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Investigation of the Tooth Profile for Screw Compressor Considering Leakages by Simulating in CFD

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Abstract: *The positive displacement twin-screw compressor is used to compress gases to modest pressures. Internal leakage is a fundamental issue with screw compressors. The design and performance are impacted by the phenomenon of leakages across various clearances. The experimental processes are costly and time-consuming to manufacture screw compressor. The performance of various tooth profiles of screw compressor can be determine by utilising CFD. The aim of study is to develop screw compressor profiles in solid modelling software and numerical analysis in CFD. The change in the flow rate, clearance, and tooth profile and by considering other parameter, like pressure velocity and temperature, the clearance between the male-female and casing can be adjusted and hence the scope of improvement considering the actual phenomena taking place within the screw compressor. By modifying the input parameter the losses between the internal leakages and the functional issues with the screw compressor can be minimize.*

Keywords: Tooth profile, Computational fluid dynamics (CFD), leakages, screw compressor.

I. INTRODUCTION

In a chamber, positive displacement compressors pull in and collect a quantity of air. The air is then compressed by reducing the chamber's capacity. Positive displacement compressors include scroll compressors, rotary screw compressors, rotary vane compressors, and reciprocating piston compressors.By reducing its volume (as in the case of positive displacement machines) or by giving it a high kinetic energy that is turned into pressure in a diffuser (as in the case of centrifugal machines), gas compressors are mechanical devices that are used to increase the pressure of gas or vapour. Industries are enormous and expensive, but they are frequently necessary for the ongoing operation of the whole process in which they are involved. Therefore, their operational dependability is at least as crucial as their effectiveness. Significant progress has been achieved in the design and production of both the major parts of these types of machines, including the rotors and bearings, as well as their minor parts during the previous several years.

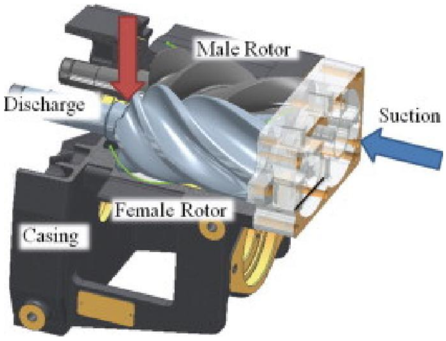

Figure 1 twin screw compressor

For the operation of the water-injected twin-screw compressor, a thermodynamic model with a thorough leakage flow model was constructed. All in all, heat and mass transfer between injected liquid and compressed vapour as well as leakage along various pathways were taken into account. For all studied operating circumstances, the projected efficiency is accurate to within 5% of experimental data [1].Helical rotors with adequate porting in a gerotors-type arrangement have been found as having possibly lower rotor contact forces and greater port areas than non-helical rotors in compressor applications. The shape, size, and wrap angle of the rotors determine important performance characteristics of a gerotortype screw compressor (volume, port flow areas, and tip leakage areas). The geometric analysis findings are compared to a traditional twin-screw compressor, and they indicate that the gerotor-type screw compressor may achieve greater axial port areas and smaller tip leakage area across the majority of the compression cycle [3]. Zhang and Hamilton investigated the impact of key geometric parameters such as compression volume curve, sealing line length, flute area, wrap angle, and blowhole area. To create the production software, mathematical models of these factors were created. Tang and Fleming [2] investigated the influence of relative blowhole area and relative contact line length on performance and proposed various strategies for optimising geometrical parameters [4-5].X.he et all illustrated the original and new twin-screw compressor flow field modelling models are developed. The FLUENT programme is used to investigate the twin-screw compressor's pressure and velocity distribution law at operational speed. The simulation findings serve as the foundation for subsequent rotor profile design refinement [6]. C wanget all Calculated and discussed to determine the implications of each leakage channel on the compressor's performance, including volumetric efficiency, indicated power, and specific indicated power at various rotational speeds. The compressor design might be improved using the prediction of leakage distribution [7]. Stotic et all have considered a delivery pressure of 8 bar (abs), the minimum measured specific power input was 5.6 k W m -3min -1 [8].Spille-Kohoffet find for a dry screw compressor compressing air from 1 bar and 300 K to 3 bar and around 470 K are presented in this study. The models account for the heating of the rotors and incorporate axial and radial gaps. The principal torque (on the male rotor) is around 18 Nm, and approximately 26 kW of total mechanical power is required. Finally, both rotors experience heat expansion in both the axial and radial directions [9]. Raneet all, demonstrates how it may be expanded even further to encompass rotor designs that aren't typical, including those with variable lead or profile variation or even internally geared machines with conical rotors. Other configurations made available by this development include dual lead, high wrap angle rotors for very high pressure differences and vacuum applications, multiple gate rotors to boost volumetric displacement, and multiple gate rotors. Its utility for thorough flow analysis and design is illustrated by a case study of a water-injected twin screw compressor [10]. The predicted indicated power from CFD projections [11] and the observed shaft power agreed well. Phase mixing, oil distribution, gas-to-oil heat transfer, as well as impacts on sealing owing to high oil concentration in leakage gaps, were all clearly depicted [12].

