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Effect of Design Parameters on Oil-Flooded Screw Compressor Performance

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EFFECT OF DESIGN PARAMETERS ON OIL-FLOODED SCREW COMPRESSOR PERFORMANCE

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ABSTRACT

The performance of oil-flooded screw compressors depends on a large number of design parameters such as rotor profile, number of rotor lobes, rotor L/D ratio, wrap angle, geometrical clearances, quantity and location of oil injection, discharge port size and tip speed. Knowledge of the effect of these parameters can help a designer to select the best performing machine for a given application. This paper systematically examines the effect of several of the above parameters on compressor size and performance for a given displacement. Proprietary computer programs (1,2) are used to calculate the geometrical characteristics of the rotors and compressor performance. Results show that the performance can be considerably improved by proper selection of design parameters. Several plots showing the relationship between these parameters and compressor size and performance are presented that could be very valuable to a designer.

INTRODUCTION

Historically, oil-flooded rotary screw air compressors have been designed with certain standardized features such as rotor profile and number of male-female rotor lobe combinations. This was because the generation of new profiles and adaptation of manufacturing methods (including the need for new tooling) was a time-consuming and expensive task. In recent times the advent of computerized highly-flexible rotor manufacturing techniques and the development of computer-based design techniques have provided new freedom to the compressor designer. The designer now has the flexibility to adjust the design parameters to their most optimum values for a specific application.
In order to exercise this freedom he must have an economical means of predicting the effect of these parametric changes on the compressor performance and size.

In 1980 the authors' company developed several computer programs to rapidly generate rotor profiles along with their supporting data (1). These programs have the flexibility of generating profiles with different shapes and number of lobes, and analyzing their associated leakage lengths and areas as well as inlet and discharge port shapes and sizes. These data are then input to a performance prediction program (2) to yield compressor BHP and capacity. These programs can be used to conduct parametric studies of the effect of geometrical clearances on compressor performance for a known profile. Such studies have been made by Singh and Patel (2) and Fugiwara et al (3) who have provided a detailed accounting of the flow and power losses as a function of clearances and wrap angle.

This paper focuses on a more fundamental aspect of the design process. A designer, using these advanced techniques, can now select a certain profile shape, and then for a given profile can select other parameters such as number of male and female rotor lobes, L/D ratio, wrap angle, discharge port opening angle, etc. While these parameters have a great influence on the machine performance, strength and reliability, meaningful data to guide the designer in proper selection of these parameters does not exist in the open literature. This paper fills this void by presenting such data for typical screw compressor applications and highlights the relative importance of various parameters.

PROCEDURE

A number of rotor designs were created and their dimensions normalized to give identical displacement per unit revolution. This was done to place all the designs on a common footing for comparative purposes. One of the primary design parameters for evaluation was the selection of the number of male and female rotor lobes. Eight designs were selected for consideration with the following male-female rotor lobe combinations: 3-4, 4-5, 4-6, 5-6, 6-6, 6-7, 7-8, 7-6. Seperate profiles were then generated for each design under a set of common guidelines: very small blow-hole area; small sealing line lengths; small rotor size with adequate mechanical strength for 100 psi application. While the profiles looked somewhat different, they shared a common design philosophy and many of the profile segments had common geometrical characteristics. In this way, the designer was allowed freedom to tailor the profiles to the number of lobes combination. Two sample profiles, 5-6 and 7-6, are shown in figure 1. The (7-6) profile resulted in highly disproportionate rotors: a stiff, large male rotor and a weak, small female rotor. It was such a poor candidate that it was dropped from further consideration.

Table 1 shows the primary geometric dimensions of the various profiles normalized to the male rotor diameter of the 5-6 profile no. 4. Since the primary purpose of this paper is to present the relative merits of different designs, such normalization has the virtue of highlighting major differences in a group of candidates. The selection of profile 4 as the base for normalization will become clear later. However, we might mention now that profile 4 turned out to be an excellent choice for typical screw compressor applications.

