ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE ENGINEERING AND TECHNOLOGY

PERFORMANCE ANALYSIS OF AIR-OIL SEPARATOR TANK IN OIL INJECTED ROTARY SCREW COMPRESSORS WITH CFD METHODS

M.Sc. THESIS

Mert ALPAGUT

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Heat - Fluid Programme

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ISTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

YAĞ ENJEKTELİ VİDALI KOMPRESÖRLERDEKİ HAVA-YAĞ AYRIŞTIRMA TANKININ CFD METODLARI İLE PERFORMANS ANALİZİ

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FOREWORD

This study was conducted under the supervision of Asst. Prof. Dr. Levent Kavurmacıoğlu. I would like to express my sincere appreciation for the support, guidance and insights that he has provided me throughout the study.

I want to thank my friends and colleagues for their help and friendship during my research.

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September 2018 Mert ALPAGUT
(Mechanical engineer)

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ABBREVIATIONS

CAGI	**:** Compressed Air and Gas Institute
CAD	**:** Computer Aided Design
CFD	**:** Computational Fluid Dynamics
DPM	**:** Discrete Phase Model
LES	**:** Large Eddy Simulation
Pneurop	**:** European Committee of Manufacturers of Compressors, Vacuum Pumps and Pneumatic Tools
RNG	**:** Renormalization Group Theory
RSM	**:** Reynolds Stress Model

SYMBOLS

a	: Inlet pipe height
b	: Inlet pipe widht
d	: Vortex finder diameter
D	: Diameter
h	: Vortex finder height
H	: Tank height
Pa	: Pascal
v	: Tangential velocity
z	: Radial position
η	: Seperation efficiency

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PERFORMANCE ANALYSIS OF AIR-OIL SEPARATOR TANK IN OIL INJECTED ROTARY SCREW COMPRESSORS WITH CFD METHODS

SUMMARY

Compressors are used in nearly all industrial sectors. Researches have shown that air compressors are consuming nearly 10% of industrial energy in the world [2-4]. Most of the countries are applying regulations for the energy efficiency of the compressors. Oil injected rotary screw compressors are the most common type that is being used in most of the industries.

One of the most important system in oil injected rotary screw compressors is separation system. There are several techniques to perform compressed air and oil separation. In this thesis study, vertical cyclonic air oil separator tank CFD analysis has performed. Nine different vortex finder diameter and length dimension combinations have analysed with three different inlet velocity. In total, twenty seven different cases has taken into account in order to optimize the design air oil separator tank.

Many studies revealed that RSM method is the most reliable method in order to predict the complex swirling flow inside air oil separator tank. Moreover, DPM method has shown best results of tracking oil droplets. Same approaches have been taken into account in this study.

Velocity inlet has implemented as inlet boundary condition as 8-10-12 m/s. Pressure outlet has given as outlet boundary condition as 700000 Pa since the operating pressure of the compressor system. In addition, inlet and outlet ports have stated as escape condtion. Walls and bottom surface of the air oil separator tank are stated as reflect and trap condition respectively. Finally, Rosin-Rammler particle size distribution has used for oil particles ranging from 1 μm to 100 μm with 1.45 spread parameter. ISO-V46 synthetic rotary compressor oil is used with 868 kg/m^3 density, 0.0085 kg/m-s dynamic viscosity and 60 L/min flowrate. Air density and dynamic viscosity is stated as 7.986 kg/m^3 and $2.11*10^{-5}$ kg/m-s respectively.

Separation efficiency and pressure drop is the most critical parameters for the performance of air oil separator tank. Inlet velocity has a big impact on pressure loss inside the air oil separator tank. Pressure loss increased rapidly with the increase in inlet velocity. Furthermore, tangential velocity increases with the increase in inlet velocity which is one of the main sources of centrifugal force acting on oil droplets so on to the separation efficiency. However, results have shown that in some cases increase in inlet velocity resulted in high separation efficiency and for some cases high inlet velocity decreased the separation efficiency. Moreover, dimensional changes of vortex finder diameter and length have more significant affect on pressure loss and separation efficiency. Increase in vortex finder diameter and length has resulted in decrease in separation efficiency and pressure loss.

It is also found out that smaller oil particles have more tendancy to escape from the outlet port. Nearly all oil particle droplets larger than 10μm trapped at the bottom surface of the tank.

After comparing the results of all twenty seven cases, it is found out that the tank design with 200mm vortex finder diameter and 300 mm vortex finder length at 8 m/s inlet velocity has shown the best performance by considering separation efficiency and pressure loss.

YAĞ ENJEKTELİ VİDALI KOMPRESÖRLERDEKİ HAVA-YAĞ AYRIŞTIRMA TANKININ CFD METODLARI İLE PERFORMANS ANALİZİ

ÖZET

Kompresörler hemen hemen tüm endüstriyel sektörlerde kullanılmaktadırlar. Yapılan araştırmalarda dünyadaki sanayi elektriğinin yaklaşık %10'u kompresörler tarafından kullanılmaktadır [2-4]. Son yıllarda kompresörlerin enerji verimliliği üzerine bir çok ülke yeni regülasyonlar uygulamaktadır. Yağ enjekteli vidalı kompresörler endüstride en yaygın olarak kullanılan kompresör tipidir.

Yağ enjekteli vidalı kompresörlerde en önemli sistemlerden biri hava-yağ ayrıştırma sistemidir. Kompresör sisteminde bulunan yağın bir çok kritik işlevi bulunmaktadır. Bu yüzden sistemde bulunan yağın basınçlı hava ile taşınması istenen bir durum değildir. Yağ tankında gerçekleştirilen ayrıştırma işleminin performansı kompresörlerin düzgün çalışması için önemli bir rol oynamaktadır. Hava ile yağı ayrıştırmak için farklı yöntemler ve ayrıştırmanın gerçekleştiği tankın farklı tasarımları bulunmaktadır. Bu tez çalışmasında, dikey siklon bir hava yağ ayrıştırma tankının hesaplamalı akışkanlar dinamiği metodları kullanılarak performans incelemesi yapılmıştır. Performans incelemesi sırasında, tank tasarımını da optimize etmek için tankın için bulunan vortex tüpünün çap ve uzunluk ölçüleri değiştirilerek toplamda dokuz farklı ölçüde analizler yapılmıştır. Geometrik değişikliklerin yanı sıra, tank girişindeki havanın hızı üç farklı değerde incelenmiştir. Toplamda yirmi yedi farklı kombinasyon için hesaplamalı akışkanlar dinamiği analizi yapılmıştır.

Kompresörlerde kullanılan hava yağ ayrıştırma tankları, siklon tip ayrıştırıcılarla kullandıkları yöntem ve geometri olarak büyük benzerlik göstermektedir. Yapılan literatür araştırmasında RSM metodunun yüksek dönel akışların hakim olduğu durumlar için güvenilir sonuçlar verdiği görülmüştür. İki fazlı akışlarda, ana faz ile ikincil fazın hacimsel debi oranlarının %10 ve aşağısı olduğu durumlarda, ikinci fazın modellenmesi için DPM methodunun diğer metodlara göre daha tutarlı sonuçlar verdiği görülmüştür. Bu tez çalışmasında da yukarıda bahsedilen metodlar kullanılarak hava yağ ayrıştırma tankı analizleri yapılmıştır.

Giriş sınır koşulu olarak üç farklı 8-10-12 m/s hız değeri parametrik olarak tanımlanmıştır. Tank çıkışında nominal çalışma basıncı olan 700000 Pa değeri girilmiştir. Ayrıca, DPM metodu sınır koşulu olarak giriş ve çıkış portu "escape" koşulu olarak tanımlanmıştır. Tankın iç yüzeyleri "reflect", alt yüzeyi ise "trap" olarak tanımlanmıştır. Tankın ayrıştırma veriminin hesaplanması için bu koşullar yüksek önem teşkil etmektedir. Son olarak, girişteki yağ partiküllerinin çapları, partiküllerin küresel olduğu kabulu ile, Rosim Rammler methodu baz alınarak 1 μm ile 100 μm arasında dağılım olarak tanımlanmıştır. Vıdalı kompresörler için özel olarak üretilen ISO V46 sınıfında, 868 kg/m^3 özkütleye, 0.0085 kg/m-s dinamik vizkositeye sahip yağ kullanılmıştır. Yağ debisi 60 L/dak olarak tanımlanmıştır. Sistemin çalışma basıncına göre havanın özkütlesi 7.986 kg/m^3, dinamik viskositesi 2.11*10^{-5}'dir. Analizler yaklaşık 4000 iterasyon sonucunda yakınsamıştır.

Hava-yağ ayrıştırma tanklarının performansı için en önemli iki parameter ayrıştırma verimi ve basınç kaybıdır. Ayrıştırma veriminin düşük olduğu hava-yağ tanklarında, yağ partikülleri basınçlı hava ile sistemin dışına taşınmaktadır. Zamanla sistemdeki yağın azalmasından dolayı kompresör veriminde düşüş ve arızalar meydana gelebilmektedir. Basınç kaybının yüksek olması ise kompresör sisteminin olağandan fazla enerji tüketmesine sebebiyet vermektedir.

Yapılan analizler sonucunda giriş hızının basınç kaybı üzerinde ciddi bir etkisi olduğu gözlemlenmiştir. Giriş hızının artması ile birlikte basınç kayıpları da hızlı bir şekilde artış göstermiştir. Ek olarak, giriş hava hızlarının artması ile birlikte santrifüj kuvvetlerin ana bileşenlerinden biri olan teğetsel hız değerleri de artmaktadır. Santrifüj kuvvetler ayrıştırma verimi için önemli bir parametredir. Ancak analizler sonucunda, teğetsel hızlardaki artış ayrıştırma verimini bazı vortex tüp dizayları için arttırırken, bazı durumlarda ayrıştırma veriminin azalmasına sebep olmuştur. Vorteks tüp çapı ve uzunluğundaki ölçüsel değişikliklerin etkisi daha net bir şekilde görülmüştür. Vorteks tüp çapındaki ve uzunluğundaki artış ayrıştırma veriminin ve basınç kaybının azalmasını ile sonuçlanmıştır. Tank içindeki yağ partiküllerinin hareketleri incelendiğinde, büyük çaptaki partiküllerin hızla tankın iç yüzey kısımlarına yöneldiği ve büyük ölçüde tankın alt yüzeyine giderek ayrıştığı görülmüştür. 10μm çapına kadar olan partiküllerin zamanla tankın çıkış port bölgesine doğru toplandığı ve bir kısmının çıkıştan kaçtığı görülmüştür.

Yirmi yedi farklı konfigürasyon birbiriyler kıyaslandığında en verimli tank tasarımının 8m/s hava giriş hızı değerine, 200 mm vortek tüp çapına ve 300 mm vortex tüp uzunluğuna sahip tank tasarımının olduğu sonucunda varılmıştır.

1. INTRODUCTION

Compressors are devices which are used for increasing pressure of a compressible fluid, mainly gaseous substances [1]. They are widely used in all industry branches, homes and in peoples' daily life. Compressors are used in many different kinds of industry branches such as; petro-chemical, electronics, automotive, food processing, machinery, glass, mining, textile, construction, etc. In addition, compressors can be found in all homes, usually for refrigeration purposes in refrigerators and air conditioners. Moreover, in our daily life, we face with compressors in transportation (cars, trains, busses, trucks), amusement parks etc. In short, people need air to live likewise, industry needs compressed air to operate.

According to the researches, it is found out that compressed air is consuming nearly 10% of industrial energy in Europe and Australia. Moreover, energy consumption of compressed air in some cases stands for 30% of industrial energy in USA [2-4]. In other words, compressed air production is one of the most electrical energy consuming utility in industrial facilities.

1.1 Compressors

Compressors are divided into two main segments according to their compression methods, namely; dynamic compressors and positive displacement compressors. Positive displacement compressors, increases pressure of the gas by decreasing the volume inside a closed chamber. There are various types of positive displacement compressors according to application and operation. Most common types of positive displacement compressors are piston and rotary screw compressors. Dynamic compressors increase the pressure of the gas by increasing the kinetic energy and converting it into pressure energy. There are 2 main types of dynamic compressors, namely; axial and centrifugal turbo compressors [5]. Basic compressor family tree is shown at Figure 1.1.

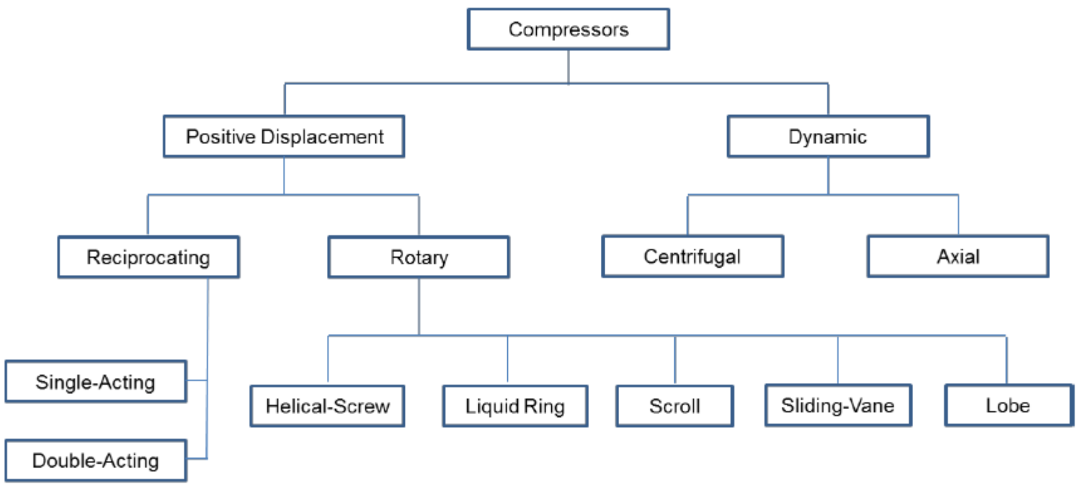


Figure 1.1 : Compressor Family Tree

1.1.1 Oil injected rotary screw compressors

Oil injected rotary screw compressors are the most common type, which is being used in nearly all industries varying from small shops to huge power plants.

In the last decade, there are many studies performed in order to increase the energy efficiency of compressors and compressed air systems. Compressed air and gas comities such as; CAGI and Pneurop are establishing new regulations in order to increase the efficiency of compressors and compressed air systems.

Life cycle costs of compressors are generally calculated for 10 years of usage. Compressors are usually seen as expensive machines while making the first investment, however, investment cost is usually around 10-15% of the total cost in 10 years' time. Commissioning and service costs are stands for around 5% and 10% of total cost respectively. Energy cost of rotary screw compressor over 10 years period takes the biggest portion with 70% of the total cost [6]. Life cycle cost of a compressor shown in Figure 1.2.

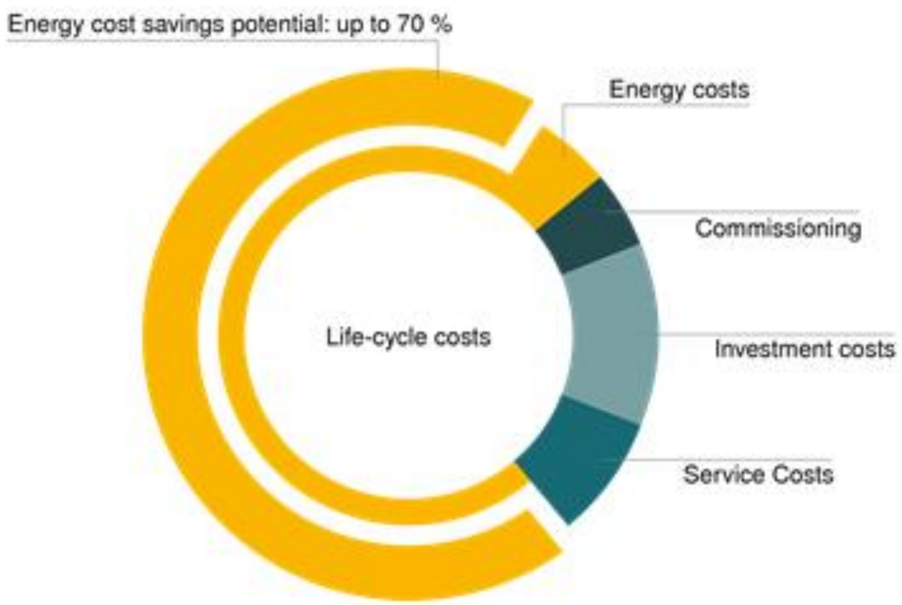


Figure 1.2 : Life Cycle Cost of Compressor

Manufacturers rely on new technologies such as variable speed drive compressors in order to save some energy. Compressed air consumption of facilities vary during the day. Variable speed compressors are capable of changing their capacity at constant discharge pressure in order to deliver the exact required compressed air flow. They are able to save around 6% of energy by just adjusting the constant pressure when compared with fixed speed compressors. Energy savings can reach up to 35% according to application [7].

Due to its operating principle, up to 93% of consumed electrical energy is converted into heat energy. Well-designed heat recovery system can recover up to 90% of the waste heat, which can be used in heating water, air, furnaces, and boilers in industry. [8].

Furthermore, during the design phase of the rotary screw compressors, there are pressure losses occur at the system components, such as; filters, piping, separator tank, cooler etc. System internal pressure losses can reach up to 0.8-1 bar at 7 bar working conditions. On the contrary, pressure losses increase in time due to clogging in filters and separators. Studies has shown that 0.14 bar pressure loss stands for 1% extra energy consumption [7].

1.1.1.1 Operating principle of oil-injected rotary screw compressors

Oil injected rotary screw compressors operates with the principle of reducing the volume of a closed chamber, which results in increase in pressure.

There are several components and systems inside of a rotary screw compressor. Basic rotary screw compressor configuration is shown in Figure 1.3 [9].

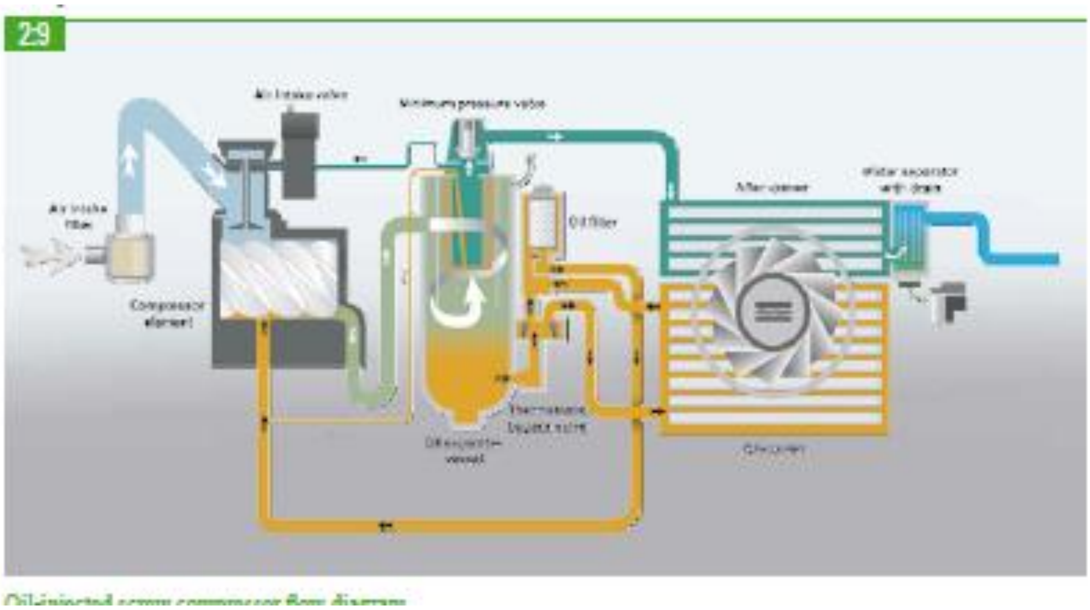


Figure 1.3 : Oil Injected Rotary Screw Compressor Scheme

Oil injected rotary screw compressors consist 5 main systems, namely; drive system, separation system, cooling system, intake control system & hydraulic system. Drive system is the most important system since the compression process performed in this system. Mechanical energy provided by an electrical motor is transmitted to compression element (airend) via belt&pulley or with an elastic coupling. Cutaway view of the airend is shown in figure [3].