Modeming of design and mesh generation

The screw compressor is made up of the 4/6 lobes where the symmetric profile with fewer edges than a male or female profile is designed, which has eight or twelve edges, respectively. It is made up of arcs in a circle. The benefit of having fewer edges is that meshing proceeds more quickly. The majority of screw compressors are still built with 4 lobes in the main rotor and 6 lobes in the gate rotor, with both rotors having the same outer diameter.

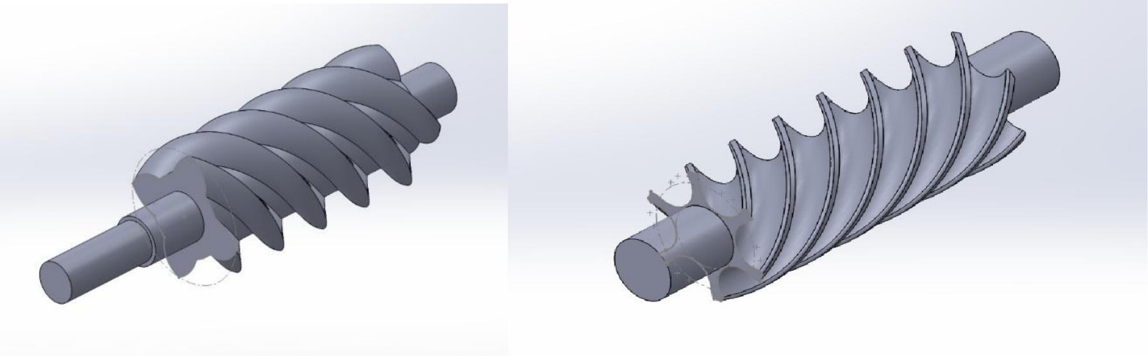

Figure 2 Design of male and female lobes

This arrangement is a compromise that has benefits for both dry and oil-flooded compressor applications and is utilised for air, refrigeration, or process gas compressors. Other combinations, such as 5/6 and 5/7, and more recently 4/5 and

3/5, are becoming increasingly common. Higher compressor pressure ratios are possible with five lobes in the main rotor, especially when paired with bigger helix angles. The 4/5 configuration has shown to be the ideal combination for oil-flooded applications with modest pressure ratios. Because of its high gear ratio, the 3/5 is used in dry settings.Any compressor case's primary duties, regardless of type, are to sustain all bearing seals and stationary internals to offer effective process fluid containment.For the screw compressor two different design of tooth profile is considered for the analysis where for the first design tooth radius is 30 mm, 4/6 lobes with the inner diameter of 30 mm and outer diameter of female is 101.4 mm and for male it is 105.8 mm, total length of screw is 170 mm. The wrap angle of male is 300mm and for female is 200mm. with axis distance 80mm. The wrap angle of male is 300mm and for female is 200mm. with axis distance 80mm.

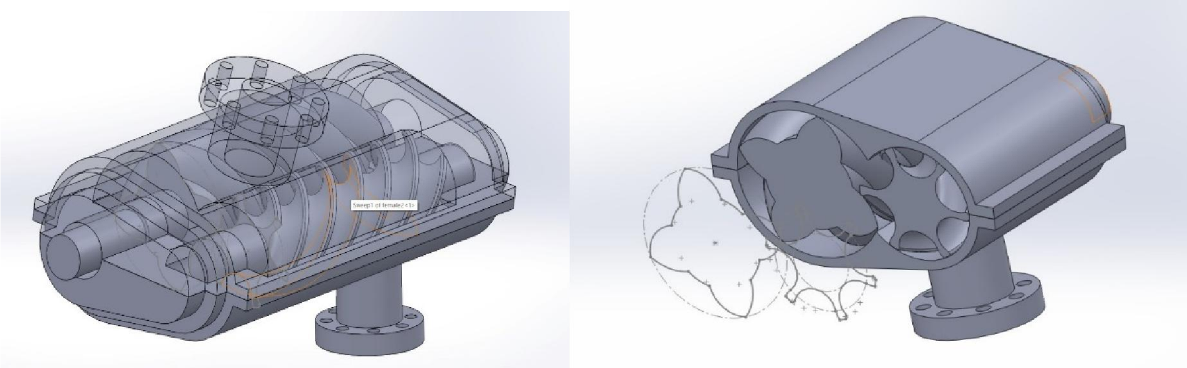

Figure 3 casing with male and female assembly

Thetooth profile the radius is 78 mm. the male outer diameter is 300mm and female is 250mm.The most crucial component of FLUENT is meshing. The pre-processor offers several distinct mesh scheme types. It is quite difficult to generate chaos with fewer grids and less skewness, though. The next step is to mesh the volume after the geometry has been produced.