In Table 1, two 4-6 profiles, 3 and 3A, are listed with the main distinction that profile 3 has unequal rotor diameters while 3A has equal diameters. Profiles of type 3A are most widely used in the industry. Thus it was included in this investigation although it violated the common design criterion of a small blow-hole. The relative merits of equal and unequal diameter rotors are more a function of manufacturing considerations than the design process. As can be seen from Table 1, only profiles with the lobe combination differing by 2, 4-6, 6-8, can be practically made to have equal rotor diameters.

This forces one to select larger female addendum which generally results in a larger blow-hole. It must be pointed out that we are talking about a larger-blow-hole only in a relative sense. For example, profile 3A has a blow-hole area of about .027 in^2 for a 6 in diameter rotor while the comparable area for profile 4 is only .003 in^2. Also, the importance of geometrical features such as blow-hole area depends on the application. As the tip speed increases, leakage areas begin to impact performance less and less while inlet and discharge port areas become more important. This is why integrated computer tools such as those described in references 1 and 2 are so important for application-oriented designs.

Table 2 lists the desired geometrical parameters that affect compressor design and performance for various profiles normalized to profile 4 values. The polar moment of inertia is based on the female rotor root diameter, $I_{zfr} = \pi D_{fr}^4/32$, which serves as a measure of the rotor's stiffness. The moment of inertias of the complete male and female rotors about their respective axis,

$$I_{zm} = \frac{1}{4} \oint_m r^2 \, (xdy - ydx)$$

$$I_{zf} = \frac{1}{4} \oint_f r^2 \, (xdy - ydx)$$

represent the stiffness of the rotor end profile including lobes. In practice, however, the contribution of lobes and the moment of inertia based on root diameter is a good indicator of the rotor stifness. From the table, it is apparent that profiles with female-male lobe number difference ($N_f - N_m$) of 2 tend to have much stiffer female rotors than those with $N_f - N_m = 1$. This indicates that $N_f - N_m = 2$ profiles may be preferable for high pressure applications where rotor loads tend to be very high.

Table 2 also shows that the overlap constant, the ratio of maximum filled volume to the theoretical filled volume, is about the same for all profiles except for the case of equal number of lobes. For the 6-6 lobe profile, cavity fill volume is only 91.2% of the theoretical volume which is also indicated by early closing of the inlet port. Inlet port closing angle decreases only slightly with the increase in number of lobes except for the 6-6 case.

The angle θ is defined as the angle of the leading edge of the male cavity and ranges from a negative value at cavity's inception to $\theta max = 360 (1+1/N_m) + \varphi_m$ at cavity's expiration. Unlike the inlet port angle, the discharge port angle is a strong function of the number of lobes. As N_m increases, the discharge port opens earlier as indicated by an increase in the port opening angle (commonly called β_m). This allows increasingly large axial and radial discharge ports with increasing N_m.

The blow-hole area for all the profiles is very small by design except for profile 3A. The blow-hole area is a strong function of the female addendum (outer radius - pitch circle radius) which is about 1.5% for most profiles except profile 3A for which it is 3%. The selection of female addendum is not completely arbitrary since it strongly affects the torque distribution between the two rotors. Very small or negative addendum can cause torque reversal at certain cavity positions, that may result in vibrations and damage.

Table 2 also shows that the contact or interlobe sealing length increases strongly with the number of lobes which has an adverse effect on performance, particularly at low-tip speeds and high pressure ratios. The maximum male and female rotor tip seal leakage lengths also increase slightly with the number of lobes but the average leakage length over the complete cycle remains about the same. In fact, the amount of tip seal leakage flow may actually decrease with number of lobes since the leakage has to pass through more cavities before reaching the inlet. While it is instructive to examine how the selection of number of lobes affects various geometrical parameters, the best comparison is obtained by examining their performance characteristics under identical operating conditions. The next section addresses the effect of these parameters on compressor performance and size.

