



PERFORMANCE EVALUATION & THERMAL ANALYSIS OF DIESEL ENGINE POWERED SCREW AIR COMPRESSOR

Bhaswar Banerjee^a, Chidanand Koshti^b, Shrihari Bondar^c, Sethuraman Kumarappan^d

^a PG Scholar, School of Mechanical Engineering, Dr. Vishwanath Karad MIT World Peace University, Pune

^b School of Mechanical Engineering Dr. Vishwanath Karad MIT World Peace University, Pune

^c School of Mechanical Engineering Dr. Vishwanath Karad MIT World Peace University, Pune

^d R&D Department, Kirloskar Pneumatic Company Limited, Pune

Abstract

Compressed air has innumerable number of applications throughout the world. In almost every industrial sector there is a requirement of compressed air. This compression of air is achieved by using air compressors. These machineries take in atmospheric air and perform compression as per the industrial requirement. In general, there are three types of air compressors on the basis of function- Centrifugal, Reciprocating and Screw Compressors. Furthermore, compressors can also be categorized based on the driving element, i.e., the component which is driving the compressor into motion in order to carry out its function. This paper represents the performance evaluation and thermal analysis of a diesel engine driven screw air compressor carried out in two main analysis namely- Air Flow analysis and Heat Flow analysis. Also, in this paper an attempt has been made to study its impact on the performance of the compressor after modification of geometry and altering the selection of fan. A simulation has been carried out using flow simulation software for the above mentioned analysis.

Keywords: Screw Compressor; Diesel Engine; Air Flow; Surface Heat Load; Heat Flux; Flow Domain.

Nomenclature

Parameter	Symbol	SI Unit
Diameter	d	mm
Density	ρ	kg/m ³
Free Air Delivery	FAD	cfm
Pressure	P	kg/cm ²
Volume Flow Rate	Q	m ³ /sec
Angular Velocity	N	rpm
Temperature	T	°C
Velocity	v	m/sec
Heat Transfer Coefficient	h	W/m ² -K
Heat load	q	W
Heat Flux	Φ	W/m ²

1. Introduction

The positive displacement rotary screw compressor uses oil injection. When air or gas is trapped in a compression chamber, its volume is mechanically reduced, which results in a proportional increase in pressure before discharge. Each rotor, which is commonly referred to as a twin screw compressor, is made up of a group of helical lobes attached to a shaft. The female rotor is the other rotor, while the male rotor is the one in question. The male lobe rolls down the female flute like a continuous piston, encasing air and continuously constricting space. The leading strip of the male lobe reaches the female groove's shape with rotation and traps the air in the previously established pocket [2], [5], [7].

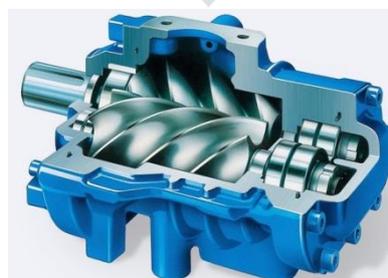


Figure 1: Oil injected rotary screw compressor

As the volume is decreased, the air is transported through the female rotor groove and compressed. When the male rotor lobe approaches the groove's end, the air that has been trapped is released from the aircend. Oil-free or oil-injected twin-screw compressors are also options. When a compressor is oil-lubricated, oil is injected. Four essential tasks are carried out by oil:

1. Cooling
2. Lubrication
3. Sealing
4. Noise dissipation [6]

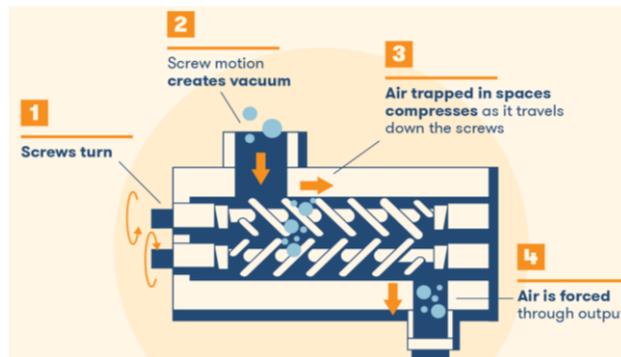


Figure 2: Layout of a rotary screw compressor

Construction and mining, general engineering, and other sectors employ KPCL Diesel Engine Driven Portable Compressors. These devices have a wide range of capacities (4.38 to 21.23 m³/min) and pressures (7 to 12.5 kg/cm²). Engines designed to give the best fuel efficiency and lowest emissions in their class power screw compressors. These engines are included in the compressors, which offer consumers long-lasting performance at a cheap cost of ownership [3].



Figure 3: Canopied Kirloskar Diesel Screw Compressor

1.1 Literature Review

A literature survey has been done in order to understand the working and schematics of a rotary screw compressor and also of diesel powered screw compressors. Further literature survey has been done in order to understand the analytical calculation methodology for estimating air flow, heat flow and heat load estimation. Following are some of the literatures taken as reference:

Junfeng Wang [1], In this paper, a full 3D transient Computational Fluid Dynamics (CFD) model of a twin screw compressor with oil injection is described in detail. Volume of Fluid (VOF) two phase flow model is used for gas and liquid phases. The rotor meshes are generated by SCORGTM grid generator and read into the CFD solver at each time step. The compression pocket is tracked using a customized post-processing technique and the pressure-volume diagram within it is also obtained and analyzed.

The simulation runs efficiently and results are obtained in about 12 hours on 24 CPU cores. The CFD prediction of air mass flow rate and indicated power has a good agreement with the experimental data measured at City University of London. Simulations demonstrate that the approaches used in this paper are robust and fast, and can be readily applied to industrial compressor systems for rapid design iterations and improvements.