Figure 1.4 : Airend

Airend consists of two intermeshing helical male and female rotors, which are contained in a housing. Male rotor is driven by the electrical motor from the main shaft and female rotor is driven by the male rotor. There is no metal-to-metal contact in between the rotors because oil creates a film layer. The clearance between male and female rotors is around 20-30 μm [10]. Compression process inside of the airend is shown at Figure 1.5 [9]. Firstly, atmospheric air is being sucked to the airend due to the vacuum created by the rotation of the rotors. The grooves of the rotors are filled with atmospheric air. Thanks to the helical shape of the rotors, air is transmitted to forward to the compression chamber. In other words, air is being stuck in the gap in between the rotors and the casing. Compression starts at the third stage in Figure 1.5. Oil injection the airend takes place in this stage. There are three main duties of oil:

- Creating an oil film in between the rotors
- Cooling
- Lubricating the rotating parts

As the rotation continues, the volume of the rotor grooves decreases while the air is trapped in between the rotors and the casing, which results in increase in pressure. Finally, the mixture of compressed air and oil leaved the airend from the discharge port. [10]

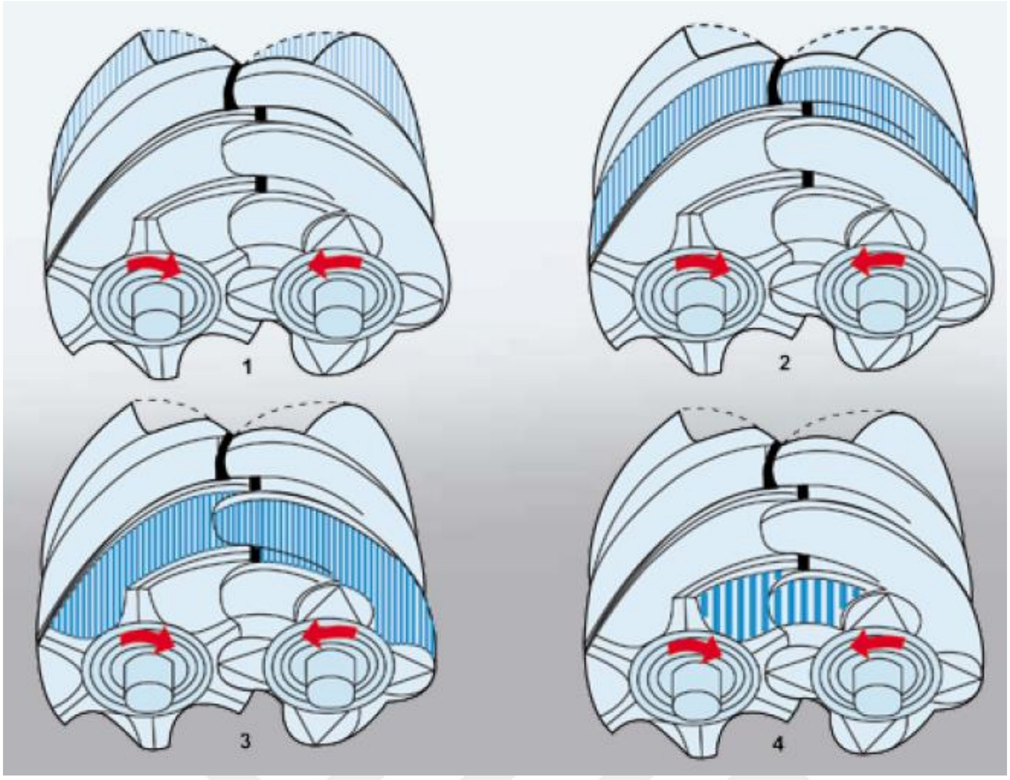


Figure 1.5 : Compression stages in Airend

Afterwards, the mixture of compressed air and oil flows through a separation vessel. There are three main duties of the separation vessel; bulk separation of compressed air and oil mixture, oil storage and creating internal pressure which provides oil circulation inside the system. During bulk separation, around 95% of oil is separated from the air and accumulated at the bottom of the separation vessel. After the bulk separation, compressed air flows through a coalescing separator, which separates the remaining small oil particles from the compressed air.

These small oil particles are fed back to the airend with the scavenger line. The residual oil content in compressed air after the separator is around 3 ppm. Then, compressed air passes through a minimum pressure valve, which ensures the minimum system internal pressure, goes to the aftercooler. Compressed air is cooled to a certain temperature in order to eliminate moisture trapped inside. Finally, compressed air is ready for usage. In typical compressed air applications, after treatment systems are used in order to increase the air quality of the compressed air.

Oil, which is accumulated at the bottom of the separation vessel flows through from an opening to the temperature, controlled mechanical valve, which is also known as thermostatic valve. By nature oil injected rotary screw compressors does not tend to operate neither at hot (>105°C) nor cold (<50°C) temperatures. Thermostatic valve ensures the oil temperature at optimum limits which is around 80°C. Oil is restricted flow through the cooler according to its temperature by the thermostatic valve. If the oil temperature is lower that optimum value, it is directed to an oil filter and back to airend. If the oil temperature is above the limits, it is directed to the oil cooler then follows the same way as in the first case [9].

As it is mentioned before, oil has very critical roles for the oil injected rotary screw compressors. Lack of oil can cause shutdowns or fatal damages on the compressor system and parts. However, some amount of oil is continuously being lost with the discharge air. The amount of lost oil is directly proportional with the performance of the coalescing air oil separator. Moreover, performance of coalescing separator is also directly proportional with the bulk separation processed in separation vessel. The better the bulk separation is, the better fine separation.

In this thesis study, performance analysis of the separation vessel will be performed where the bulk separation is being handled.

1.1.1.2 Separation vessel

There are various types of separation vessels are being used in gas-liquid separation systems. Separation vessels are mainly used in oil-gas industry, chemical plants, compressors etc. Vessel sizes are very large in industrial applications. However, in compressor systems separator vessel sizes has to be very small since compressors consist an enclosed canopy. There are two main types of separator vessels according to the physical separation principles namely, vertical and horizontal separation vessels.

Gravity (knock out) drums

Gravity drums are simple horizontal type of separators. Basically, the mixture of gas and liquid enters from upper side of the vessel and travels to the opposite upper end of the vessel. Gravity is the main force, which separates gas and liquid particles [11]. Due to the sudden expansion in inlet of the vessel, velocity of the stream decreases. Velocity of the gas stream is very important because separation efficiency is much better with low flow velocity and it is proportional to vessel diameter. Gravity Drums are less efficient when compared with other type of separators. They are capable of doing bulk separation up to 300µm of liquid particles. Additional fine separators should be used afterwards [12].

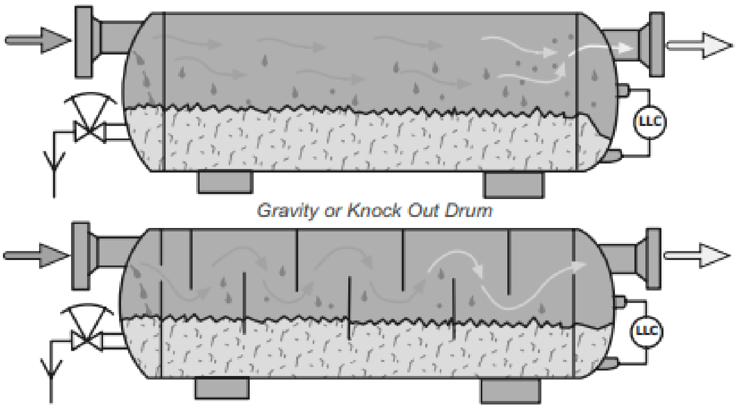


Figure 1.6 : Gravity (Knock out) Drums

Vertical separators

Inpingement and cyclone separators are common types of vertical separators, which uses different techniques in order to separate the liquid particles.

Inpingement separators

Inpingement separators have an inlet port at the middle of the vessel. They usually consist a diverter just after the inlet port where the first gross separation occurs. Gas-liquid mixture hits the diverter as it enters the vessel and liquid flow is directed to the bottom of the vessel [13]. Whereas, gas particles flows to the outlet port is located at the top of the vessel. Gravitational for separates relatively small liquid particles from the gas stream while it is traveling to outlet port. Moreover, just before the outlet port mesh pads are being used inside the inpingement separators which separates liquid particles up to 1µm [12]. Cutaway photo of an inpingement separator can be seen in Figure 1.7 [11].

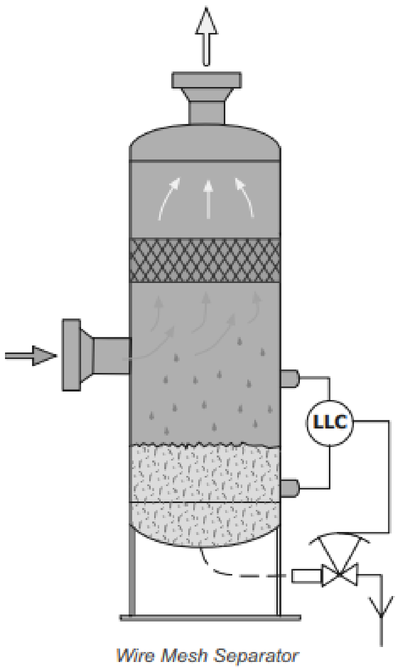


Figure 1.7 : Inpingement Separator

Cyclone separators

Cyclone separators use the principles of centrifugal and gravitational forces to separate liquid droplets [13]. Velocity is the key parameter which effects the separation efficiency like other type of separators. However, high velocity is desired on cyclone separators which is directly related with efficiency and minimum droplet size [12]. Centrifugal force plays a major role for the separation. Centrifugal force, which effects the liquid droplets, may range 5 times for the larger diameter droplets and may reach up to 2500 time for the small diameter droplets [13]. Well designed cyclone separator can separate liquid particles up to 10µm [11].

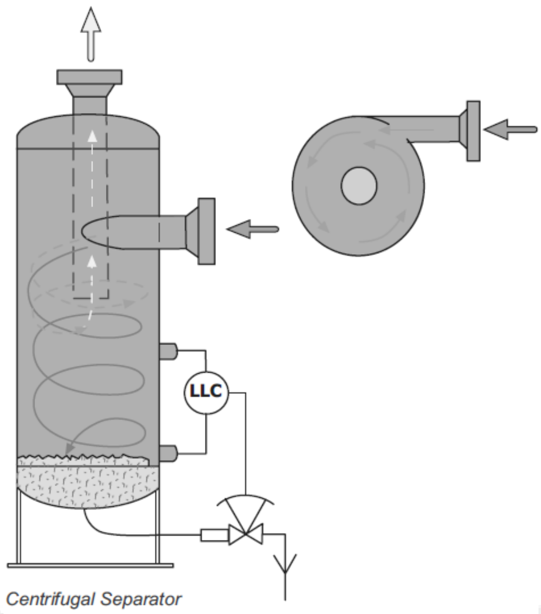


Figure 1.8 : Cyclone Separator

1.2 Literature Review

Cyclone separators are widely used in oil injected rotary screw compressors in order to separate oil from the compressed air. Manufacturers prefer using cyclone type separators because they are easy to manufacture & low cost, provide good separation efficiency in small dimensions.

People have tried to understand the complex flow inside the cyclone separators with mathematical modelling, by doing experiments and performing numerical studies with CFD tools. Mathematical models can make rough estimation about the complex flow inside the cyclone separators. However, it is not enough to make the best cyclone design. Experimental studies can predict efficiency of the cyclone separators precisely; however, it is not easy to visualize the flow inside the cyclone separators in compressors due to their operating conditions. CFD approach to the cyclone separator has become popular in the last decade due to the developments in technology. CFD approach offers reasonable, time saving and cost effective solutions while designing cyclone separators.

Many studies have been performed in order to increase the efficiency of cyclone separators with CFD modelling. Matsuzaki et al. have studied the effect of the height of the outlet pipe by using Large Eddy Simulation (LES) model on a Smagorinsky cyclone. They have analyzed particle motion with Langrangian method and made comparison with the experimental results [14]. Slack et al. has tried to show the differences by using LES and Reynolds Stress Model (RSM) methods in order to analyze the flow field inside a Stairmands cyclone model. The results show that both models can predict the flow field precisely. However, RSM model gives same results with a coarser mesh, which results in saving computational power and time [15]. Another study has performed in order to show the effect of vortex diameter size on cyclone efficiency by using different types of solvers. Results of the study has shown that k-ε and RNG k-ε solver methods are failed to predict the flow. RSM solver has shown consistent results when compared with the experimental data [16].

Elsayed et al. have investigated cyclone separators with different inlet pipe dimensions by using CFD approach. They have used RSM and Discrete Phase Model (DPM) methods in order to solve the flow field and particle motion respectively. The results shown that optimum ratio of the height and width of the inlet pipe should be in between 0.5 – 0.7 [17]. Another study has conducted in order to show the effect of vortex finder height by Xiang et al. They have investigated a cyclone with different vortex finder height and diameter. They have found out that diameter of the vortex finder does not affect tangential velocity and separation efficiency. However, increased height of vortex finder reduces the tangential velocity, which has a negative effect in separation efficiency [18].

Moreover, the effect of inlet pipe size and velocity on the efficiency of hydrocyclone has been studied Bo TANG et al. It is understood that high inlet velocity increases the amount of trapped fine particles at the bottom of the cyclone. It is also shown that, increase in inlet diameter decreases the separation effiency of coarse particles due to chaotic and short circuit flow around the vortex finder [19].

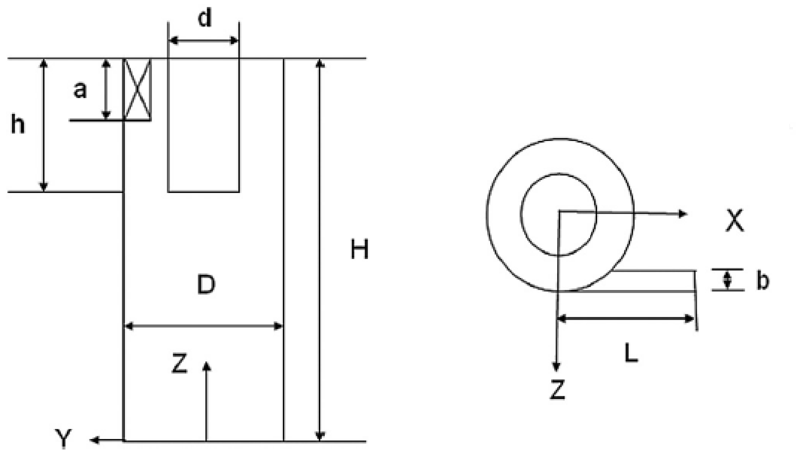
Another study has performed on different shapes of vortex finders by Kuo-Jen Hwang et al. They have studied effect of straight and conical shapes of vortex finder designs on separation efficiency, particle diameter cut size and pressure drop. It has been seen that conical shape of vortex finder design improves the separation efficiency relatively high when compared with conventional straight vortex finder results. However, conical designs has a slight negative effect of pressure drop occuring inside the cyclone [20].

2. OIL AIR SEPARATOR TANK ANALYSIS

There will be no experimental studies performed in this thesis study in order to validate the numerical solutions.

2.1 Comparison & Validation Model

At first, another study, which has been done by Gao et al., has taken into account as reference to show that CFD solver is working properly. Gao et al. has investigated the performance analysis of an oil injected rotary screw compressor separator vessel both numerically and experimentally with the effect of breakup of oil droplets [21]. They have studied on a relatively small separator tank (D=65mm) which can be seen at Figure 2.1.



Dimension	Cyclone diameter *D*	Cyclone height *H*	Central channel diameter *d*	Central channel height *h*	Inlet pipe height *a*	Inlet pipe width *b*
Dimension/D	1.0	3.0	0.5	1.0	0.5	0.15

Figure 2.1 : Separation Vessel Dimension

Validation of the CFD solver has performed according to the 14.4 m/s inlet velocity with different size of mesh. The results are inconformity with the studies of Gao et al. Slight differences between the curves are due to mesh size and the parameters which are not explained in the article such as; turbulent viscosity ratio etc. Figure 2.2. shows the comparison of tangential velocity vs radial position with 14.4 m/s inlet velocity at the 55 mm height from the bottom of the tank of CFD results.

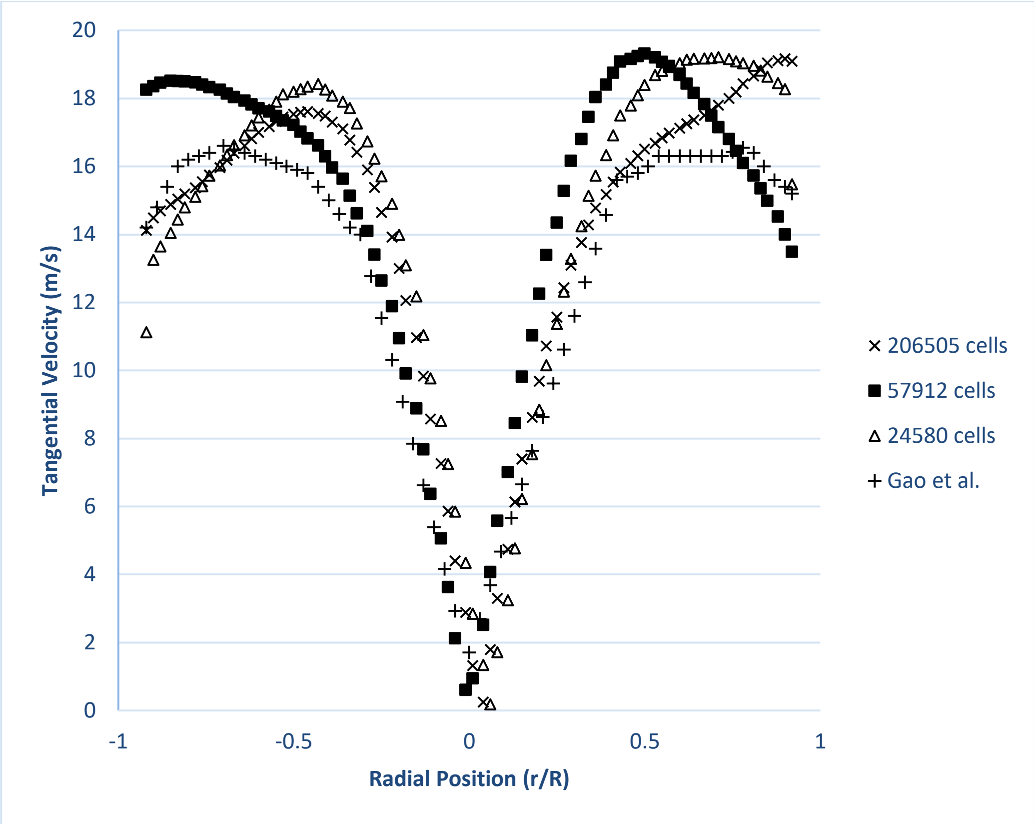


Figure 2.2 : Tangential Velocity Comparison (z = 55 mm; v = 14.4 m/s)

Solution with 206505 shows similar results with the solution of Gao et al.

The results of tangential velocity, static pressure drop and tangential velocity contuars are shown in figures 3.3,3.4,3.5.