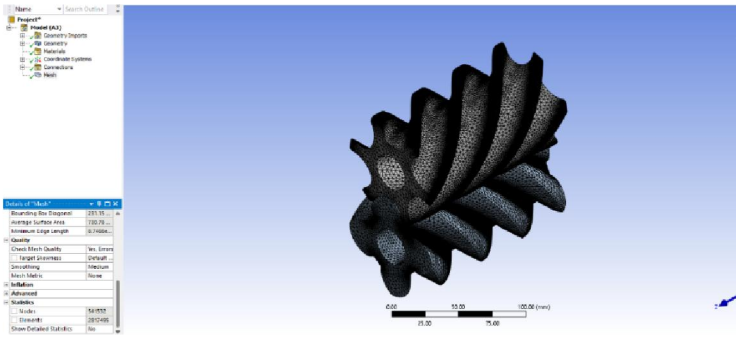

Figure 4 mesh generation

The element quality and aspect ratio of a suitable mesh are examples of quality criteria that match your demands (analysis kind, level of accuracy, time). The best mesh may be obtained by knowing your geometry and employing controls; after all, a better product design results from this.

Grid generation test

In CFD, "grid generation" refers to a collection of methods for creating a numerical mesh throughout the whole system that will be modelled. The accuracy and resolution of the simulation results will be determined by the grid used for CFD simulations, which will also have an impact on the calculation time and amount of detail in the results.

Type of geometries	Element order	Total number of elements	Total number of nodes
Hexagonal	Linear	411429	2029404
Hexagonal	Quadratic	2987141	2028731
hexagonal	tetrahedral	5728154	30335055

Table 1 grid generation test

In comparison to tetrahedral elements, hex or "brick" elements typically produce more accurate results at lower element counts. The optimum option may be tetrahedral elements if the geometry is intricate. There are further methods that can offer you more mesh control, but these default or automated meshing techniques could be sufficient

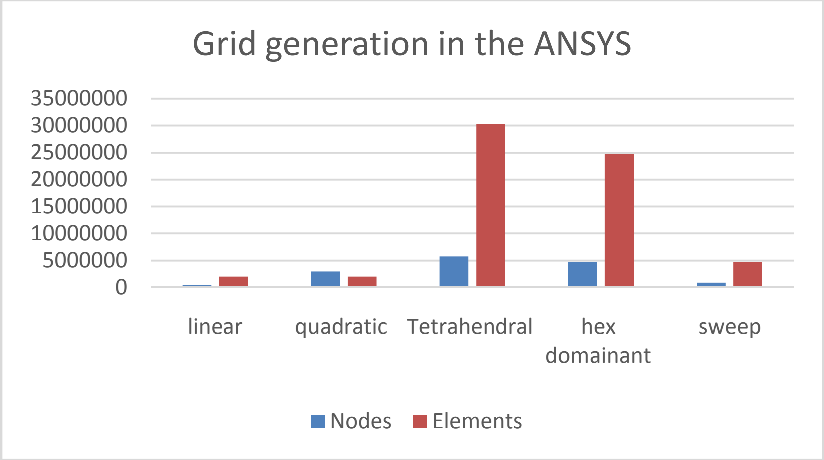

Figure 5 comparison of the generated mesh

Boundary circumstances:

The compressor's provided faces are male and female in their respective ways, and the intake and outflow conditions are pressure inlet and pressure outlet, respectively.

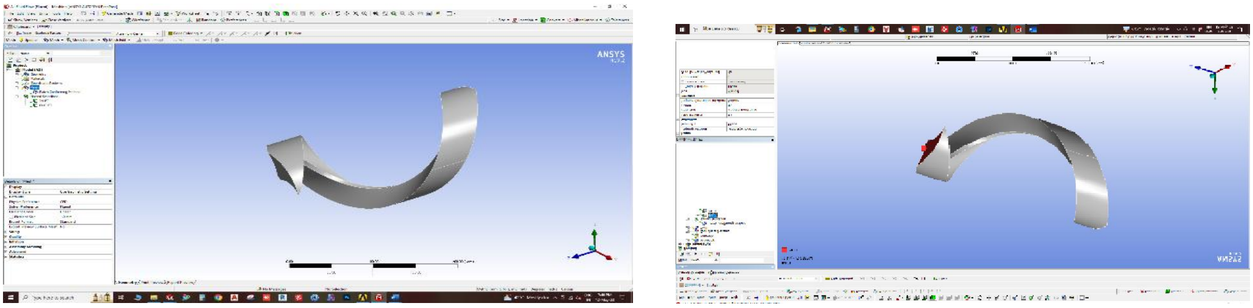

Figure 6 boundary condition of the flute

Analytical solving technique

In general, FLUENT solves the governing integral equations for mass and momentum conservation, as well as (where suitable) energy and additional scalars like turbulence and chemical species. Typically, a control volume-based technique is used, which consists of the following steps: Dividing the domain into discrete control volumes using a computational grid; and integrating the governing equations on the individual control volumes to construct algebraic equations for discrete dependent variables ("unknowns") such as velocities, pressure, temperature, and conserved scalars. Linearization of the discretized equations and solution of the resulting linear equation system to get updated values for the dependent variables. For the analytic solution of the screw compressor the lead angle helix angle are considered.