Camelia POPA [2], A suitable procedure for optimization of the screw compressor shape, size, dimension and operating parameters is described here, which results in the most appropriate design for a given compressor application and fluid. Some optimization issues of the rotor profile and compressor parts are discussed, using 5/6 screw compressor rotors to present the results. It is shown that the optimum rotor profile, compressor speed, oil flow rate and temperature may significantly differ when compressing different gases or vapors or if working at the oil-free or oil flooded mode of operation. To determine the most effective compressor design for any particular duty, a comprehensive multivariable optimization of screw compressor geometry and operating circumstances was carried out. This was accomplished using a computer program the authors created to simulate compressor processes. This program provides a general description of the lobe segments in terms of a number of important parameters, can generate different lobe shapes, and simultaneously calculates compressor thermodynamics.

Xiangjing Liang [3], Under unloading conditions, the twin screw air compressor's operation was examined. Based on the principles of ideal gas and conventional thermodynamic relations, a mathematical model representing the operation of an oil-injected twin screw compressor under unload conditions was developed, taking into account the impacts of oil injection, gas-oil heat transfer, internal leakage, etc. By comparing the outcomes of the simulation model and testing, the experiment was conducted to validate the model. The simulation and test findings also demonstrated that, when the discharge pressure was decreased, volumetric efficiency rose with an increase in rotating speed. As discharge pressure and rotating speed grew, so did the shaft power. The flow rate rose as the rotating speed and discharge pressure were reduced. This model offers a reference for the highly efficient and energy-efficient functioning of screw compressors and can be used to estimate performance under unload scenarios.

2. System Description & Design

The term “Compressor Package” refers to the combination of all the components along with the bare compressor itself that ultimately forms the final finished product. The components that generally comprises of a Diesel Engine driven Compressor Package are:

- a) Bare Compressor (also called as Airend)
- b) Diesel Engine
- c) Air Oil Separator Tank
- d) Radiator type oil cooler
- e) Fuel Tank
- f) Battery
- g) Safety and Discharge Valves
- h) Pressure and Temperature switches

Apart from the above components there are two main components namely- the baseframe, this forms the base of the entire package on which all the components are mounted, and the canopy, which forms as an outer covering for the package. The canopy is placed for noise reduction and ease of maintenance while servicing. [3]

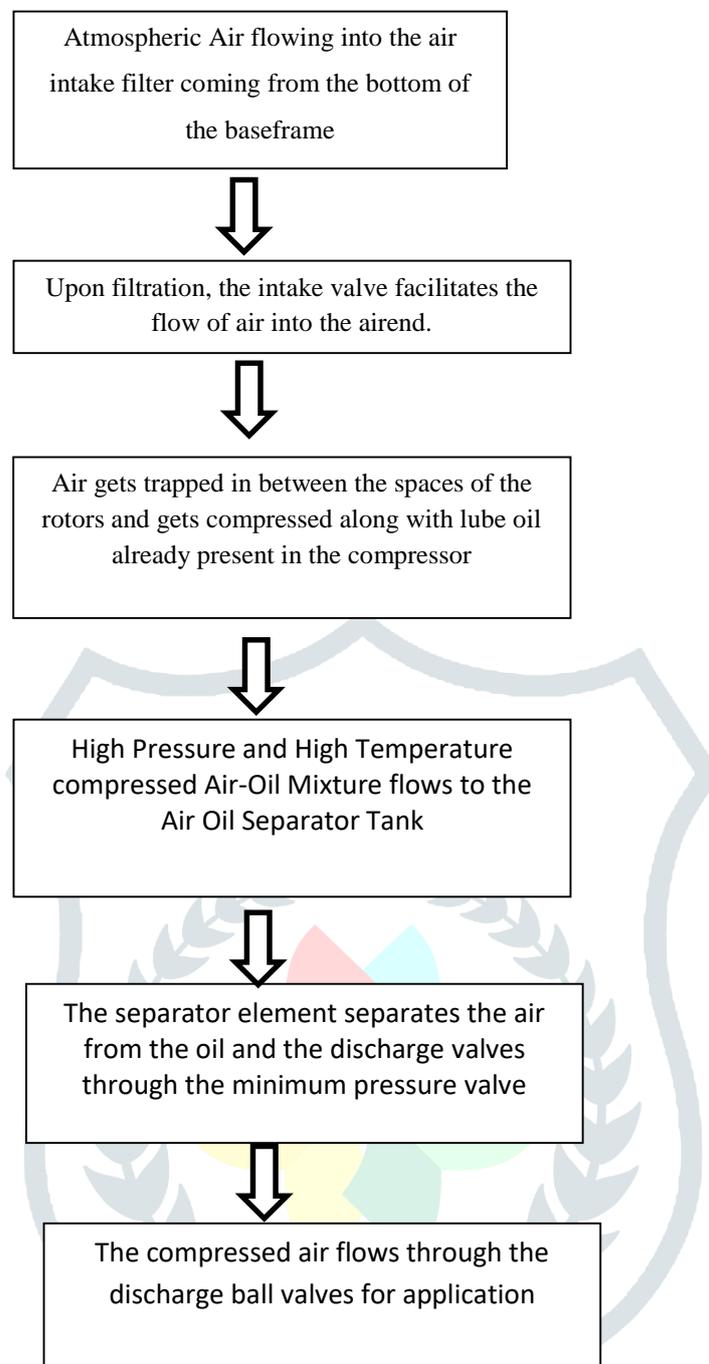


Figure 4: Oil injected screw air compressor package layout

The above images show the layout of a diesel engine driven screw compressor package. There are two main flow circuits in a compressor package; Air flow circuit and Oil flow circuit.

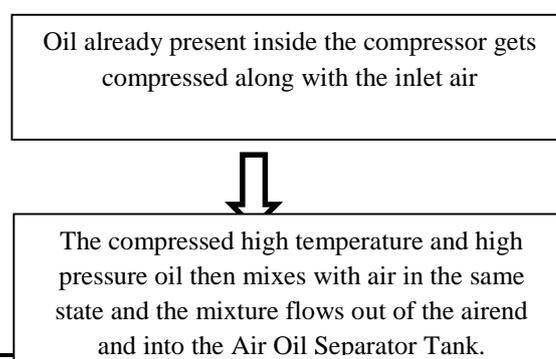
2.1 Air flow Circuit

While in operation, all the canopy doors remain closed. The intake air is taken as atmospheric air which flows from below the baseframe into the intake filter. Upon filtration, the intake valve facilitates the flow of air into the air end. Air trapped in between the rotor spaces get compressed along with the lube oil already present inside the compressor. After compression, the high pressure and high temperature mixture of air and oil flows into the Air-Oil Separation Tank, where the separator element present inside the tank separates the Oil from the Air. Subsequently, the oil free high pressure air is then passed on to the discharge valves after going through the minimum pressure valve. Discharge ball valves channelize the air for application [2], [8].



