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The Effects of Diffuser Exit Velocity and Distance Between Supply and Return Apertures on the Efficiency of an Air Distribution System in an Office Space

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THE EFFECTS OF DIFFUSER EXIT VELOCITY AND DISTANCE BETWEEN SUPPLY AND RETURN APERTURES ON THE EFFICIENCY OF AN AIR DISTRIBUTION SYSTEM IN AN OFFICE SPACE

BY

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This thesis is approved as a creditable and independent investigation by a candidate for the Master of Science degree and is acceptable for meeting the thesis requirements of this degree. Acceptance of this thesis does not imply that the conclusions reached by the candidate are necessarily the conclusions of the major department.

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CONTENTS

ABBREVIATIONS

CFD	Computational Fluid Dynamics
HVAC	Heating, Ventilation and Air Conditioning
ASHRAE	American Society of Heating, Ventilation & Air Conditioning Engineers.
CFM	Cubic feet per minute
CFH	Cubic feet per hour
IN	Inch
FT	Feet
BTU	British thermal unit
kW	Kilo Watt
kWh	Kilo Watt Hour
kPa	Kilo Pascal
atm	Atmospheric pressure
m	Meter

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ABSTRACT

THE EFFECTS OF DIFFUSER EXIT VELOCITY AND DISTANCE BETWEEN SUPPLY AND RETURN APERTURES ON THE EFFICIENCY OF AN AIR DISTRIBUTION SYSTEM IN AN OFFICE SPACE

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The distance between supply air diffusers and return air grilles is an important design parameter that has not been properly investigated. Sometimes, design and installation constraints force the engineer or the contractor to place the return duct close to the supply duct which can have a huge impact on the overall efficiency of air distribution in the zone. When the return grille is too close to the supply diffuser, air by-passes the room and flows directly back into the return duct. By-passing of the air in a zone of a building will prevent the heat transfer and the circulation that are needed for the HVAC system to work properly. This has an impact on the system's overall efficiency and results in wasted energy. The work presented here studies the effect of the distance between supply and return air openings at various exit velocities on the overall efficiency of an HVAC system.

The study modeled air distribution in a typical office space that was assumed to have negligible leakage and was adiabatic. The density of air was assumed to be constant. Also, for the scope of this study, temperature was assumed to be constant since the interest is only in proper duct placement and thermodynamic effects were not being considered.

The study investigated both square and round diffusers. The diffusers and return grilles were placed equidistant from the center of the ceiling at lengths of 12, 24, 36, 48 and 60 inches away from each other. They were coincident within the same x plane. Each placement configuration was modeled at three different flow rates: 200, 350 and 500 cfm that represented low, medium and high air flow for a typical VAV system.

The analysis showed that round diffusers are more efficient than square diffusers when they are placed less than 60 inches away from the return grille. At the designed flow rate of 350 cfm and at 12, 24, 36, 48 and 60 inches away from the return grille, the square diffuser showed short-circuiting percentages of 23.4, 21.9, 17.2, 10.9 and 5.6 percent respectively. On the other hand, at the same flow rate of 350, the round diffuser showed short-circuiting percentages of 10.9, 9.4, 7.5, 6.6 and 5.8 percent when the supply and the return openings were placed 12, 24, 36, 48 and 60 inches apart respectively.

INTRODUCTION

In the HVAC design industry, the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE) provides standards and guidelines for all aspects of system design. When it comes to the placement of the return air grille with respect to the supply air diffuser, the guideline is vague and does not place any limits on the maximum and minimum distances between the two openings. Chapter 20, Space Air Diffusion, of the ASHRAE Handbook of Fundamentals has a short paragraph for the location of the return grille; it states “Openings should be located to minimize short-circuiting of supply air. If air is supplied by jets attached to the ceiling, exhaust openings should be located between the jets or at the side of the room away from the supply jets” [18]. Other sources like Price Industries, have a similar guideline when it comes to the location of return air grilles which is “The return air inlet has very little effect on room air diffusion, regardless of inlet type or location. However, return air inlets should be located a sufficient distance from the supply outlet so that short-circuiting of supply air does not occur.” [21]

In many cases, design and installation constraints force the engineer or the contractor to place the return duct too close to the supply duct, which might have a huge impact on the overall efficiency of the HVAC system in the zone. When the return duct is placed too close to the supply diffuser, air by-passes the zone and flows directly into the return duct. By-passing of the air in a zone will prevent the heat transfer and the circulation that are needed for the HVAC system to

work which means a negative impact on its overall efficiency and results in wasted energy.

While ASHRAE and other lead HVAC societies do not put any limiting guidelines on the minimum distances between supply and return apertures, consultants and contractors do not have a clear context for evaluating the magnitude of efficiency loss produced by non-ideal placement. The objectives of this study were to (1) demonstrate the relevance of the distance between the supply diffuser and the return grille on the overall efficiency of an HVAC system and to (2) estimate the energy losses due to an air by-pass in the system.

The scope of this study was limited to an office space or a small zone; no obstacles were placed in the room due to the fact that the scope of the study was to investigate the direct by-passing of the air that happens in the upper area of the zone, between to supply and return apertures.

The study utilized a Computational Fluid Dynamics (CFD) software, Star CCM+, to model the fluid dynamics in the office space. The two most common diffusers, square and round, were studied. Five distances between supply and return ducts were modeled at three different flow rates. The five distances represented a wide range of distances while the three flow rates represented low, medium, and high VAV system flow rates. The zone was considered sealed and adiabatic. The density of air was assumed to be constant. Also, for the scope of this study, temperature was assumed to be constant since the interest is only in proper duct placement.

Model Description and Review of Related Literature

Project Background

The HVAC system in a building can consume a considerable portion of the total energy use of a building. According to the U.S. Department of Energy's website, 32% of the energy consumed in office buildings is from heating and cooling [22]. The system is dynamic and responds to the ever changing needs of the occupants and is designed to maintain consistent conditions regardless of external climate. Even a small inefficiency in operation may cause large changes in energy consumption. It is possible that the placement of the return grille is not always in the optimal proximity to the supply duct but due to convenience of design or attempts to reduce material costs. The effect of the distance on the placement of supply and return apertures need to be quantified and addressed in order to reduce the HVAC systems inefficiencies caused by misplacements of the ducts. This analysis will study the relevance of the distance between the supply diffuser and return air grille; the study will model air distribution in a zone with set parameters with the variables being the supply diffuser and the return air grille placements along with incrementally-varying flow rate at each placement. Square and round diffusers will be modeled. When the return is too close to the diffuser and by-pass occurs, the system supplies more air than necessary to maintain the desired thermal requirements, this equates to an inefficiency which can be quantified monetarily.

Literature Review

Return Grille Location

The existing literature and guidelines from professional journals and associations provide a vague guidance in regards to the return air grille placement. According to Dave Fetters, the influence of the return duct is limited to approximately one diffuser length in all directions. [1] The air leaving through the diffuser is at a low velocity and therefore the pressure differential from the room to the return grille is very small as well so as long as the return grille is outside of said distance away from the inlet, it should not affect air flow in the room. ASHRAE provides the standards for HVAC design in the United States, their guideline states "*If the outlet is selected to provide adequate throw and directed away from returns or exhausts, supply short-circuiting is normally not a problem*." [2] Additionally, in another ASHRAE journal, Dan Int-Hout states "In the real world, air movement toward return openings is imperceptible except within a few inches of the opening." [3] Chapter 20, Space Air Diffusion, of the ASHRAE Handbook of Fundamentals has a short paragraph that has the guideline for the location of the return grille and it states "Openings should be located to minimize short-circuiting of supply air. If air is supplied by jets attached to the ceiling, exhaust openings should be located between the jets or at the side of the room away from the supply jets". [18] Other sources like Price Industries, have a similar guideline when it comes to the location of return air grilles which is "The return air inlet has very little effect on room air diffusion, regardless of inlet type or location.

However, return air inlets should be located a sufficient distance from the supply outlet so that short-circuiting of supply air does not occur." [21]

Other references to placement of the return duct offered similar advice. The advice is rather vague, which may be intentional because it is believed that as long as the return grille is not placed directly next to the supply diffuser it does not present any undesired air flow issues. However, these guidelines need to be verified and the magnitude of the effects of the return grille location on the HVAC system need to be addressed properly as poor guidelines can lead to a poor design which will produce inefficiency in the system.