PARAMETRIC EVALUATION

Effect of Number of Lobes on Airend Size.

Figure 2 shows the percent change in male & female rotor cylinder volume $[\pi (D_f^2+D_m^2) \times L/4]$ relative to profile 4 as a function of the number of male rotor lobes. It is clear that smaller number of lobes reduces rotor volume and thus the airend size.

Also, there is a distinct difference between N_m-N_f = 2 and N_m-N_f = 1 profiles. The former require considerably larger rotors for a given displacement. This happens because, for a part of the revolution, one of the female cavities is idling and not taking part in the suction or compression process.

Effect of Number of Lobes on Rotor Stiffness

Rotor stiffness is an important parameter since it may limit the maximum operating pressure of the compressor. Figure 3 demonstrates the effect of number of lobes on female rotor deflection based on root diameter inertia I_{zfr}. While stiffness is a strong function of the selection of lobe depth, the profiles with N_f-N_m = 2 have female rotors almost equal in size to the male rotors and thus are inherently much stiffer.

Effect of Number of Lobes on Discharge Port Velocity

The maximum gas velocity through the discharge port is plotted in Figure No. 4 with the profile 4 taken as the norm. This velocity is indicative of the port losses that are proportional to velocity squared. The plot shows that the velocity decreases with an increase in the number of lobes. This is clear from the trend in total discharge port area shown in table 2. It also means that the discharge port losses are generally higher for profiles with less number of lobes. Thus at high tip speeds where discharge port losses become significant, profiles with a lower number of lobes will have poor performance.

Effect of Lobe Combination on Performance

Figure 5 shows the volumetric efficiency (VE) and normalized specific power (BHP/100 CFM) for various profiles over a tip speed range of 10 to 50m/sec. The performance is based on wrap angle φ_m = 300°, L/D = 1.65, moderate clearances, typical oil injection rates, inlet pressure = 14.5 psia, 7.8 pressure ratio, 7° early discharge port opening angle and air as the compressed fluid. All profiles except 3A show specific power gradually increasing with tip speed. The profile 3A which does not share the common design philosophy of the rest has an optimum performance at about 25m/sec. This is because at low tip speeds, the large blow-hole area begins to adversely affect performance through inter-cavity leakage and recompression. Among other profiles, the 3-4 combination has the worst performance at 50 m/sec due to its restricted discharge port.

Profiles with lobe difference of 1 have marginally better performance than those with differences of 2 at all tip speeds. While profile 6-7 shows the best performance in the diagram, all profiles with N_m = 5 to 7 can be considered to have equivalent performance since the differences are minor from a practical point of view and are within the programs' accuracy. It should also be clear by now that the 5-6 profile has all the attributes of a good profile in terms of performance, strength, and size and thus was selected as the base profile for normalization. It must be emphasized again that these and other results are valid only for the listed conditions and should be used for relative purposes only. For example, high tip speed performance of 3-4 profile can be improved considerably by opening the discharge port even earlier than the 7° used here. Thus each design should be optimized according to the individual application.

Effect of Wrap Angle on Performance

The change in wrap angle has two primary effects: The discharge port size increases and the overlap con- stant decreases with wrap angle. Table 3 lists various important geometric characteristics of profile 4 for wrap angle varying from 200° to 400°. Figure 6 shows the performance of profile 4 with L/D - 1.65 and π_i = 7.8 at various wrap angles φ_m. The poor performance of the φ_m = 200° design at high tip speeds is again caused by overly restrictive discharge port. The best performance is indicated by 300° wrap angle. It is interesting that the generally poor performing φ_m = 200° design also has the best VE which decreases with increase in wrap angle. This is because the leakage lengths are smaller at lower wrap angles (table 3). In the limit, one can think of φ_m = 0 which will then become a Roots compressor. In the Roots compressor, interlobe and radial seal drive lengths are equal to the rotor length but the built-in compression ratio is zero and the axial discharge port if one can be conceived is extremely small.