The lead angle and helix angle

$Rp^1 = (N1/ N1+N2) * Dcc$……………………………………………………………...…1.1.1

$Rp^2 = 1-rp^1 = (N1/ N1+N2) * Dcc$………………...…………..1.1.2

$A1 = r0^1 - Rp1$……………………………………….……………………………………1.1.3.

$A2 = r0^2-Rp^2$ ………………....………………………………………………………...1.1.4

$\lambda= \tan^{-1} (L_{d1}/2\pi r_{p1})$…………….……....………………………………………........1.1.5

$\beta = (\pi/2) - \lambda$.……………………………………………..…………………………...1.1.6

Some remaining parameters of the female rotor are defined based on the constant speed ratio, M12, between the male and female rotor as followes

$m_{12} = (w_1 / w_2) = (\phi 1 / \phi 2) = N2/N1$ ……………………………………………...…………..….1.1.7

$L_{d2} = m_{12} \, L_{d1}$ …………………………………………………………………………...……..1.1.8

$T_{w2} = T_{21}/m_{12}$ …………………………………………...……………………………………..…1.1.9

$P_{1} = (L_{d1}/2\pi) = (L_W/T_{w1}) = Rp^1 \tan \lambda = (Rp^2/ m_{12}) \tan \lambda = (p2/ m_{12})$ ……………...……..1.2.0

The constant angular velocity ratio between rotors, the relation between these two angles is

$\phi 2 = (\phi 1/ m_{12})$ ………………………..…………………………………………….…...1.2.1

The envelope theory is then applied to find the profile of the rotor. The male rotor profile is transformed onto the coordinate system by the transformation matrix

$$M_{21} = M_2 * M_1 = \begin{matrix} cos\phi & -sin\phi & 0 \\ sin\phi & cos\phi & 0 \\ 0 & 0 & 1 \end{matrix} = \begin{matrix} cos\phi & sin\phi & C \\ -sin\phi & cos\phi & 0 \\ 0 & 0 & 1 \end{matrix} \text{ ……………..……………………1.2.2}$$

The comparison of machine performance uses integral values as specified. Therefore, a theoretical study of the twin screw compressor may be done

CFD analysis in ANSYS

It is assumed that the leakage is unaffected by the flow caused by the rotor motion and that the leakage may be measured by a steady flow at a variety of rotor locations with a certain pressure drop. Even though FLUENT offers a reasonable level of physical model accuracy, experimental data is still required to verify the model's numerical output.

Moreover the atmospheric pressure and temperature is considered where it found that the modelling is perfect to get analysed on CFD. For the modelling the tooth profile, pressure and temperature is changed as per the requirement with considering the two parameters of the tooth profile.

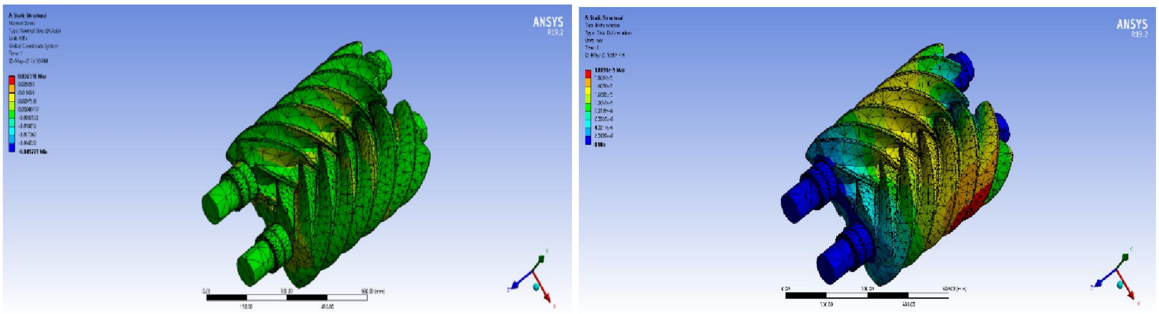

Figure 7 static structure analysis and total deformation of the twin screws

The impact of constant (or static) loading on a structure is determined via a static structural simulation. A structure's stress, strain, and deformation can be examined under various loading scenarios. This aids in the early identification of weak spots with poor strength and durability to assist designers prevent costly failures. Inertia and damping effects, such as those brought on by time-varying loads, are not included in a static structural analysis, which analyses the impact of constant (or static) loading conditions on a structure.

