, $+10^0$. The results were compared with previously obtained results for a machine fitted with radial vanes (0^0 inclination).

The results of this work were used to effect a comparison with calculated results obtained from a computer based simulation and optimisation study which is reported in a companion paper.

1.0 INTRODUCTION

The use of small rotary sliding vane compressors in air compression , refrigeration , and automotive air conditioning systems has been studied by various investigators e.g [1,2,3,4,5] . Such studies have helped to further the understanding of the operational behaviour of this type of machine and have contributed to the range of design improvements which have taken place over recent years. Rotary sliding vane machines fitted with radially sliding vanes have an inherent advantage over reciprocating compressors in that they are better balanced machines being capable of operation at higher rotational speeds and suffer less from vibrational effects. Sliding vane machines however have their own limitations. They are prone to internal leakage effects which can be overcome using oil injection at the expense of the need to provide oil injection and oil separation components. As single stage machines they tend to have a limited range of operating pressure ratios. At high rotational speeds the effects of friction between the tips of the sliding vanes and the stator surface over which the vanes slide become much more pronounced. References [1,4,5] highlight some of the effects of vane tip friction whilst references [2,3] illustrate how simulation and optimisation studies may be employed to bring about design improvements.

Theoretical studies of a sliding vane machine fitted with non-radial (inclined vanes) have been reported by Ooi [2]. The work reported in this paper is a follow up to that of Ooi in that it reports the findings of tests performed on a number of sliding vane machines fitted with inclined vanes. The companion paper [6] to be presented at this conference gives further details of the simulation and optimisation procedures used to study inclined vane machines and shows a comparison of a selection of experimental and theoretical results.

2.0 DESCRIPTION OF SLIDING VANE COMPRESSORS

Following a series of simulation and optimisation studies , four sliding vane machines were constructed and subjected to a series of experimental tests. The machines shared a common stator unit but utilised four different rotors having vane slots of different inclination. Two compressors had vanes with a forward ($+5^0$, $+10^0$) inclination whilst a further two machines had vanes with a backward (-5^0 , -10^0) inclination. The vane inclination is defined with respect to the direction of rotation of the rotor as shown in figure 1 and a radial vane machine would be said to have a 0^0 vane inclination. . All the machines tested used a common set of vanes. Table 1 gives details of the principal dimensions of the machines.

Table 1 Principal Compressor Dimensions

PROPERTY	ABBREVIATION	DIMENSIONS	UNITS
Stator Radius	R_s	46.524	mm
Rotor Radius	R_r	40.048	mm
Eccentricity	ε	7.354	mm
Suction Port Opening Angle	β_1	64.754	deg
Suction Port Closing Angle	β_2	148.929	deg
Discharge Port Opening Angle	β_3	316.565	deg
Discharge Port Closing Angle	β_4	326.332	deg
Vane Height	H_v	22.545	mm
Rotor Length	L_r	170.000	mm
Discharge Port Axial Length	L_d	70.061	mm
Suction Port Axial Length	L_s	13.115	mm
Rotor Slot Depth	D_s	22.830	mm
Vane Thickness	t_v	3.945	mm
Mass per Vane	m_v	0.064	kg
Vane length	L_v	85.000	mm
Number of Vanes	N_v	6	
Sealing Arc Clearance	C_s	0.0775	mm
Vane Tip Radius	R_v	5.000	mm
Non - radiallity	σ	-0 , -5 , 0 , +5 , +10	deg

3.0 EXPERIMENTAL EQUIPMENT

The four test compressors were tested in the test rig which had previously been used to test radial vaned machines. This test rig is illustrated in figure 2 . It was thus possible to test the compressors at specified rotational speeds, set discharge pressures and various suction conditions according to the prevailing atmospheric conditions.

Measurements were made of the suction pressure and temperature , the discharge pressure and temperature , the rotational speed , the shaft input torque , and the volume of air delivered per unit time . Additional measurements were made to determine the flow rate of the oil which was injected into and subsequently removed from the compressor together with the heat transfers which occurred in the oil cooler and the air after-cooler . Figure 2 shows clearly the location of the various measuring devices .

Temperature measurements were made using copper-constantan thermocouples which were calibrated in the laboratory . Steady pressures were recorded using standard Bourdon test gauges . Dynamic pressure measurements were made at a number of locations using piezo-electric and piezo resistive pressure transducers.