Air Diffusion Models

A review of the actual air diffusion in office spaces with a ceiling diffuser is necessary to validate the results obtained from the CFD software. Air diffusion in spaces with ceiling diffusers follow the Surface or Coanda effect. As shown in figure 1 below, which is obtained from Chapter 21 of the ASHRAE Handbook of Fundamentals [18], supply air moves along the upper ceiling of the zone besides the low pressure region.

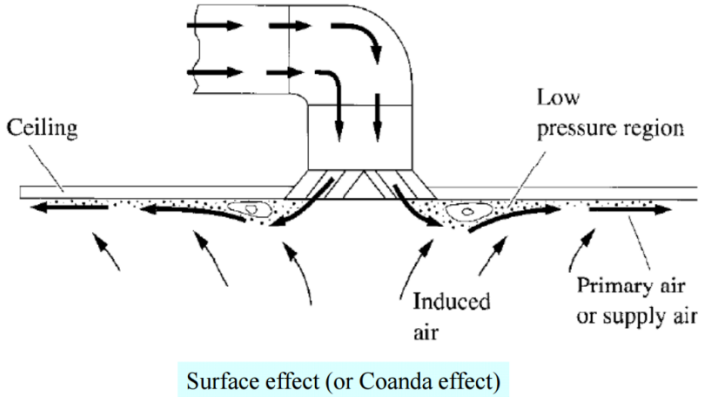

Figure 1: Surface or Coanda effect

Figure 2 below shows the space air diffusion over a zone using a square diffuser.

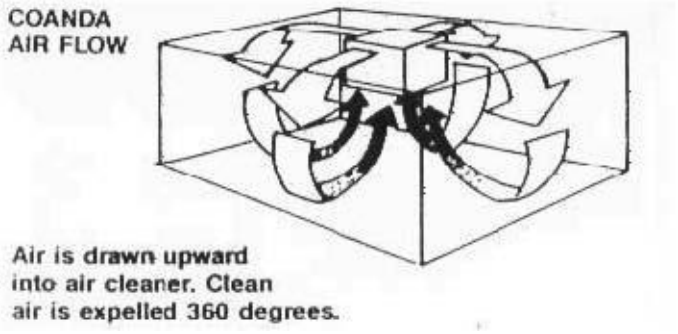

Figure 2: Air Space Diffusion; square diffuser

Computational Fluid Dynamics

Computational fluid dynamics is a branch of fluid mechanics that utilizes numerical analysis and algorithms to analyze fluid flows. CFD software allows engineers and scientists to model fluid flows in a specific environment without having to build that environment. CFD software uses several inputs to analyze fluid flow; the researcher needs to model the geometry of the space that the fluid will flow in, and create a mesh model that divides the space into cells. Each cell has a unique location that the software uses in order to calculate the fluid flow at that specific location.

CAD model Development

Modeling Approach

A Solidworks model of a rectangular prism 15 ft squared and 9 ft in height serves as a small office space with a single supply diffuser and one return grille. The modeled room had two recessed square faces. One face served as an inlet and the other served as an outlet location to mate the diffuser model. Each type of the two diffusers and the return grille was placed equidistant from the center of the ceiling at lengths of 12, 24, 36, 48 and 60 inches apart from each other. They were coincident within the same x plane. The square and round diffusers were modeled with four shells and a 45 degree throw angle. Using a cavity function, the diffusers were modeled as a negative volume to represent the fluid volume.

The designed flow rate, 348 cfm, of the model was found using the energy equation; refer to the Governing Equations section. According to the Engineering Tool Box website [23], the maximum recommended air velocity in the low and medium pressure ducts is 490 feet per minute, which was set to compensate for the unacceptable noise and frictional losses inside the ducts. Based on the designed flow rate of 348 cfm and the maximum recommended exiting velocity of 490 fpm, the neck area of the diffuser was calculated to be around 0.71 square feet; that is equivalent to 10.2 inch x 10.2 inch square or 11 inch radial diffusers. Diffuser sizes will use the English unit system throughout the study as they are commonly found in different manufacturers' catalogs.

Diffuser manufacturers, like Price Industry, have even sizes available for both square and round diffusers, so a 10 inch x 10 inch square and a 10 inch round diffusers were chosen for the analyses. Since the neck area of the two types of diffusers are not the same, exit velocities were not the same as well; the exiting velocities are similar for both diffusers and are listed in Table 2. The small variation in the exit velocities of the diffusers did not cause a significant effect on the results.

In order to study the effect of different exit velocities on the distance between supply and return apertures flow rates of 200 cfm and 500 cfm used. The 200 cfm flow would represent a lower VAV system flow which is dissipated in a case of low system load while the 500 cfm flow would represent a high flow due to a high VAV system load.

Figures 3 and 4 below show the square diffuser, while figures 5 and 6 while the round diffuser.