The most frequent use of static structural measures in verification and/or validation analysis does not imply that these measurements are always simple. These have evolved significantly to include complex internal interactions (contacts, joints, and composites), links to other fields of physics (fluid structure, electromagnetic structure), and other factors. Static simulation may provide significant performance insight into component and assembly performance.

Deformation in physics and continuum mechanics is the change in a body's reference configuration to its present configuration. A configuration is a collection of all the lotions of a body's constituent components. External loads, intrinsic activity (such as muscular contraction), bodily forces (such gravity or electromagnetic forces), changes in temperature, moisture content, chemical reactions, etc. can all induce deformation.

In this case, it was found that the entire deformation of the lobes was advantageous for the normal. Stress under the load of the normal stress. While it is also discovered that the total lobe distortion is below the safety factor, certain lobe segments have somewhat larger deformation factors that are acceptable for the design process.

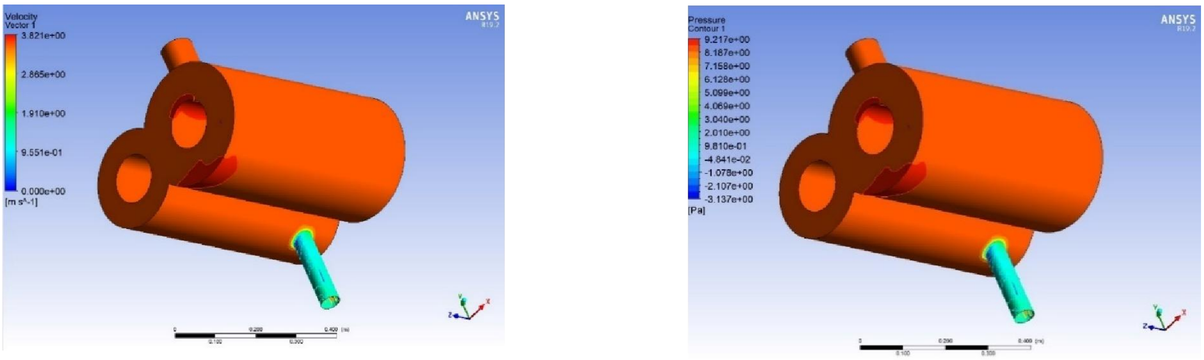

Figure 8 analysis of tooth profile of model-1

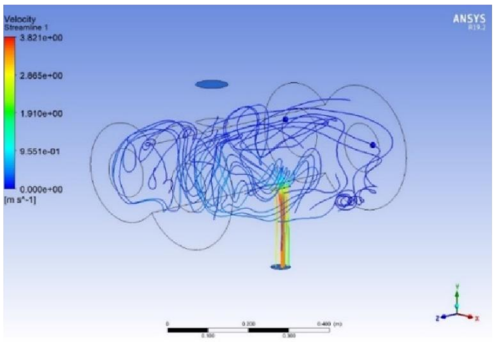

Figure 9 velocity streamline

The CFD analysis is done with considering the atmospheric pressure which is around 101,325 Pa and the velocity is considered as in the ms^{-1}. The tooth profile is considered as radius of 30 mm. Based on this the result of the screw compressor found that the pressure loss in the compressor is not more than the 1.5 PA which is considered as a good compressor for the industrial propose. Apart from this the velocity is also found around 2.865 to 3.821 ms^{-1}. The velocity stream line for the flow of the air have a smooth flow in the compressor. There is a generation of the vacuum in the compressor casing but neglecting that the flow is found smoother till the end of the outlet.

Analysis of the male and female flute

The male and female flute, the one tooth of the screw lobes are analysed. The tooth profile radius is 28 mm used for the CFD analysis. The meshing is generated of 1 mm tetrahedral meshing which gives the best meshing nodes and edges for the design considering the patch conforming algorithms. The boundary condition for the flute is shown in the figure 6. From the results it has been found that the temperature does not have a higher variation in the profile. The pressure at the initial stage found to be at low and then started continuous rise in the tooth profile. The highest pressure difference is found at the outlet of the flute where the high pressure air is passed to the outlet section.