The shaft input torque and the shaft rotational speed were measured using a Vibrometer type TG-20/BP torque transducer located between a variable speed drive and the compressor. The variable speed drive (Kopp Variator) was driven by a constant speed electric motor and allowed a wide range of compressor operating speeds to be achieved . It was thus relatively simple to determine the compressor input power .

A flow nozzle manufactured according to BS 1571 : Part II was used to measure the discharge air flow rate . The pressure drop across the flow meter was measured using a water manometer whilst the air temperature downstream of the meter was measured using a mercury in glass thermometer .

Oil for lubrication and internal sealing was injected into the machine at one specific angular location on the stator. The injection process depended on the difference in pressure between the oil separator located on the discharge side of the machine and the pressure at the injection point in the compression chamber. The greater the pressure difference the greater can be the oil injection rate into the machine. A limited control was possible on the oil injection rate by means of a control valve located between the oil separator and the oil injection manifold.

The compressor discharged a mixture of air and oil and the oil separator located on the discharge side of the machine was an essential part of the system. The separator which acted as an oil reservoir or accumulator was located between the compressor discharge pipe and the air after-cooler. Air leaving the oil separator passed to the cooler, then to the metering nozzle before being discharged to atmosphere. The oil separated in the separator could be passed via an oil cooler to the oil injection manifold and thence into the compressor.

4.0 EXPERIMENTAL RESULTS

4.1 Effects of Speed and Nominal Discharge Pressure on Free Air Delivery.

Each of the compressors was tested over a range of speeds and nominal discharge pressures. The compressor speed could be set at any value between 1000 rev/min and 2000 rev/min but a test series would generally utilise speed increments of 200 rev/min starting from the base speed of 1000 rev/min. For each speed condition a succession of constant discharge pressures was employed. The lowest discharge pressure used was 6.21 bar gauge and the highest discharge pressure used was 9.66 bar gauge and meant that for a nominal suction pressure of 1 bar (the typical atmospheric pressure) the machines would be operated over a range of absolute pressure ratios such that $10.66 > (P_d / P_s) > 7.21$. Figures 3a, 3b show the variation of the free air delivery with speed and discharge pressure for a machine having vanes inclined at -10^0. As would be expected for a positive displacement machine having a very very small clearance volume and a fixed volume compression ratio the free air delivery is directly proportional to the rotational speed. Internal leakage effects would appear to be almost constant although close scrutiny of the free air delivery versus discharge pressure graphs show that the free air delivery decreases very slightly with increasing discharge pressure. Very similar trends were observed for machines having vane inclinations of -5^0, $+5^0$, $+10^0$.

4.2 Effects of Speed and Nominal Discharge Pressure on Shaft Input Power.

Figures 4a, 4b, show the variation of input shaft power with speed and nominal discharge pressure respectively for the machine fitted with vanes inclined at -10^0. For a fixed discharge pressure the input shaft power appears to increase virtually linearly with the rotational speed. Close scrutiny of the various lines however reveals that as the speed increases the shaft input power increases more rapidly than a simple linear relationship might imply and is consistent with the argument that higher rotational speeds are also associated with larger vane tip / stator surface contact forces and thus greater frictional effects. The shaft input power also increases with increasing values of the nominal discharge pressure and is consistent with the fact that the indicated cycle work must increase as the discharge pressure is increased whilst the rotational speed and the suction pressure are kept constant.

4.3 Effects of Speed and Nominal Discharge Pressure on the Volume Throughput per Unit of Power Input (litres/kWs).

Figures 5a, 5b, which are again for a machine with -10^0 vane inclination show the variation of the free air delivery per unit of power input i.e the variation of the parameter FOPT. Figure 5a shows that FOPT appears to be virtually constant with respect to speed but decreases as the discharge pressure increases. This trend is confirmed in figure 5b where the curves for the various speeds are almost co-linear. Figure 5b also indicates a linear decrease in the values of FOPT with increase of the discharge pressure. Once again corresponding tests with machines having vane inclinations of -5^0, $+5^0$, $+10^0$, revealed broadly similar behaviour and trends.