Effect of L/D Ratio on Performance

After the lobe combination, the next parameter normally selected is the L/D ratio. The impact of L/D ratio on airend performance for a 300° wrap angle, 5-6 profile is plotted in Figure 1, while the normalized geometric characteristics are listed in Table 4. L/D ratio of 1.15 generally shows the worst performance in figure 7 both for VE and specifice power, particularly at the low tip speeds. A small L/D ratio means larger diameter rotors for a given displacement and larger leakage areas. At higher tip speeds, the leakage areas begin to have less influence and performance for all L/D ratios tend to cluster together. L/D ratios of 1.65 and 1.8 show the best overall performance. It is perhaps no coincidence that the commonly used values of wrap angle of 300° and L/D = 1.65 based on experience also show up so well in this study.

Effect of Opening The Discharge Port Early

One method of improving the performance is to open the discharge port early, i.e., to reduce the built-in pressure ratio. This has the effect of increasing the discharge port size at the cost of some backflow compression. However, at high tip speeds, gas inertia tends to overcome this backflow effect and significant improvement in performance can be achieved. The optimum built-in pressure ratio and other parameters such as opening angle is a function of the profile shape, number of lobes, wrap angle and L/D ratio. Figure 9a shows the effect of 0°, 7° and 14° early opening on performance of the 5-6 profile. The data indicates that specific power improves with early opening at 50 m/sec while getting worse at 10m/sec. Even at 50m/sec it appears that any further early opening will make the performance worse. However, with 200° wrap angle, the effect on specific power is significant even at 18.7° early opening (Figure 9b) since the discharge port is very restrictive at the built-in pressure ratio as shown in Table 3. Thus it is necessary to determine the optimum opening angle according to profile shape and operating conditions.

CONCLUSIONS

Considerable data relating the effect of certain important parameters to the performance of oil-flooded rotory screw compressors have been presented. These data can be effectively used by a compressor designer as a guide in selection of proper parameters for a given application. The following significant conclusions can be drawn from these data.

(i) The number of rotor lobes has a major influence on compressor performance, size and rotor stiffness. Low number of lobes can lead to a smaller airend but also poorer performance, particularly at higher tip speeds and pressure ratios. A 5-6 lobe combination offers a good overall design but leads to unequal rotors. The best lobe number selection depends on the application and other considerations beyond performance such as ease of manufacturing and maintenance of tolerances.

(ii) A wrap angle of around 300° and L/D ratios of 1.5 to 1.9 offer good performance for typical air compressor applications.

(iii) Performance can be improved by opening the discharge port early. The amount of early opening is a function of rotor profile and operating conditions.

(iv) The computer programs used here can be a powerful aid in optimizing compressor geometery according to the application.

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TABLE 1

Normalized Profile Dimensions

Profile No.	Lobe Combination	Diameter Male	Diameter Female	Center Dist.	Wrap Angle Female	Overlap Constant	% Female Addendum
1	3-4	0.972	0.746	0.633	250	0.960	1.51
2	4-5	0.994	0.769	0.685	240	0.973	1.04
3	4-6	0.992	0.934	0.757	200	0.976	1.35
3A	4-6	0.972	0.972	0.763	200	0.970	2.94
4	5-6	1.0	0.798	0.709	250	0.984	1.56
5	6-6	1.055	0.748	0.720	300	0.912	1.87
6	6-7	1.017	0.829	0.746	257.1	0.989	1.51
7	6-8	1.016	0.957	0.812	225	0.986	1.51
8	7-8	1.028	0.843	0.767	262.5	0.992	1.51