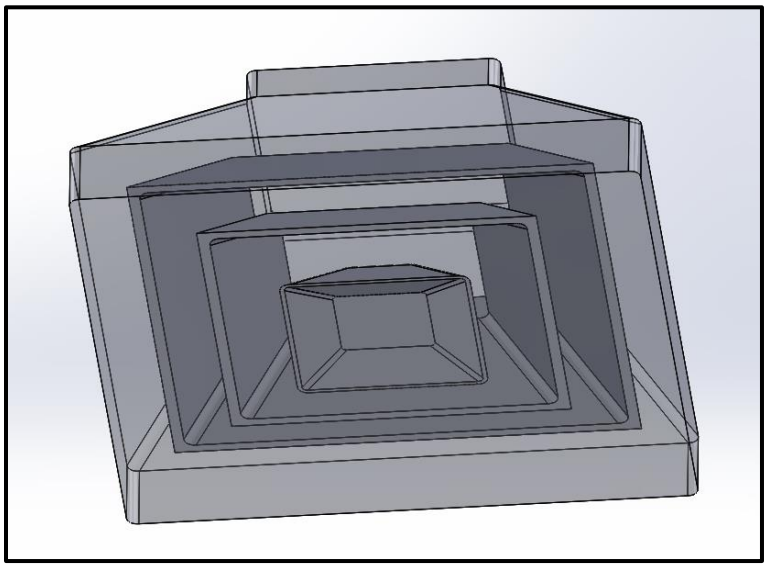

Figure 3: Solidworks Model of Square Diffuser; 45 degree throw angle, 4 shells

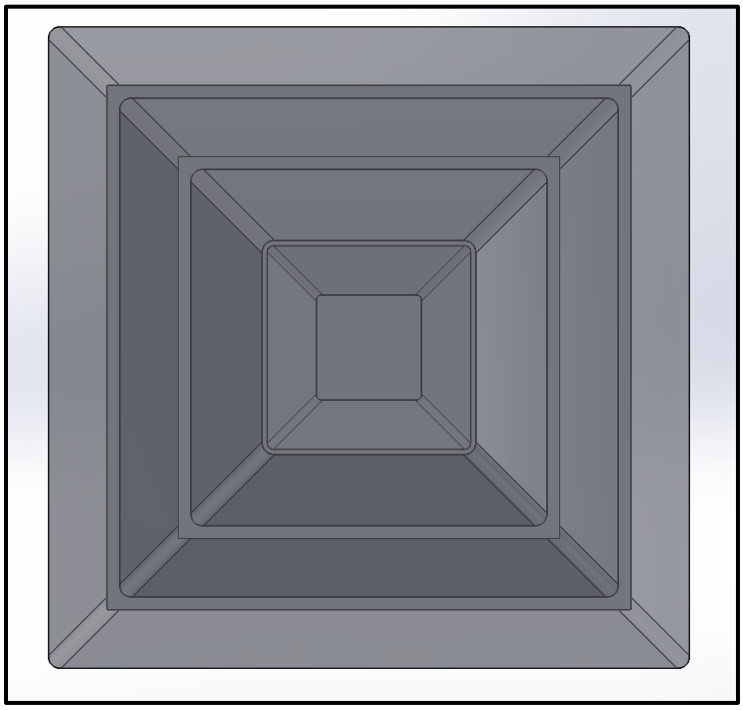

Figure 4: Solidworks Model of Square Diffuser; 45 degree throw angle, 4 shells

Figure 5: Solidworks Model of Round Diffuser; 45 degree throw angle, 4 shells

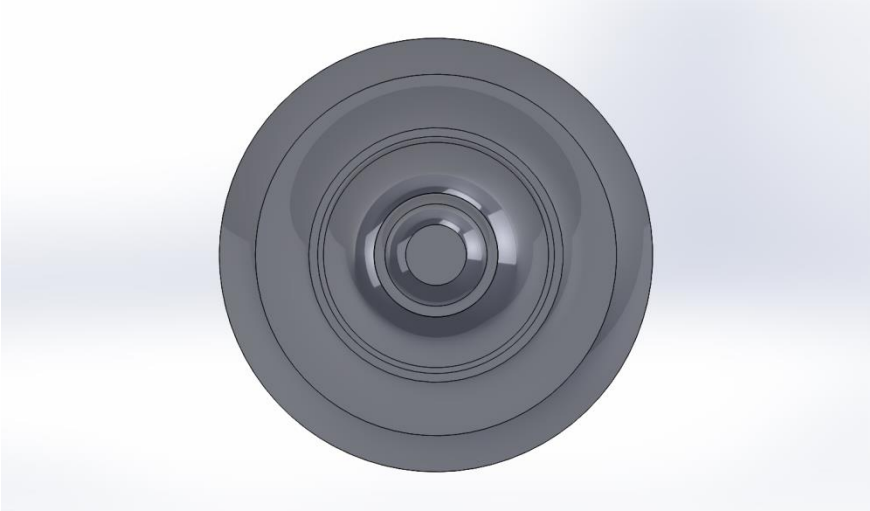

Figure 6: Solidworks Model of Round Diffuser; 45 degree throw angle, 4 shells

CFD model development

Figures 7, 8, 9, 10 and 11 below show the geometry models of the office space, with the square diffuser at 12, 24, 36, 48 and 60 inches away from the return grille.