4.4 Effects of Non-radiallity on Free Air Delivery

Figure 6a shows the variation of the free air delivery with vane inclination for various speeds at a constant discharge pressure of 6.21 bar gauge. The graph also inludes data for a machine having zero vane inclination. The variation of free air delivery with vane inclination is marginal but the values for the machine having zero vane inclination are perceptibly higher than those for the machines which employed inclined vanes . Figure 6b shows precisely similar trends for the slightly higher discharge pressure of 6.89 bar gauge .

4.5 Effects of Non-radiallity on Shaft Input Power .

Figure 7a shows the variation of the shaft input power with vane inclination for various rotational speeds and a constant discharge pressure of 6.21 bar gauge . In general this graph shows a decrease in the shaft input power as the vane inclination changes from -10^0 to $+10^0$. The changes are however quite small . Figure 7b shows corresponding results for a slightly higher discharge pressure of 6.89 bar gauge . In this case the results for the rotational speed of 1800 rev / min show a minimum shaft input power for zero vane inclination .

4.6 Effects of Non-radiallity on Volume Throughput per Unit of Power Input FOPT (litres / kWs).

Figure 8a and its companion figure 8b are perhaps the most revealling features of this paper. Each figure shows the variation of the parameter FOPT with vane inclination (non-radiallity) for a range of operating speeds and a fixed discharge pressure . Both figures show that FOPT exhibits a maximum value at a value of a vane inclination between 0^0 and $+5^0$. This feature indicates that a small positive vane inclination is required to produce optimum performance in terms of the volume throughput per unit of power input . Exactly similar trends were observed as the discharge pressure was increased to 9.66 bar gauge .

5.0 CONCLUSIONS

A comprehensive series of steady state tests on a number of rotary oil injected sliding vane compressors have shown for machines having a small clearance volume and a fixed volume compression ratio that when these machines are operated at constant speed varying the discharge pressure has little influence on the free air delivery . In all cases the volume throughput depended directly on the rotational speed of the compressor . Internal leakage increased very slightly as the discharge pressure was raised at a constant rotational speed . Changes in speed and discharge pressure both result in a changes of the required input shaft power . An increase in either speed or discharge pressure raises the compressor power requirement.
Tests have shown that for a given machine , a parameter FOPT , the throughput per unit power input , remains virtually constant as the machine is operated at a fixed nominal discharge pressure but over a range of rotational speeds . The parameter FOPT decreases as the discharge pressure is increased whilst the suction pressure remains constant .

Tests performed on machines having inclined vanes have shown that the vane inclination has an effect on the value of the parameter FOPT, the volume throughput per unit power input . The parameter exhibits a maximum value as the vane inclination is changed from -ve values (backward facing vanes) to +ve values (forward facing vanes) . This observation was confirmed over a range of operating speeds and operating pressure ratios . For the machines tested the maximum value of the parameter FOPT was observed to occur with vanes having a vane inclination of between 0^0 and $+5^0$ however the increase in the value of FOPT compared with the value of FOPT for 0^0 vane inclination was so small that manufacturing considerations would really dictate that machines with no vane inclination i.e radial vane machines are to be preferred .

6.0 ACKNOWLEDGEMENT

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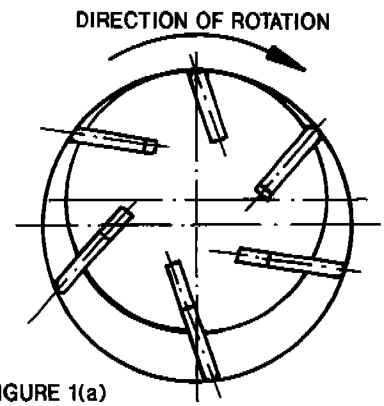