NOTE: Male Rotor Wrap Angle = 300°

TABLE 2

Normalized Geometrical Parameters

Profile No.	Lobe No. NM-Nf	Female Root Inertia	Leakage Contact	Leakage Radial	Blow Hole Area	Port Angles Inlet	Port Angles Disch ßm	Total Disch Port Area
1	3-4	0.24	0.714	0.932	0.7	378	22.8	0.546
2	4-5	0.65	0.872	0.974	0.9	366	39.5	0.806
3	4-6	2.40	0.843	0.966	0.7	366	43.1	0.962
3A	4-6	3.04	0.879	0.954	9.0	372	39.6	1.01
4	5-6	1.00	1.0	1.0	1.0	360	48.0	1.0
5	6-6	0.71	1.07	1.026	0.6	324	49.7	1.101
6	6-7	1.67	1.11	1.035	1.0	354	53.8	1.190
7	6-8	4.42	1.26	1.029	1.2	354	54.5	1.208
8	7-8	2.11	1.29	1.055	1.0	350	58.1	1.424

TABLE 3

Effect of Change in Wrap Angle on Normalized

Geometrical Parameters of Profile 4

Wrap ∠ (Deg)	Overlap Constant	Port (Deg.) Inlet	Port (Deg.) Disch	Blow Hole Area	Leak. Lengths Contact	Leak. Lengths Radial	Discharge Port Area
250	0.9998	336	37.6	1.08	0.904	1.090	0.692
276	0.9952	354	43.3	1.04	0.953	1.045	0.858
300	0.9839	360	48.0	1.0	1.0	1.0	1.0
324	0.9628	372	51.8	0.95	1.048	0.928	1.130
350	0.9310	384	55.1	0.91	1.102	0.868	1.225

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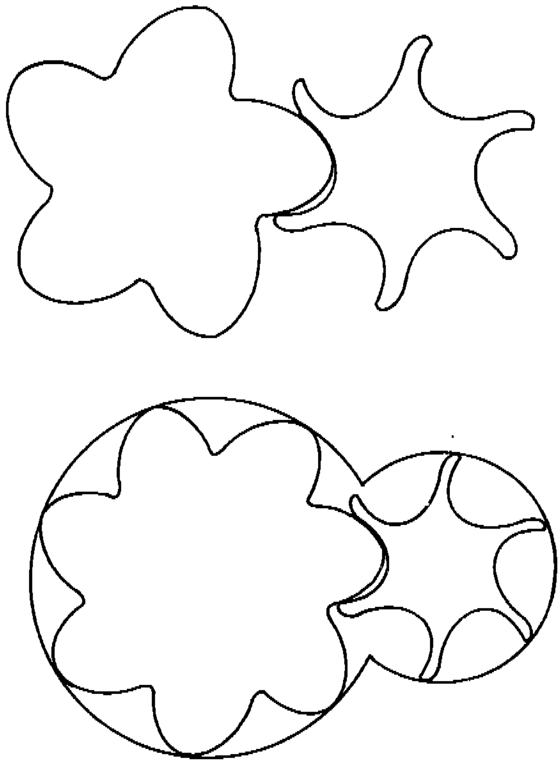
SYMBOLS

N = Number of lobes
D = Outside diameter
Iz = Polar moment of inertia
L = Rotor length
r = Profile radius
x,y = profile coordinates in x,y plane
θ = Cavity position angle
β = Discharge port opening angle
φ = Wrap angle
π_i = Built-in pressure ratio
$\oint$ = Integration around the profile