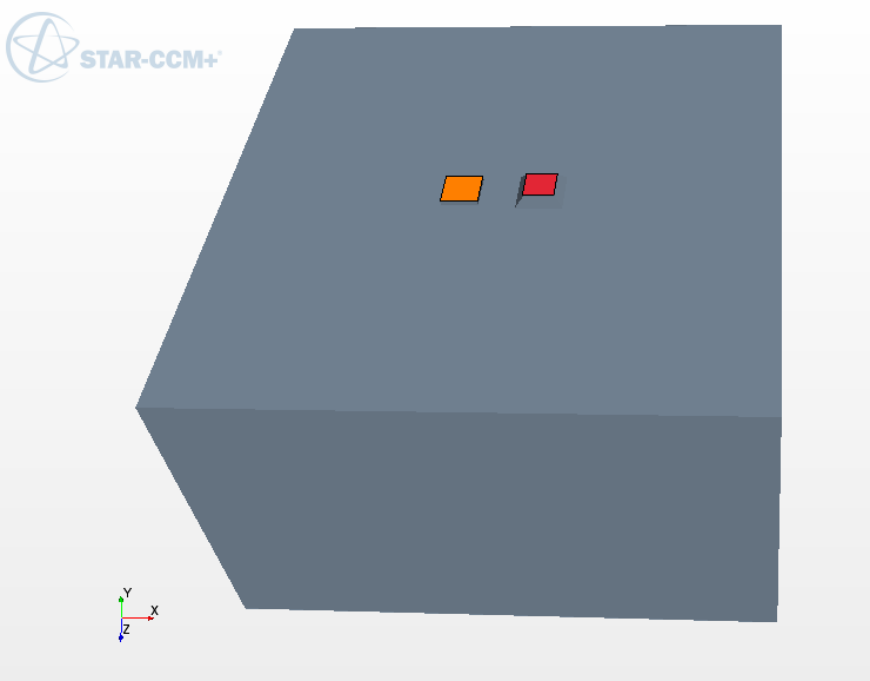

Figure 7: Office Space with Square Diffuser at 12 inches away from return

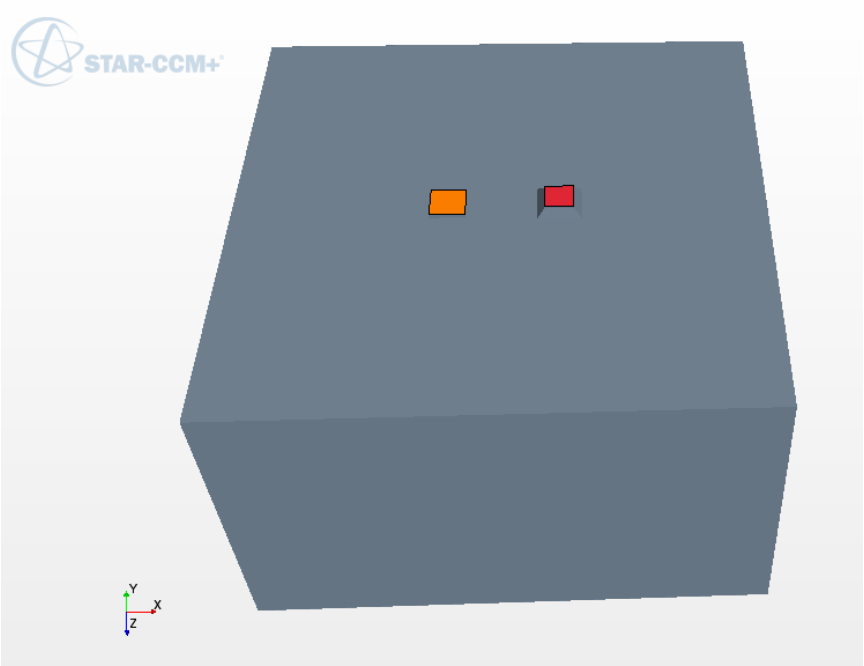

Figure 8: Office Space with Square Diffuser at 24 inches away from return

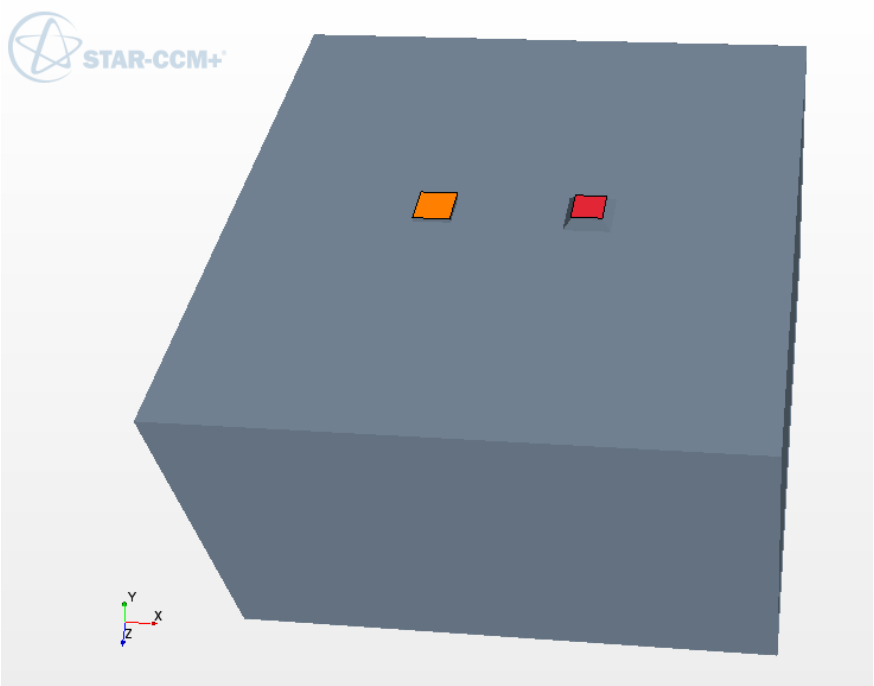

Figure 9: Office Space with Square Diffuser at 36 inches away from return

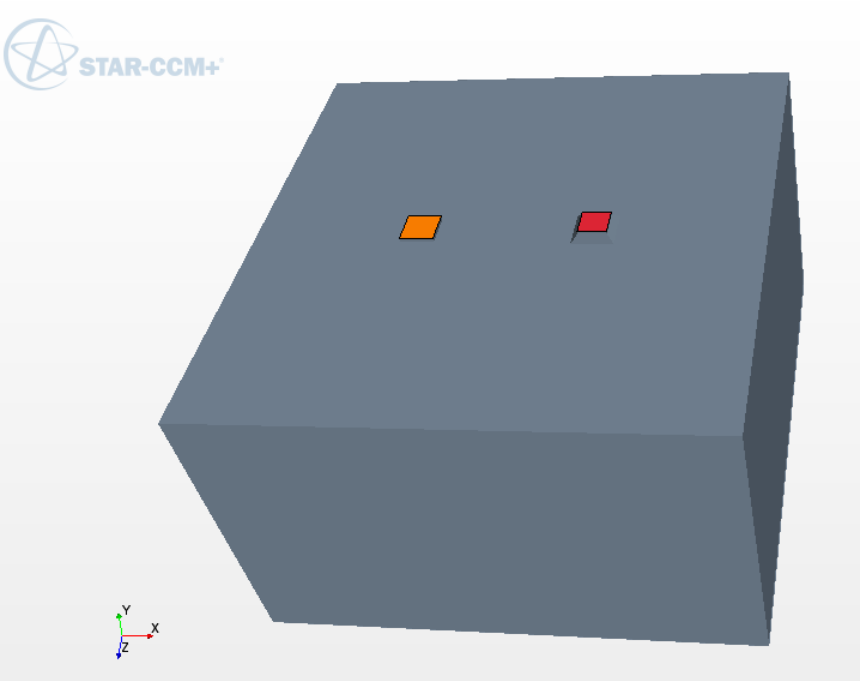

Figure 10: Office Space with Square Diffuser at 48 inches away from return

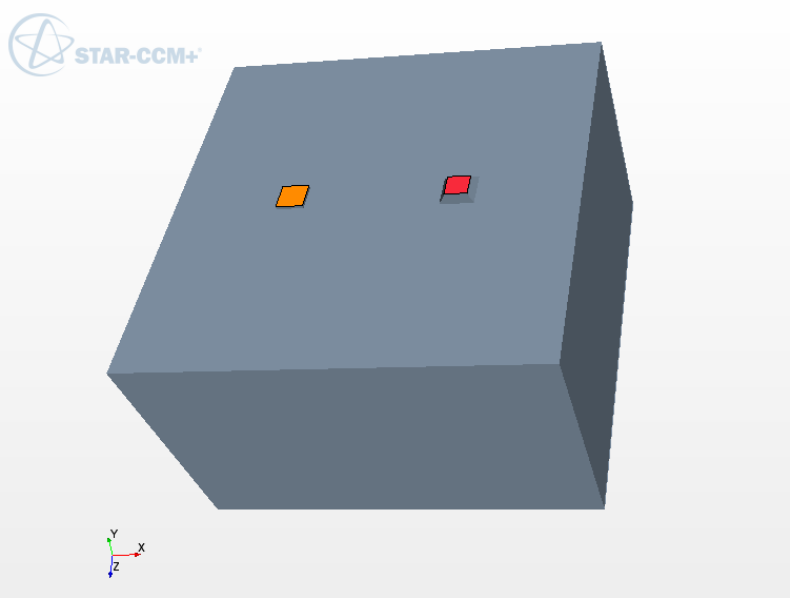

Figure 11: Office Space with Square Diffuser at 60 inches away from return

Figures 12, 13, 14, 15 and 16 below show the geometry models of the office space, with the round diffuser at 12, 24, 36, 48 and 60 inches away from the return grille.

