Investigation On Twin Screw Compressor Rotor Profile

Abhishek G. Yewale, (Dr.) Ratnakar R. Ghorpade, Suraj K. Abdan.

Abstract:The screw compressor is a favorable rotating displacement compressor with a screw-shaped piston. The primary components are male and female rotors that move towards each other while the volume between them and the housing reduces. In an industry, the compressors are one of the highest power consuming machines. Hence, to decrease the power consumption, the efficiencies of these machines need to be at the highest level at all operating conditions. There are different geometric and installation parameters of the screw compressor that influence screw compressor efficiency such as inlet and outlet ports, rotor size, lobe mixture, rotor length to diameter ratio, clearance and primarily rotor profile. There are very limited or no literature available with open access for the design of a screw rotor profile. This paper focuses on the initial steps to be followed during the design of screw rotor profile. Various curves have been used to generate rotor profile. The curve used in a particular area of the main profile is the sine curve. The study presents profile generation using above mentioned curve. An analysis and comparison of the performances of this profile with widely used profile in the commercial market. Profile used here for comparison is “N” profile developed by City, University of London, UK

Index Terms:Female Rotor, Lobe Ratio, Male Rotor, Sine-curve, Trochoid, Rotor Profile, Twin Screw Compressor.

1 INTRODUCTION

SCREW compressors are basically easy rotary positive displacement machines capable of high velocity operation at high efficiencies over a broad range of working pressures and flow rates. [1], [2], [3]. A screw compressor essentially consists of a pair of interlocking helical rotor blades that rotate within a fixed housing that encloses them completely, as shown in fig. 1 the space between any two consecutive lobes of each rotor, and around it the housing forms a separate chamber. The size of the chamber changes as it rotates due to the displacement of the line of contact between the two rotors. This is maximum if the entire length between the lobes is not limited to the other rotor.

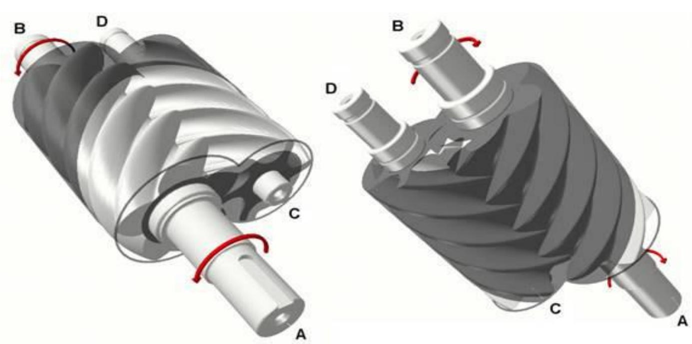

Fig. 1. ***Screw Compressor Rotors.[1]***

Its minimum value is zero when the second rotor is finally in complete meshing contact. The two mating rotors actually make up a pair of helical gears, with the protrusions acting as teeth. These are normally described in the fig. 1 as the male or main rotor (A-B) and the female or gate rotor (C-D) respectively.

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Despite these machines popularity, public knowledge of the rotor profile generation's science foundation is still confined. This subject is well covered by three textbooks, written in Russian. Sakun, 1960, provides complete information of the generation of circular and elliptic models and a Russian asymmetric profile called SKBK. Andreev, 1961, helped produce contours for rotating instruments, while Amosov et al, 1977, was the first one to show how asymmetric SRM contours could be generated. Rinder, 1979, suggested an asymmetric SRM profile in a German-language textbook. Some engineering aspects of screw compressors were announced by Konka in 1988. There are two English textbooks on screw compressors, namely on Industrial compressors, O'Niell, 1993, and Arbon, 1994, on twin-shaft compressors [2]. A significant amount of patents on multiple elements of screw compressors, especially their profiles, have been released. SRM a Swedish compressor firm and City University of London, pioneers in the studies, growth and design of screw compressors, are presenting the most significant of these. Recent progress in mathematical modeling and software modeling has boosted the quest for fresh profiles and enhanced them. These analytical techniques can be coupled to create a strong tool to analyze and optimize procedures through cumbersome tests and mistake proofs, eliminating earlier visual modifications. Hence, this rotor blade contour optimization design method has been developed extensively over the past few years and is likely to lead to further improvements in machine performance in the near future. In this paper an attempt is made to make new rotor profile using the general methods of screw compressor rotor profiling [3], [4], [5], [6], [7], [8],[9]. The procedure used to develop the rotor profile is mentioned in this paper.

2 PROFILE CONSTRUCTION

Various parameters are required to define the construction of the rotor profile shown in

Fig. *2*. Here rotor profile is constructed in transverse direction with reference to general methods of screw rotor profiling [1],

Fig. *2* shows the screw rotor profile in transverse directions. For rotor profile construction, some steps are to be followed as the profile is a set of combination of different shapes like circle, ellipse, cycloid, trochoid etc. Plotting the complete profile in a single curve is impossible due to the use of multiple curves in one shape.