2.2 Oil Flow Circuit

Oil already present inside the compressor gets compressed along with the inlet air. Consequently, the temperature and pressure of the lube oil increases. This compressed high temperature and high pressure oil then mixes with air in the same state and the mixture flows out of the airend and into the Air Oil Separator Tank. The separation process is carried out by the separator element. The oil droplets being heavier and denser hit the element and fall down to the base of the tank. After the separation process is complete the oil being of high temperature and cannot be reused for lubrication, is channeled through to the oil cooler. A radiator type oil cooler is used. Upon cooling, the oil is recirculated back into the airend and the cycle continues. An Oil Stop Valve (OSV) is used to regulate the flow of oil from the separation tank to the oil cooler [4].



3. Methodology

To find out the flow velocity of air inside the package during operation we need the operating parameters of the fan and subsequently calculate the velocity at the flow domain of the fan. The operating parameters of the fan are as follows:

Blade diameter = 939.8 mm (38 inches)

No. of blades = 4

Shroud Type: Nozzle

Rotational Speed: 2024 rpm

Air Density: 1.09 Kg/m³

Air Flow Rate: 12.51m³/sec

The velocity at the flow domain of the fan is calculated by the expression

$$U = \frac{\pi d N}{60}$$

After the evaluation of the fan flow domain, the velocity of flow within the package during operation needs to be calculated. This value will be the maximum flow of air that carries out surface cooling for each and every component, while the canopy door remains closed. This velocity is calculated by the expression:

$$Q = \frac{\pi}{4} d^2 v$$

Upon calculating the above two velocity parameters, the following expression has been used to compute the mass flow rate of the air flow:

$$\dot{m} = Q \times \rho$$

Now, after calculating the above parameters the software simulation has been carried out. The air flow analysis has been done in order to understand the behavior of air over the surfaces and package domain when the entire package is in operation.

The analysis has been carried out by CFD simulation where the Mass flow rate of air, velocity of air within the package and the fan flow domain velocity has been considered as boundary conditions. A 3D flow domain was created at first which consisted of two surface domains- the air domain and the fan domain. Upon creation of flow domain the meshing of the domains was carried out. Fine meshing was done in order to get more accurate results.

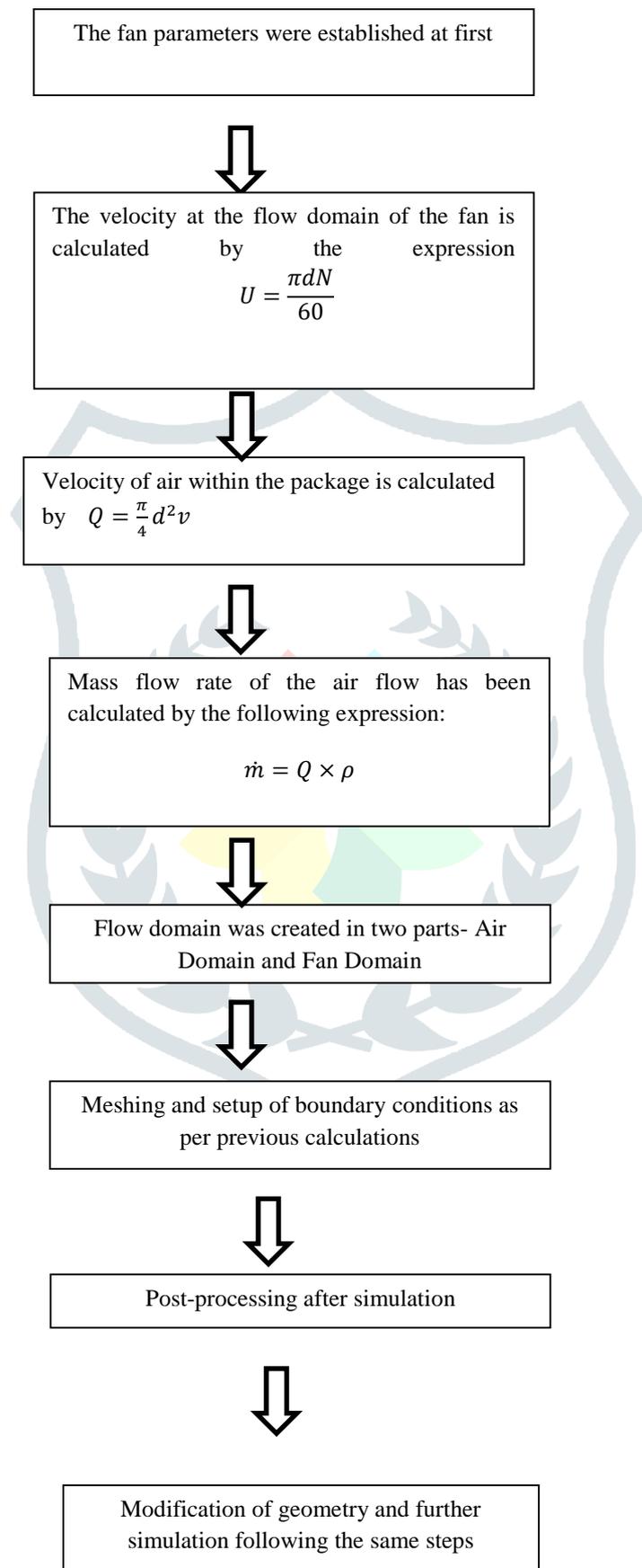
Changes were made to the shape after the initial simulations were done in order to enhance the package's air flow and maximise the fan, so that the power requirement of the fan can be reduced. Another set of simulation has been run with the modified geometry to find the impact of the modifications on the air flow.