FIGURE 1(a)
NEGATIVE VANE INCLINATION

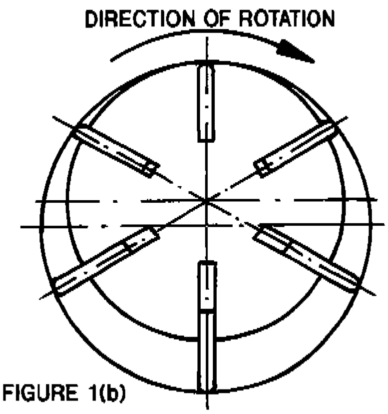

FIGURE 1(b)
RADIAL VANE GEOMETRY

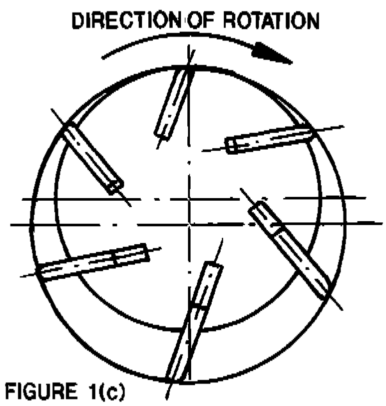

FIGURE 1(c)
POSITIVE VANE INCLINATION

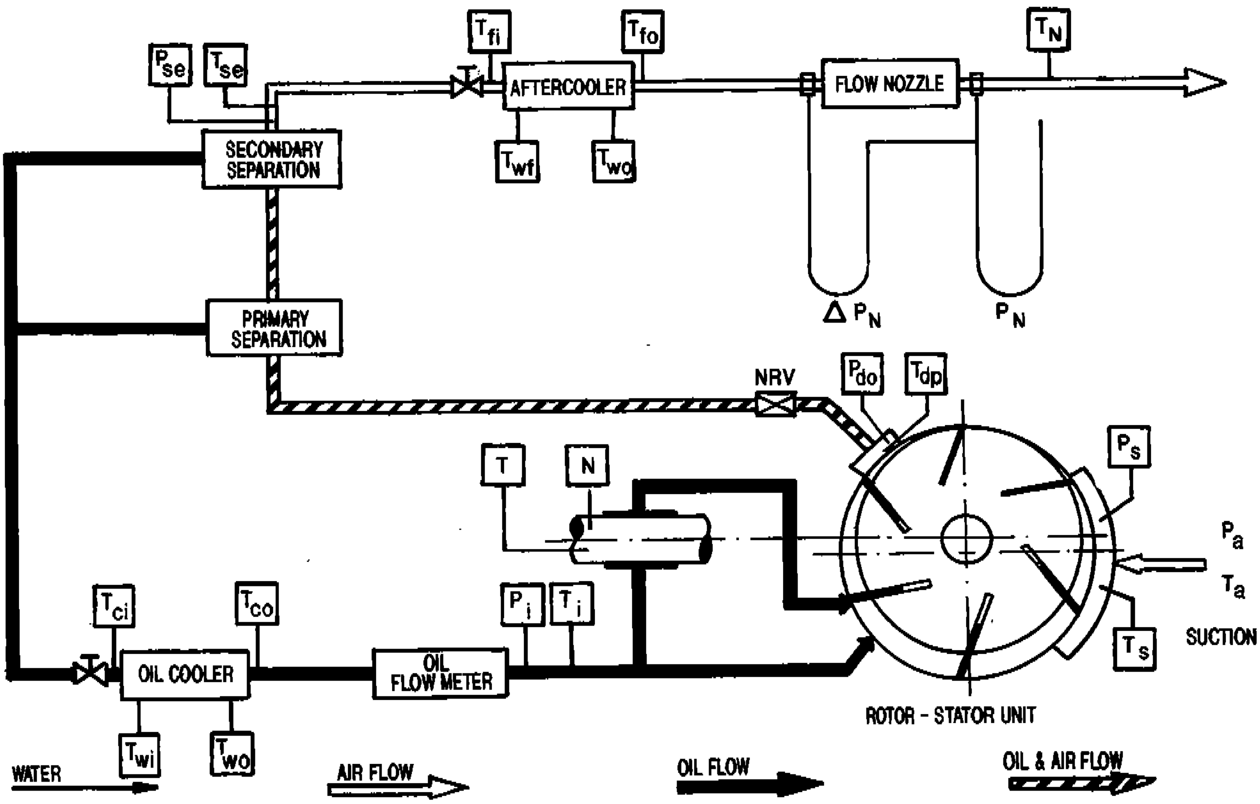

FIGURE 2. SCHEMATIC DIAGRAM FOR THE EXPERIMENTAL TEST RIG AND ITS ASSOCIATED INSTRUMENTATION

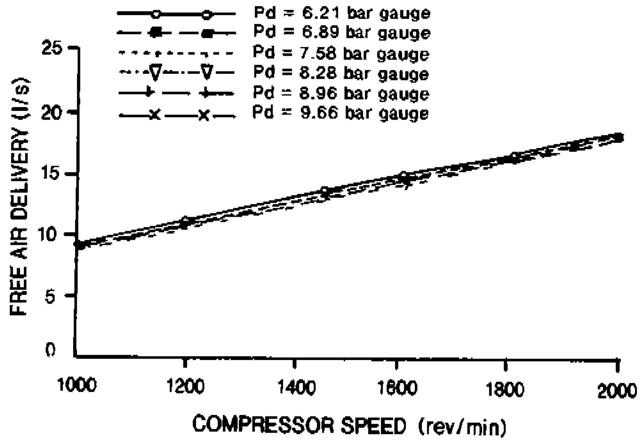