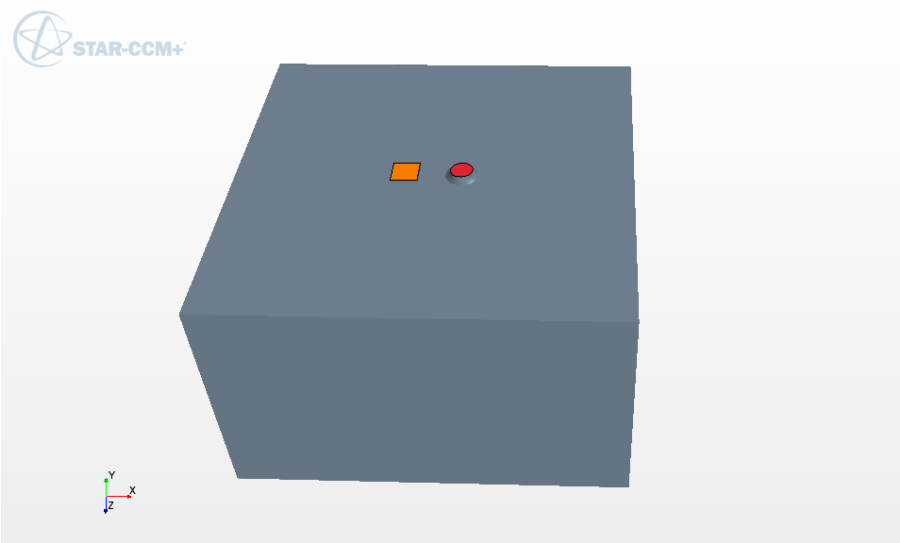

Figure 12: Office Space with Round Diffuser at 12 inches away from return

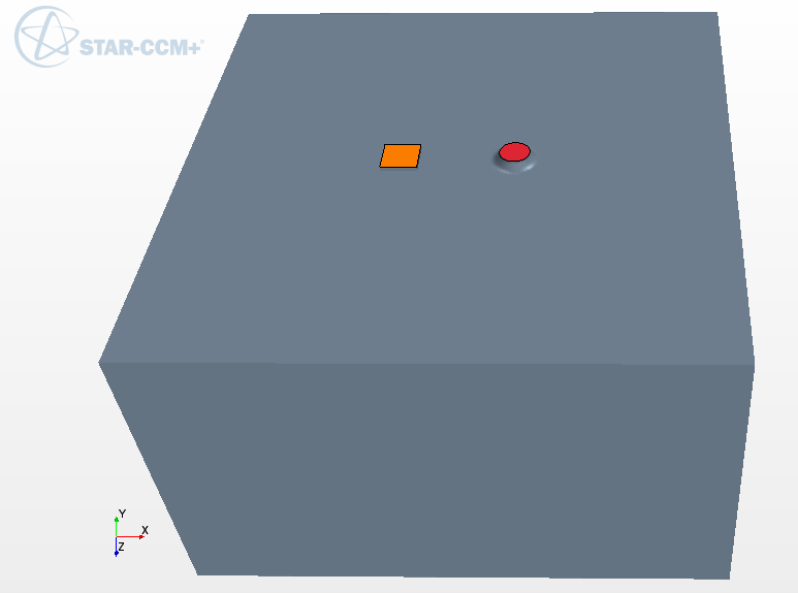

Figure 13: Office Space with Round Diffuser at 24 inches away from return

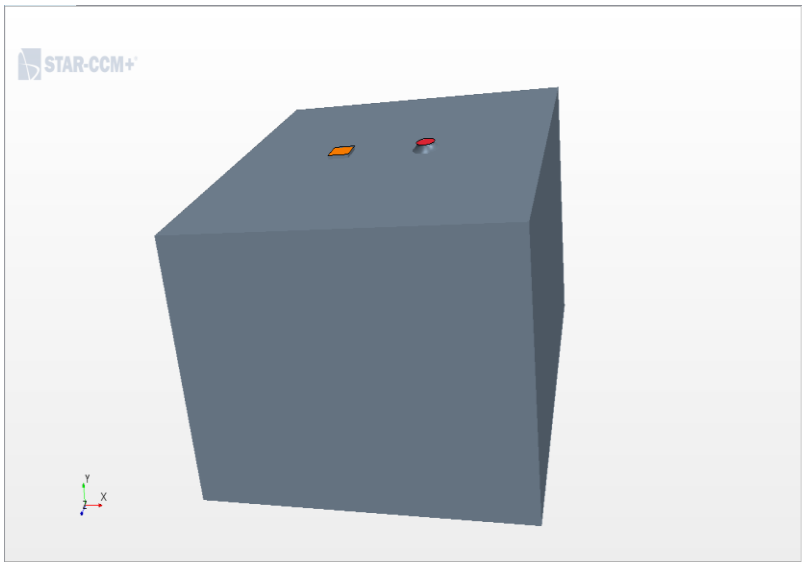

Figure 14: Office Space with Round Diffuser at 36 inches away from return

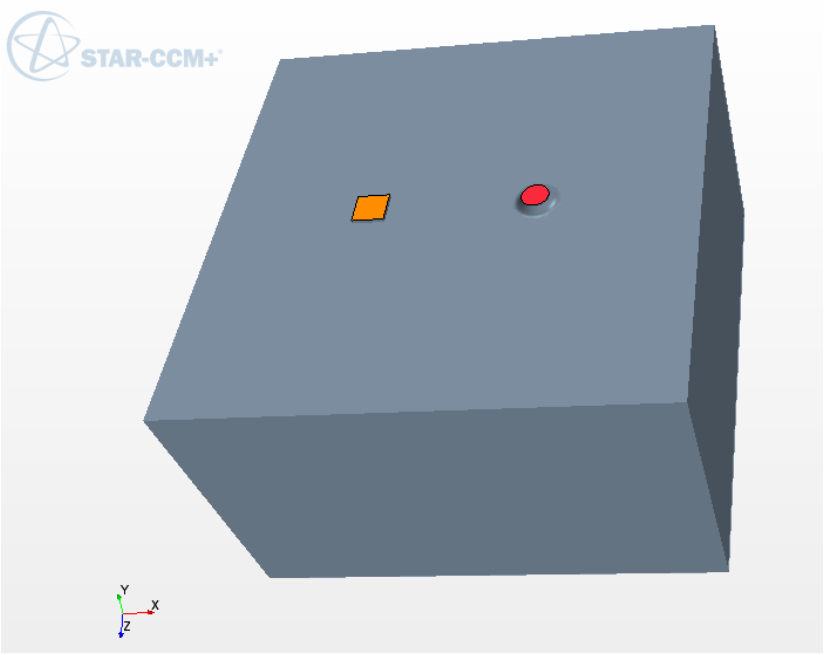

Figure 15: Office Space with Round Diffuser at 48 inches away from return

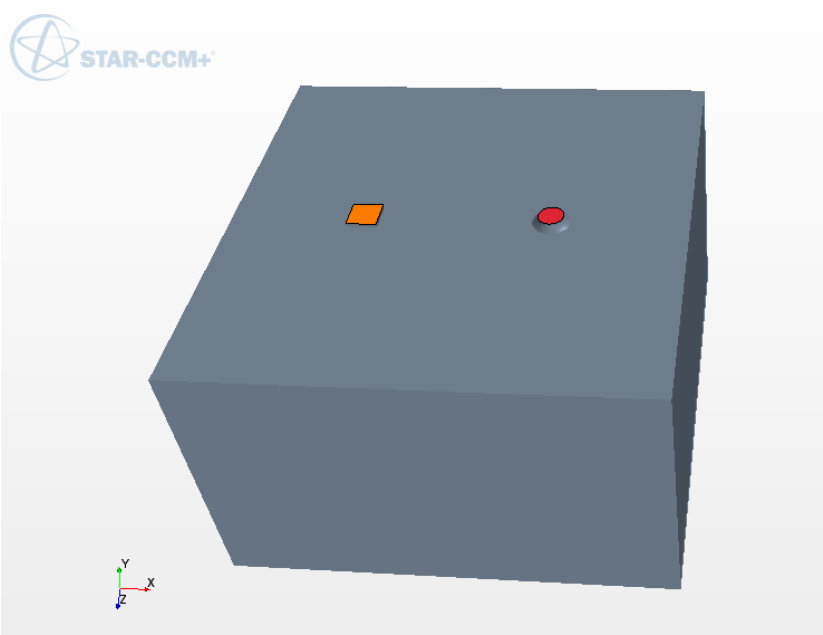

Figure 16: Office Space with Round Diffuser at 60 inches away from return

Approach for Problem Solving

Using a model of a small office space, the placement of one supply diffuser and one return grille was varied. Using the velocity and pressure profiles of the analysis, the relevance of the distance between supply and return ducts was analyzed. Since HVAC design requires low air flow velocity for reducing noise and requires a pressure differential large enough to move the air through the system, the ideal placement was a balancing act between those two parameters. To further explain: too close together would allow for small pressure difference and low velocity but poor air exchange which is an ineffective design, too far apart would allow for good air diffusion but would require high velocity and a large pressure difference which would cause more noise and uses more energy.

Physical Conditions

Table 1 below shows the physical models that were used in the CFD software to simulate a real life fluid dynamics condition. Table 1 has a justification column that contains the reason for selecting each physical model.

Table 1: Physical Conditions Chosen

Properties	Model	Justification
Space	3 dimensional	Meshes are 3 dimensional
Motion	Stationary	Zone is stationary
Time	Steady	Iterations are required to reduce the residual errors and to get converged solutions
Material	Air	Working fluid inside the room is Air
Flow	Segregated	Stratified, wavy and annular flow of Air
Density	Constant density	Density is invariant throughout the continuum
Viscous regime	Turbulent	Flow of Air with amplifications of infinitesimal disturbances
Navier Stokes values	Reynolds averaged Navier Stokes	To breakdown the instantaneous pressure and velocity values to mean value and a fluctuating component
Turbulence models	K-epsilon turbulence	Effects like convection and diffusion of turbulent energy as Re number is greater than 4000
Wall surface	Smooth flow	Smooth Scalar view
Force	Gravity	Acceleration due to gravity in y direction
Wall treatment	Two Layer all y+ wall treatment	To assume near-wall cell lies within the logarithmic region of the boundary layer
Turbulence models	Realizable K-Epsilon Two layer	Improvisation in K-Epsilon model to have turbulent flows in sub layers

The initial conditions and reference values that were used in the simulations are listed in Table 2 below. Note that the velocities at the neck of the the two types of diffusers are not the same; this is due to the fact that the study chose the closest standard diffuser neck sizes of 10 inch by 10 inch for the square diffuser and 10 inch in diameter for the round diffuser. The velocities were close and did not cause an effect on the results.

Table 2: Initial Boundary Conditions

Initial Boundary Conditions		
Type	Flow rate (cfm)	Velocity (fpm)
Square	200	366
	350	641
	500	917
Round	200	287
	350	504
	500	720

Model Assumptions

Assumptions and Justifications

To simplify the analysis, the room was assumed to be empty. The room was considered sealed (no air exchange across the boundaries except at the diffuser outlet and return inlet) and adiabatic. The density of air was assumed to be constant. Also, for the scope of this study, temperature was assumed to be constant since the interest is only in proper duct placement and thermodynamic effects were not being considered. Initial simulations omitted a diffuser to further simplify the model and thus the analysis. The diffuser was added once the model was properly defined and functioning. The reference pressure value was 14.7 psi. The inlet and outlet boundary pressure values were 0 psi guage.

Governing Equations

Velocity Input Design:

The sensible or cooling load of the office was assumed to be 7500 Btu/hr. Equation 1 below shows how the flow rate of supply air, in cfm, was calculated for the CFD Model. The office set temperature was chosen to be $75°\mathrm{F}$ while the supply air temperature was chosen to be $55°\mathrm{F}$.