SUBSCRIPTS

m = Male rotor
f = Female rotor
fr = Female rotor root diameter



5+6 Profile

7+6 Profile

Figure 1 - Two Sample Profiles Used In The Parametric Evaluation

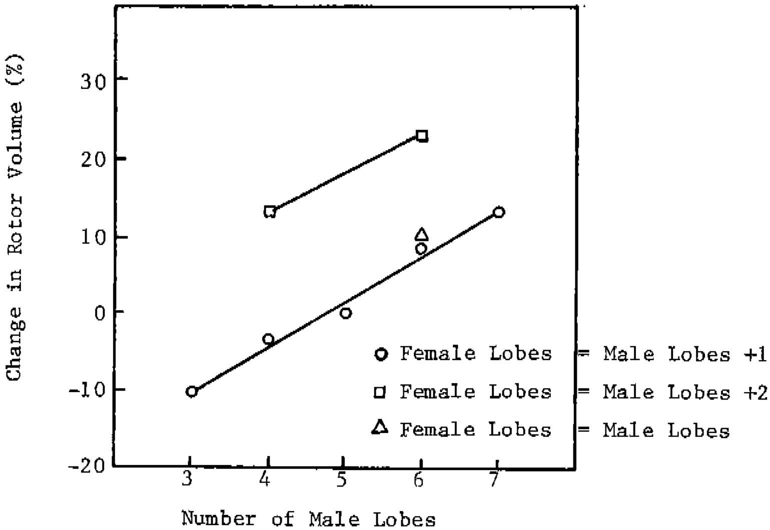


Figure 2 - Percent Change In Rotors Volume Compared To Profile 4.

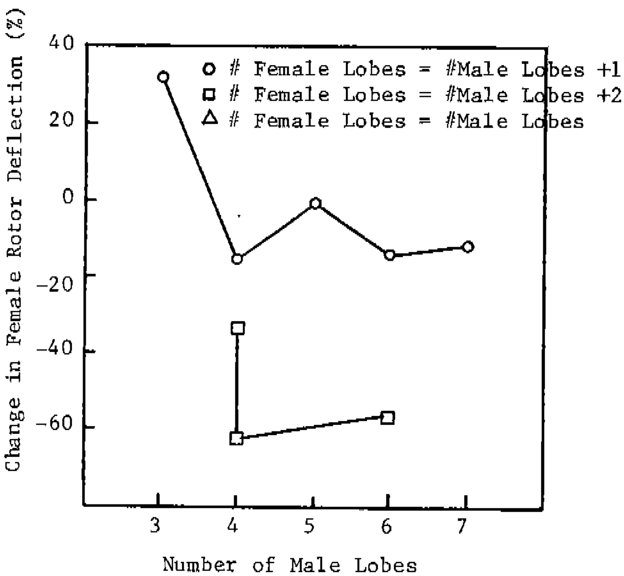


Figure 3 - Percent Change In Female Rotor Deflection Compared To Profile No. 4 (Based On Female Rotor Root Dia.).

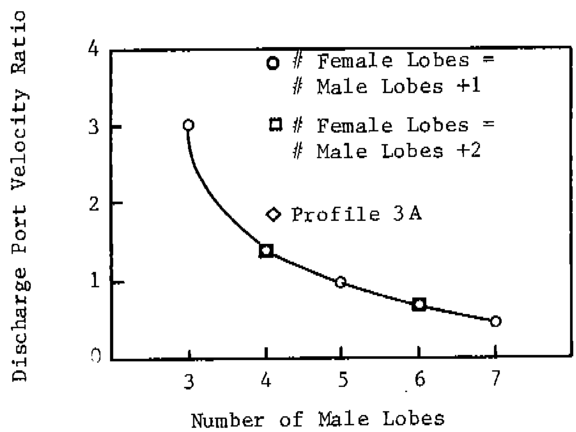


Figure 4 - Maximum Discharge Port Velocity Normalized To Profile 4.