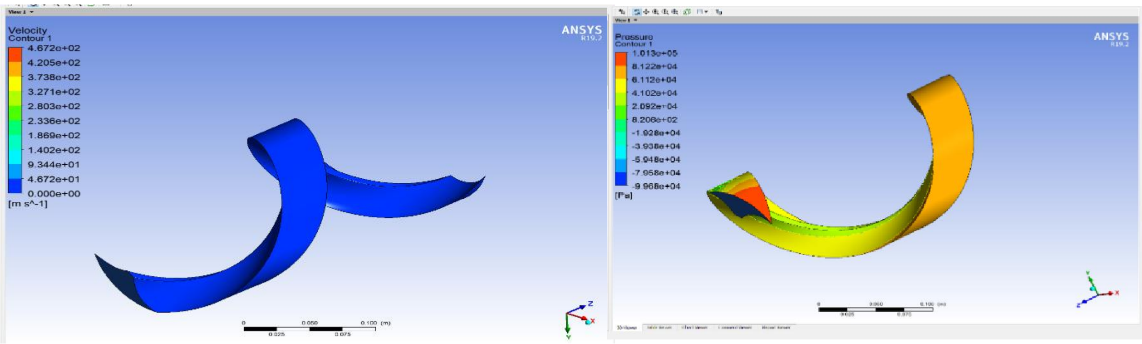

Figure 10 CFD analysis of the velocity and pressure for male flute

The velocity stream line is from the CFD analysis found that it have a smoother flow of the air form the flute. Almost more than 90% of the air is flow form the tooth profile with the increment in the pressure.

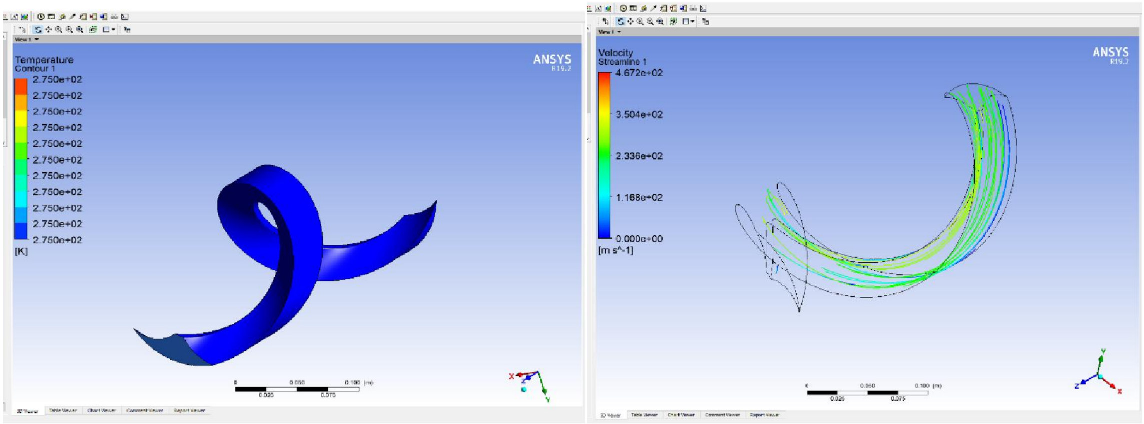

Figure 11 CFD analysis of the temperature and velocity stream line for male flute

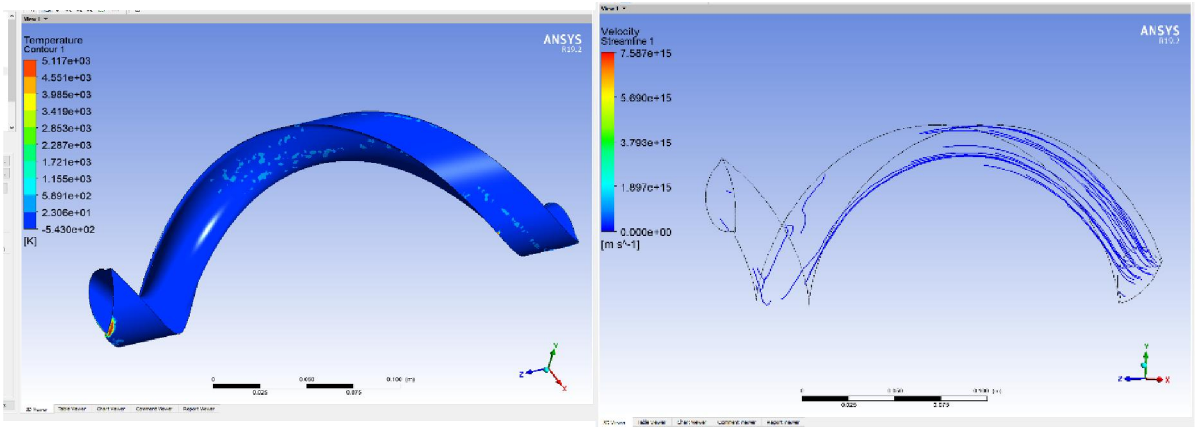

Figure 12 CFD analysis of the female flute

For the female flute the more improvement found in the flute. The velocity and the pressure flow in the flute is high than the male flute. The stream line of the velocity and applied pressure on the flute have a more accurate smooth flow then the male and more pressure ratio found. This can be conclude that the female lobe have a less losses in the casing during the operation than the male lobes. The constant physical force that anything (a fluid like air) in contact with an item applies to or against it is known as pressure. If the fluid isn't flowing or if you are moving with the fluid, you are under static pressure. You would feel the same amount of air pressure from all sides.