FIG. 3(a) VARIATION OF FREE AIR DELIVERY WITH COMPRESSOR SPEED

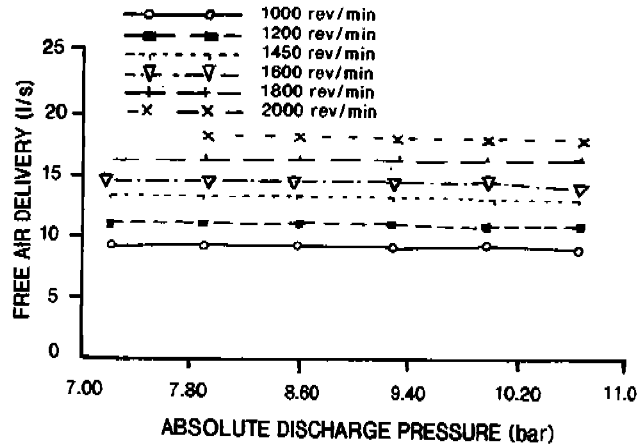

FIG. 3(b) VARIATION OF FREE AIR DELIVERY WITH NOMINAL DISCHARGE PRESSURE

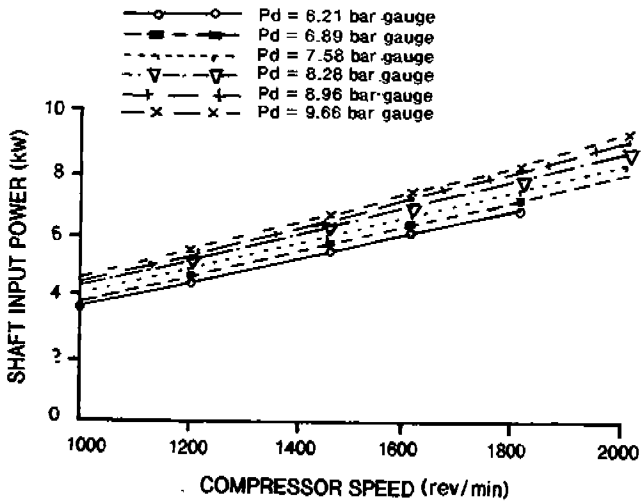

FIG. 4(a) VARIATION OF SHAFT INPUT POWER WITH COMPRESSOR SPEED

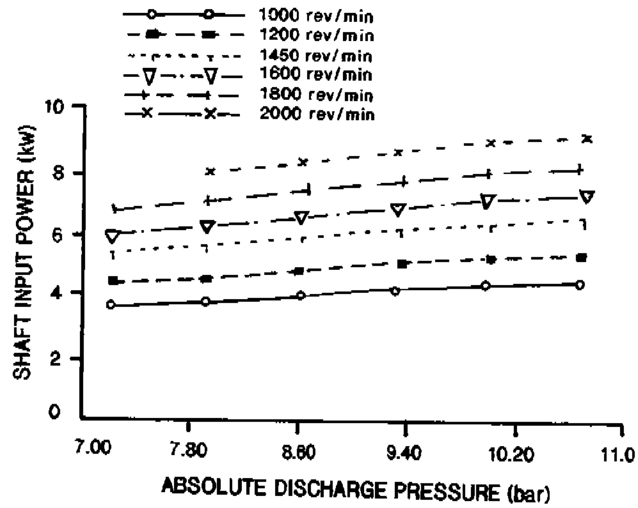