Fig. *3*. shows the coordinate system of twin screw rotor, where subscript 1 represents the male rotor and subscript 2 denotes female rotor. According to General Theory of Rotor profiles [3-9], initially the male rotor curve is drawn using the following equations

$$X_d = X(\phi) = R\cos(\phi) \quad (1)$$

$$Y_d = Y(\phi) = R\sin(\phi) \quad (2)$$

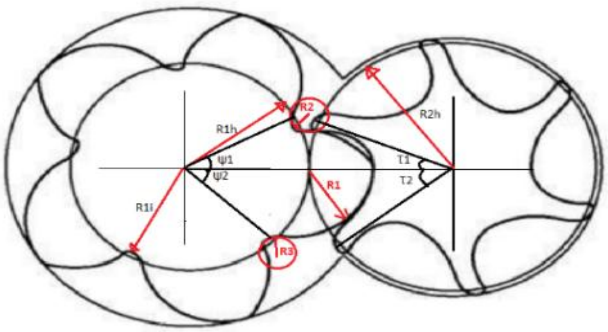


Fig. 2. ***Parameters of profile***.

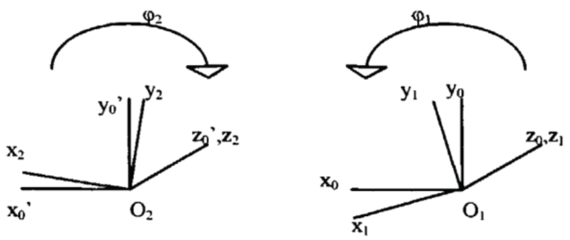


Fig. 3. ***Coordinate systems of the two Rotors.***

The female rotor is the conjugate of male rotor so conjugate curve is drawn by the following equations,

$$X = x(\phi,\psi) = -A\cos\left(\frac{\psi}{i}\right) + X_d(\phi)\cos k\psi + Y_d(\phi)\sin k\psi \quad (3)$$

$$Y = y(\phi,\psi) = A\sin\left(\frac{\psi}{i}\right) - x_d(\phi)\sin k\psi + y_d(\phi)\cos k\psi \quad (4)$$

Using this theory, various types of rotor profiles are made and they are classified as below and shown in Fig. 4.

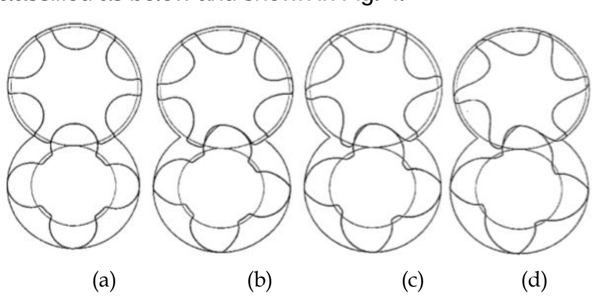


Fig. 4.(a) Circular Profile (b) SRM "A" Profile (c) D Profile (d) N Profile

3 New Profile

From the above given equations (1), (2), (3) and (4), a new asymmetric profile is proposed using multiple curves as shown in

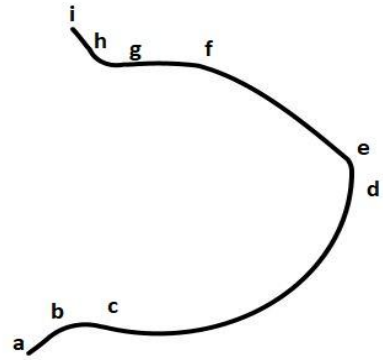


Fig. *5*, which includes a circle at its leading edge (c-d). This circle at the leading edge ensures proper force transmission between the two rotors. At trailing edge, a combination of ellipse and sinusoidal curve is used (e-f and f-g). In the case of N rotor profile trochoid is used on trailing edges of the rotor while in the newly developed profile, the combination of ellipse and sinusoidal curve is used which enhances breathing capacity of the profile. The newly developed profile is shown in the fig. 5. Using the above curve 3D model of the Male rotor generated is as shown in the Fig. 6. Curves used in newly developed profile are mentioned in the Table I.

Table I

Curves	
Segment	***Curve***
ab	Circle
bc	Circle
cd	Circle
de	Circle
ef	Sine Curve
fg	Ellipse
gh	Circle
hi	Circle

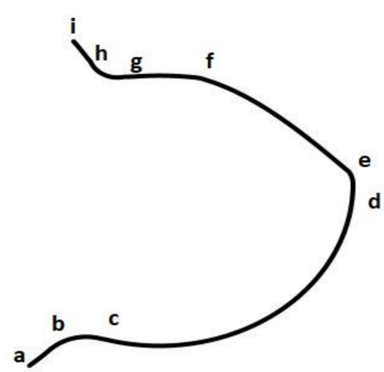


Fig. 5.Newly Developed Rotor Profile

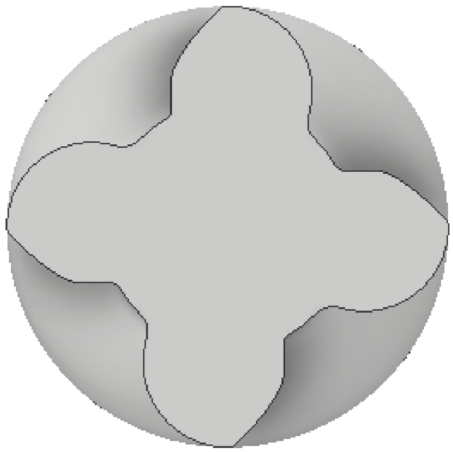
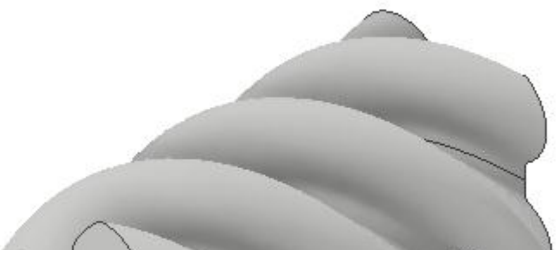