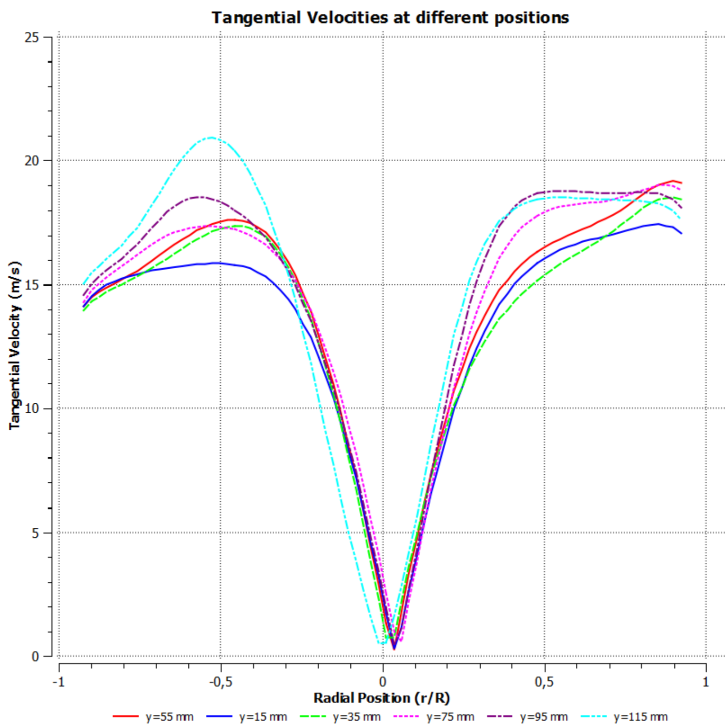


Figure 2.3 : Tangential Velocity vs Radial Position (y=15,35,55,75,115 mm)

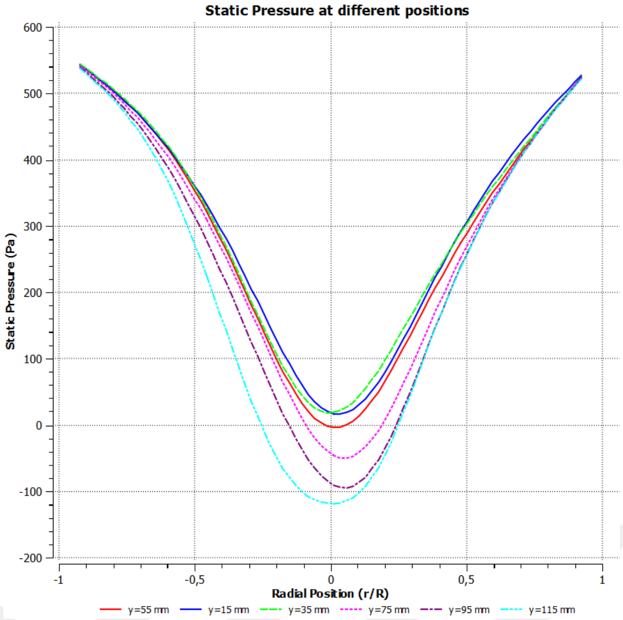


Figure 2.4 : Static Pressure vs Radial Position (y=15,35,55,75,115 mm)

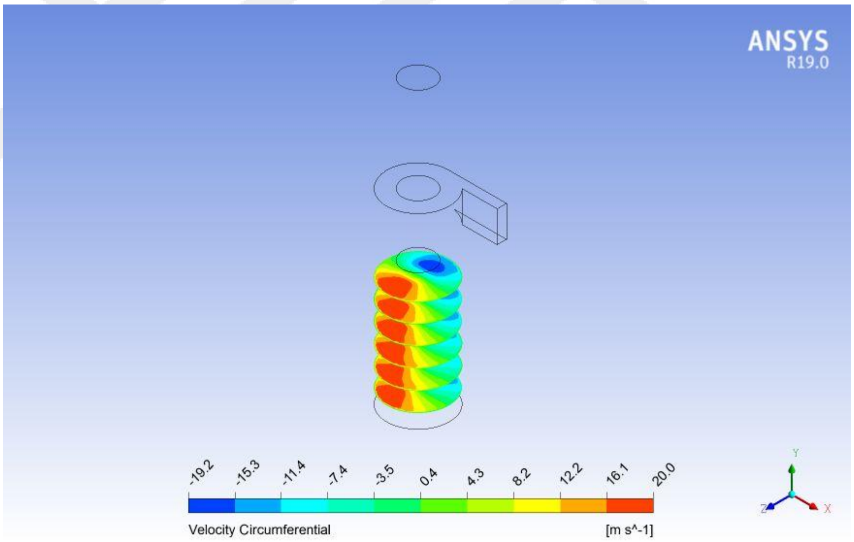


Figure 2.5 : Tangential Velocity Contuar vs Radial Position (for y=15,35,55,75,115 mm)

2.2 Analysis of HDD 22 Compressor Air Oil Separator Tank

The aim of this thesis is, performing CFD analysis of a Air Oil Separator tank which is being used in 22 kW compressor manufactured by Dalgakıran Compressor.

Results of the comparison study shows that the methods that are used in CFD analysis gives reasonable solution with previous studies. Same approach will be applied to the real case study.

2.2.1 Geometry

Oil separator tanks in compressors has similar separation method with cyclones. Only difference is cyclones are usually desinged as cylinder on cone geometry in order to have more effective separation efficiency. However, due to pressure vessel directives it is not possible to design oil separator tanks in a similar logic due to operation under pressure. Air oil separator tanks are considered as pressure vessels due to their volume and operating pressure. They usually consists of 4 main parts, namely, torispherical end, body, top flange and cover. Torispherical end, body and top flange are welded eachother according to the standards. Those parts consist some chamfers, joggles and fillets in order to ease the manufacturing porcesses. Moreover, body part has many nozzles with different sizes, which are being used for the connections of safety valves, oil level gauges, transmitters etc.

CAD drawing air oil separator tank is shown in Figure 2.6.

Figure 2.6 : CAD Drawing of Air Oil Separator Tank

In this study, geometry of the air oil separator tank is simplified from some details so a better quality mesh can be applied.

Flow geometry created in Space Claim software. Geometry has divided into many bodies in order to have swep mesh all around the flow geometry, which will ease the convergence of the solution and provide more accurare results. Flow geometry is shown in Figure 2.7.

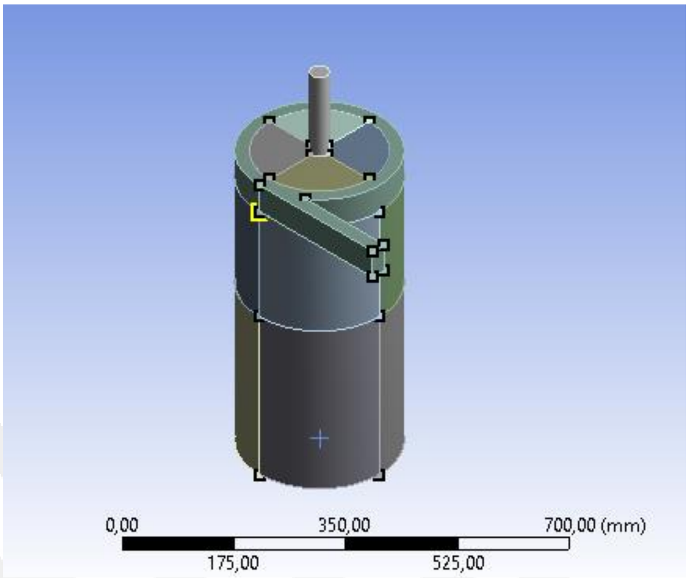


Figure 2.7 : Flow Geometry

In this study, nine different vortex finder dimension cases has taken into account in order to come up with most efficient configuration of the oil separator tank design. Cross sectional view of the oil separator tank is shown in Figuıre 2.8.

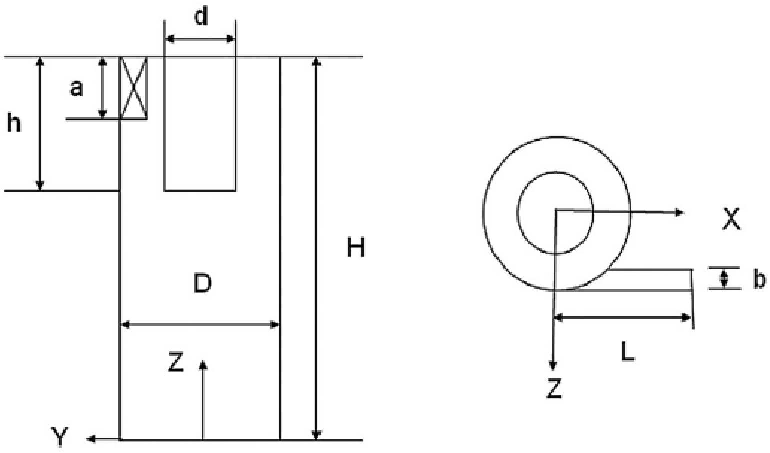


Figure 2.8 : Oil Separator Tank Cross Sectional View

Dimensional parameters of the oil seaprator tank with the different vortex finder dimensions is shown in Table 2.1.

Table 2.1 : Oil Separator Tank Dimensions (mm).

Case Number	Inlet Pipe Height & Width (a x b)	Diameter (D)	Tank Height (H)	Vortex Finder Diameter (d)	Vortex Finder Height (h)
A1	50 x 25	265	550	200	250
A2	50 x 25	265	550	200	300
A3	50 x 25	265	550	200	350
B1	50 x 25	265	550	219	250
B2	50 x 25	265	550	219	300
B3	50 x 25	265	550	219	350
C1	50 x 25	265	550	240	250
C2	50 x 25	265	550	240	300
C3	50 x 25	265	550	240	350

2.2.2 Mesh

Flow geometry has divided 20 bodies in order to have sweepable mesh on the geometry. Hex mesh has been performed on the geometry. Divided geotmetry and meshing has shown in Figure 2.9 and 2.10 respectively.

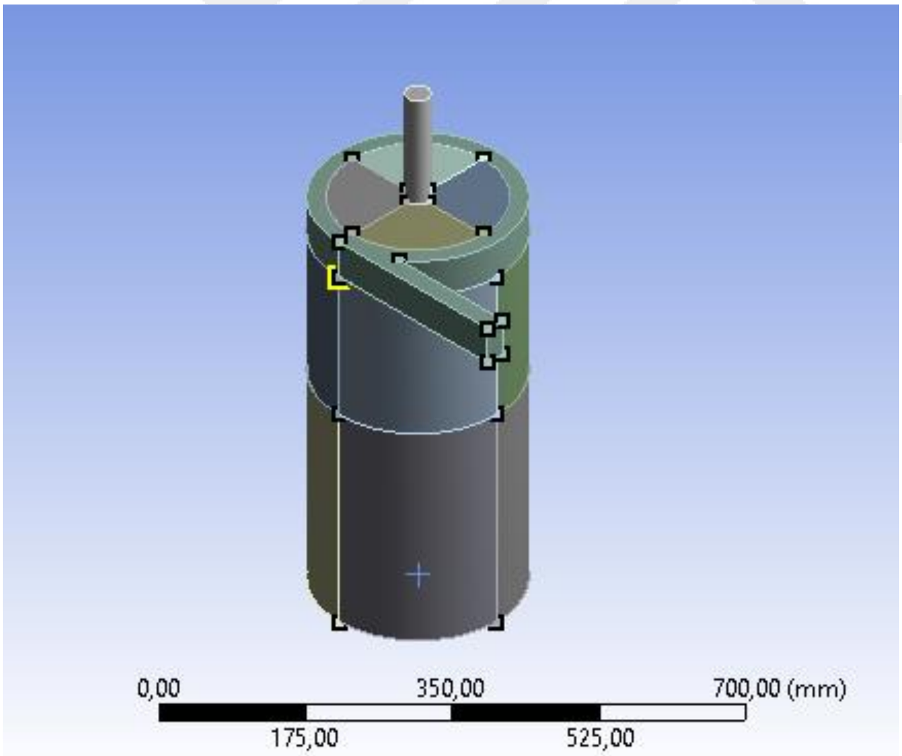


Figure 2.9 : Divided Flow Geometry

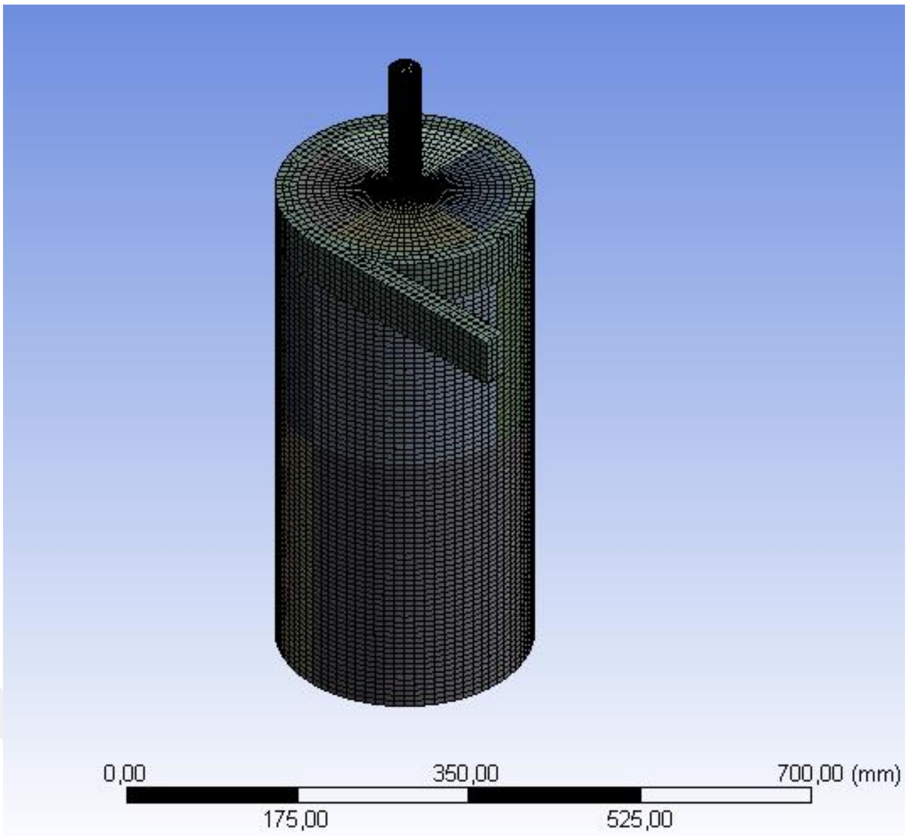


Figure 2.10 : Mesh

As it is mentioned above, nine different cases with different vortex finder diameters have been investigated in this study. Due to dimensional differences, there are slight differences in number of meshes. Moreover, finer mesh has been performed for a single case in order to show the independency of the results with different mesh size. Table 2.2 shows the differences of the results with respect to different size of meshes.

Table 2.2 : Mesh Independency

Number of Mesh	Static Pressure Loss (Pa)	Difference from Min.
220553	2632	6.1%
129848	2480	0%
71785	2347	-5.6%

2.2.3 CFD solver

Ansys 19.0, Fluent software has been used in order to solve the flow inside the air oil separator tank. Firstly, airflow has been computed then, oil particles with different sizes has injected to the computational domain.

Gravity is enabled in order to take account the effects of gravitational forces acting on oil particles. However, the effect of centrifugal forces much larger than gravitational forces acting on oil particles.

Kaya & Karagöz has studied the various numerical schemes and turbulence models to solve the highly dominant swirl flows inside the tangential inlet cyclones. They have investigated k-ε, RNG k-ε, RSM methods and show the differences of the tangential velocity, axial velocity and pressure drop inside the cyclones. Moreover, they have compared the results with experiemental and numerical data, which is given in the literature. As a result, RSM has given most reliable and accurate results when compared with other models [22]. In this study, same approach has taken into account. Standard – Scalable wall functitons has been used due to weak pressure gradient near walls of the oil separator tank which is reported in Fluent User Manuals [23].

Volumetric concentration of oil particles is less than 10% in total flow. Discrete phase modelling is used in order to track of the oil particles inside the flow domain with the assumption of one way coupling where continous phase is not effected from the motion of oil particles due to low volumetric concentration. According to operating conditions, special type of synthetic oil is used in oil injected rotary screw compressors with the density of 868 kg/m^3 and dynamic viscosity of 0.0085 kg/m-s. Oil flow rate is 60 L/min (0.868 kg/s). Operating conditions of the system is 7 bar.g, air density and dynamic viscosity has stated as 7.986 kg/m^3, $2.11*10^{-5}$ kg/m-s.

Oil particles from 1 μm to 100 μm with 50 μm mean diameter has investigated in this study. The spread parameter is taken as 1.45 according to the experimental studies of Xiang,Gao et al [24]. Particle size distribution has calculated according to Rosim Rammler distribution method. Cumulative mass fraction distribution according to particle size is given in Figure 2.11.

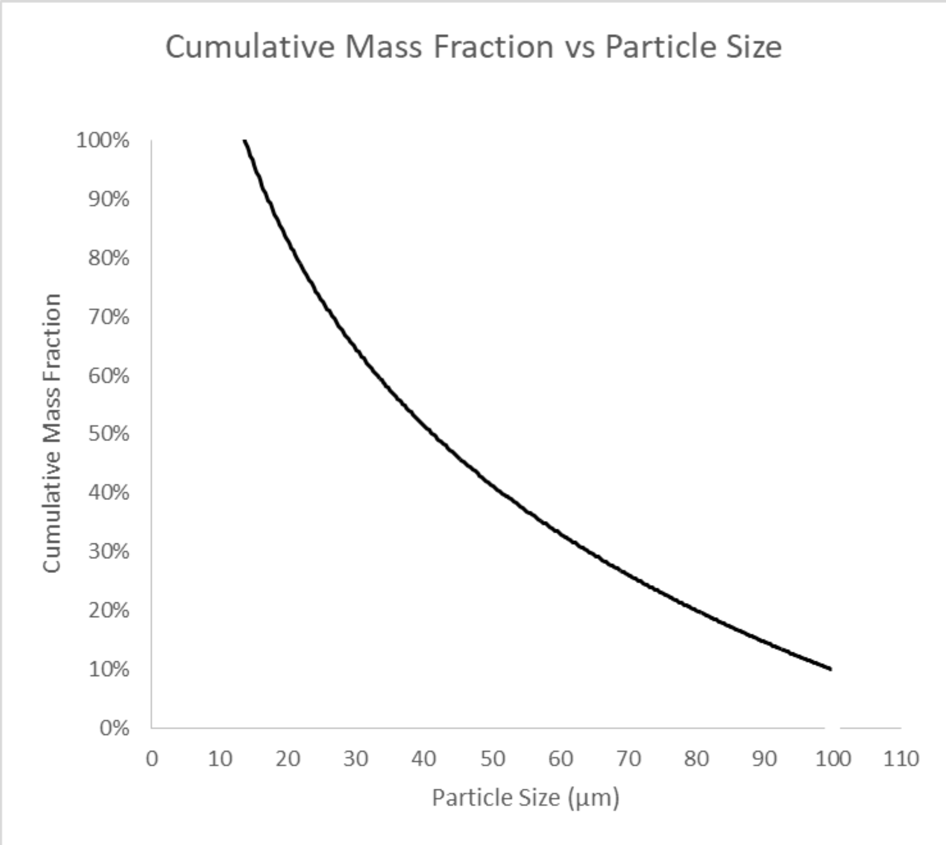


Figure 2.11 : Cumulative Particle Size Distribution

Velocity inlet and pressure outlet boundary conditions has implemented for inlet and outlet of the oil separator tank respectively. Three different velocities (8 – 10 – 12 m/s) have taken into account parametrically for each vortex finder sizes to optimize the performance of the oil separator tank. In total, 27 different cases have been analyzed in this thesis study. The walls of the oil separator tank has stated as reflect boundary condition. Under normal operating conditions, bottom of the oil separator tank is filled with oil up to certain level. Top level of oil is assumed as wall and trap boundary condition has applied. Pressure outlet has given 700000 Pa and escape condition has assinged to the outlet.

Kaya & Karagöz has also studied effects of different solver methods in a gas cyclone. They have tried five different cases of solution methods. In the end, their results show that pressure interpolation scheme is critical in swirling flows. Presto solver method has given most reliable solutions. There are no significant difference in between the numerical schemes for momentum equation however, quick solver method has stated as the best. Likewise, for turbulent equations first and second order upwind schemes has given identical results. Due to high swirling flow higher order upwind schemes has shown negative effect on convergence. They stated that second order upwind scheme and first order upwind scheme can be the best choise for turbulen kinetic energy and Reynold stresses respectively. Finally, simplec algorithm has decided to be the most advantageous for pressure-velocity coupling [22]. Solver methods which have been used in this study, is shown in Table 2.3.

Table 2.3 : Solver Methods

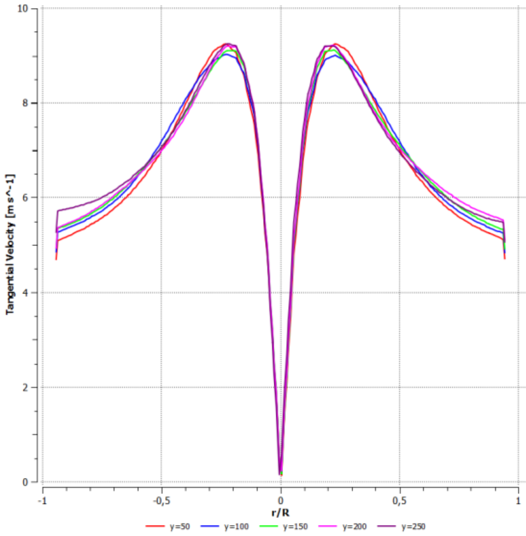
	Solution Method
Pressure-Velocity Coupling	SIMPLEC
Pressure	PRESTO
Momentum	QUICK
Turbulent Kinetic Energy	Second Order Upwind
Turbulent Dissipation Rate	Second Order Upwind
Reynold Stress	First Order Upwind

Hybrid initilization has performed and calculation has run. All of the 27 cases has convergerd.

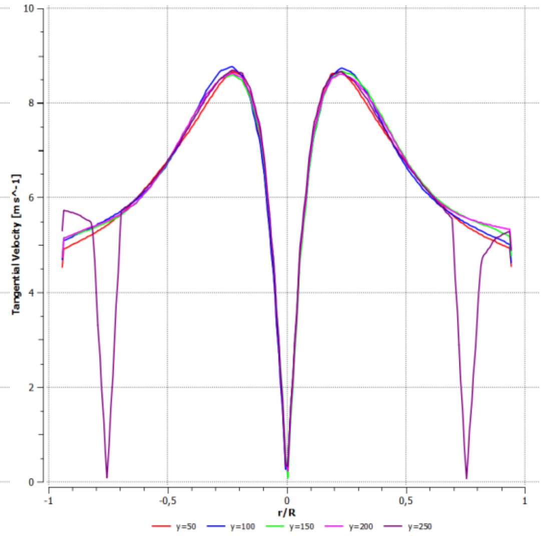
2.2.4 Results and discussion

Validation of the numerical model has performed and details are given in the previous chapter. Numerical model which is being used in this study has given acceptable results with the previous study that has been done by Gao et al.