The load calculation is as follows:

$$\dot{q} = \dot{m} \cdot c_p \cdot (T_a - T_{supply}) \qquad (1)$$

$$\dot{q} = 7500 \frac{Btu}{hr} = 1.08 \cdot \dot{V} \cdot 20$$

$$\dot{m} \cdot c_p = \rho \cdot \dot{V} \cdot c_p \approx 1.08 \cdot \dot{V} \; for \; air$$

$$\boldsymbol{\dot{V}} = 348 \; CFM$$

$$\dot{q} = load \left[\frac{Btu}{hr}\right]$$

$$\dot{m} = mass \; flow \; rate \; of \; supply \; air \left[\frac{lb_m}{hr}\right]$$

$$c_p = specific \; heat \; capacity \; of \; air \left[\frac{Btu}{lb_m \cdot °\mathrm{F}}\right] = 0.241 \frac{Btu}{lb_m \cdot °\mathrm{F}}$$

$$T_a = desired \; ambient \; room \; temperature \; [°\mathrm{F}]$$

$$T_{supply} = temperature \; of \; supply \; air \; [°\mathrm{F}]$$

$$\dot{V} = volumetric \; flow \; rate \left[\frac{ft^3}{min}\right]$$

The office dimensions are:

$$V = l \cdot w \cdot h \qquad (2)$$

$$l = 15 \; ft, w = 15 \; ft, h = 9 \; ft$$

$$\boldsymbol{V} = 15 \cdot 15 \cdot 9 = 2025 \; ft^3$$

Reynolds number calculation:

$$Re_L = \frac{\rho v L}{\mu} \qquad (3)$$

$$\rho = density\ of\ air = 0.0749\ \frac{lb}{ft^3}$$

$$v = mean\ velocity\ of\ the\ fluid\ at\ 350\ cfm = 10.7\ \frac{ft}{s}$$

$$L = length\ of\ diffuser, at\ the\ discharge = 0.833\ ft$$

$$\mu = dynamic\ viscosity = 1.23 * 10^{-5}\ \frac{lbs}{ft^2}$$

$$Re_L = 54{,}275$$

The Reynolds number at this flow rate is 54,275 which means a turbulent flow regime.

Mesh Generation

Mesh Models:

To be able to simulate air flow in the office space, mesh generation was needed. The mesh divides the office space into small cells so the CFD software can calculate the flow at each cell of the mesh. The study chose a small mesh base size of 0.05 m; the smaller the mesh base size, the more accurate the simulation will be. Overall, the office space was divided into approximately 615,000 cells. Table 2 below shows the chosen mesh models.

Table 2: Mesh Models Chosen

Meshing Type	Chosen Mesh Function
Surface Mesh	Surface Remesher
Volume Mesh	Polyhedral Mesher
Volume Mesh	Prism Layer Mesher
Attributes	**Chosen Values**
Base size	0.05 m
No of prism layers	5
Prism Layer Stretching	1.5
No of faces	3672421 approx.
No of cells	614646 approx.

Figures 17, 18, 19, 20 and 21 below show the mesh models of the office space, with the square diffuser at 12, 24, 36, 48 and 60 inches away from the return grille.