Fig. 6. 3D model of the Male rotor for the Developed profile



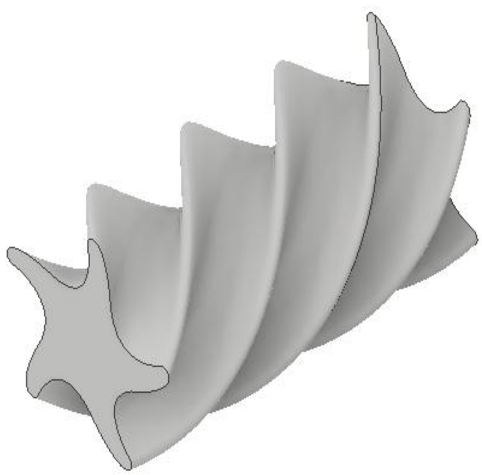


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Fig. *11.Isometric view of the 3D model of the rotor*

4 Performance Evaluation

Screw compressor manufacturing within tolerances and testing of its performance is costly and time consuming. So to predict the performance of screw rotor profile empirical [16], analytical and numerical methods are used. The performance of newly developed rotor profile is compared with "N" rotor profile, when the rotor dimensions and operating conditions are kept same. A proprietary software, DISCO and SCORG, designed and developed by City, University of London for the performance estimation of different rotor profiles is used for comparison. A flow chart which demonstrates the process for the use of the software is Fig. 12. For the For the comparison of newly developed rotor profile with available theoretical prediction of "N" rotor profile, the rotor lobe combination and L/D ratio used is presented in Table II. The working condition of the rotor profile are mentioned in the Table III.

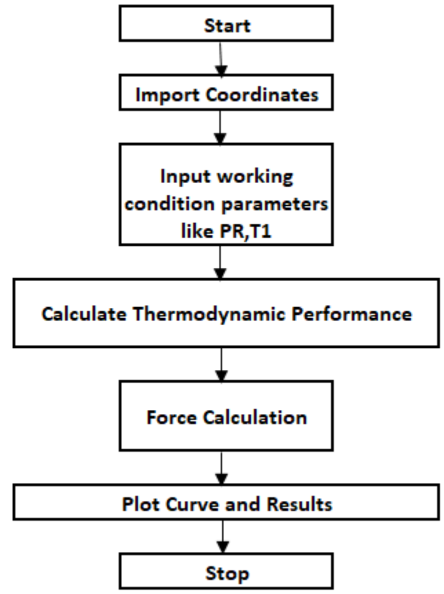

Fig. 12. Flow of the Process

Table II

Sr No.	Parameter of the Rotor	
	Symbol	***Value***
1	Z_1	4
2	Z_2	5
3	L/d	1.55

Table III

Sr No.	Working Conditions	
	Parameter	***Value***
1	Pressure Ratio	8.5
2	Volume Index	4.2
3	Suction Inlet Temp (0c)	35^{o} C

5 Results and Discussions

5.1 Thermodynamic Performance

5.1.1 Volume v/s Rotation Angle

Fig. 13. Volume v/s Rotation Angle

Use of combination of sine curve and ellipse instead of trochoid on the trailing edge the new developed profile shows higher displacement volume as shown in

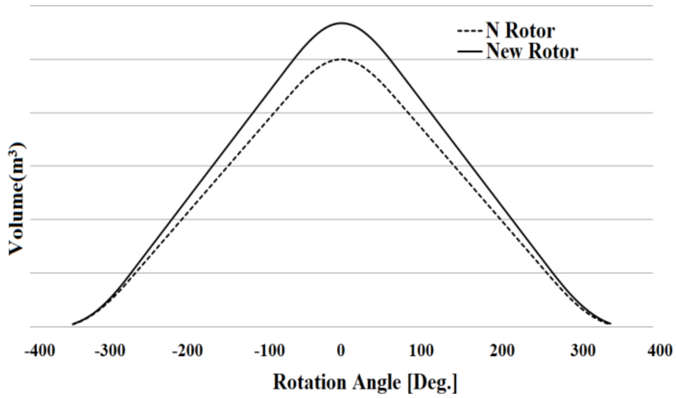

Fig. *13*

5.1.2 Pressure V/s Rotation Angle Diagram

Pressure is built in with respect to the rotation angle. Comparison of both the Rotor Profile is shown in

Fig. *14*.