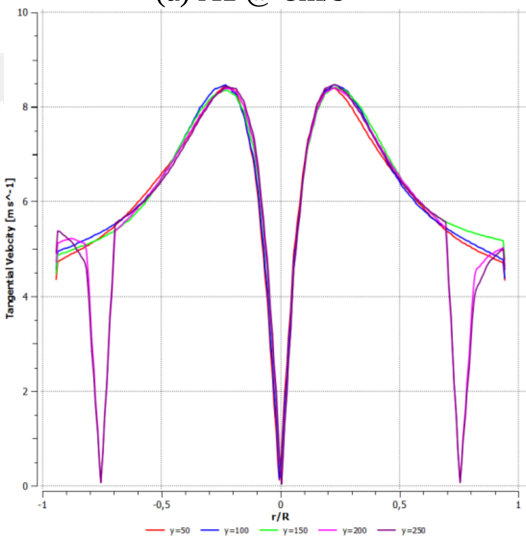
Tangential velocity is the most critical parameter for the flow field inside cyclones. It is the main source of centrifugal forces which effects the particle motion. Figure 2.12 - Figure 2.13 - Figure 2.14 illustrate the tangential velocity variance from the walls of the separator tank to the center with different sections from the bottom of the separator tank.



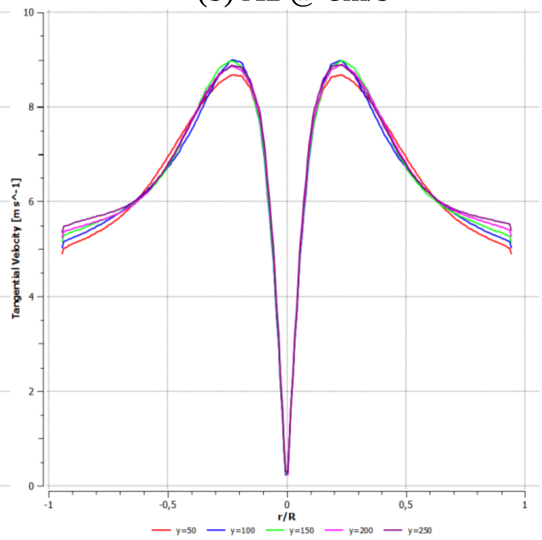
(a) A1 @ 8m/s



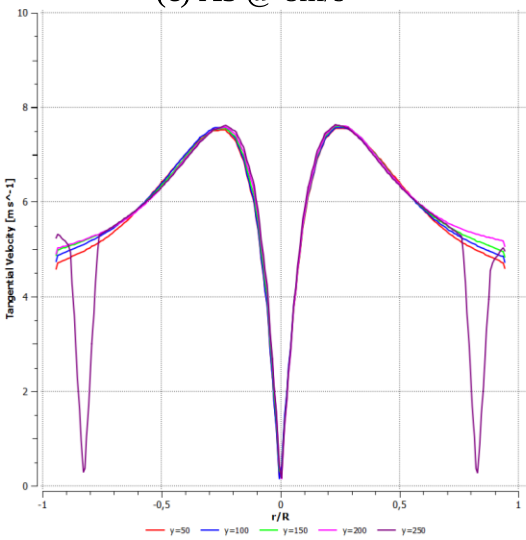
(b) A2 @ 8m/s



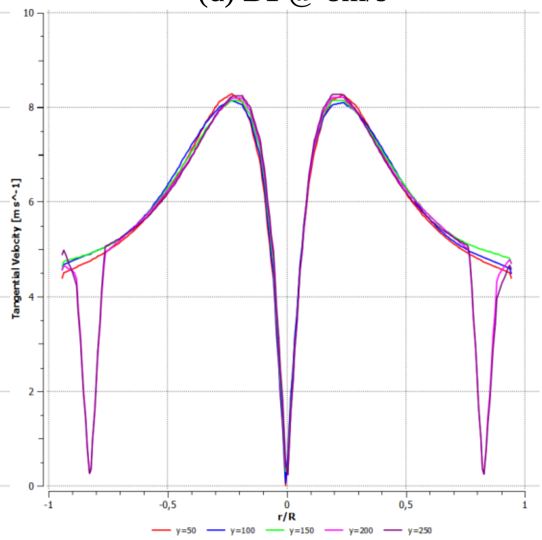
(c) A3 @ 8m/s



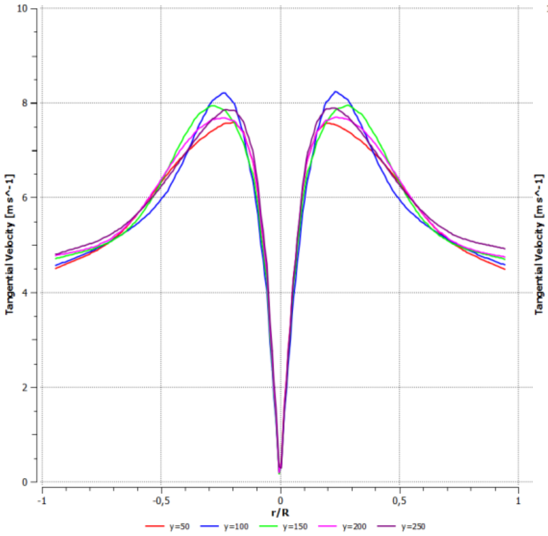
(d) B1 @ 8m/s



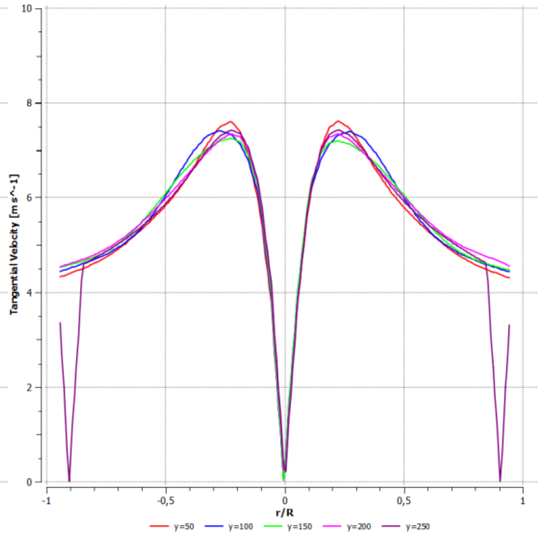
(e) B2 @ 8m/s



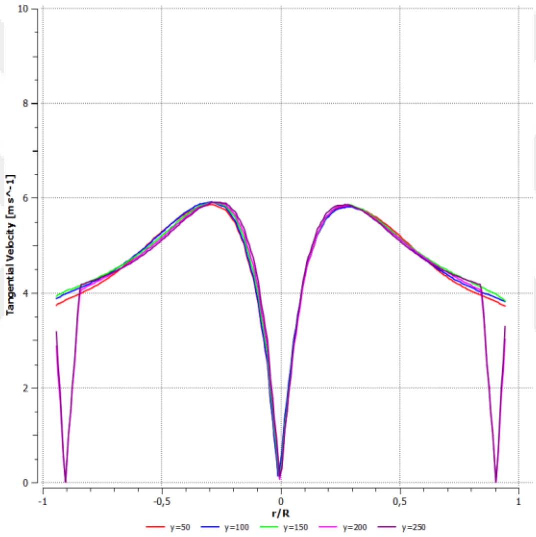
(f) B3 @ 8m/s



(g) C1 @ 8m/s

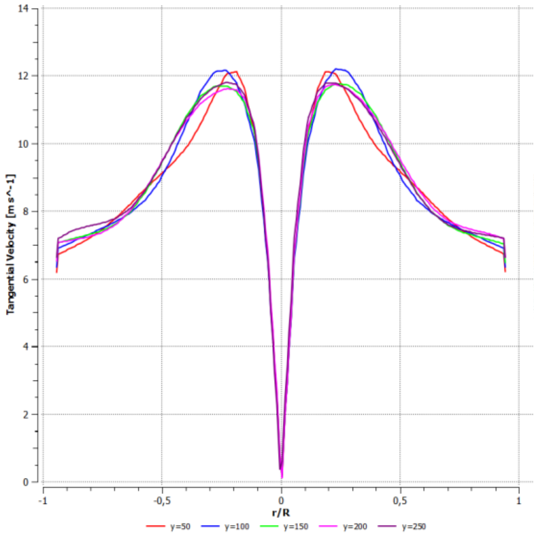


(h) C2 @ 8m/s

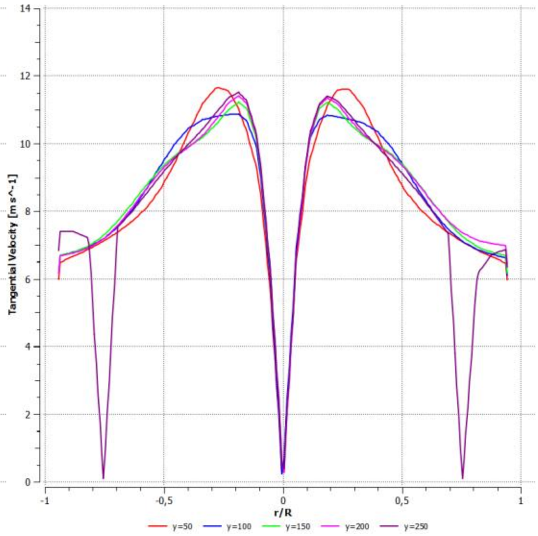


(i) C3 @ 8m/s

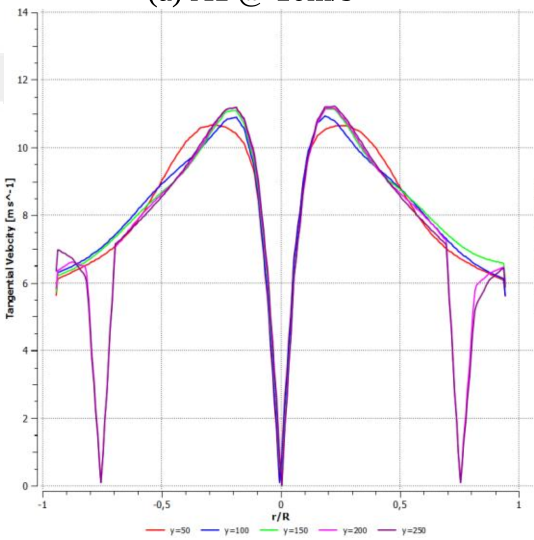
Figure 2.12 : Tangential Velocity vs Radial Position Charts @ 8m/s Inlet Velocity for (y=50,100,150,200,250 mm)



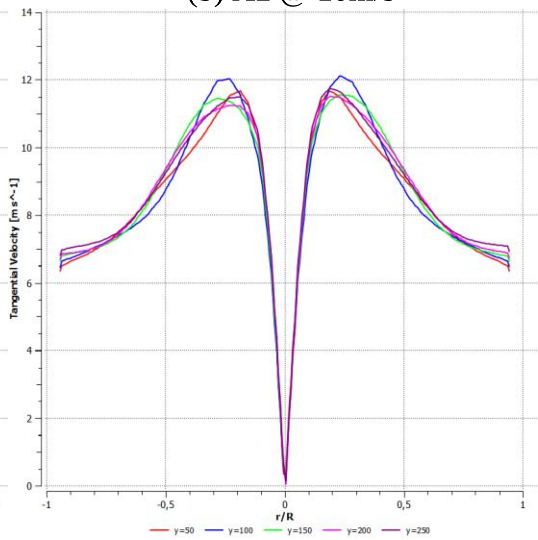
(a) A1 @ 10m/s



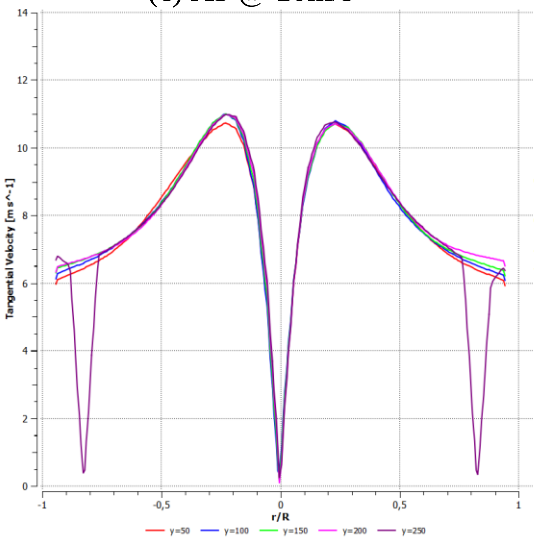
(b) A2 @ 10m/s



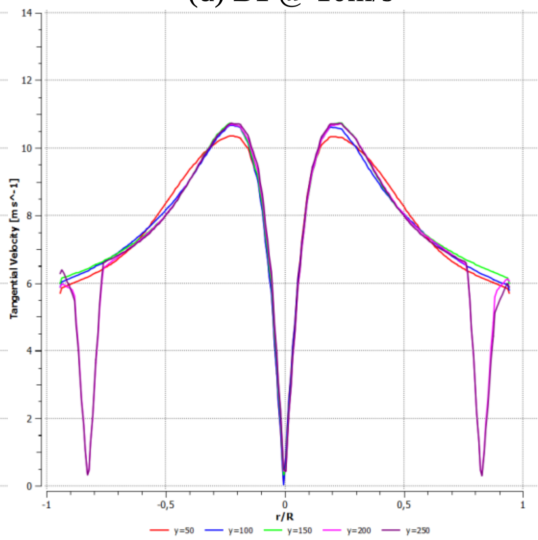
(c) A3 @ 10m/s



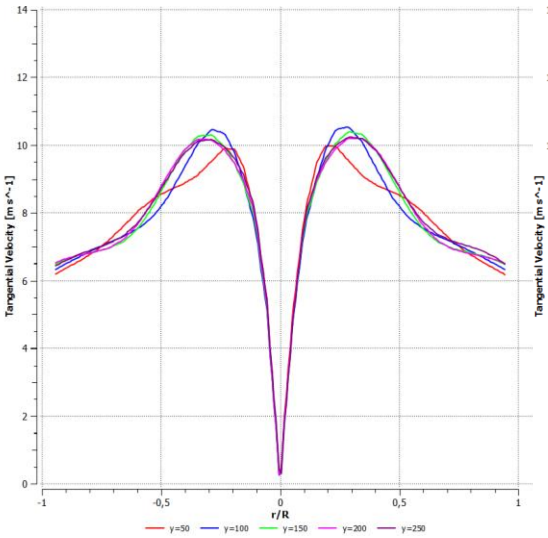
(d) B1 @ 10m/s



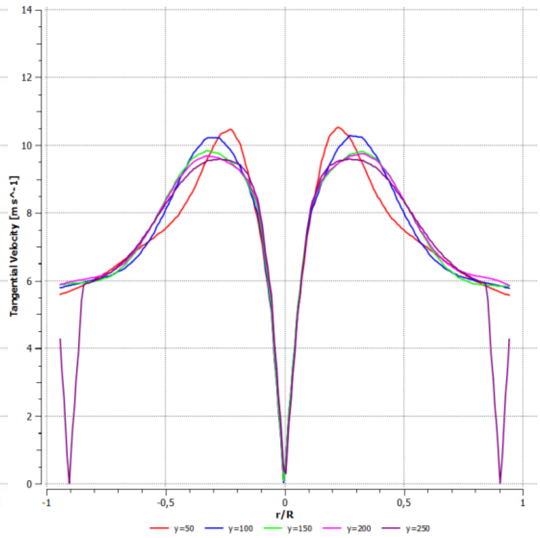
(e) B2 @ 10m/s



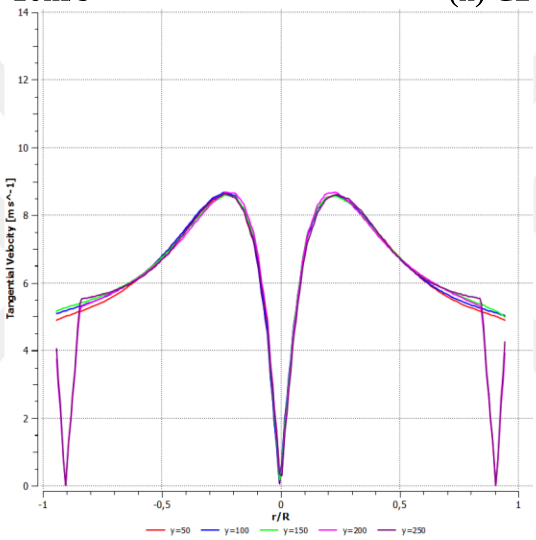
(f) B3 @ 10m/s



(g) C1 @ 10m/s

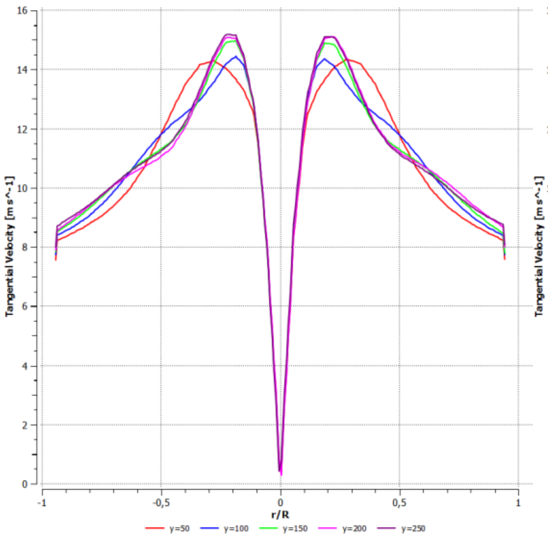


(h) C2 @ 10m/s

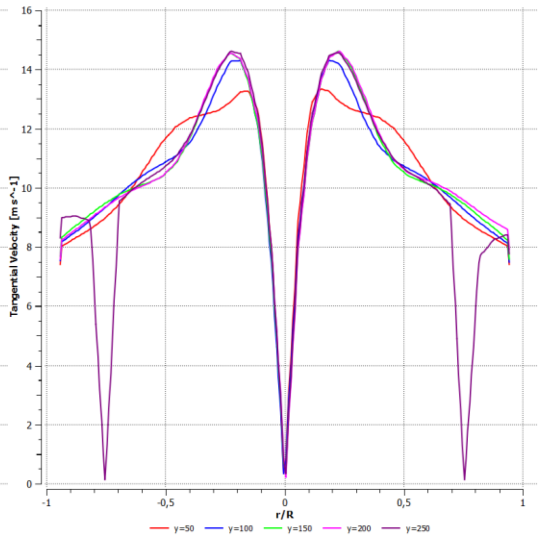


(i) C3@ 10m/s

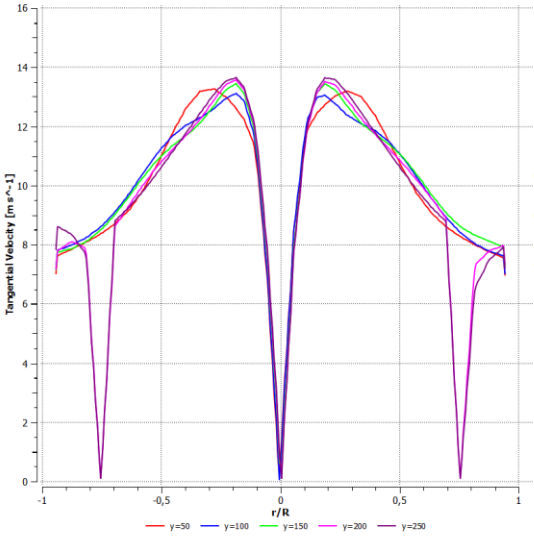
Figure 2.13 : Tangential Velocity vs Radial Position Charts @ 10m/s Inlet Velocity for (y=50,100,150,200,250 mm)