FIG. 4(b) VARIATION OF SHAFT INPUT POWER WITH NOMINAL DISCHARGE PRESSURE

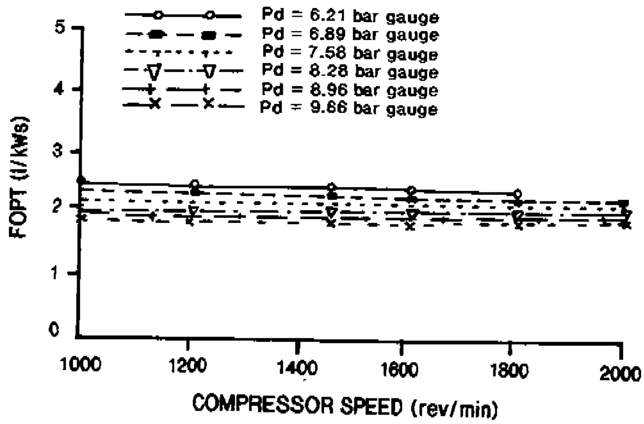

FIG.5(a) VARIATION OF VOLUME THROUGHPUT/kW WITH COMPRESSOR SPEED

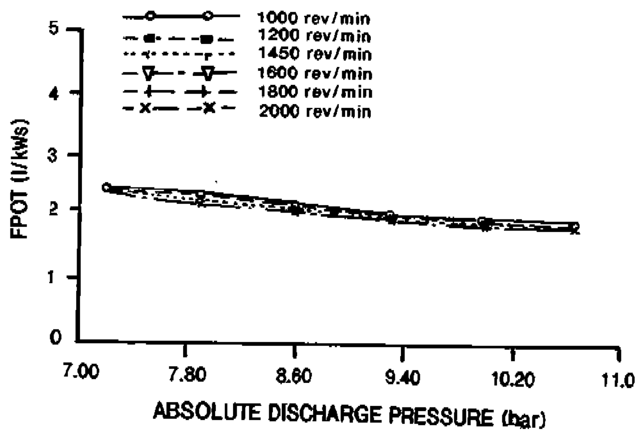

FIG.5(b) VARIATION OF VOLUME THROUGHPUT/kW WITH NOMINAL DISCHARGE PRESSURE

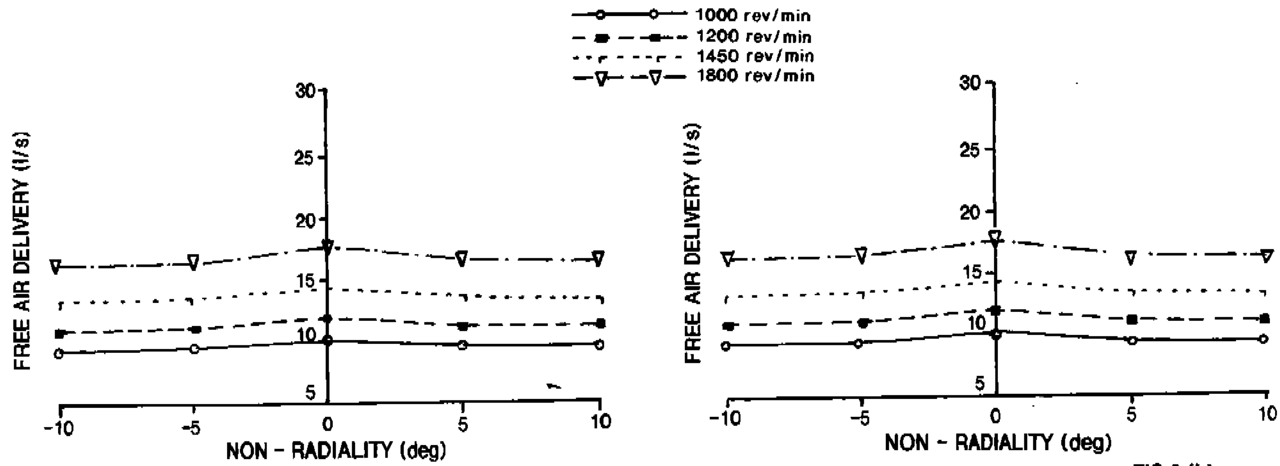

FIG 6 (a)

FIG 6 (b)