The setup or the pre-processing of the meshed components for the second set of simulations consisted of the following inputs:

1. There are 4 different air inlet passages into the package with one chief outlet.
2. The outlet is at the end of the package (Radiator Oil Cooler) where the flow of air is channeled out along with the exhaust air of the engine.
3. Atmospheric air at 25°C and 1 atm pressure is assumed as inlet air..
4. K-Epsilon Turbulence Model is used.
5. For Fan, the domain motion is rotational and the angular speed is 2420 rpm.
6. The Mixing Model for the fan interference regions with air inlet and outlet is taken to be Frozen Rotor.
7. Air mass flow rate inside the package during operation is 12.51 m³/sec.
8. Air mass flow rate at the outlet is 14.0112 m³/sec. [3]

After the above mentioned input values and boundary conditions, the simulation has been carried out. The simulation consisted of 1000 iterations and the results were observed. Post-processing of the acquired result has been carried out to visually represent the data in an understandable way.

The following schematic will give an overview of the methodology followed:



4. CFD Simulations

As stated previously, a specific path of operations were followed for carrying out the CFD simulations for the compressor package as specified. This gave the understanding of the flow and the behavior of air over the surfaces which can actively affect the process of surface cooling. The initial procedure followed was as per the procedure followed in [1].

1. Flow Domain Creation

The 3D geometry of the package was simplified and processed so as to obtain only the area of where the air flow will be prevalent. All the standard fasteners, bolts, nuts, rivets and screws were removed and a geometric clean-up was carried out. Holes and pipes were also removed and only the components that are majorly affected by overheating and require surface cooling are kept.



Figure 5: 3D Geometry of the compressor package

The flow volume is divided into two domains- Air and the Fan Domain. All the components inside the package i.e., the engine, compressor, separator tank, fuel tank and intake filter are kept in the air domain whereas, a separate domain was created for the fan where the air interference with the fan and fan blade is only considered as a separate domain.

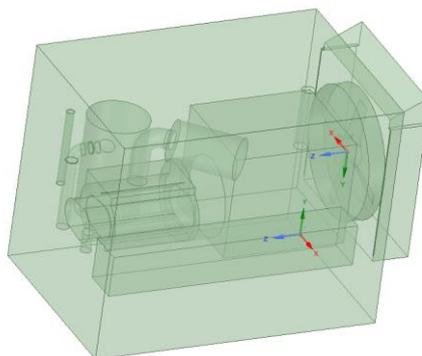


Figure 6: Flow Volume of the package

2. Meshing

Meshing is often used in software-based simulation for Finite Element Analysis (FEA) and Computational Fluid Dynamics (CFD). It can significantly impact the accuracy of the simulation and the resources required to perform the simulation.

A very fine mesh has been generated in this simulation in order to obtain a more accurate result.

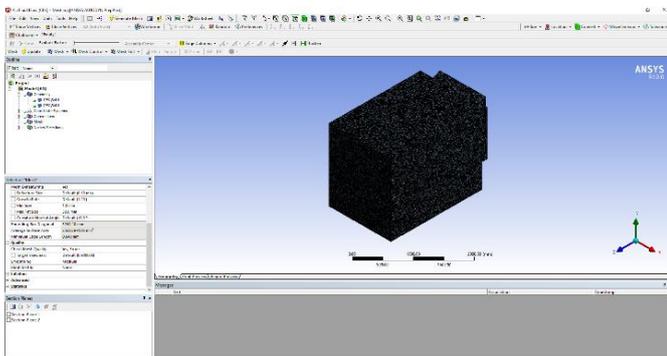


Figure 7: Meshed Geometry

Advanced	
Number of CPUs for Parall...	4
Straight Sided Elements	
Number of Retries	0
Rigid Body Behavior	Dimensionally Reduced
Triangle Surface Mesher	Program Controlled
Topology Checking	Yes
Pinch Tolerance	Default (0.90 mm)
Generate Pinch on Refresh	No
Statistics	
<input type="checkbox"/> Nodes	1013418
<input type="checkbox"/> Elements	5440984

Figure 8: Mesh Quality

Mesh elements allow governing equations to be solved on predictably shaped and mathematically defined volumes. Typically, the equations solved on these meshes are partial differential equations.

3. Setup and pre-processing

Pre-processing is the first step of CFD simulation. Here, all the necessary boundary conditions, inlet and outlet conditions and analysis model selection are carried out. The data input in the solver directly affects the output of the simulation.

For the first set of simulations, we have considered only one inlet of air (through the base frame) and one primary outlet (behind the fan).

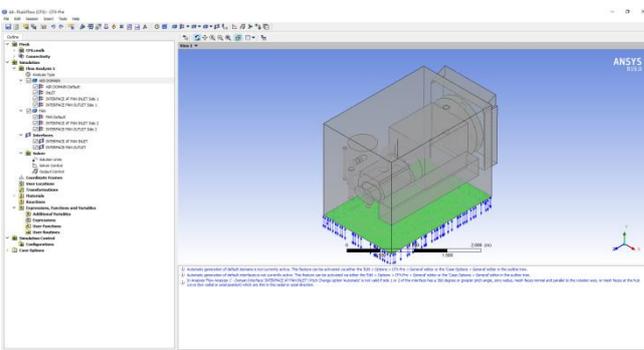


Figure 8: Inlet Specification

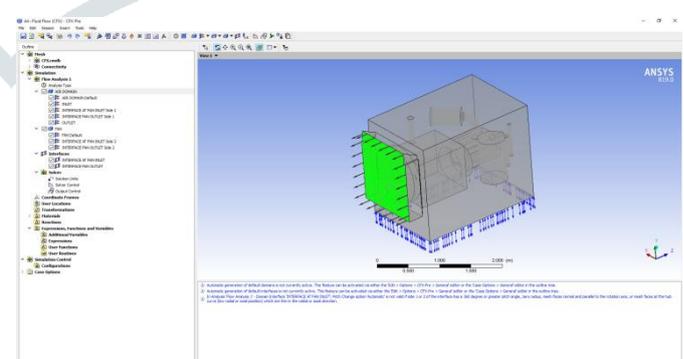


Figure 9: Outlet Specification

The Turbulence model chosen here is K-Epsilon Model and the mixing model is Frozen Rotor, since we need to calculate the velocity of the air flow and the domain of the fan has to remain stationary, even if the blades are rotating at an angular speed of 2420 rpm.