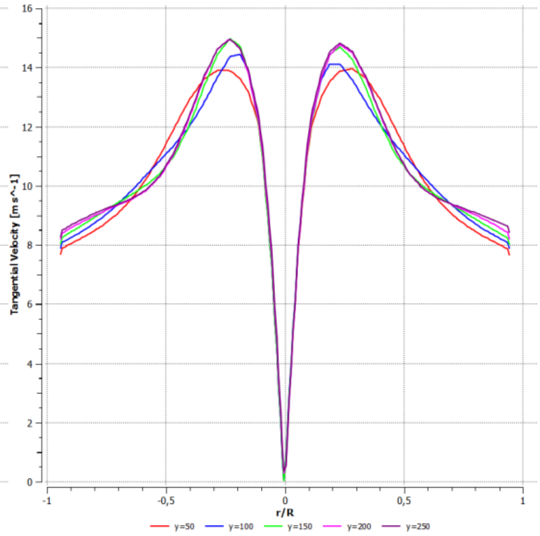
(a) A1 @ 12m/s



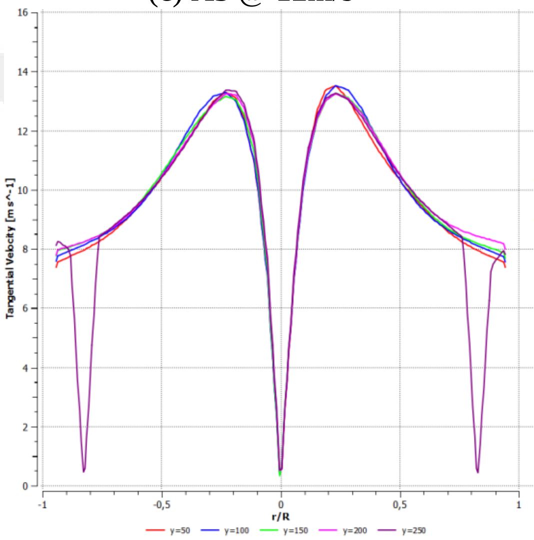
(b) A2 @ 12m/s



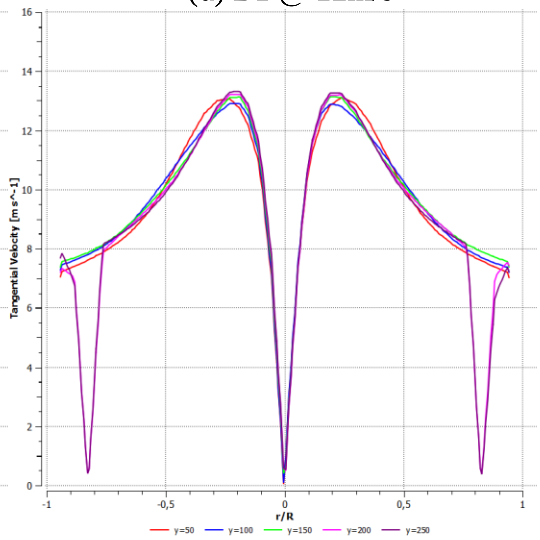
(c) A3 @ 12m/s



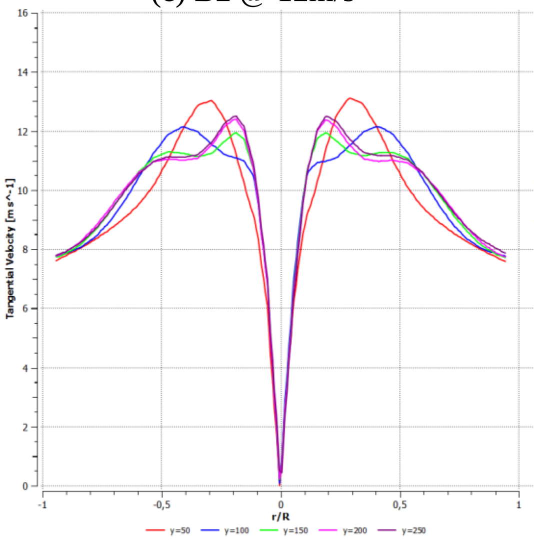
(d) B1 @ 12m/s



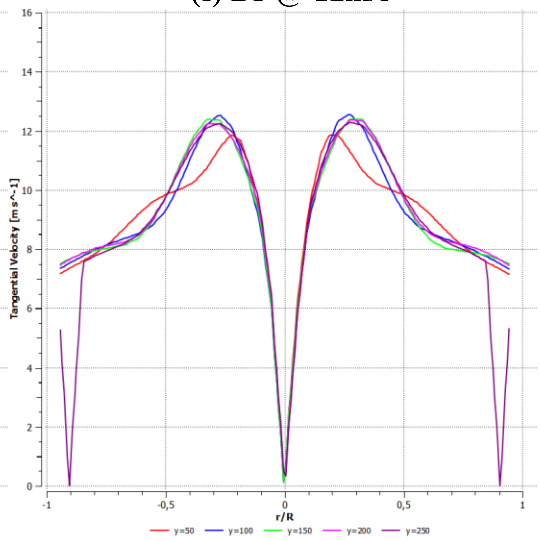
(e) B2 @ 12m/s



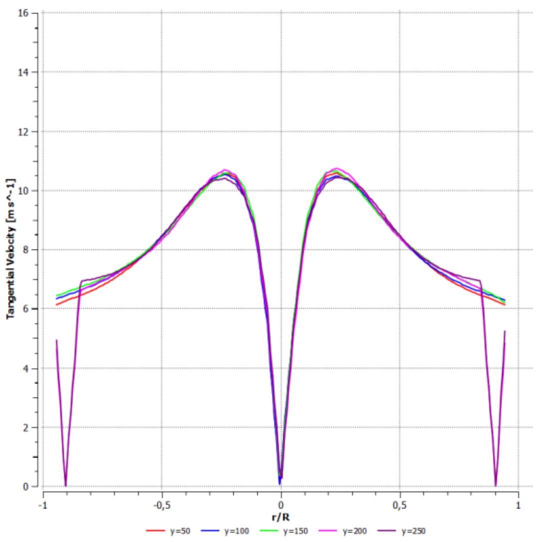
(f) B3 @ 12m/s



(g) C1 @ 12m/s



(h) C2 @ 12m/s



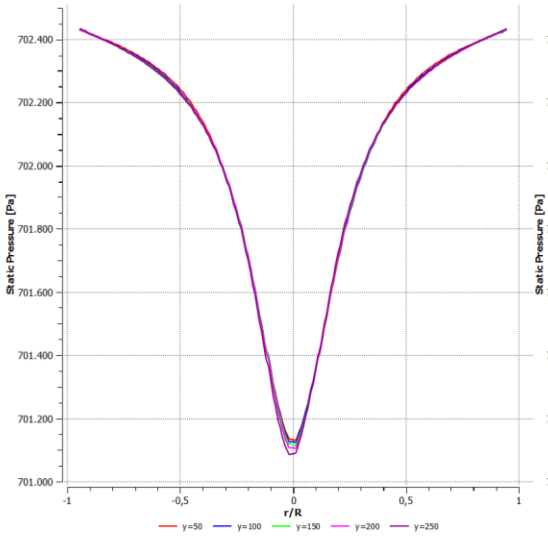
(i) C3@ 12m/s

Figure 2.14 : Tangential Velocity vs Radial Position Charts @ 12m/s Inlet Velocity for (y=50,100,150,200,250 mm)

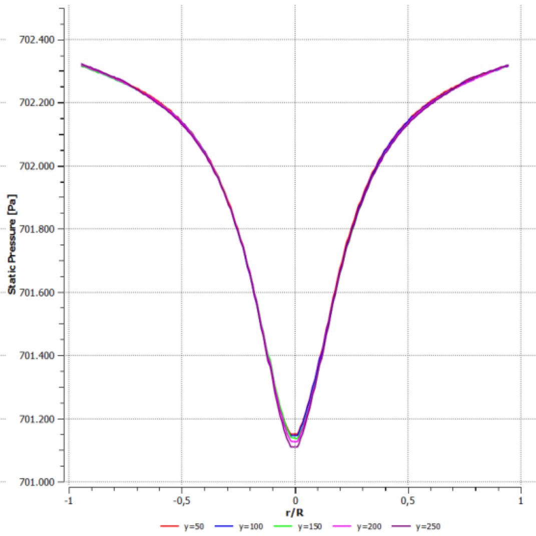
There are mainly two swirling flow occurs inside the air oil separator tank like in most of the separators (Rankine vortex) . Tangential velocity of the outer vortex (downward direction) is relatively slower when compared with the inner vortex (upward direction). For the cases A2,A3,B2,B3,C2 and C3 it can be seen that tangential velocity goes to zero velocity for a certain distance from the outer wall. This is due to the tangential velocity is plotted at five different sections and some sections are above the bottom surface of the vortex finder, so tangential velocity is zero at the vortex finder wall. In addtion, tangential velocity is zero at the center of the separator tank for all sections due to there is no swirl at the center. The results show that tangential velocity decreases when the vortex finder diameter and length increases. Moreover, increasing the inlet velocity increases the magnitude of tangential velocity remarkably. Tangential velocity reaches 20% higher than the inlet velocity in some cases.

As it is mentioned at previous chapters, pressure loss is another vital parameter for the flow inside air oil separator tank. Pressure loss inside the compressor systems directly effect the compressor performance.

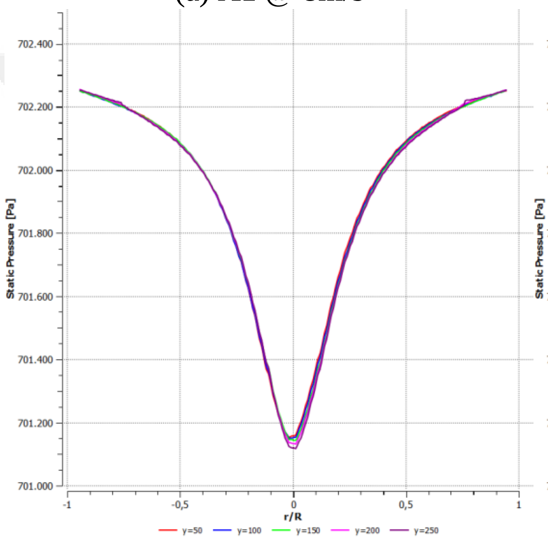
Static pressure variance from the walls of the separator tank to the center with different sections from the bottom of the separator tank is shown in Figure 2.15, Figure 2.16 and Figure 2.17.



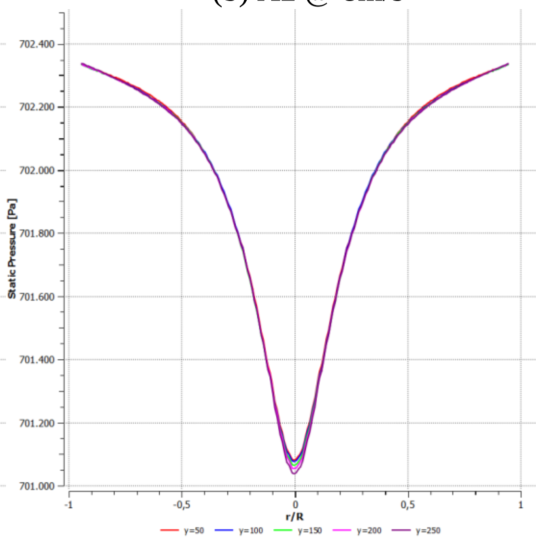
(a) A1 @ 8m/s



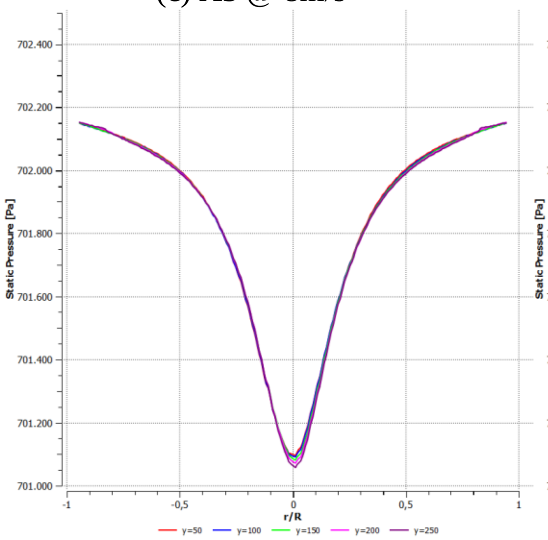
(b) A2 @ 8m/s



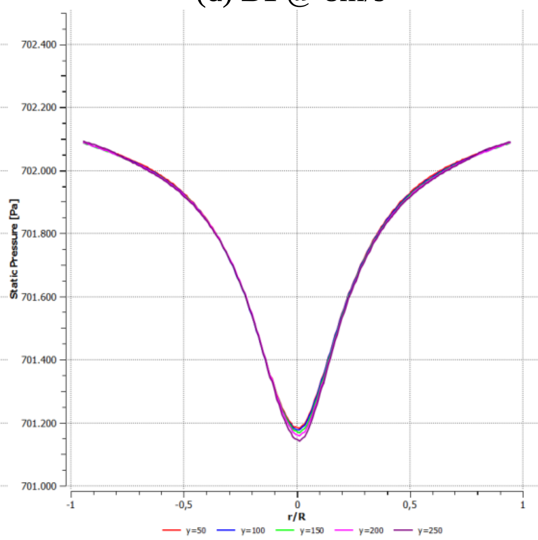
(c) A3 @ 8m/s



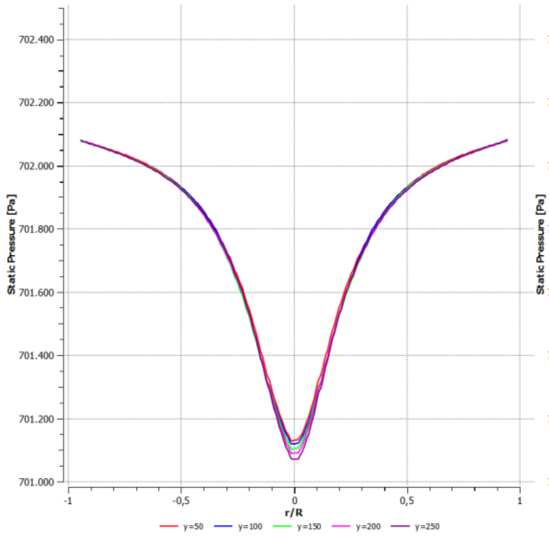
(d) B1 @ 8m/s



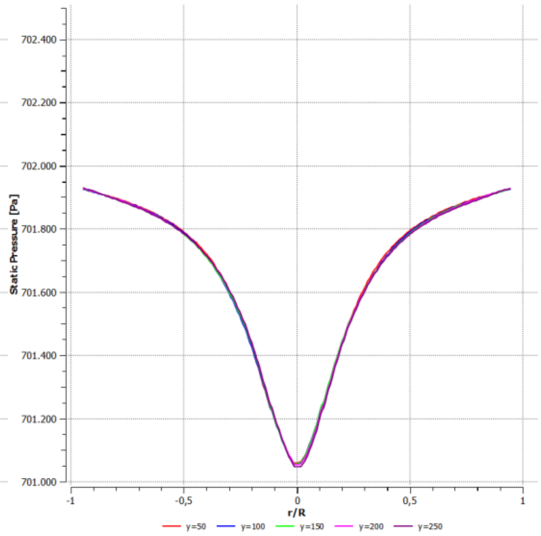
(e) B2 @ 8m/s



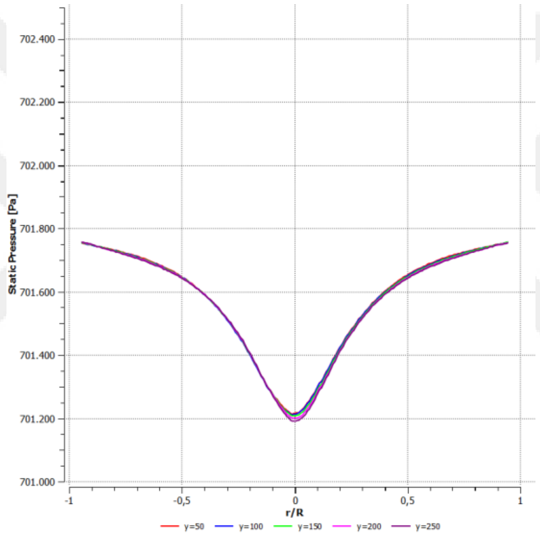
(f) B3 @ 8m/s



(g) C1 @ 8m/s

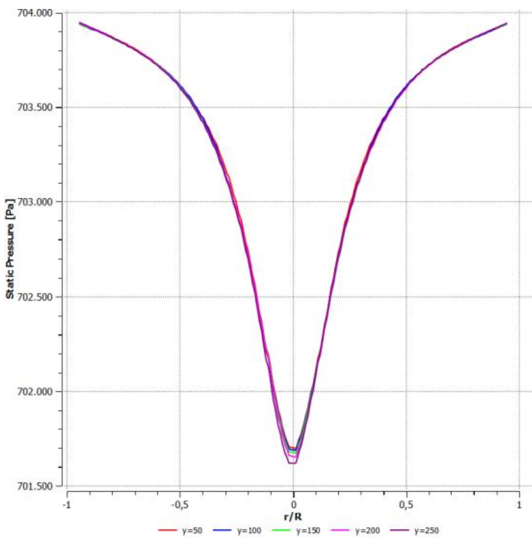


(h) C2 @ 8m/s

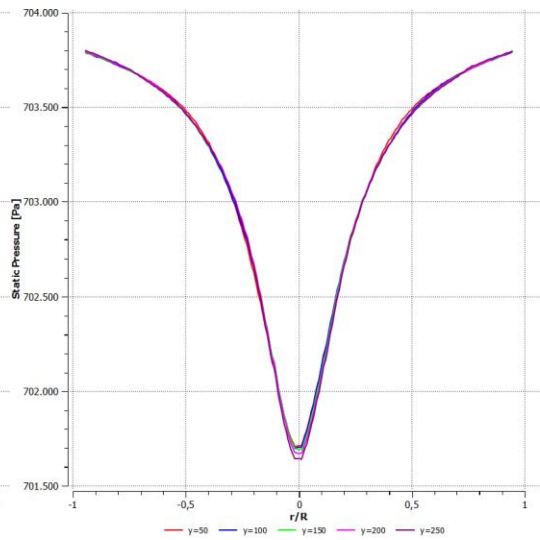


(i) C3@ 8m/s

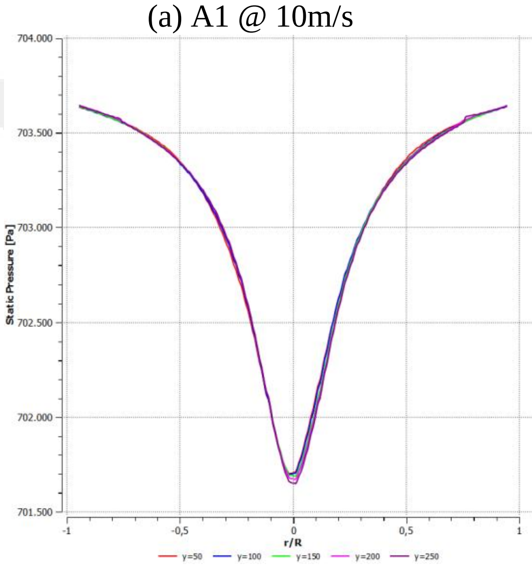
Figure 2.15 : Static Pressure vs Radial Position Charts @ 8m/s Inlet Velocity for (y=50,100,150,200,250 mm)