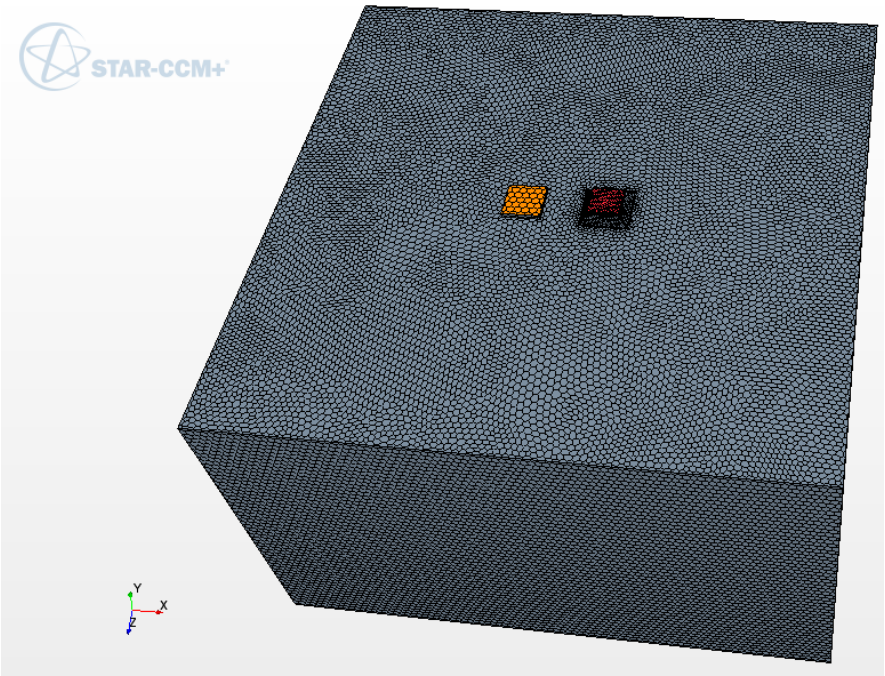

Figure 17: Mesh Model of Office Space at 12 inches apart, square

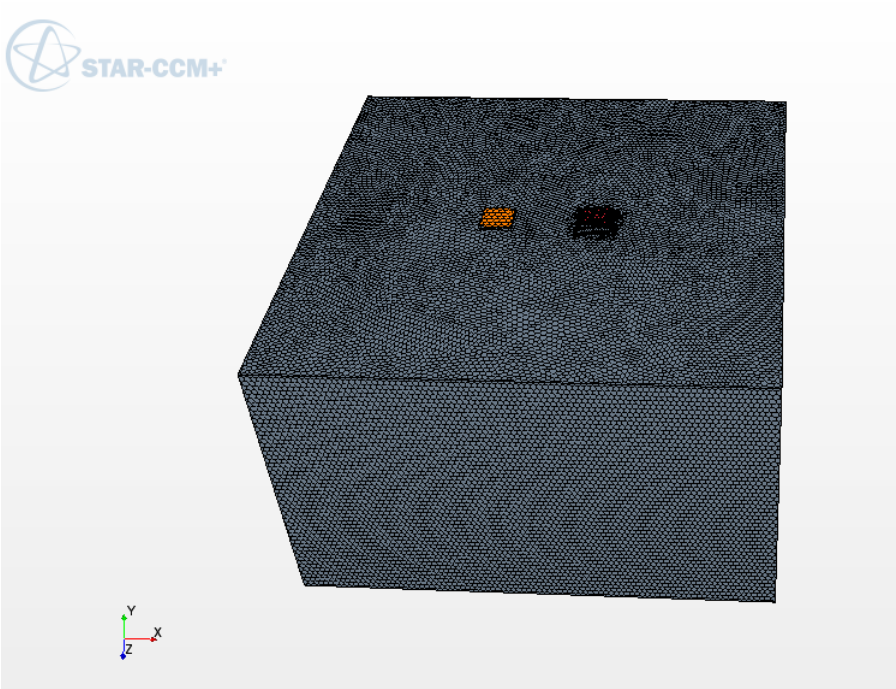

Figure 18: Mesh Model of Office Space at 24 inches apart, square

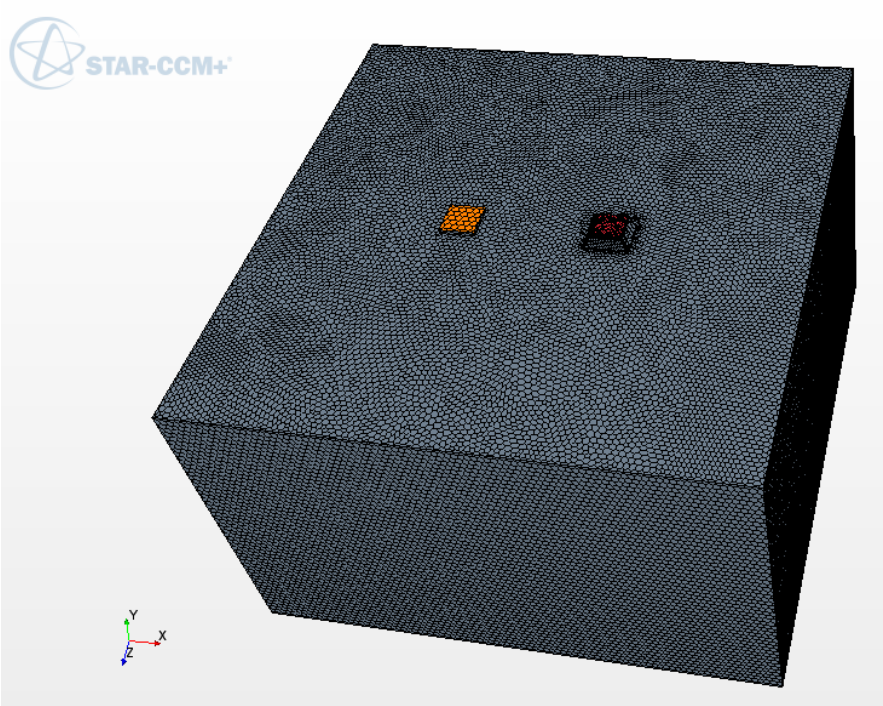

Figure 19: Mesh Model of Office Space at 36 inches apart, square

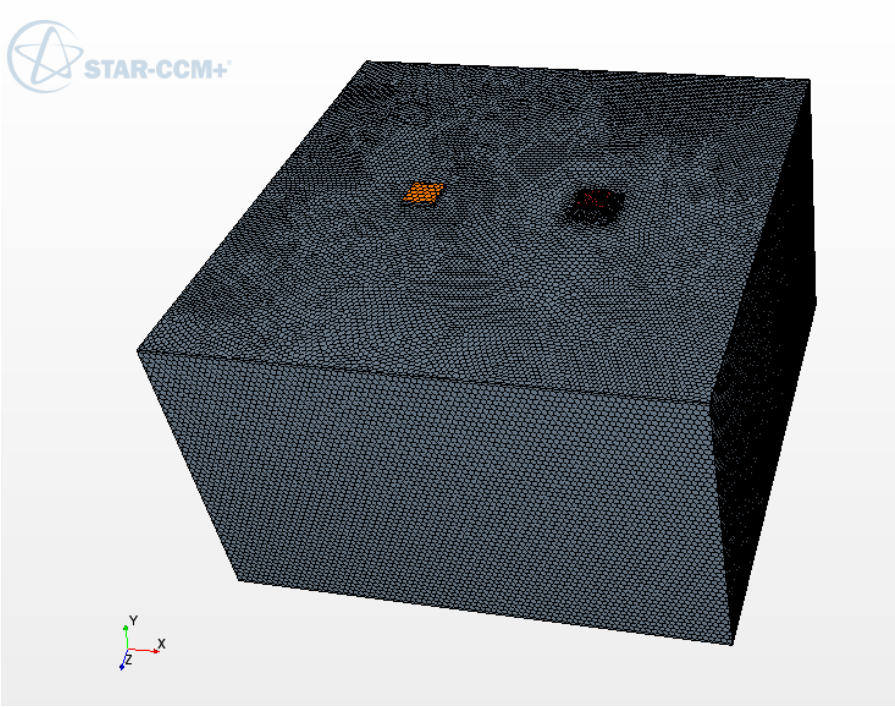

Figure 20: Mesh Model of Office Space at 48 inches apart, square

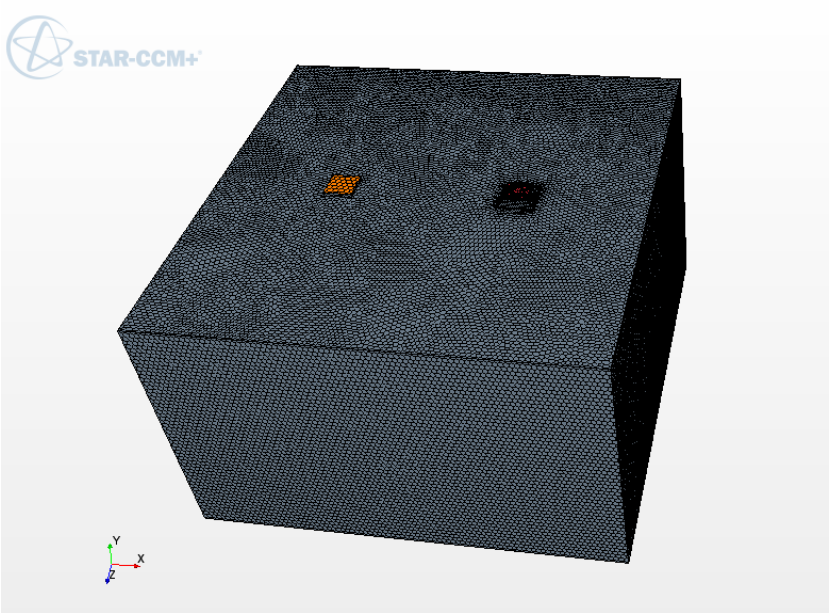

Figure 21: Mesh Model of Office Space at 60 inches apart, square

Figures 22, 23, 24, 25 and 26 below show the geometry models of the office space, with the round diffuser at 12, 24, 36, 48 and 60 inches away from the return grille.