VARIATION OF FREE AIR DELIVERY WITH NON - RADIALITY

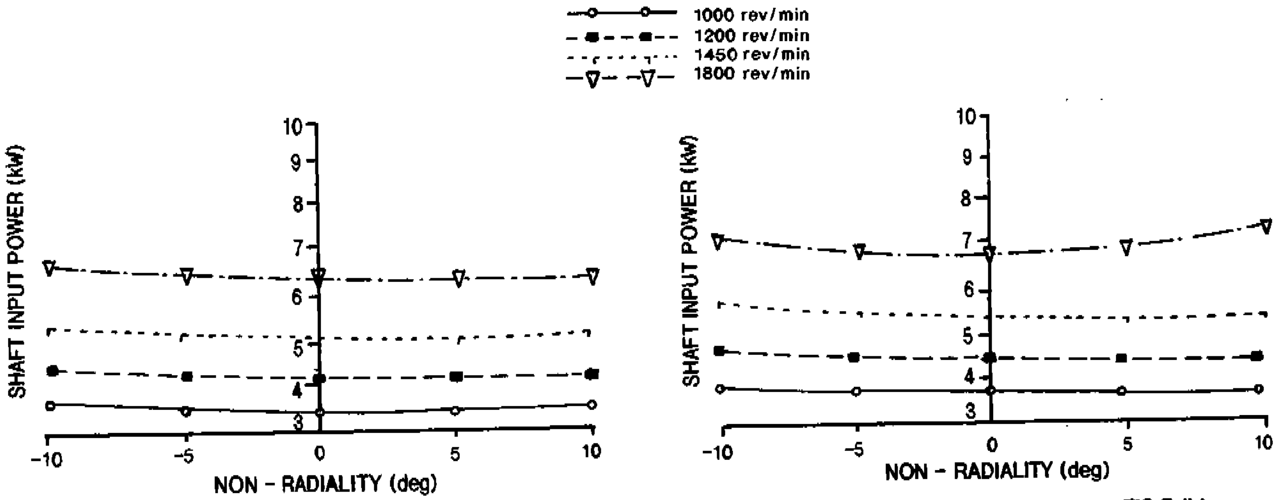

FIG 7 (a)

FIG 7 (b)

VARIATION OF SHAFT INPUT POWER WITH NON - RADIALITY

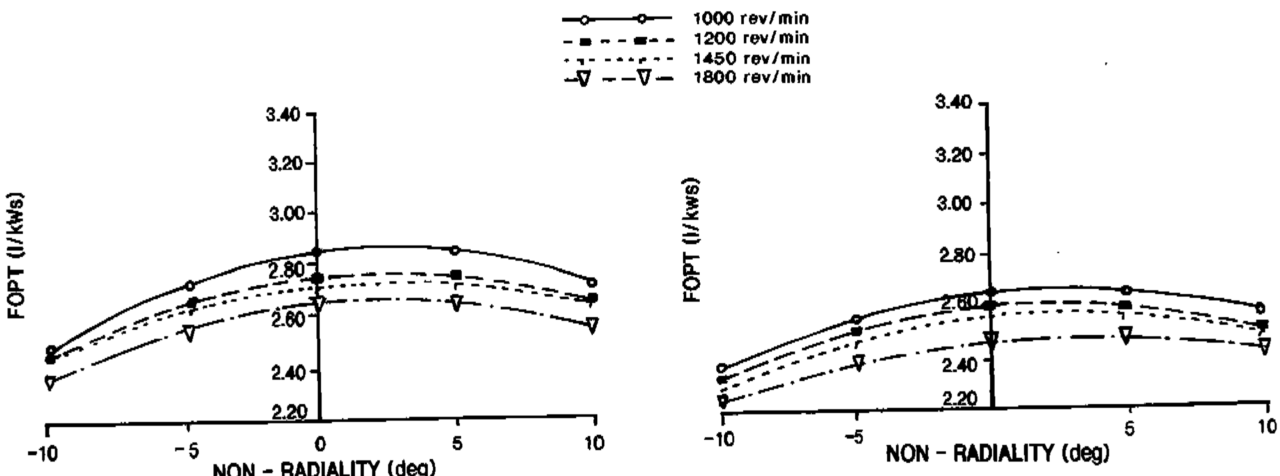

FIG 8 (a)

FIG 8 (b)

VARIATION OF VOLUME THROUGHPUT/ kW WITH NON - RADIALITY