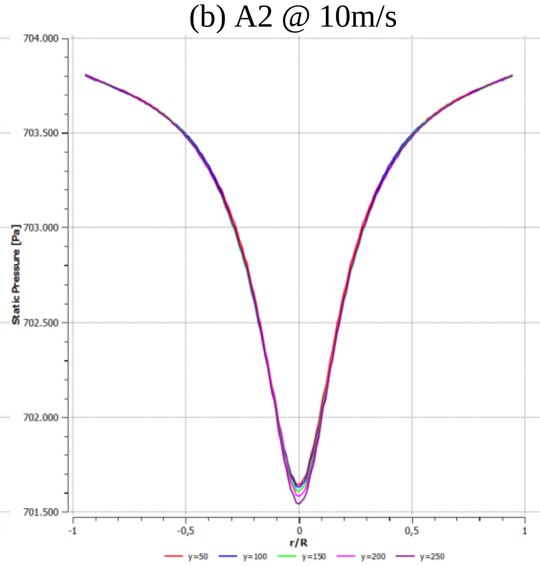
(a) A1 @ 10m/s



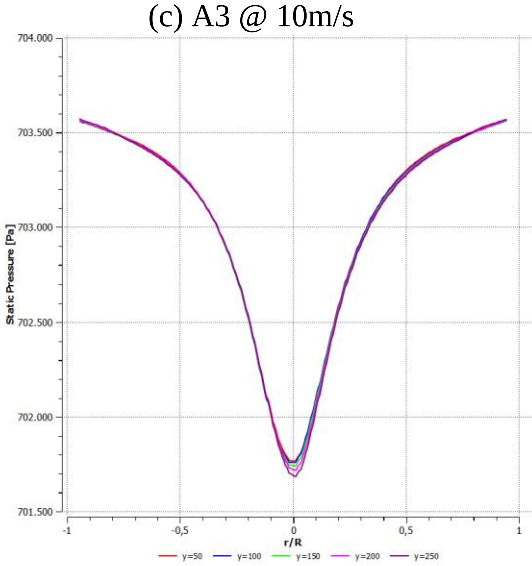
(b) A2 @ 10m/s



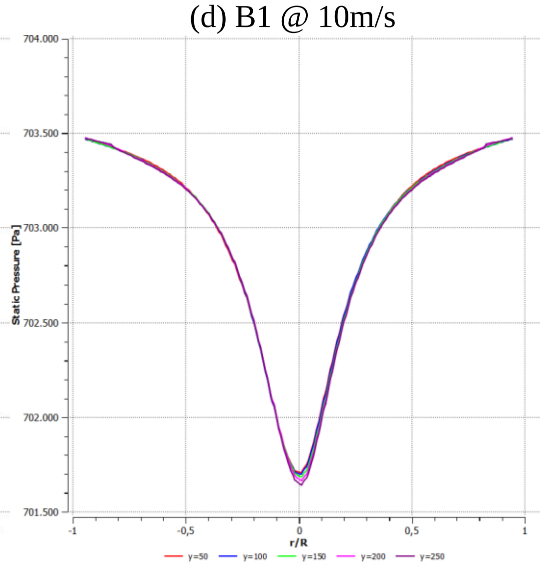
(c) A3 @ 10m/s



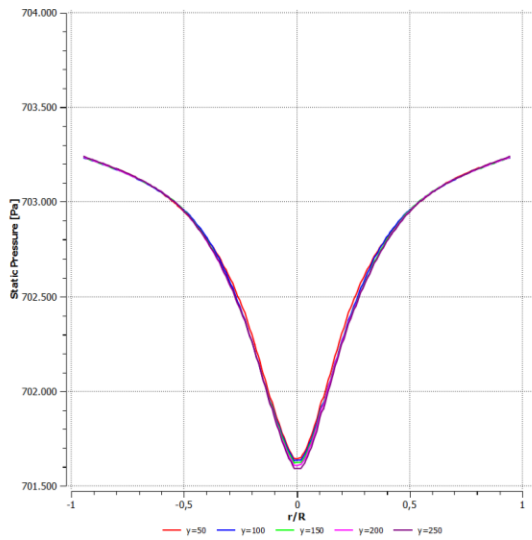
(d) B1 @ 10m/s



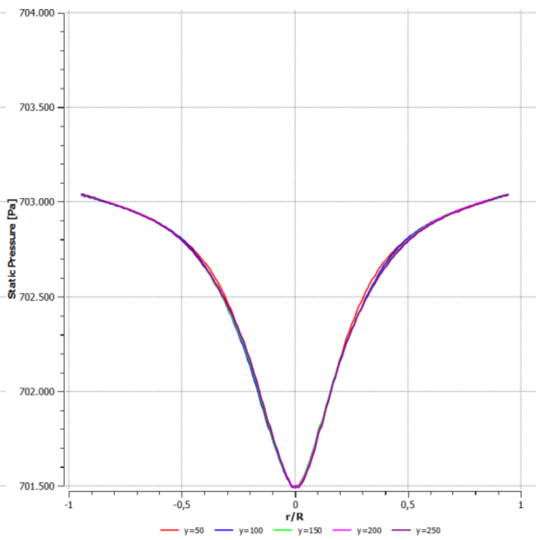
(e) B2 @ 10m/s



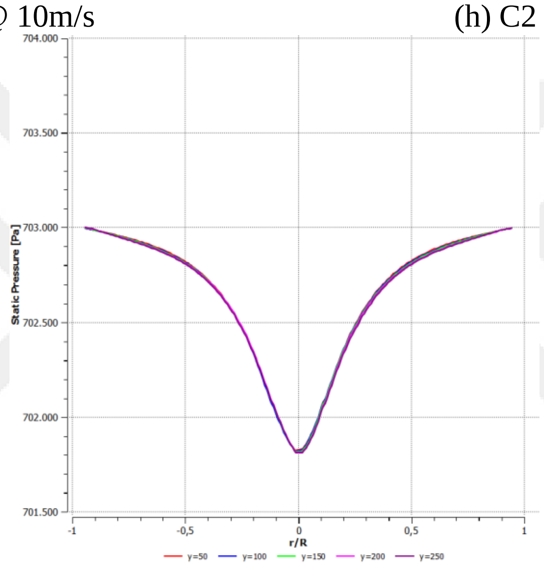
(f) B3 @ 10m/s



(g) C1 @ 10m/s

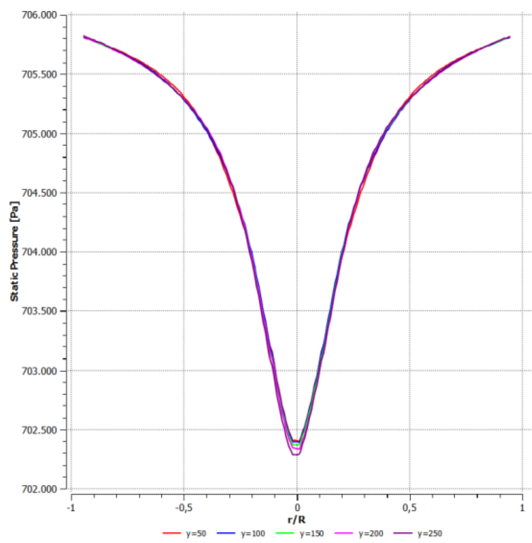


(h) C2 @ 10m/s

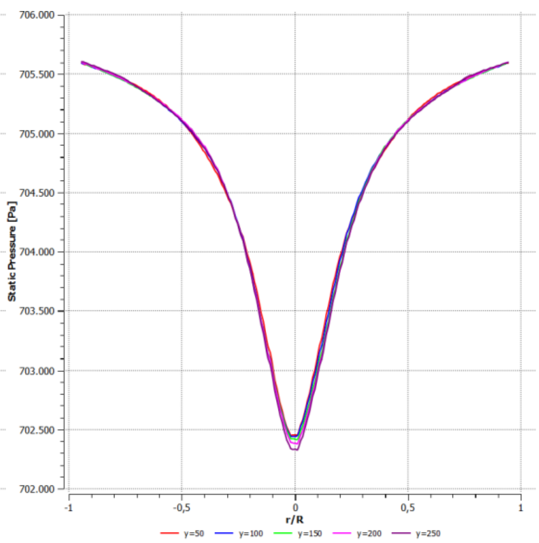


(i) C3@ 10m/s

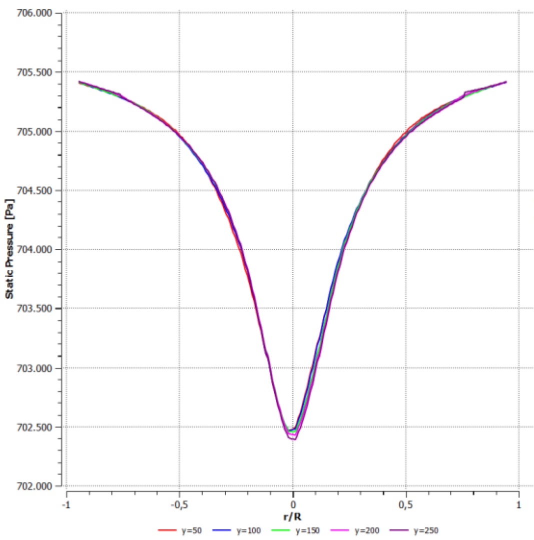
Figure 2.16 : Static Pressure vs Radial Position Charts @ 10m/s Inlet Velocity for (y=50,100,150,200,250 mm)



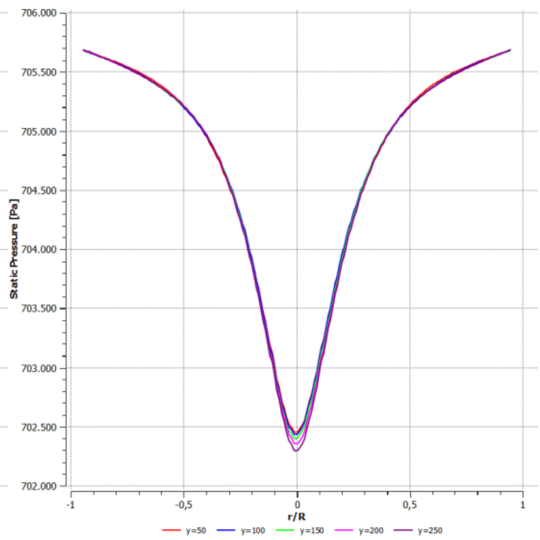
(a) A1 @ 12m/s



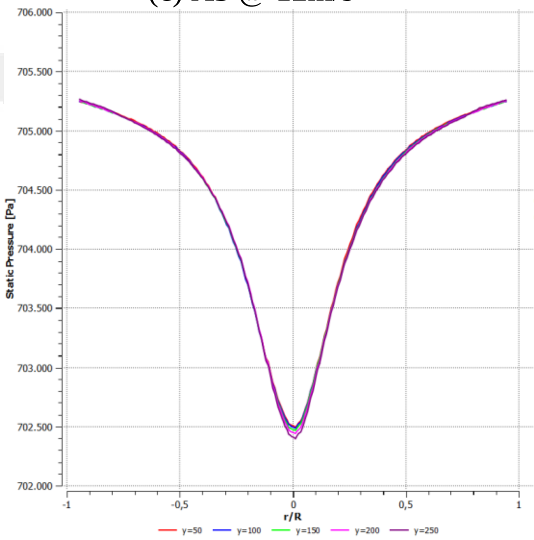
(b) A2 @ 12m/s



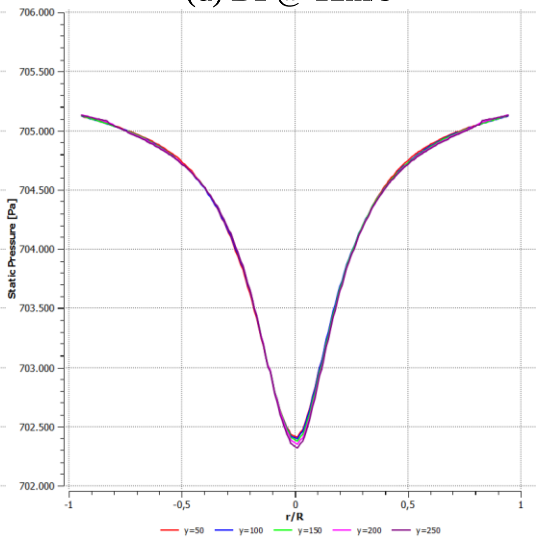
(c) A3 @ 12m/s



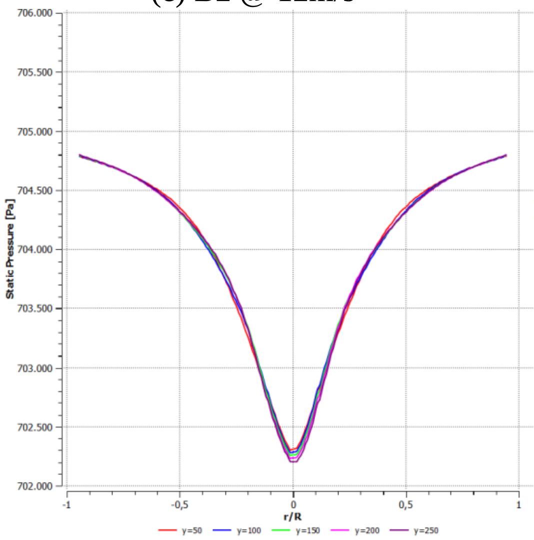
(d) B1 @ 12m/s



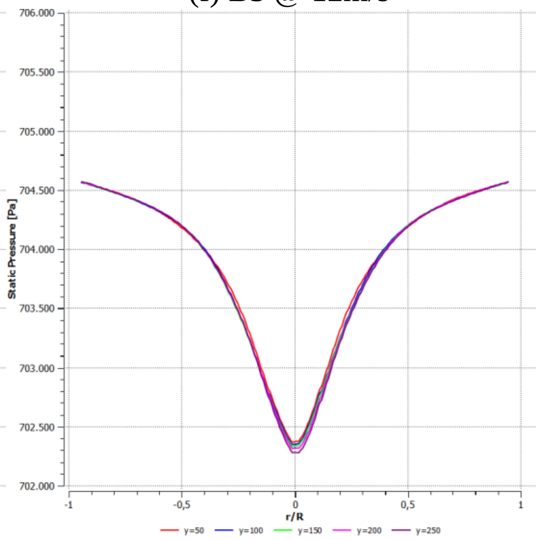
(e) B2 @ 12m/s



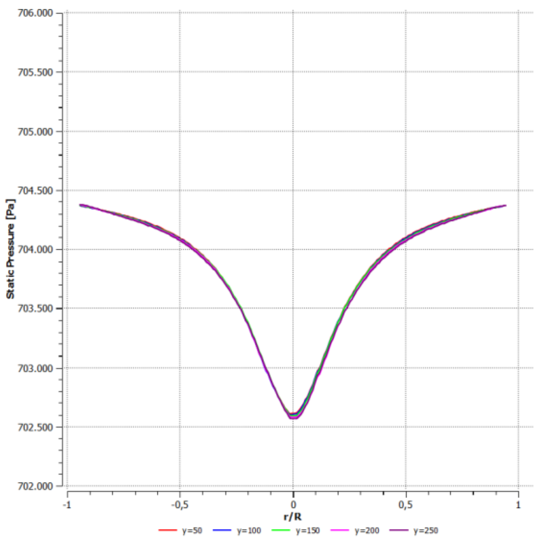
(f) B3 @ 12m/s



(g) C1 @ 12m/s



(h) C2 @ 12m/s

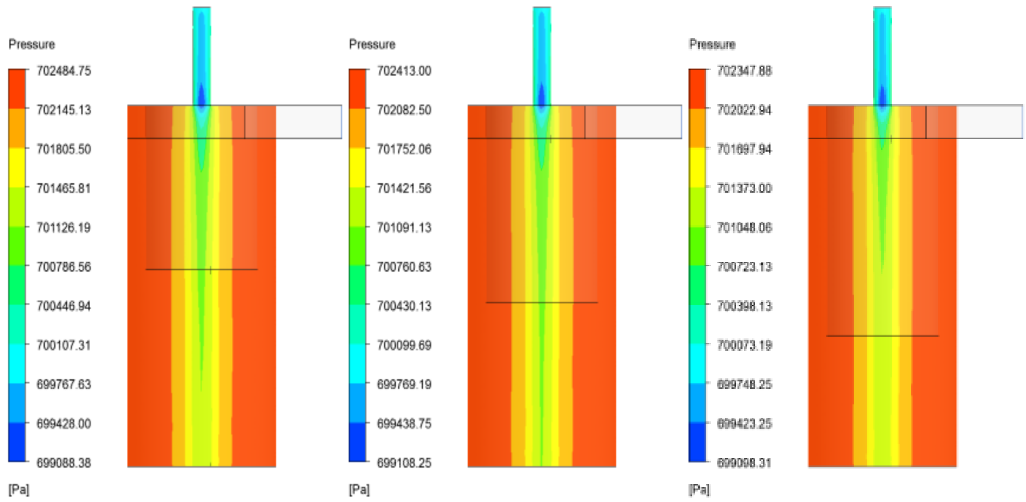


(i) C3@ 12m/s

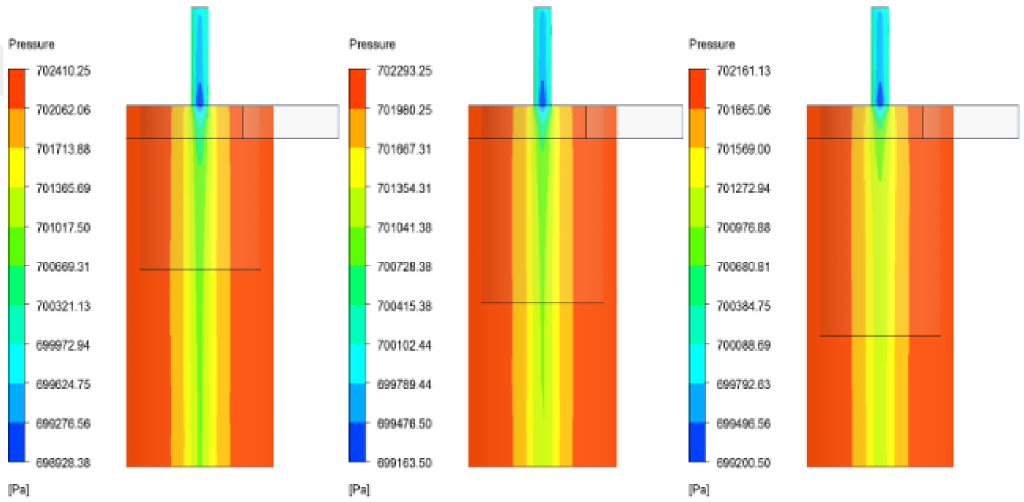
Figure 2.17 : Static Pressure vs Radial Position Charts @ 12m/s Inlet Velocity for (y=50,100,150,200,250 mm)

Static pressure curves showed symmetrical behaviour from the center axis of the air oil separator tank. Static pressure decreased sharply from the walls of the tank to the center.Pressure drop has increased dramatically with the increment in inlet velocity. All of the sections has shown nearly same static pressure values due to there is no disturbance on the flow which would result in pressure loss in axial direction. Studies has shown that negative pressure has created at the center of the cyclones. [25-26] However, negative pressure has not been formed in this study. That might be due to there was no opening at the bottom of the tank and it consists pressurized air inside. Vortex finder diamater has relatively more effect on pressure loss than the height. Increase in vortex finder diamater and length decreases the static pressure loss which can be stated as a positive effect for air oil separator performance.

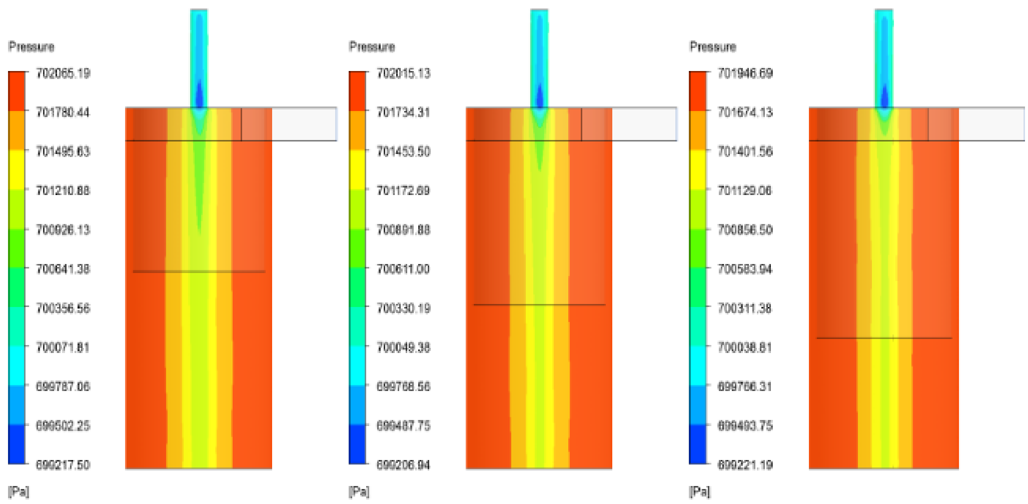
Static pressure contuars on a vertical plane has for different inlet velocities has shown in Figure 2.18, Figure 2.19, Figure 2.20.