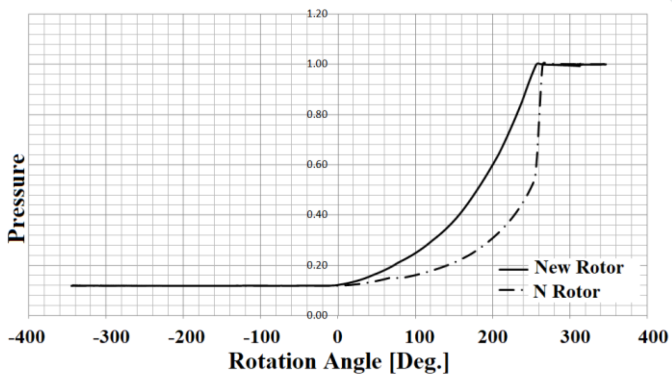

Fig. 14.Pressure V/s Rotation Angle

The pressure distribution over the rotor profile generated using SCORG is shown in

Fig. *15*.

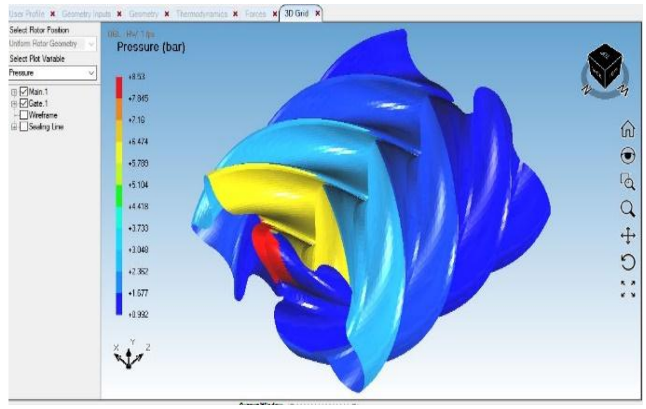

Fig. 15 Pressure Distribution over the New Developed rotor Profile

5.1.3 Specific Power

The specific power of the screw compressor is defined as the shaft power required by the compressor for unit mass flow. The newly developed profile has lower specific power as compared to "N" rotor profile and it decreases with speed of rotation

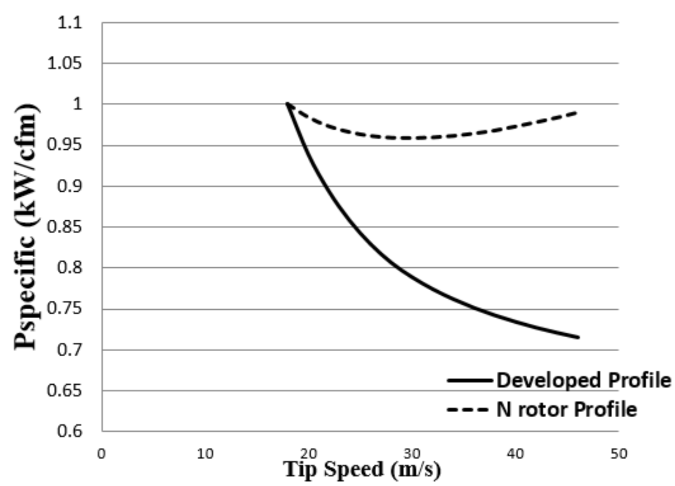

Fig. 16. Specific Power

5.1.4 Volumetric Efficiency

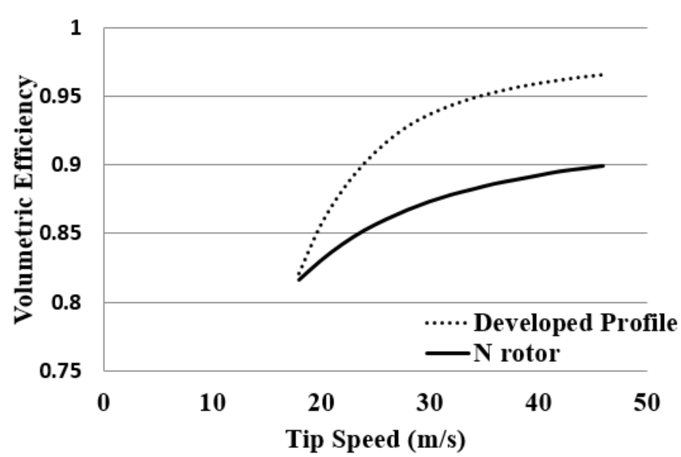

Fig. 17. Volumetric Efficiency

As from

Fig. *13* it can be seen that the displacement volume of the new profile is greater than the N rotor profile. So, at the same speed and the same dimensions shown in Table I and the working condition mentioned in Table II for both the profile, new profile will have higher displacement volume and thus will deliver higher volume flow rate The comparison of volumetric efficiency is shown below in

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Fig. *13*, the displacement volume of the new profile is higher than "N" rotor profile. So, at same speed and rotor diameter, new rotor profile gives higher volumetric flow rate. The comparison of volumetric efficiency is shown in

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5.2 Forces on the rotors

5.2.1 Radial Force

The meshing of rotors and operating pressure and speed, generate to axial and radial forces. The radial forces are transferred the housing through radial/roller bearings while axial forces are transferred through axial/ball bearings. The comparison of radial forces generated by new profile and "N" rotor profile is presented in

Fig. *18* and

Fig. *19* respectively for a rotor tip speed of 24 m/s. The comparison shows that the radial forces arising in both the rotor profiles are in almost the same