The mass flow rate of air at the outlet is calculated to be 14.0112 m³/sec and the inlet condition of air is atmospheric air at 25°C and 1 atm pressure.

5. Results

After running a simulation of 1000 iterations, the following results have been obtained for the first set of simulations.

Figure 10: Velocity of airflow in the package (Mid-plane)

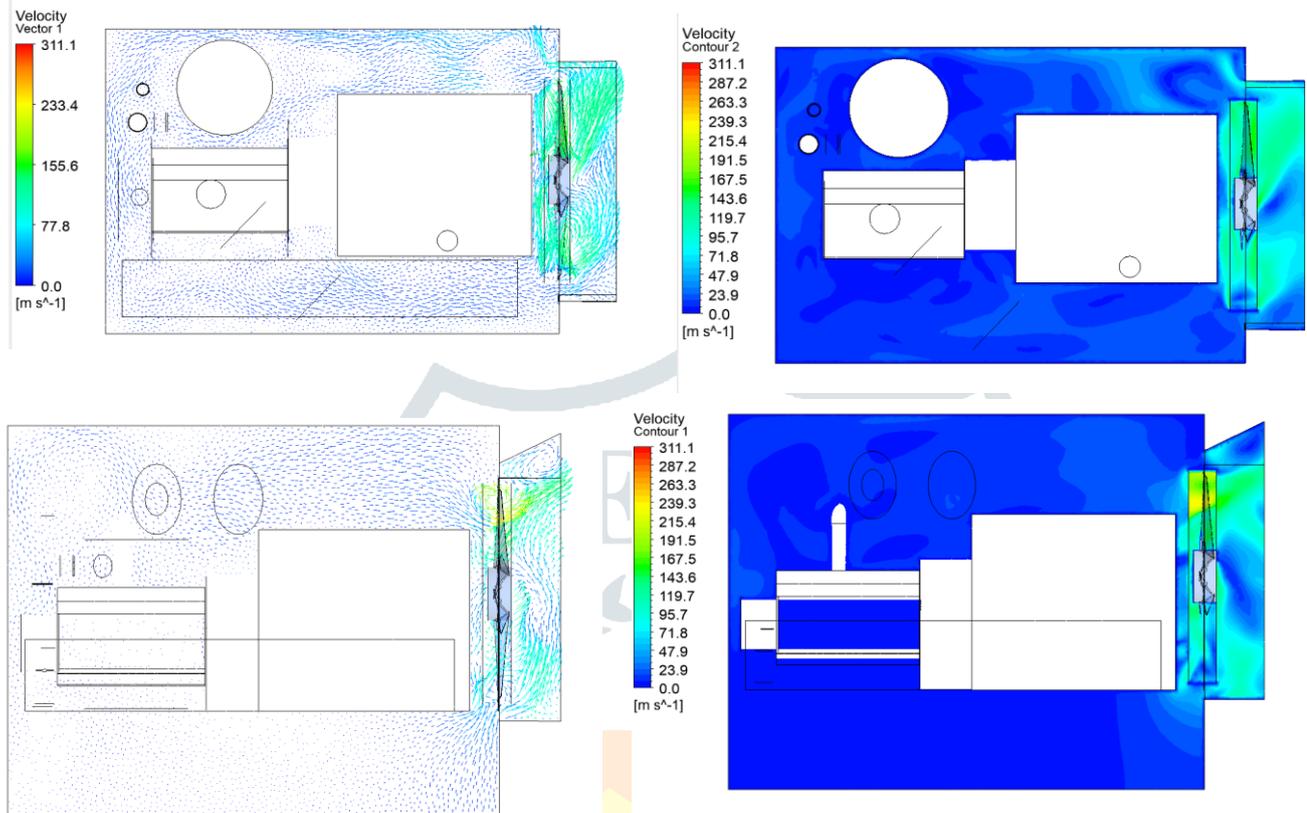


Figure 11: Velocity of airflow in the package (Center Plane)

As seen in the above simulation, the maximum velocity of air inside the package is 311 m/sec, which is excessively high. As a result, the fan is required to draw more power from its motor which eventually makes the fan even bigger and the cost also increases. Also, the air flow is much turbulent and needs to be streamlined. Hence, an alternate solution is provided in order reduce the fan power and rapid air flow within the package. This solution will reduce the power consumption of the fan as well as streamline the flow of air for much better surface cooling effect for each of the components.

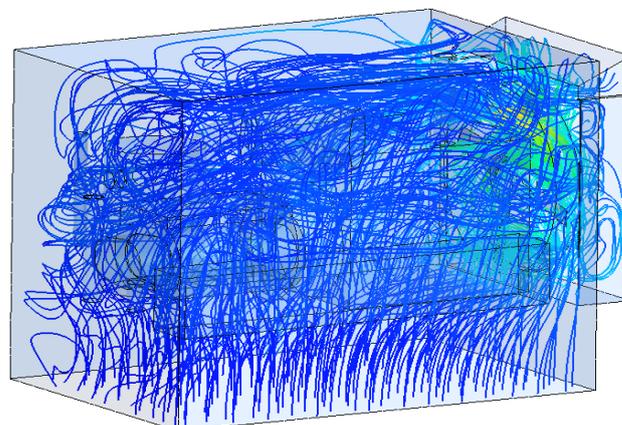


Figure 12: Velocity Streamline

Modification in geometry

1. Two slits of dimension (230 x 150) mm are created on either side of the aircend on the canopy by removing the material.
2. A square cutout of dimension 460 mm is provided at the front of the canopy above the aircend.
3. Atmospheric Air is allowed to enter through the above mentioned openings as inlet air needed for compression function.