(a) A1@8m/s (b) A2@8m/s (c) A3@8m/s

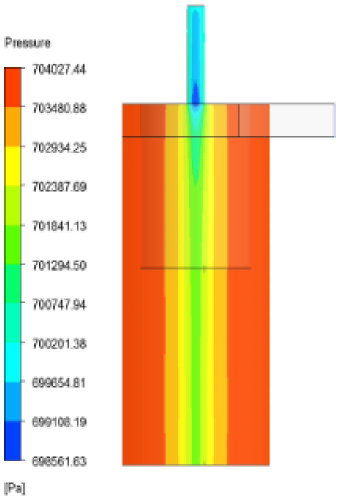


(d) B1@8m/s (e) B2@8m/s (f) B3@8m/s

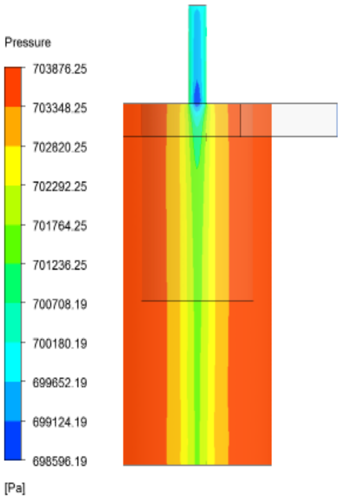


(g) C1@8m/s (h) C2@8m/s (i) C3@8m/s

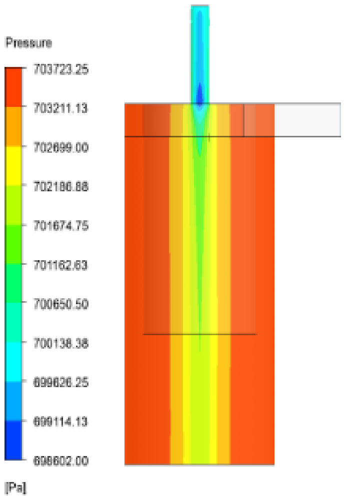
Figure 2.18 : Static Pressure Contuar Plots @ 8 m/s Inlet Velocity



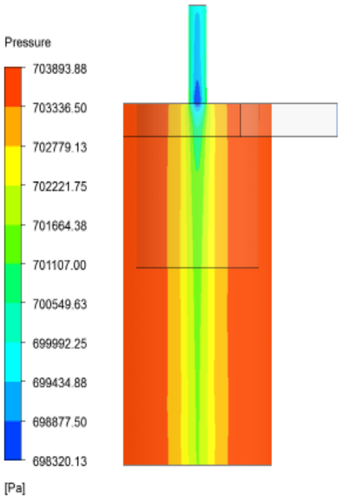
(a) A1@10m/s



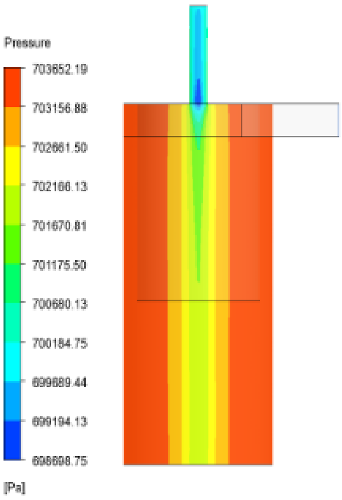
(b) A2@10m/s



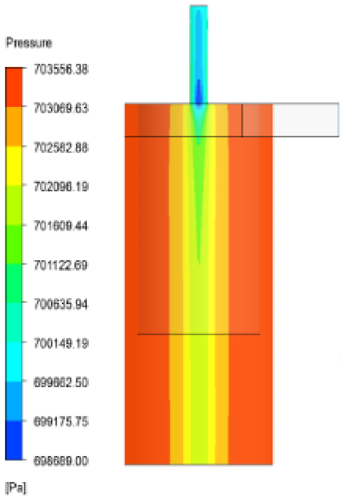
(c) A3@10m/s



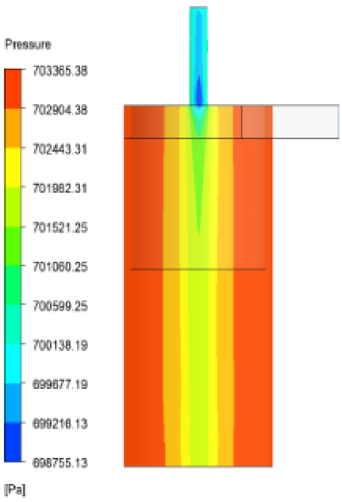
(d) B1@10m/s



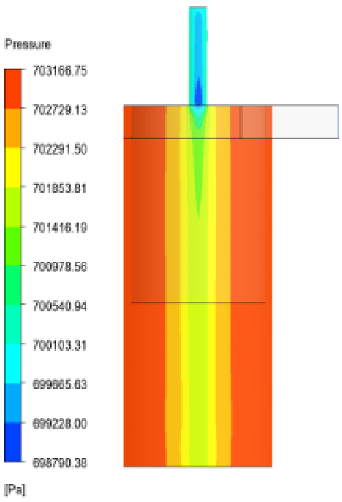
(e) B2@10m/s



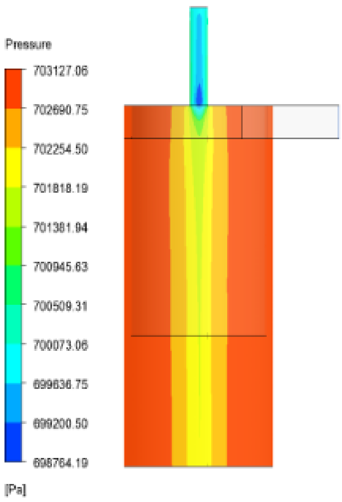
(f) B3@10m/s



(g) C1@10m/s

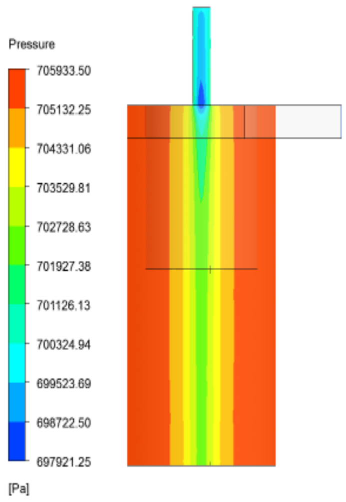


(h) C2@10m/s

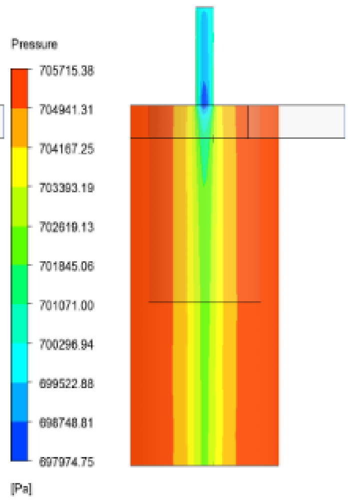


(i) C3@10m/s

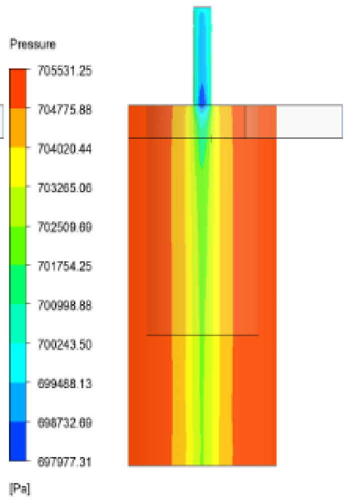
Figure 2.19 : Static Pressure Contuar Plots @ 10 m/s Inlet Velocity



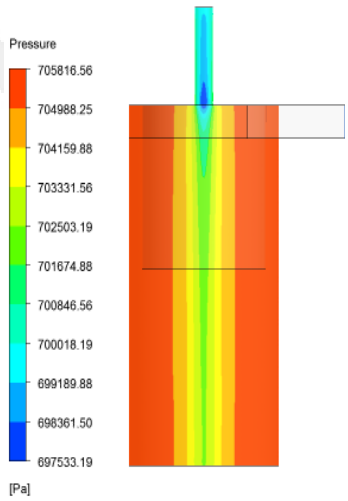
(a) A1@12m/s



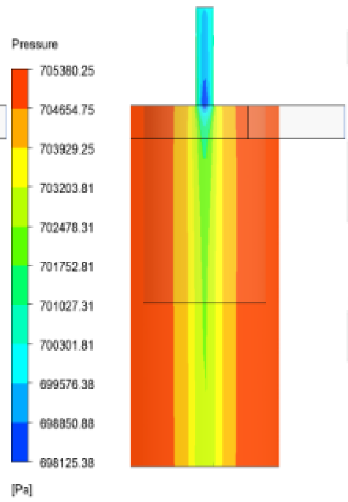
(b) A2@12m/s



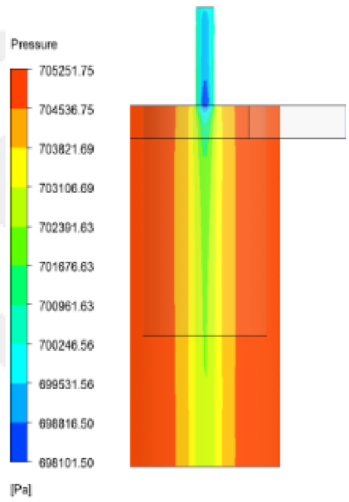
(c) A3@12m/s



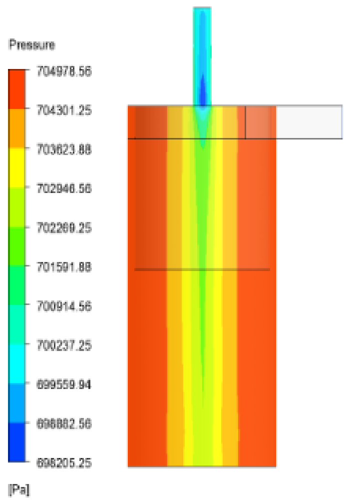
(d) B1@12m/s



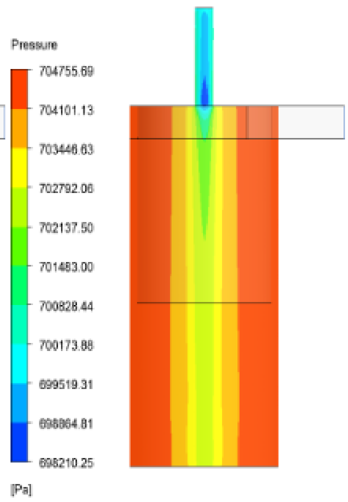
(e) B2@12m/s



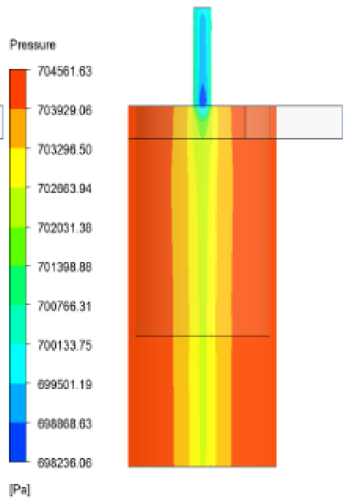
(f) B3@12m/s



(g) C1@12m/s



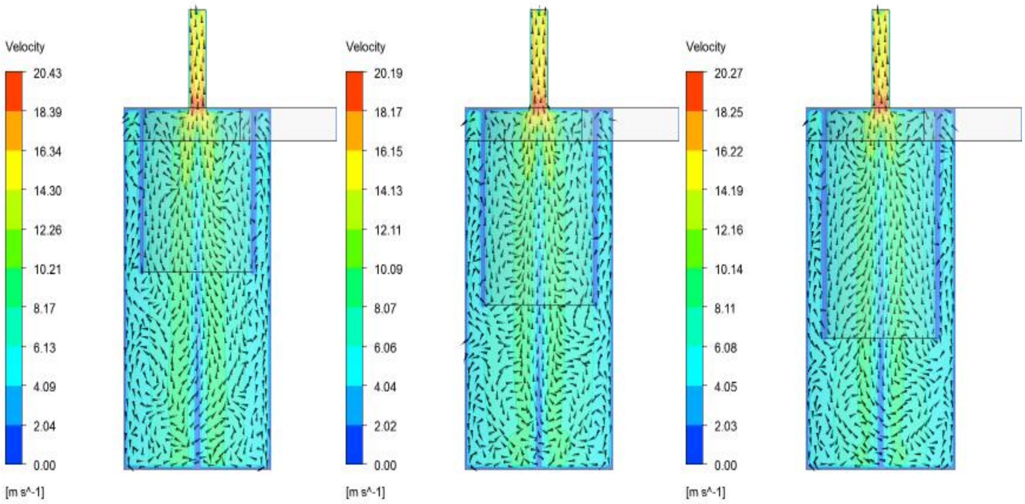
(h) C2@12m/s



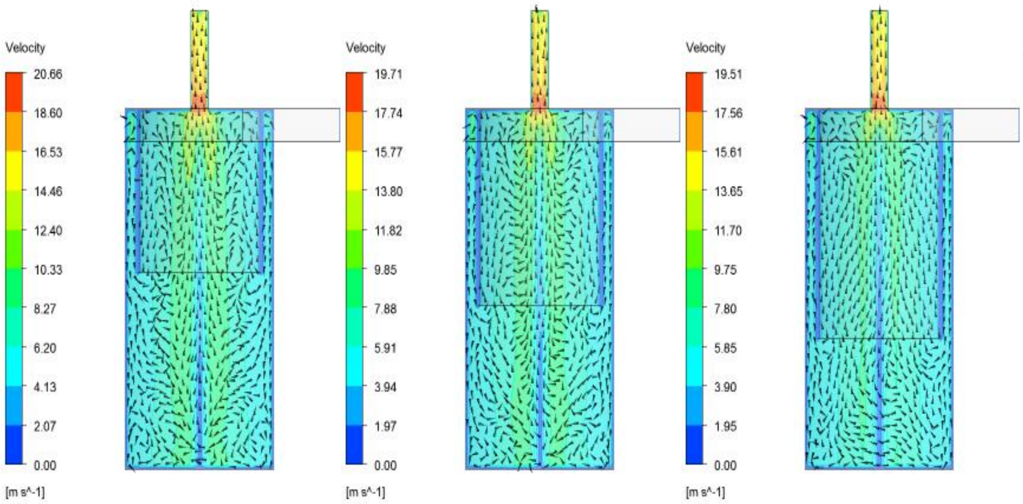
(i) C3@12m/s

Figure 2.20 : Static Pressure Contuar Plots @ 12 m/s Inlet Velocity

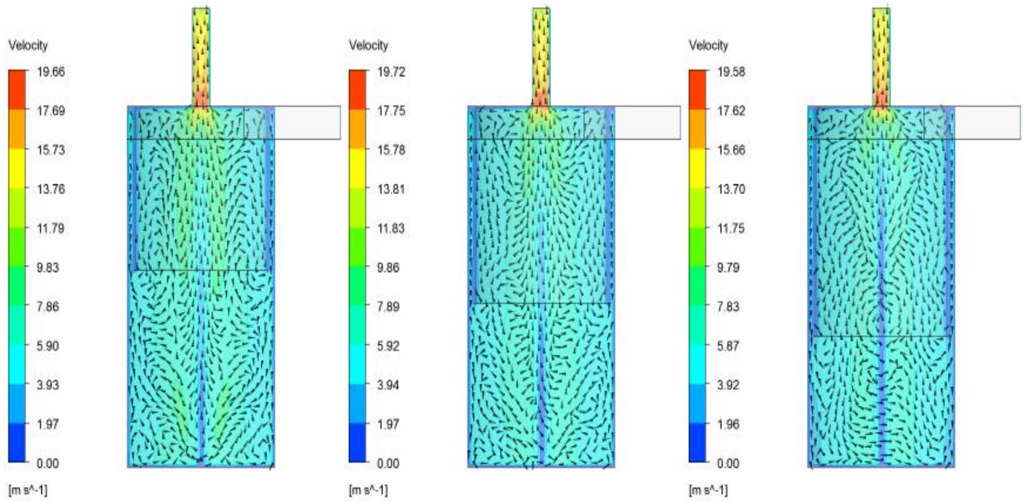
Static pressure loss is higher at high inlet velocities. At the center of the separator tank static pressure drops down due to inner vortex. Vacuum area has occured at the outlet port of the tank for all of the cases. Some further studies can be performed on the diameter of the outlet port in order to lower the static pressure drop at this area. As it is mentioned above increase in vortex finder diamater and height reduces the static pressure drop.



(a) A1@8m/s (b) A2@8m/s (c) A3@8m/s

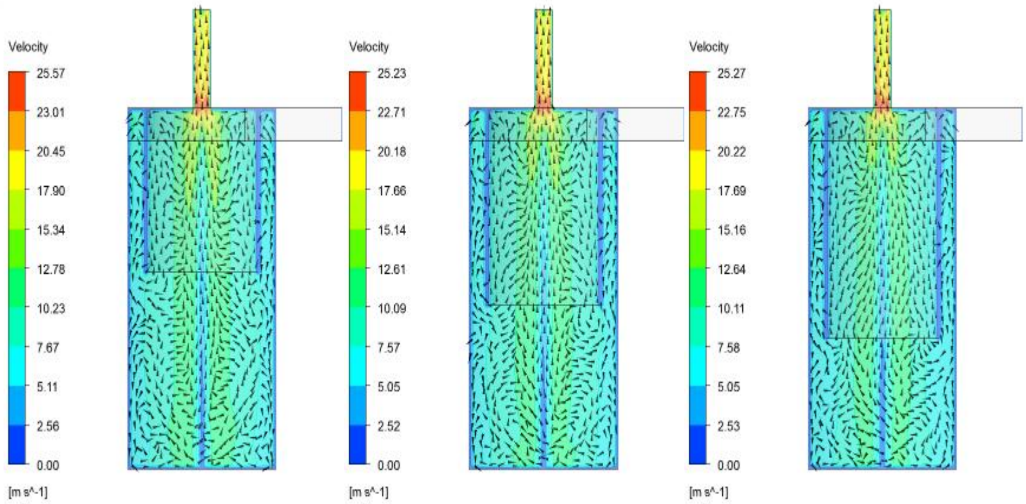


(d) B1@8m/s (e) B2@8m/s (f) B3@8m/s

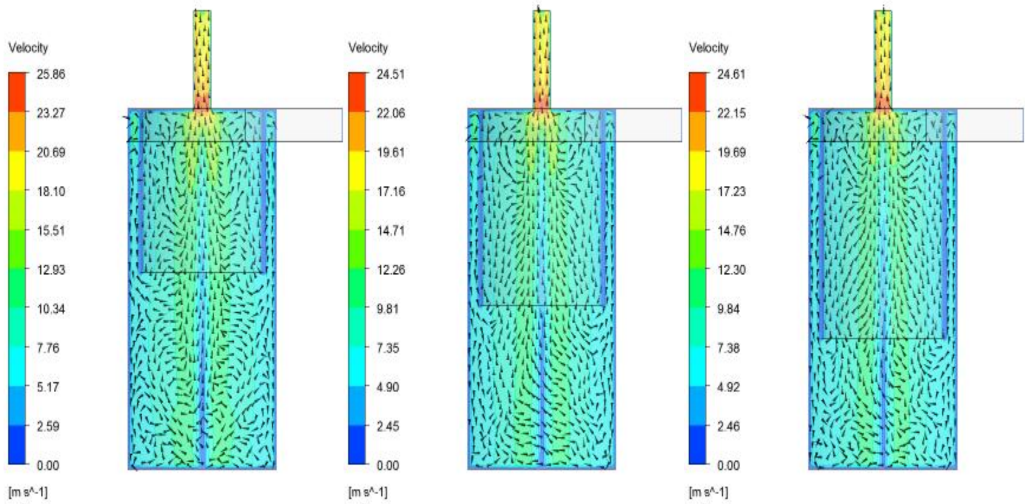


(g) C1@8m/s (h) C2@8m/s (i) C3@8m/s

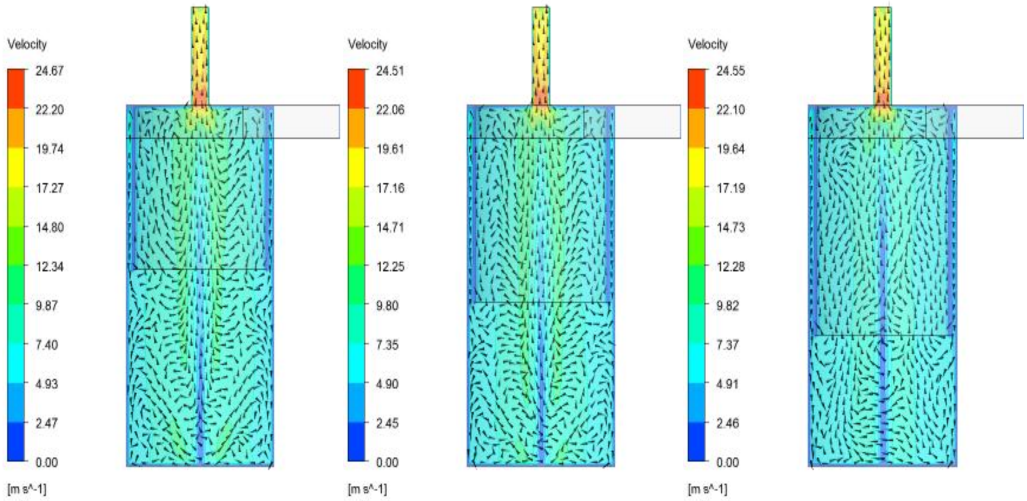
Figure 2.21 : Velocity Magnitude and Vectors @ 8 m/s Inlet Velocity