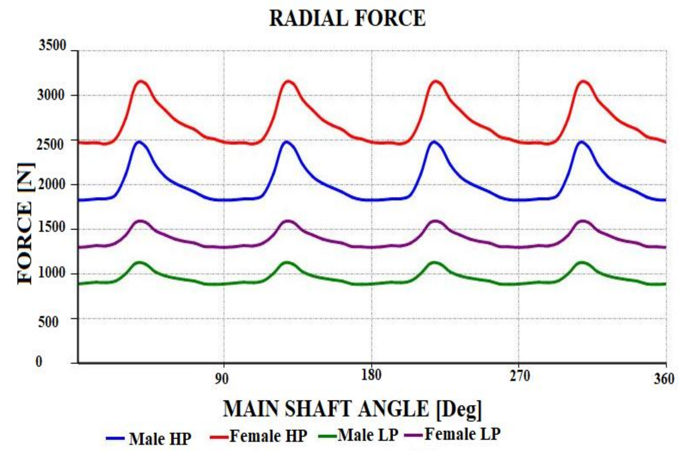

Fig. 18 Radial force for New Profile

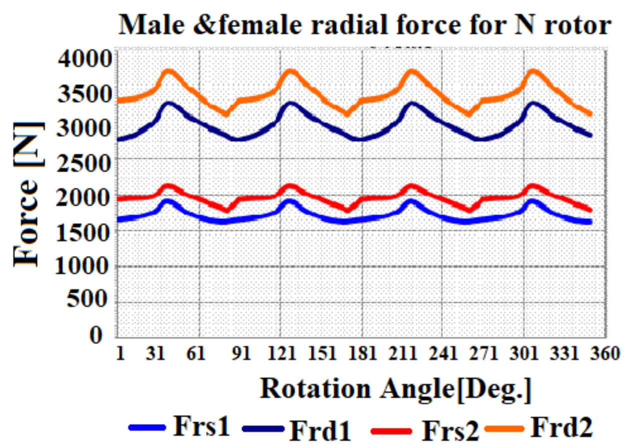

Fig. 19Radial force for N Rotor Profile

5.2.2 Axial Forces

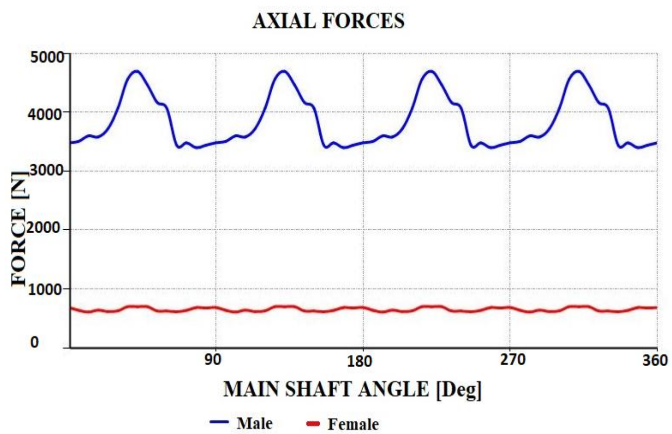

Fig. 20Axial Forces in new profile

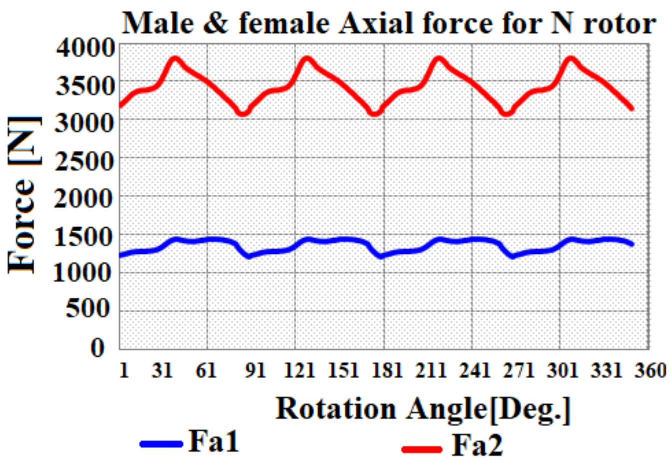

Fig. 21. Axial Forces in N Rotor Profile

The axial force arises due to the pressure arising at the discharge end. The axial forces arising in new profile is shown in Fig. 20 and axial forces in "N" rotor profile is shown in

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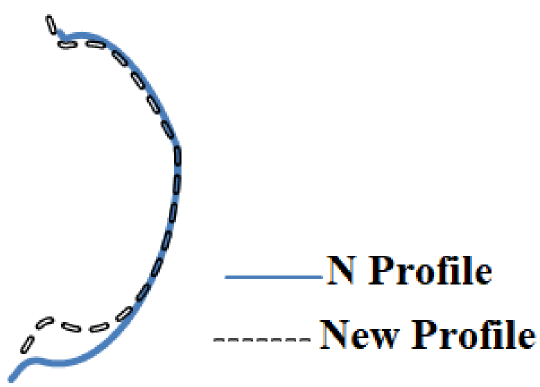


Fig. 8. Comparison of developed profile with N Rotor Profile.