Now, after implementing the above modifications, the following is the new flow domain that is obtained for new set of simulations needed:

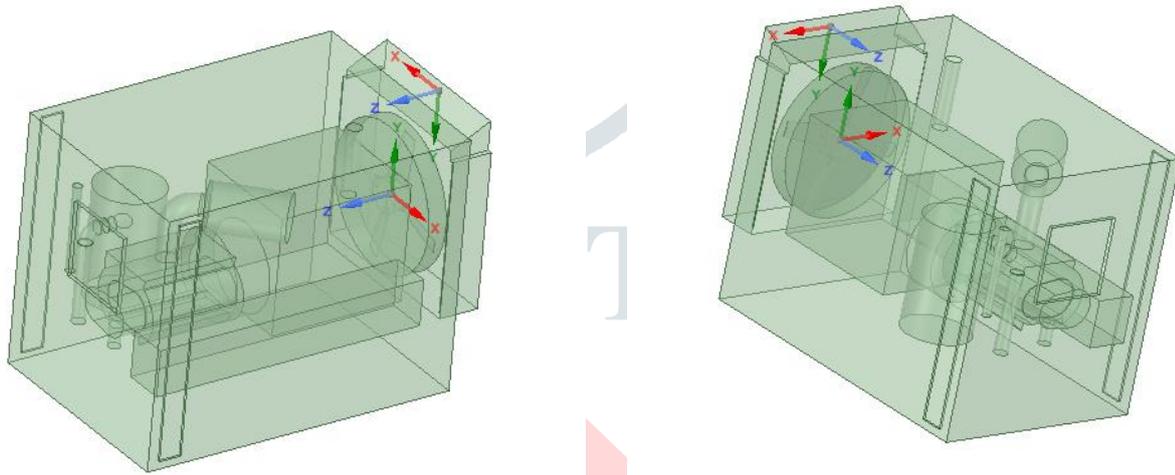


Figure 13: Flow domain for modified geometry

Now meshing of the geometry has been generated. Here also, a fine mesh has been created so that the output results are much more accurate.

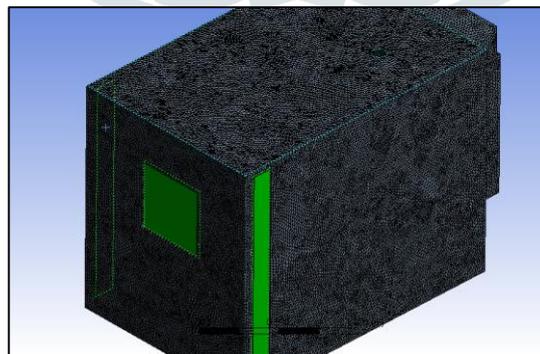


Figure 14: Meshed Geometry (Modified)

Sizing	
Size Function	Curvature
<input type="checkbox"/> Max Face Size	2.e-002 m
Mesh Defeaturing	Yes
<input type="checkbox"/> Defeature Size	Default (1.e-004 m)
<input type="checkbox"/> Growth Rate	Default (1.20)
<input type="checkbox"/> Min Size	1.e-003 m
<input type="checkbox"/> Max Tet Size	5.e-002 m
<input type="checkbox"/> Curvature Nor...	Default (18.0 °)
Bounding Box Di...	3.24410 m
Average Surface ...	0.22960 m ²
Minimum Edge L...	6.4e-004 m

Statistics	
<input type="checkbox"/> Nodes	997039
<input type="checkbox"/> Elements	5347446

Figure 15: Mesh Quality

For setup and pre-processing, the following conditions have been taken as input for simulation of the modified geometry:

1. There are 4 different air inlet passages into the package with one chief outlet.
2. The outlet is at the end of the package (Radiator Oil Cooler) where the flow of air is channeled out along with the exhaust air of the engine.
3. Atmospheric air at 25°C and 1 atm pressure is assumed as inlet air..
4. K-Epsilon Turbulence Model is used.
5. For Fan, the domain motion is rotational and the angular speed is 2420 rpm.
6. The Mixing Model for the fan interference regions with air inlet and outlet is taken to be Frozen Rotor.
7. Mass flow rate of air inside the package during operation is 12.51 m³/sec.
8. Mass flow rate of air at the outlet is 14.0112 m³/sec. [3]

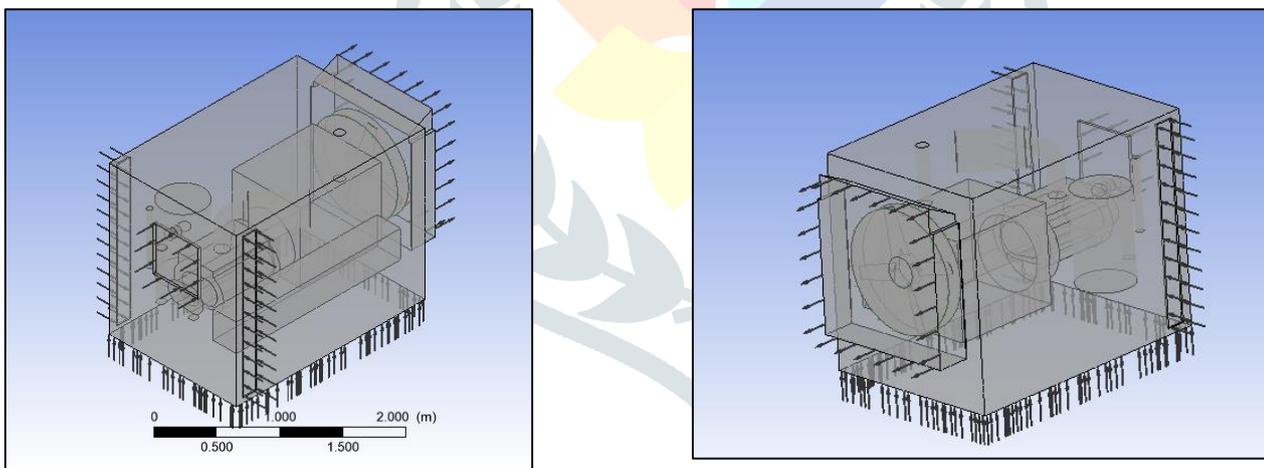


Figure 16: Inlet and Outlet Specifications for modified geometry

Finally, we obtain the results for the final simulation with the modified geometry:

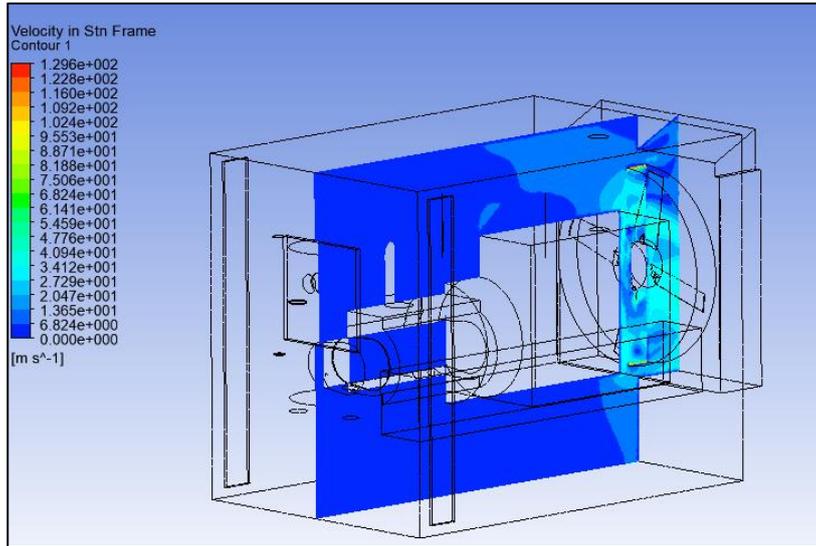


Figure 17: Velocity of air flow in the package (Center Plane)

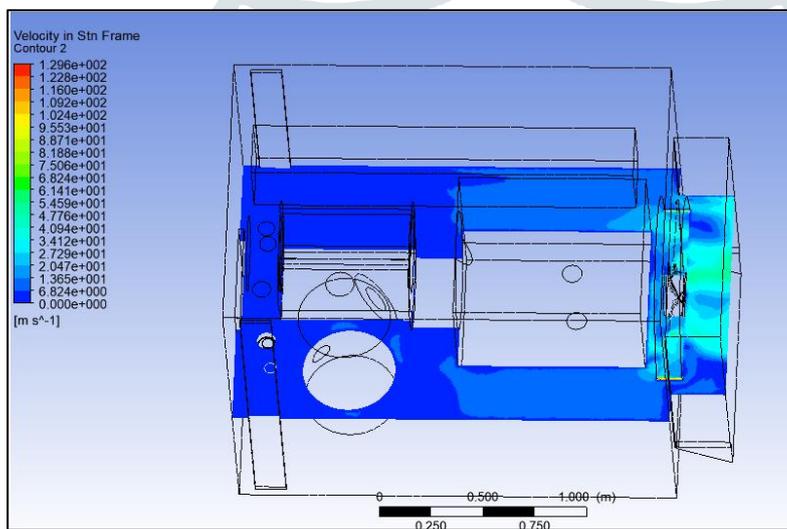
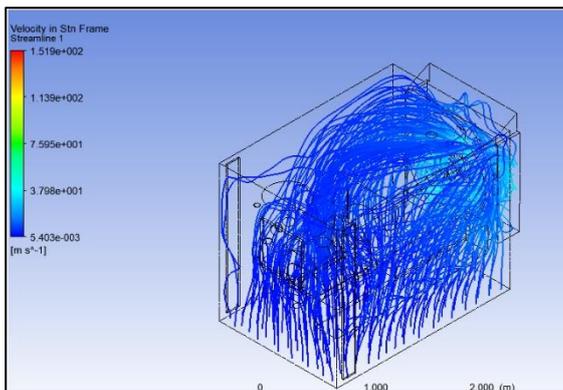
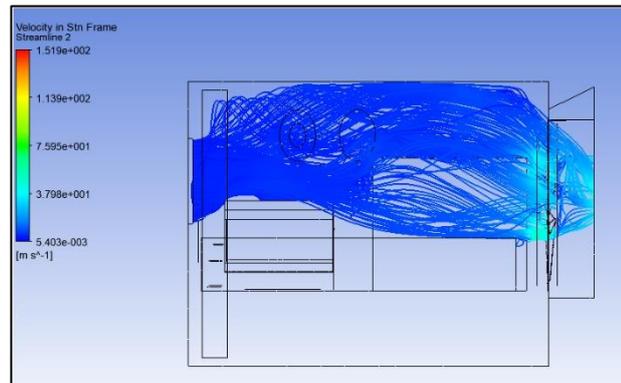


Figure 18: Velocity of air flow in the package (Mid-Plane)