(a) A1@10m/s (b) A2@10m/s (c) A3@10m/s

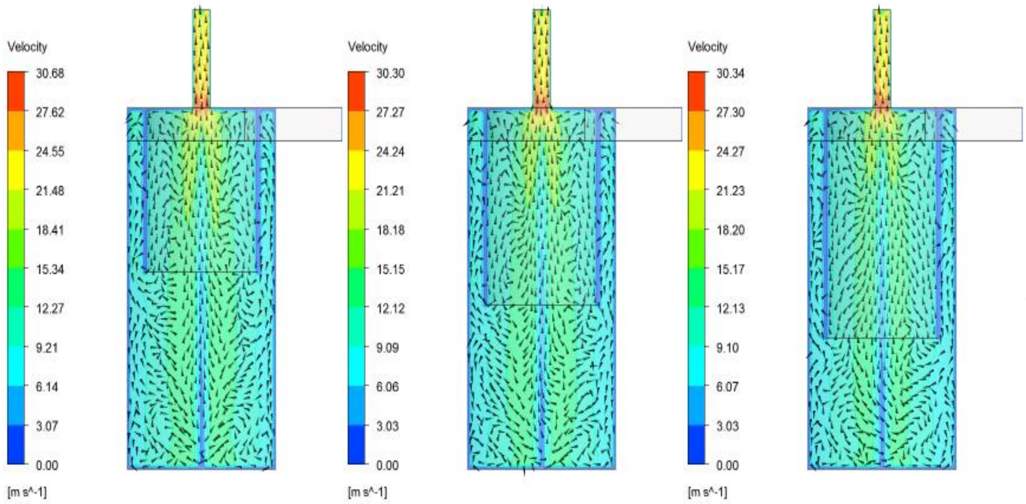


(d) B1@10m/s (e) B2@10m/s (f) B3@10m/s

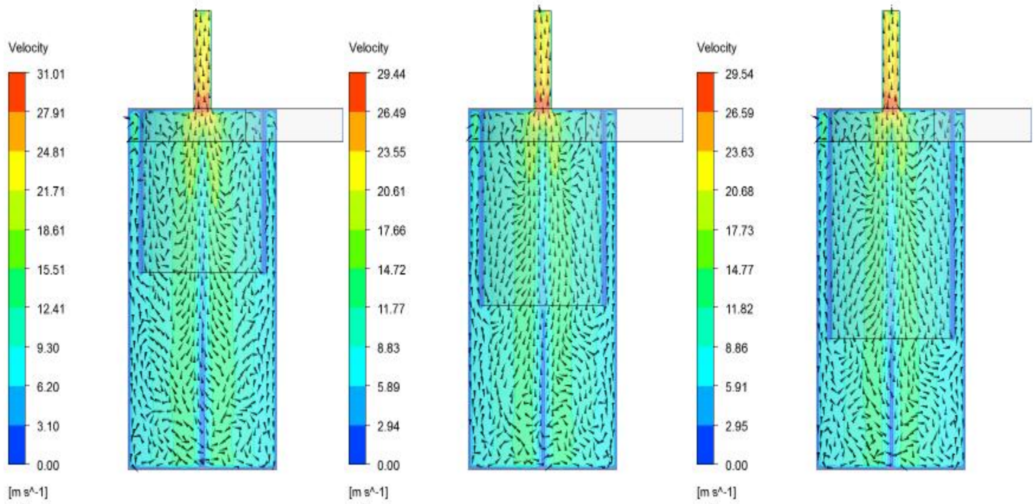


(g) C1@10m/s (h) C2@10m/s (i) C3@10m/s

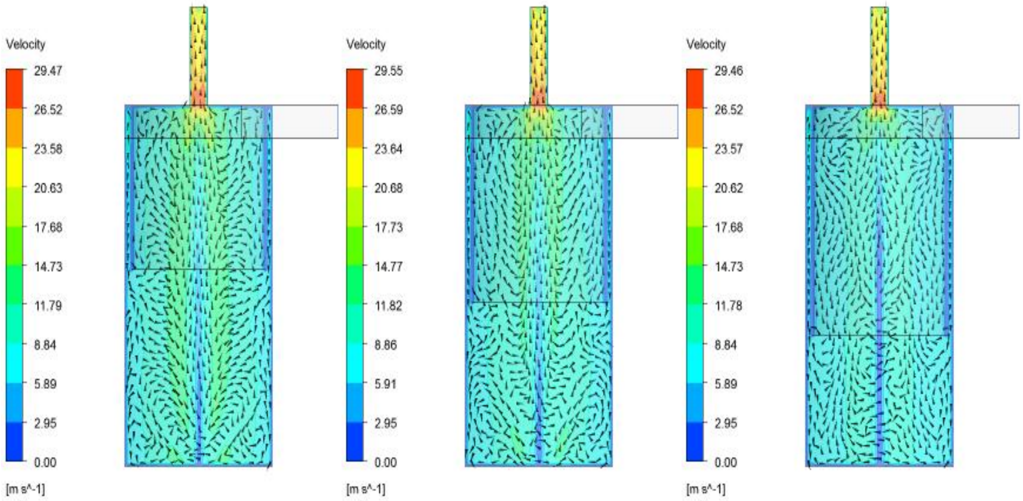
Figure 2.22 : Velocity Magnitude and Vectors @ 10 m/s Inlet Velocity



(a) A1@12m/s (b) A2@12m/s (c) A3@12m/s



(d) B1@12m/s (e) B2@12m/s (f) B3@12m/s



(g) C1@12m/s (h) C2@12m/s (i) C3@12m/s

Figure 2.23 : Velocity Magnitude and Vectors @ 12 m/s Inlet Velocity

Figure 2.21, Figure 2.22 and Figure 2.23 illustrates the velocity magnitude and vectors at the center plane of the air oil separator tank. It can be seen that for all cases small vortex have occured at the bottom right and left corners of the tank. Increase in vortex finder diamater resulted in short cut flow just after the bottom surface of the vortex finder. Air is directed to outlet without completing its rotation. Some amount of small oil droplets can be carried to outlet due to short circuit flow. Trefz and Muschelknautz has found out that nearly 10% of flow short circuits the cyclones [27]. Lastly, due to the small diameter of the outlet port velocity magnitude increases nearly 2-3 times of the inlet velocity that creates extra pressure drop.

Different cases have been compared with eachother according to separation efficiency and pressure loss which has been occured from inlet to outlet. Separation efficiency is calculated as follows:

$$\eta = \left(1 - \frac{m_{oil,out}}{m_{oil,total}}\right) * 100\%$$

Separation efficiency of different vortex finder dimensions with different inlet velocities is shown in Figure 2.24.

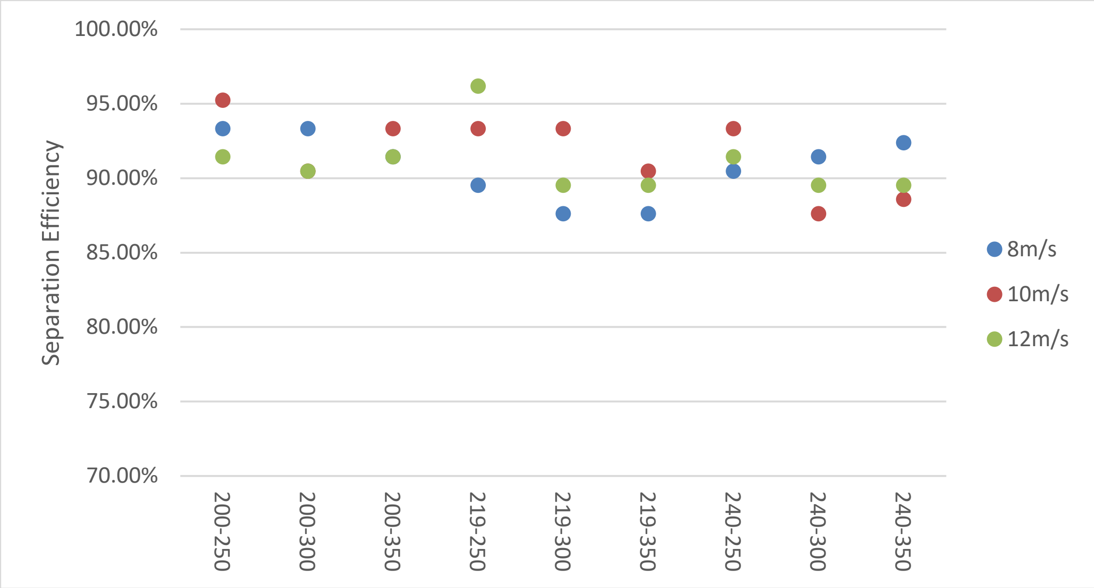


Figure 2.24 : Separation Efficiency vs Vortex Finder Dimensions (d - h)

Due to dimensional and inlet velocity changes it is hard to say that separation efficiency followed a trend. In some cases, increase in inlet velocity decreases the separation efficiency and some cases showed the opposite. However, the effect of dimensional changes were more remarkable. Increase in vortex finder diameter and height decreased the separation efficiency. Maximum separation efficiency has reached at 219 mm vortex finder diameter and 250 mm vortex finder height with 12 m/s inlet velocity.

Furthermore, static pressure loss is the another vital parameter for air oil separator tanks in oil injected rotary screw compressors. Compressors consume too much energy in order to produce compressed air. Therefore, even a slight pressure drop inside the compressor system can result in waste of energy in time. Static pressure loss of studied cases has compared in Figure 2.25.

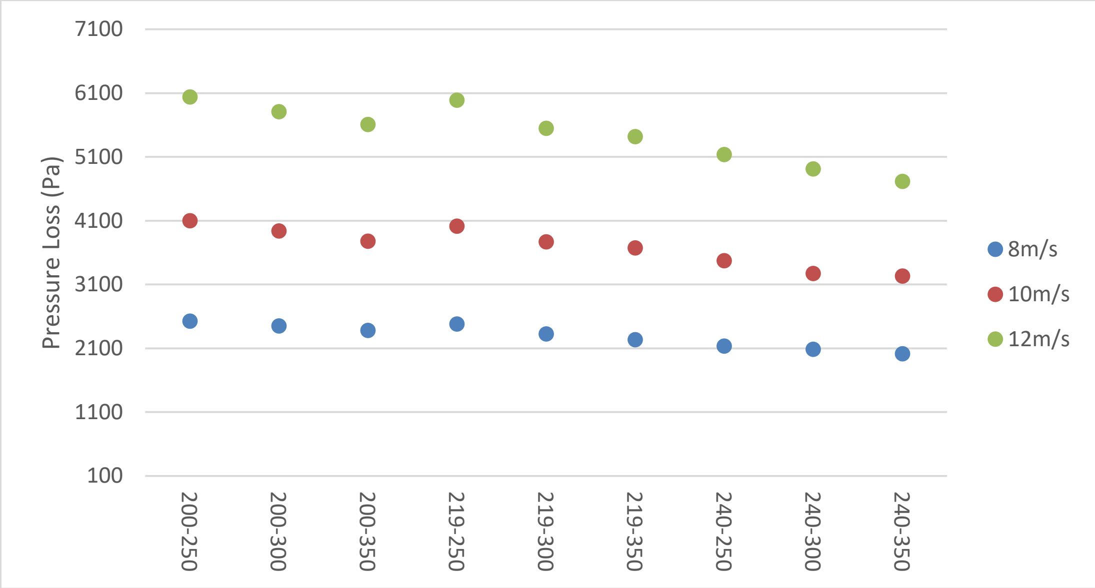


Figure 2.25 : Pressure Loss vs Vortex Finder Dimensions (d - h)

Pressure loss has increased exponantially with the increased velocity. Vortex finder dimensional differences changed pressure drop slightly. It was found out that static pressure loss has decreased when the vortex finder diameter and height has increased. Minimum pressure drop has achieved at the case of vortex finder diameter 240 mm and height 350 mm with 8 m/s inlet velocity.

It is understood that change in vortex finder dimensions has shown opposite effect on separation efficiency and pressure loss. Optimization has performed by taking the percentage differences from maximum separation efficiency and minimum pressure loss inside the oil separator tank. Table 2.4 illustrates the percentage differences from the most efficient point of separation efficiency and pressure drop inside the air oil separator tank.

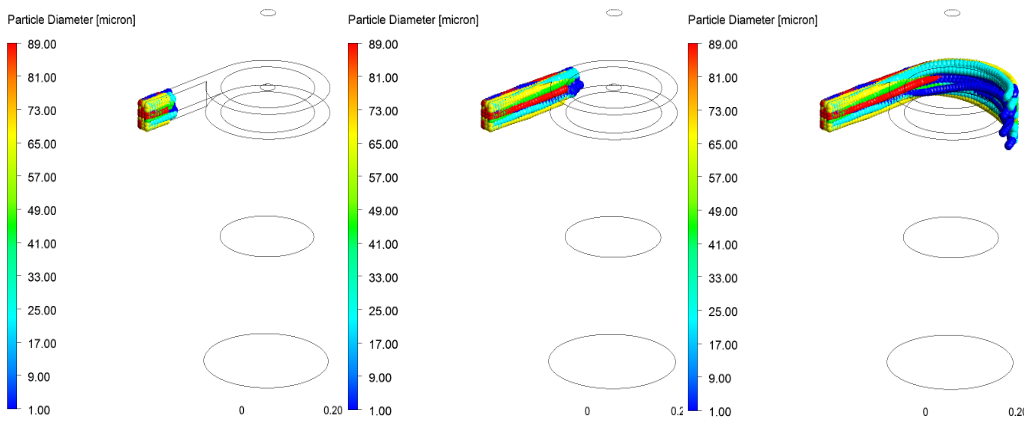
Table 2.4 : Separation Efficiency & Pressure Drop Results

Vortex Finder Diameter (d – h)	Inlet Velocity (m/s)	Separation Efficiency (%)	Pressure Loss (Pa)	Difference Separation Efficiency from Max.	Difference Pressure Loss from Min.
200-250	8	93.3	2527	-3 %	25 %
200-300	8	93.3	2451	-3 %	22 %
200-350	8	91.4	2380	-5 %	18 %
219-250	8	89.5	2480	-7 %	23 %
219-300	8	87.6	2326	-9 %	15 %
219-350	8	87.6	2235	-9 %	11 %
240-250	8	90.5	2133	-6 %	6 %

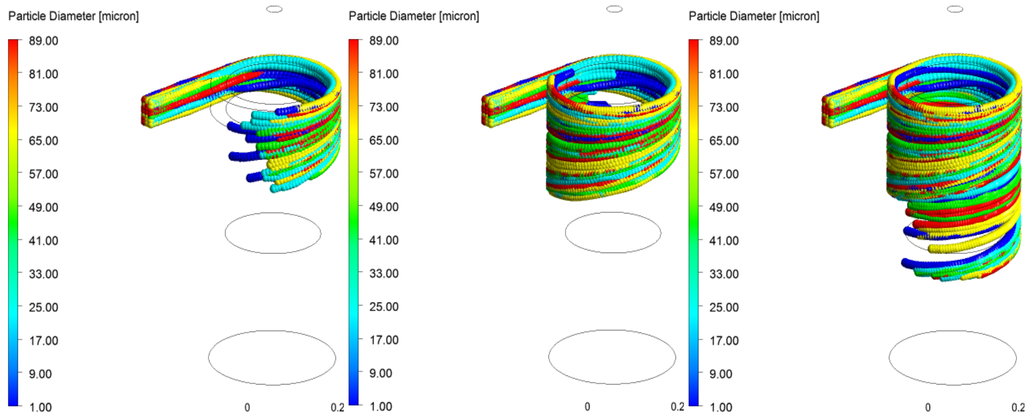
240-300	8	91.4	2083	-5 %	3 %
240-350	8	92.3	2015	-4 %	0 %
200-250	10	95.2	4099	-1 %	103 %
200-300	10	90.5	3938	-6 %	95 %
200-350	10	93.3	3776	-3 %	87 %
219-250	10	93.3	4013	-3 %	99 %
219-300	10	93.3	3768	-3 %	87 %
219-350	10	90.5	3671	-6 %	82 %
240-250	10	93.3	3473	-3 %	72 %
240-300	10	87.6	3274	-9 %	62 %
240-350	10	88.6	3235	-8 %	61 %
200-250	12	91.4	6041	-5 %	200 %
200-300	12	90.5	5808	-6 %	188 %
200-350	12	91.4	5611	-5 %	178 %
219-250	12	96.2	5989	0 %	197 %
219-300	12	89.5	5547	-7 %	175 %
219-350	12	89.5	5418	-7 %	169 %
240-250	12	91.4	5136	-5 %	155 %
240-300	12	89.5	4912	-7 %	144 %
240-350	12	89.5	4717	-7 %	134 %

Separation efficiency and pressure loss of 27 cases has evaluated. Separation efficiency is considered as most critical parameter while deciding on the most efficient case. It is observed that vortex finder diameter of 200 mm and height of 300 mm with 8 m/s inlet velocity, has showed a good performance on both separation efficiency and pressure drop.

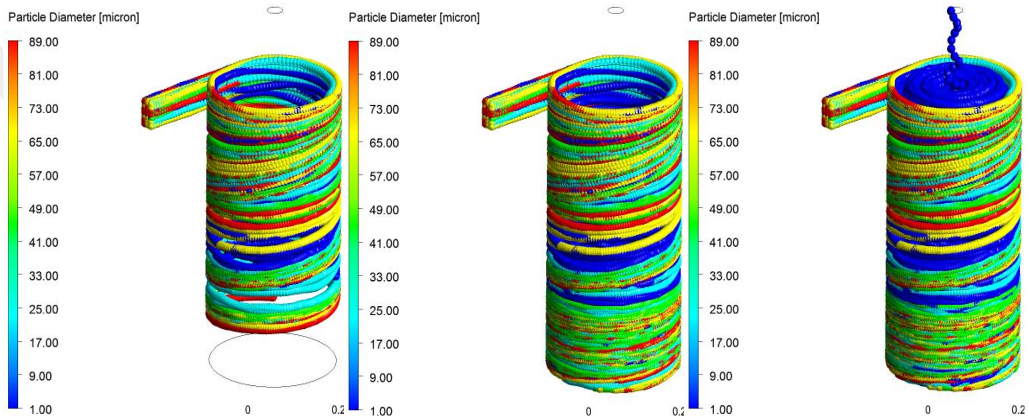
DPM particle tracks for different particle diameters with different time steps has illustrated in Figure 2.26 for Case A2 with 8m/s inlet velocity.



(a) t=0.01s (b) t=0.03s (c) t=0.05s



(d) t=0.1s (e) t=0.15s (f) t=0.25s



(g) t=0.5s (h) t=1.0s (i) t=3.0s

Figure 2.26 : DPM Particle Track for Case A2 @ 8 m/s

DPM particles have relased from inlet with 0.001seconds time interval. Positions of the particles with different time steps has shown in Figure 2.26. Centrifugal forces has started acting on particles at 0.03s after injection. It can be seen that larger particles have started moving closer to the outer wall whereas smaller particles were closer to the outer vortex finder wall. Smaller particles are moving faster than larger particles due to frictional forces which have higher impact on larger particles. At 0.25seonds nearly non of the small particles can be seen at the outer wall. In between 0.5 & 1 seconds particles have reached the bottom surface of the tak where the separation started due to trap boundary condition. Small particles has started to accumulate inside of the vortex finder and some of them are escaped from the outlet at t=3.0s. In this study, particle distribution is varying from 1µm to 100 µm. It can be seen that particles in between 1 to 10 μm is only reaching the outlet port. In other words, separation efficiency of larger particles is showed better results.

3. CONCLUSIONS

Air oil separator tank of a 22 kW oil injected rotary screw compressor has been analysed in this thesis study. Separation efficiency and pressure losses are the key parameters while designing of an air oil separator tank. The effect of inlet velocity, vortex finder diamater and length on separation efficiency has been investigated in order to optimise the air oil separator tank design for a specific case.

Inlet velocity has a big influence on static pressure. Static pressure increased sharply with the increment in inlet velocity. Moreover, increase in inlet velocity increases the magnitude of tangential velocity. It was expected to have better separation efficiency at high inlet velocities however, in some cases high inlet velocity showed high separation efficiency whereas in some cases separation efficiency reduced with the increase in inlet velocity.

It has been understood that vortex finder diamter and length is very critical while designing a air oil spearator tank. Static pressure and separation efficiency has changed remarkably with the changes in vortex finder diamater and length. Increase in vortex finder diamater has resulted in decrease in static pressure. Magnitude of tagential velocity has dropped down with the increase in vortex finder diamater. In most of the cases, separation efficiency is reduced with the increase in vortex finder diamater. Likewise, increase in vortex finder length has resulted in decrease in static pressure, tangential velocity and separation efficiency.

DPM particle tracks have revealed that only some portion of small particles tend to escape from the outlet. It is observed that mostly particles around 10 μm are escaping from the outlet port. Larger particles have been separated more efficiently due to higher centrifugal forces acting on them.

Moreover, some further studies can be conducted in order to improve the air oil separator tank design more efficienctly. Most of the numerical studies about cyclones has shown high separation efficiency when compared with the experimental results. It is thought that using the effect of breakup and collision models show better compability with the experimental results. Breakup and collision models can be investigated for future studies. In addition, it has been observed that mostly small particles tend to escape from the outlet. Particle sizes can be reduced in order to have a better overview to estimate the cut-off diamater. Lastly, pressure loss at the outlet port were relatively higher than the other sections. Improvement studies can be performed for outlet section too.

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