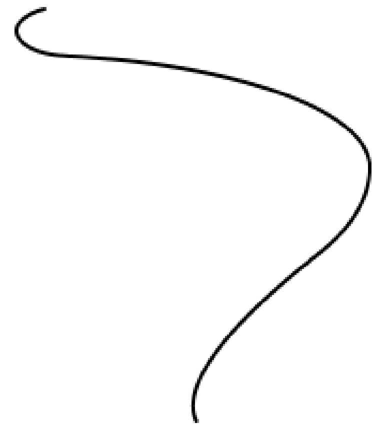


Fig. 9. Female Rotor Curve

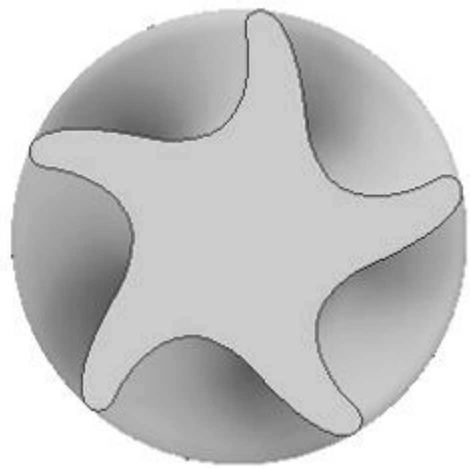
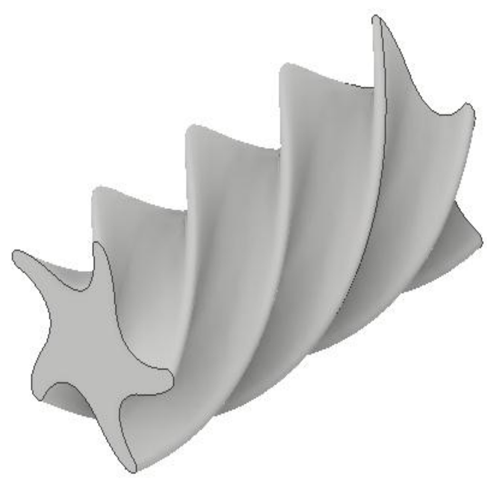


Fig. 10.3D model of the female rotor for the Developed profile



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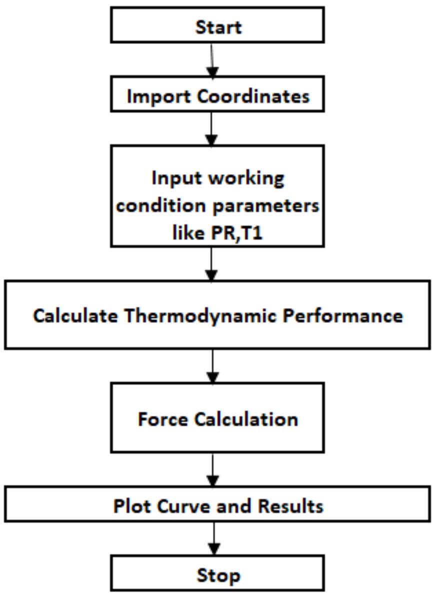

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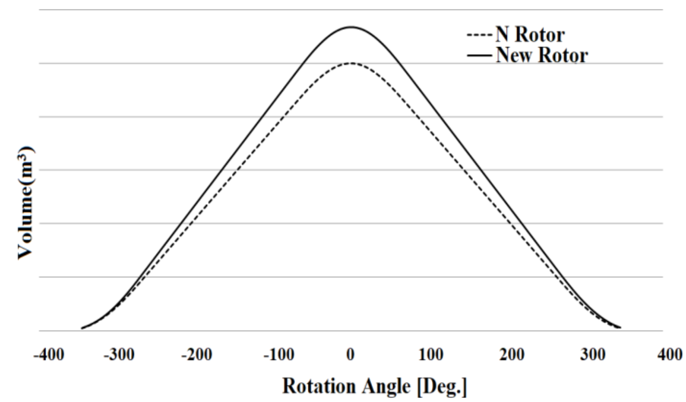

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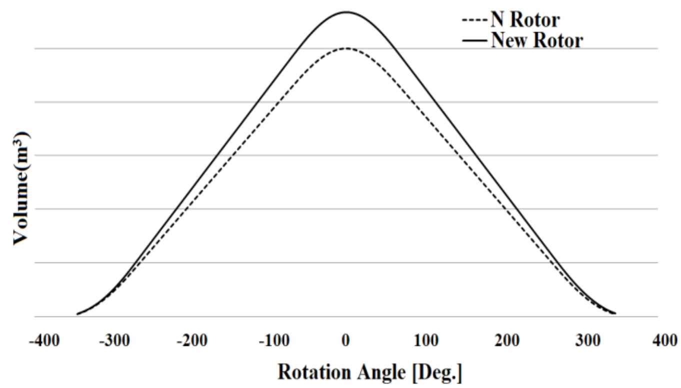