On the basis of the two type of the rotor tooth profile for the 30 radius profile the pressure and temperature difference is compared with male and female profile as shown in the graph

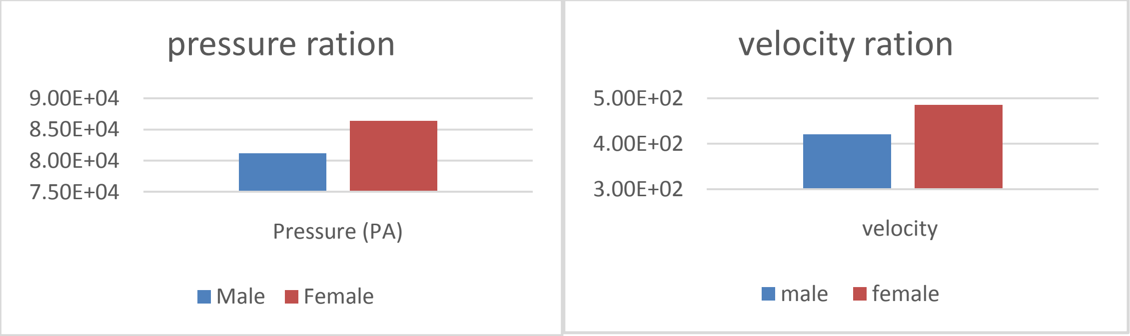

Figure 13 comparison of the screw compressor male and female lobe

Now, with the consideration of the parameters of the compressor where the air is used as a medium for the twin screw compressor. In this profile tooth radius is taken as 78 mm. The pressure is around the 1 bar to 25 bar where it is found that the pressure increased from 1 bar to nearly 15 bar. In this the condition of the tooth is found smooth with the flow of the air.

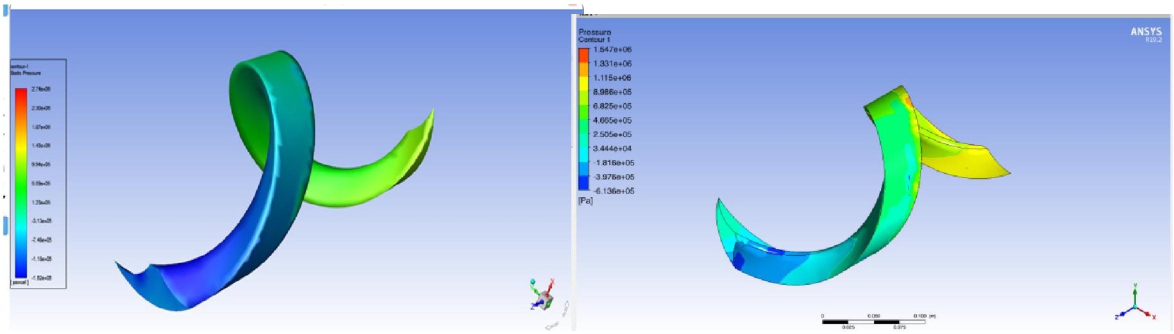

Figure 14 Male and female pressure difference

Apart from this, the total temperature is choose is around 25 c to 150 c, the flute has shown at first stage the normal temperature ratio around 1c to high at 150 with the sudden change in temperature. The static pressure is smooth as we have found in the result we obtained. The pressure contour is have a less flow ratio in comparison to the female, as shown in the figure. The female flute has shown the good pressure ration and have a less losses from it, the losses ration in comparison to the male is not more than the 2 bar. The temperature ration is also less than the male rotor where the results shows that the overall losses in the twin screw compressor is not higher in ration and the overall efficiency is also higher in this model.

The performance of the female and male rotor have a good efficiency, the changes during the analysis in male and female is also found satisfactory which gives the best result to the screw compressor.

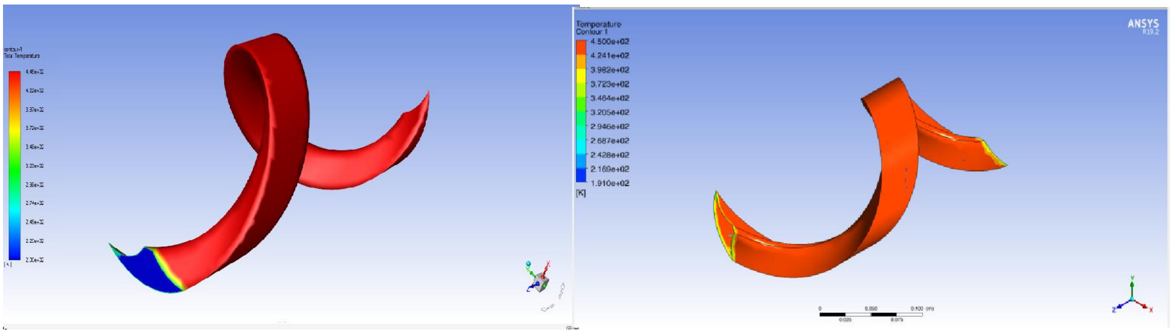

Figure 15 Male and female temperature difference

With the other tooth profile of the screw compressor the comparison of the rotor is generated where it is found that the female have less losses then the male one. The inlet and outlet losses in the compressor is found satisfactory. Apart from this the casing of the compressor have is considered for the losses with the male and female of the twin screw compressor.