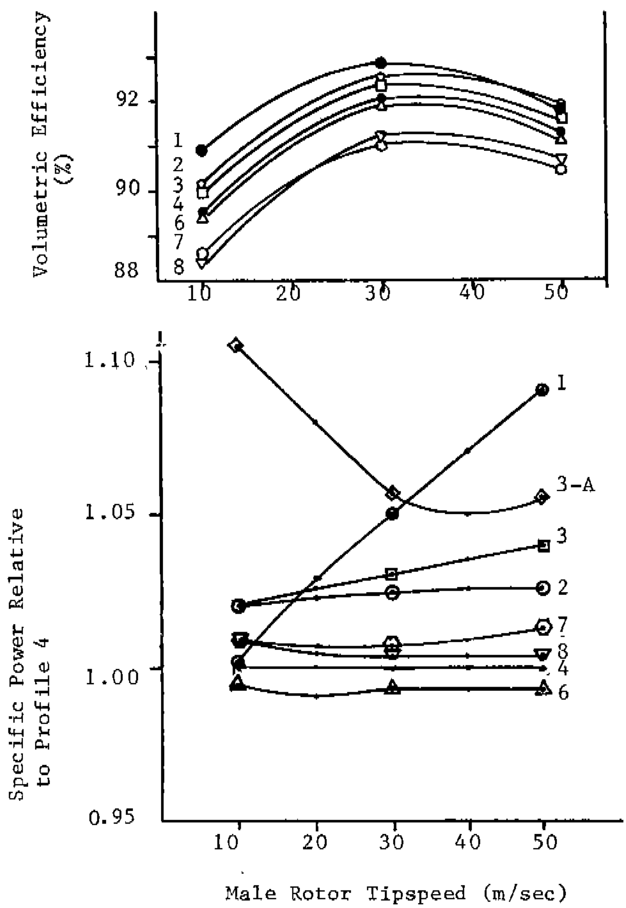


Figure 5 - Normalized Specific Power And Volumetric Efficiency For All Profiles Versus Male Rotor Tipspeed.

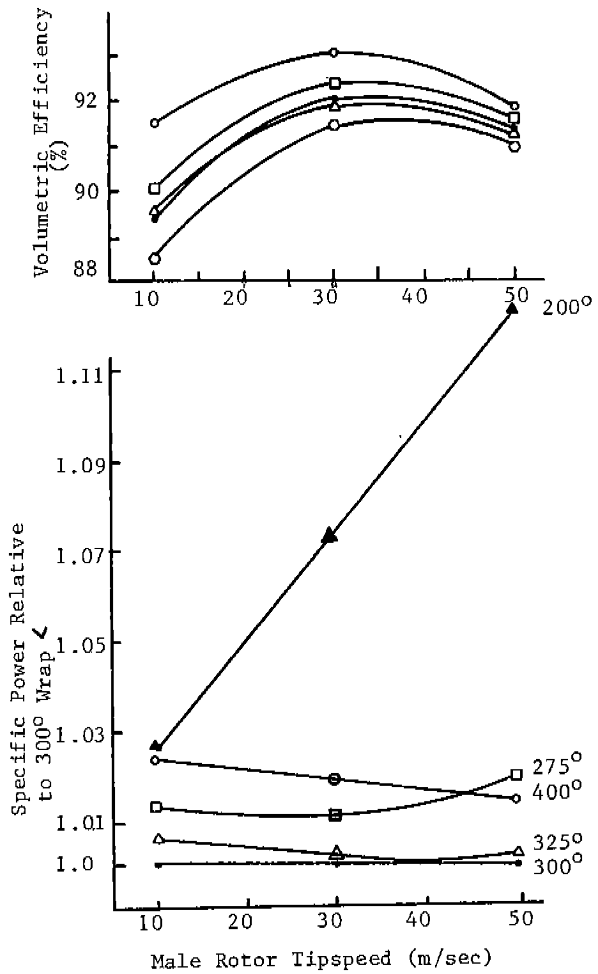


Figure 6 - Effect of Wrap Angle on Specific Power And Volumetric Efficiency of Profile 4.