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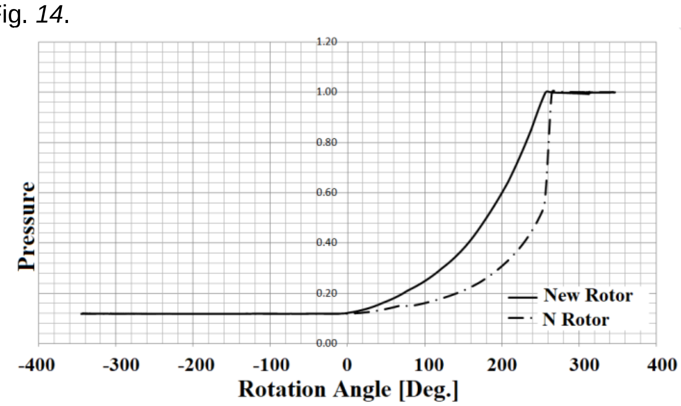

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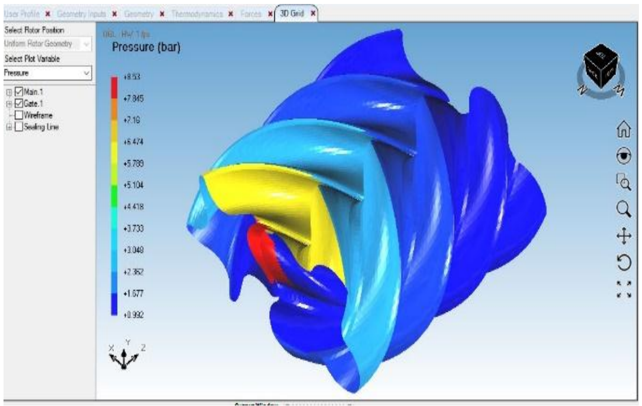

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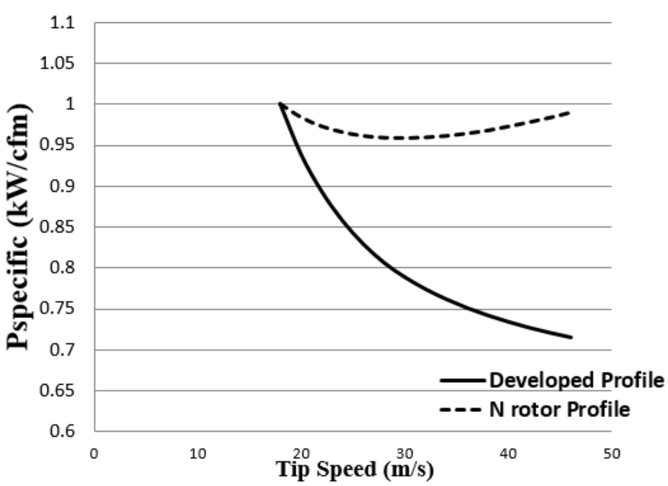
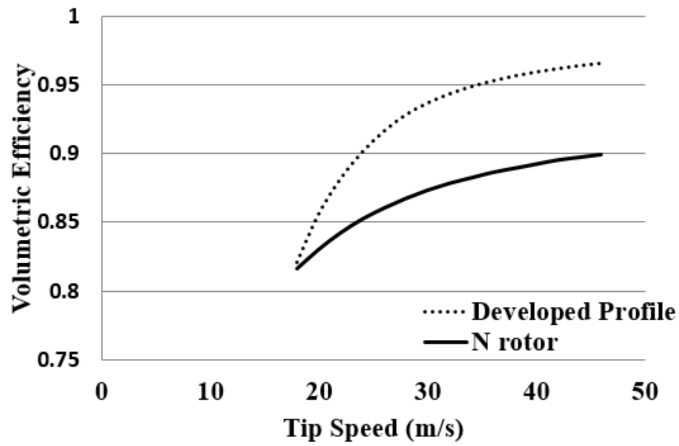


Fig. 16. Specific Power

5.3.4 Volumetric Efficiency



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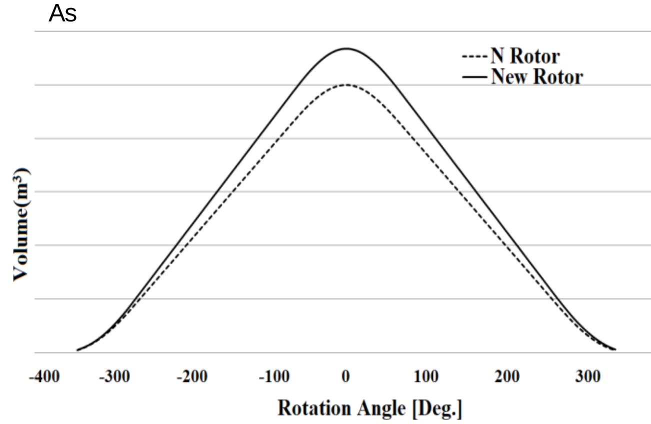