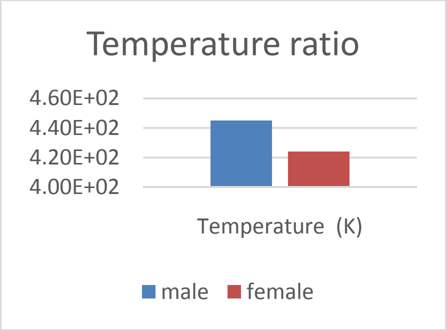

Figure 5.13 comparison the temperature of both the lobes

Degree of the tooth profile	Male	Female
0	1.25E+05	3.44E+04
90	6.34E+05	4.665E+05
180	9.82E+05	8.966E+05
270	1.87E+06	1.331E+06
360	2.25E+06	1.574E+06

Table 2 the male and female pressure difference at the different profile angle

Based on the obtained results the graph shows the variation of the pressure and velocity from the 0 to 360 degree. Where this shows the variation from the inlet to the outlet of the screw compressor. For the case of moderate over compression, pressure and temperature plots were shown for an arbitrary rotation angle. Here, the mass flow at the outlet showed a strong oscillation due to the periodic opening of the chambers. The usage of the thermal expansion results as a deformation of the rotors as well as the casing will be used in ongoing projects. Furthermore, additional work is needed to simulate oil or water injected screw compressors since multi-phase flows require very high mesh qualities for stable simulations.

II. CONCLUSION

In this paper, the rotor profiles and clearance distribution of process gas screw compressors have a significant impact on their performance. In this work, more faces are needed to create geometry if there are more edges used to create the screw compressor profile. We must cut back on the amount of edges while still optimizing the shape. The analysis of the screw compressor's various tooth profiles reveals that the female lobes experience significantly lower losses than the male lobes. Overall, there is only a 1.00e+05 pa difference across the profiles. The lobes' velocity ratio ranges from 4 to 5.0e+02 m s -1. The temperature difference from the suction, compression to exhaust overall changes is from 2.80e+02 k to 3.85e+02 k. According to the analysis, the male to female, female to casing, and male-female losses differences are all good, and the velocity streamline on the lobes exhibits a smooth flow across the whole streamline. Without taking into account tiny cavities and according to the stream line, the fluid flow through the compressor is much more effective.

The most important criterion for measuring screw compressor performance is clearance. This makes it possible for the rotors' clearances to come together and form suitable contacts with the casing. The profile and its clearance distribution must be developed in a way that will prevent damage or seizure should rotor contact which is highly likely with such tiny clearance occur.

As a result, the twin screw compressor design time and total cost might be reduced by using CFD simulation, which offers useful forecast for the component performance with small adjustment

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Nomenclatures

Computational fluid dynamics (CFD)
Vapour injection (VI)
Kinetic energy- epsilon ($K-\varepsilon$) turbulence
Pressure volume (P-V) indicator diagram
M^3/h.- meter^3/ hour
M^3/min - meter^3 / minute
Revolution per minute (RPM)
Kg/s - kilogram/second
Finite element analysis (FEA)
The centre-to-centre distance between axes of rotors, D cc,
Outer radii of male and female rotors, ro1 and ro2,
Numbers of teeth of male and female rotors, N1 and N2,
Wrap angle of the male rotor and female rotor, tw1 and tw2,
The rotor length, LW,
The pitch circle radii of the male and female rotors rp1 and rp2,
The addendums of the male and female rotors a1 and a2,
The lead of the male rotor and female rotor Ld1 and Ld2,
λ And β are the lead angle and helix angle of the rotor,
The constant speed ratio, m12,

$\omega 1$ and $\omega 2$ are the angular velocities of the male and female rotors,
P1 and P2 are the screw parameters of the male rotor and female rotor,
The contact of rotors externally in different directions $\phi 1$ and $\phi 2$,
Profile of the rotor - M21.

Abbreviation
Computational fluid dynamics (CFD)
Vapour injection (VI)
Kinetic energy- epsilon ($K-\varepsilon$) turbulence
Pressure volume (P-V) indicator diagram
Revolution per minute (RPM)
Finite element analysis (FEA)

BIOGRAPHICAL NOTES

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