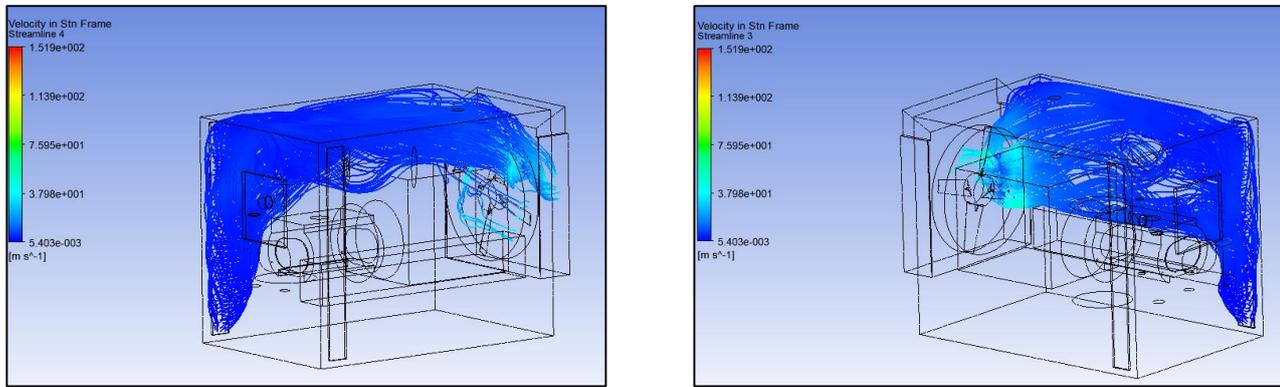
As seen in the above images, the maximum velocity has been reduced from 311 m/sec to about 130 m/sec. This is due to the influx of more amount of air through the various inlets with the modified geometry. Also, the following images show the velocity streamline inside the compressor package.



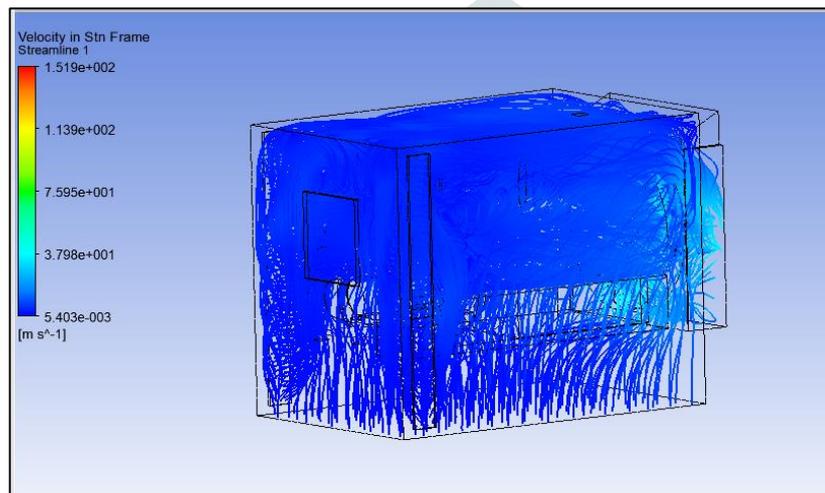
(a) Baseframe Inlet



(b) Cutout Inlet



(c) Slit 1 and slit 2 air inlet

Figure 19: Velocity Streamline (Modified geometry)**Figure 20:** Overall Velocity Streamline for modified geometry

6. Conclusion

The modified geometry gives a much improved streamline flow of air inside the package with the fan operating at the stated rpm. This will allow a better surface cooling for each and every component inside as the air flow is not erratic and it is properly channelized. For example, the engine gives the highest surface heat load inside the package. Hence, a cutout has provided in the frontal face of the canopy so that the air coming in from that inlet will directly flow over the engine and as a result will cool the surface down faster. Initial procedure was as per [1].

Moreover, the velocity of flow inside the package has been reduced. The maximum velocity of air inside the package has been reduced to 151 m/sec in stationary frame. Hence, there is a reduction of about 51% in the velocity. This will directly reduce the rpm required for the fan and consequently, the power consumption.

7. Acknowledgement

The author would like to thank Kirloskar Pneumatic Co. Ltd., Hadapsar, Pune, India, for supporting this research. The author would also like to thank all the co-authors and be grateful for their contribution.

8. References

- [1] **CFD Analysis of A Twin Screw Compressor with Oil Injection**, Junfeng Wang Simerics Inc, jw@simerics.com Hui Ding Simerics Inc Sham Rane Ahmed Kovacevic University of London, 25th International Compressor Engineering Conference at Purdue, May 24-28, 2021 Kovacevic, A., Stolic, N., Mujic, E., & Smith, I. K. (2005, January). *Design Integration for Screw Compressors*. In *ASME International Mechanical Engineering Congress and Exposition* (Vol. 42118, pp. 29-33).
- [2] **Influence of oil injection parameters on the performance of diesel powered screw air compressor for water well application**, [K.K.Dhayanandh^a](#), [K.Rameshkumar^a](#), [A.Sumesh^a](#), [N.Lakshmanan^b](#), Department of Mechanical Engineering, Amrita School of Engineering, Coimbatore, Amrita Vishwa Vidyapeetham, 641112, India, Team Lead – Technology, ELGi Equipments Ltd, Coimbatore, India
- [3] **Technical Reports, KPCL**
- [4] **The Optimization of Internal Processes from a Screw Compressor with Oil Injection to Increase Performances**, Sorin Neacșu, Cristian Eparu* , Adrian Neacșu Petroleum - Gas University of Ploiesti, 39 Bucharest Blvd., Ploiesti (100680), Romania Corresponding Author Email: cristian.eparu@gmail.com, *International Journal of Heat and Technology* Vol. 37, No. 1, March, 2019, pp. 148-152 Journal homepage: <http://iieta.org/Journals/IJHT> Švigler, J. (2010). *A Treatise on the Theory of Screw Machines*. Chapter 7&8, Západočeská Univerzita.
- [5] **Screw compressors: Theory, design and application**. Xing, Z. W. (2000), Chapter 3 and 6, China Machine Press, Beijing, China.
- [6] **Study of Multiphase Flow at the Suction of Screw Compressor, Proc. Int. Compressor Conf. at Purdue, Paper 1353**. Arjeneh M., Kovačević A., Gavaises M., Rane S., (2014).
- [7] **Boundary Adaptation in Grid Generation for CFD Analysis of Screw Compressors, Int. J. Numer. Methods Eng., Vol. 64: 401-426**. Kovačević A., (2005).
- [8] **Oil as a design parameter in screw type compressors: oil distribution and power losses caused by oil in the working chamber of a screw type compressor**. Deipenwisch R and Kauder K., (1999). *IMECHE Transactions*; 6; 49-58; *Int. Conf. on Compressors and their systems, London*.