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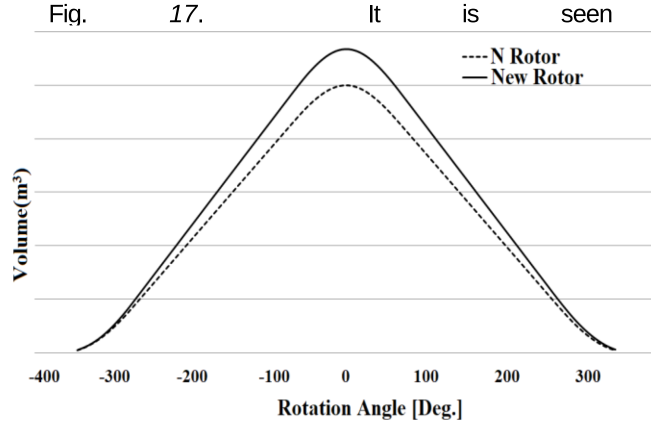

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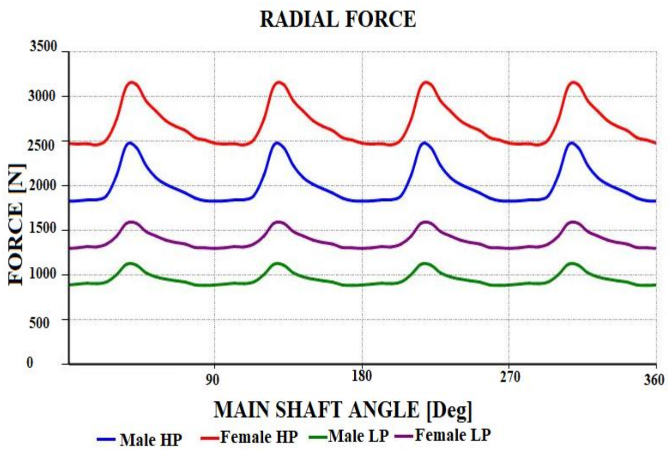

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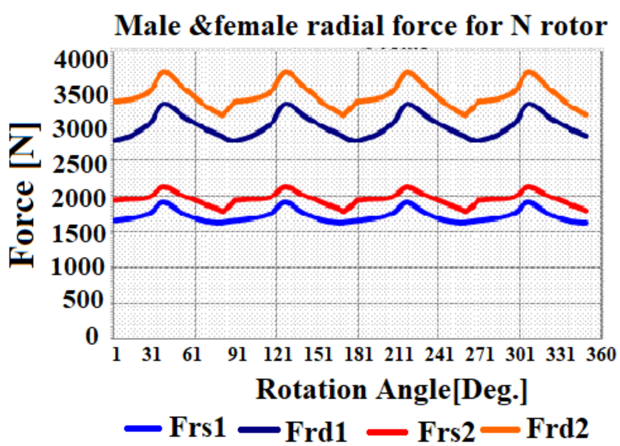

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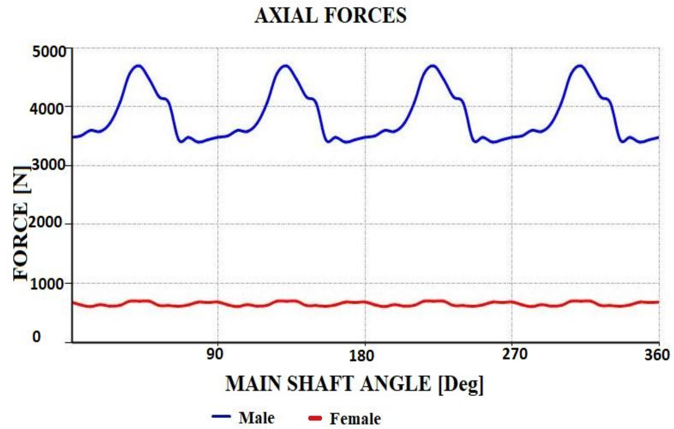
5.4.2 Axial Forces



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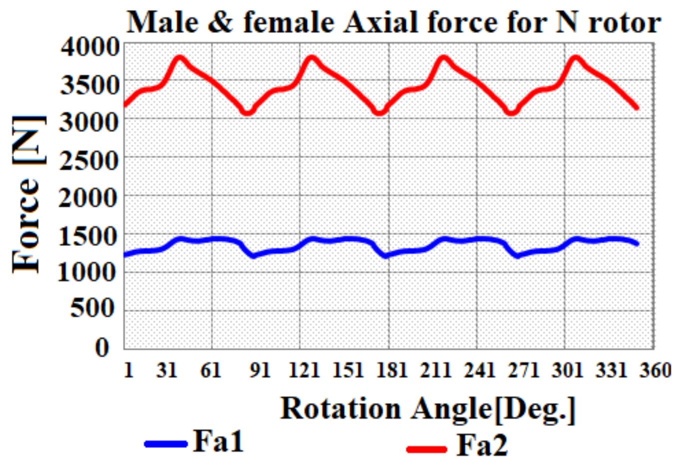

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4 Conclusion

Through the investigation of available data on the screw compressor rotor profile, a new rotor profile is proposed and the comparison with standard N rotor profile is given in the paper. A major difference in both the profile is the trailing edge curve. Use of a combination of sine curve and ellipse shows a better result than the trochoid curve. Change of trochoid to combination of ellipse and circle results increased displacement volume for the newly developed profile. Thus at the same power input the newly developed rotor profile will result have higher discharge thus resulting to decreased specific power for new Rotor profile. The volumetric and adiabatic efficiencies at tip speed ranging from 18m/s to 46 m/s are much better than N Rotor Profile.

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