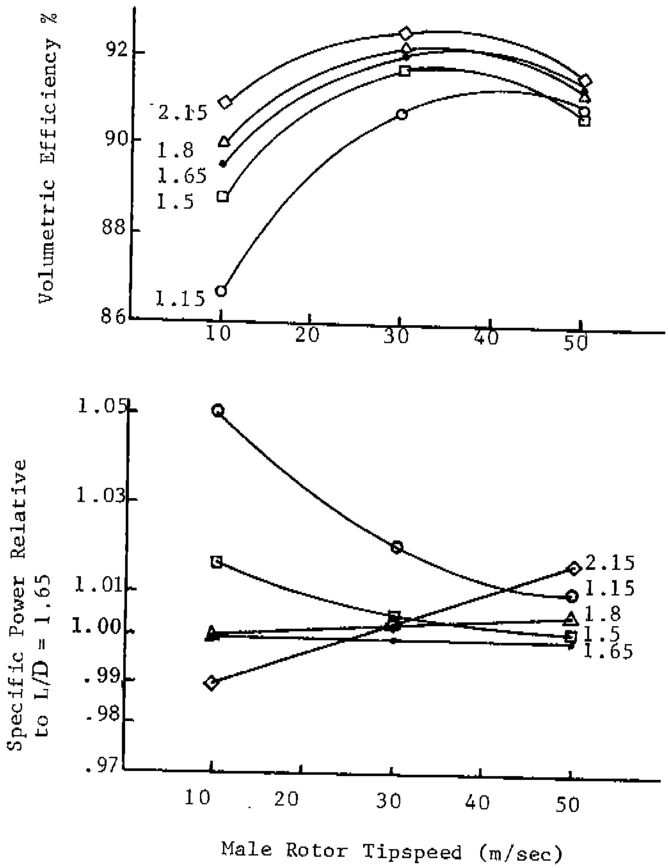


Figure 7 – Effect of L/D Ratio On Specific Power and Volumetric Efficiency For Profile 4.

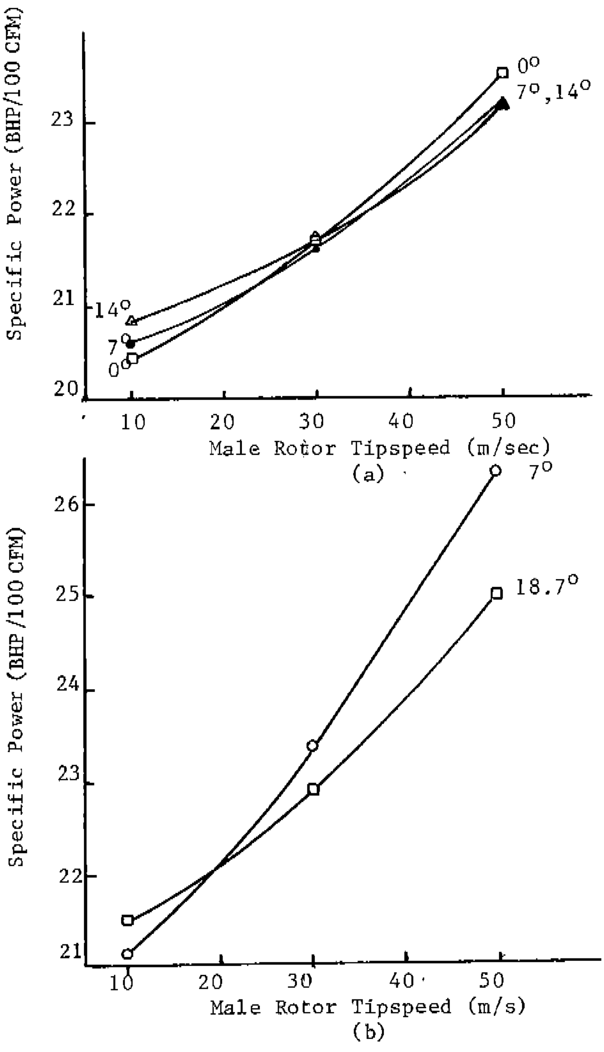


Figure 9 - Effect of Early Opening of Discharge Port on Specific Power for Profile 4.
(a) Wrap Angle - 300°.
(b) Wrap Angle - 200°.