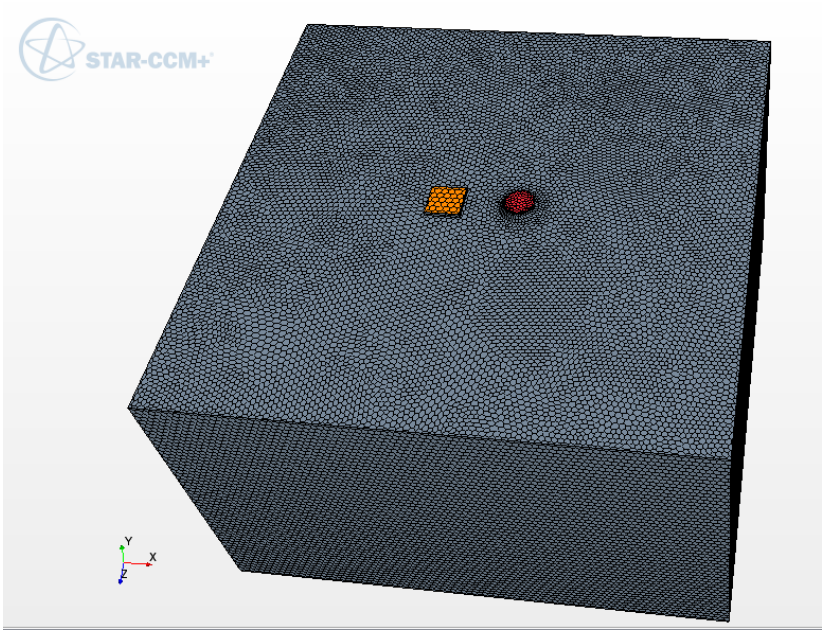

Figure 22: Mesh Model of Office Space at 12 inches apart, round

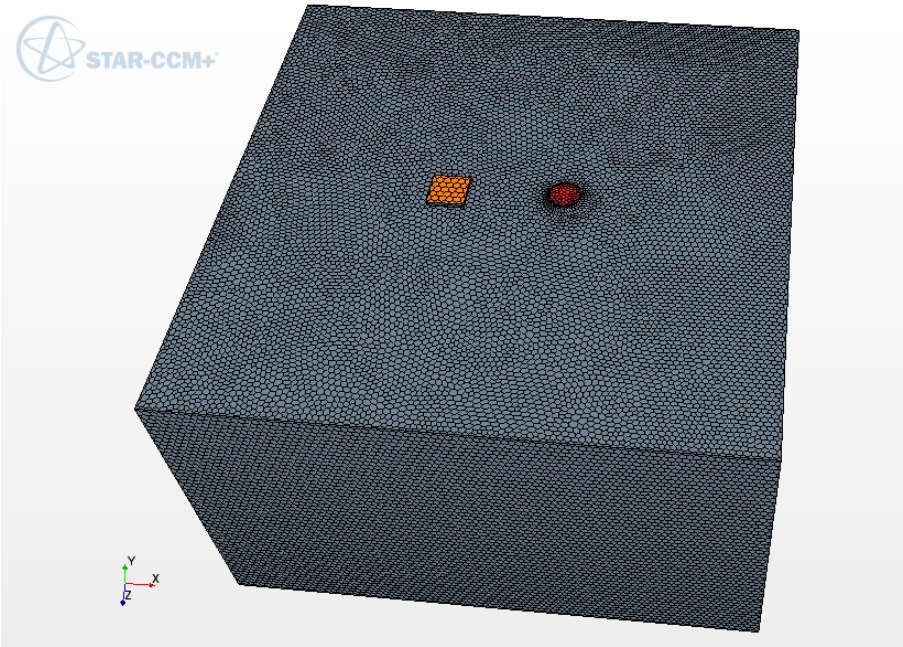

Figure 23: Mesh Model of Office Space at 24 inches apart, round

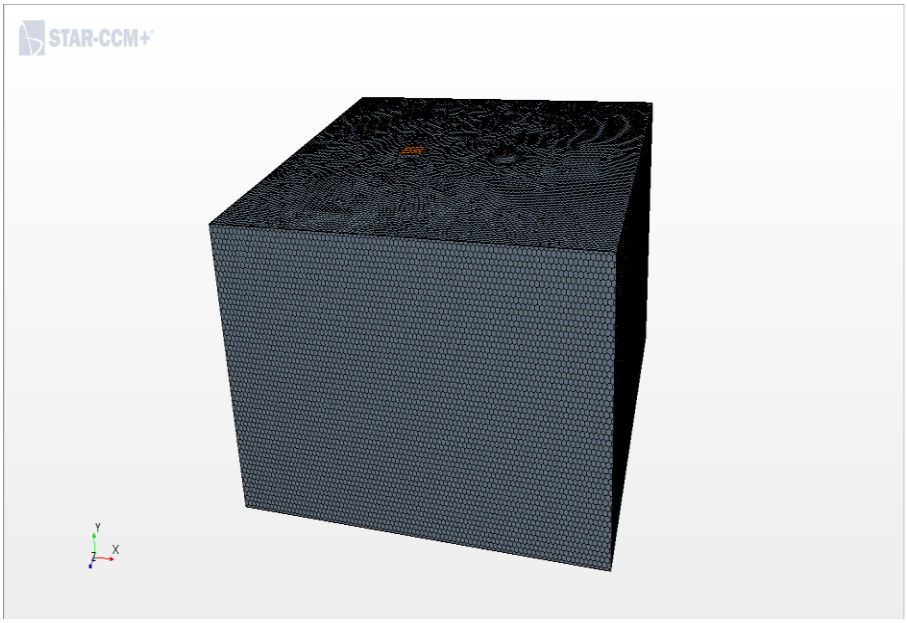

Figure 24: Mesh Model of Office Space at 36 inches apart, round

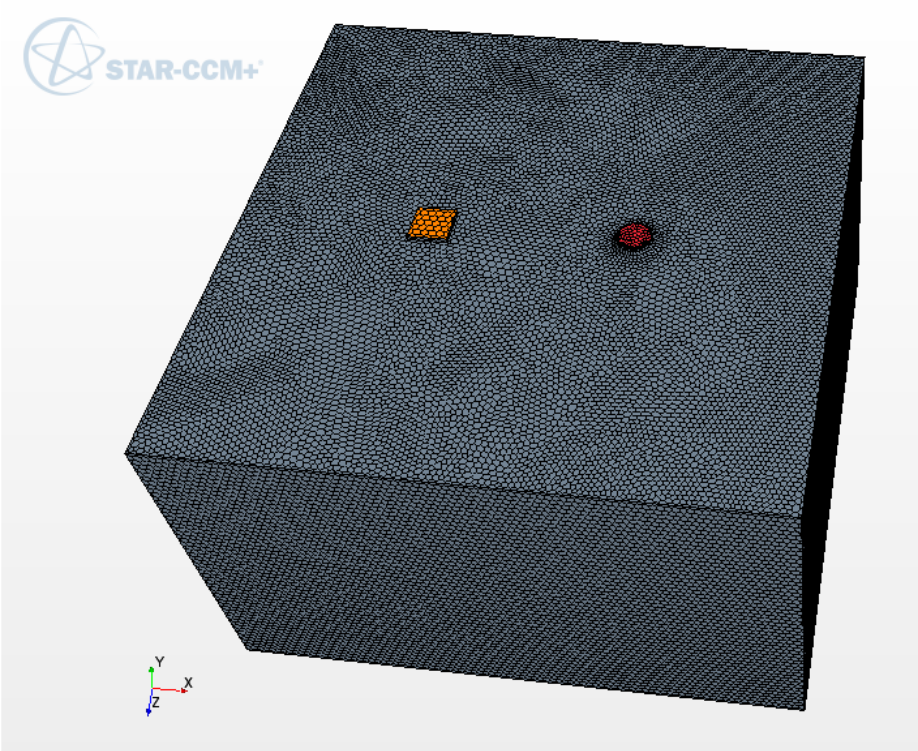

Figure 25: Mesh Model of Office Space at 48 inches apart, round

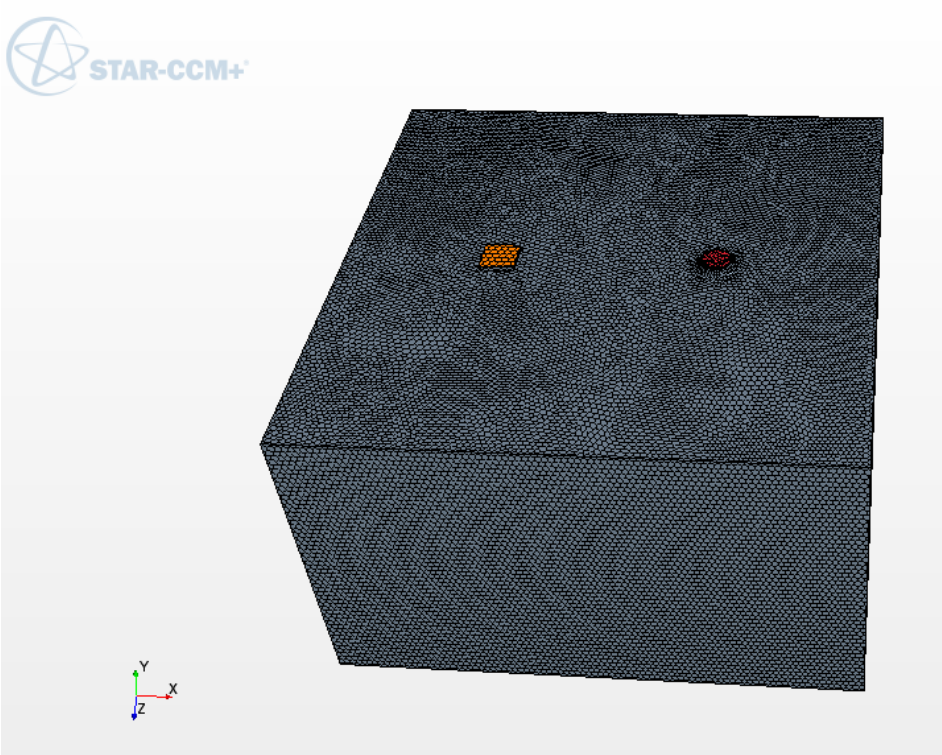

Figure 26: Mesh Model of Office Space at 60 inches apart, round

Results

The following results show flow simulations of models with the return air grille set at the selected distances, 12, 24, 36, 48 and 60 inches, away from the square or round diffusers. Each configuration was subjected to a volumetric flow rate of 200, 350 and 500 cfm. Each simulation is shown with a two-dimensional velocity scalar view, a velocity streamline view and a plot of the residuals. The scalar view shows the characteristic flow of air from the diffuser but does not give a clear picture on whether or not air by-passes the room. The streamline view provides a useful metric to determine if the distance between inlet and outlet apertures is at an optimal length based on minimal bypass; streamlines are patches of fluid that represent its flow and speed. Streamlines are used in this investigation to determine the percentage of air by-pass in the system; the number of the short-circuiting streamlines was divided by the total number of streamlines. The three-dimensional figures below show the short-circuiting streamlines.

Part One: Square Diffuser

Square Diffuser: 12 inches away from the return; 200 cfm

As can be seen in Figure 27, the air that is diffused from the right side of the diffuser is not circulating past the return grille; it is by-passing the zone and going directly back into the return grille. Figures 28, 29 and 30 show the three dimensional streamlines that represent the circulation of air; the figures show that

approximately 12 of the 64 stream lines, 18.8%, that were diffused in the room by-passed the system and did not circulate in the room.

A plot of the simulation residuals is shown in Figure 31. The plot shows that all 6 residuals were relatively converged at or below the 0.003 level; the residuals plot show how precise the results of the simulation are. Residuals are the difference between the iterated value from the software and the exact solution.

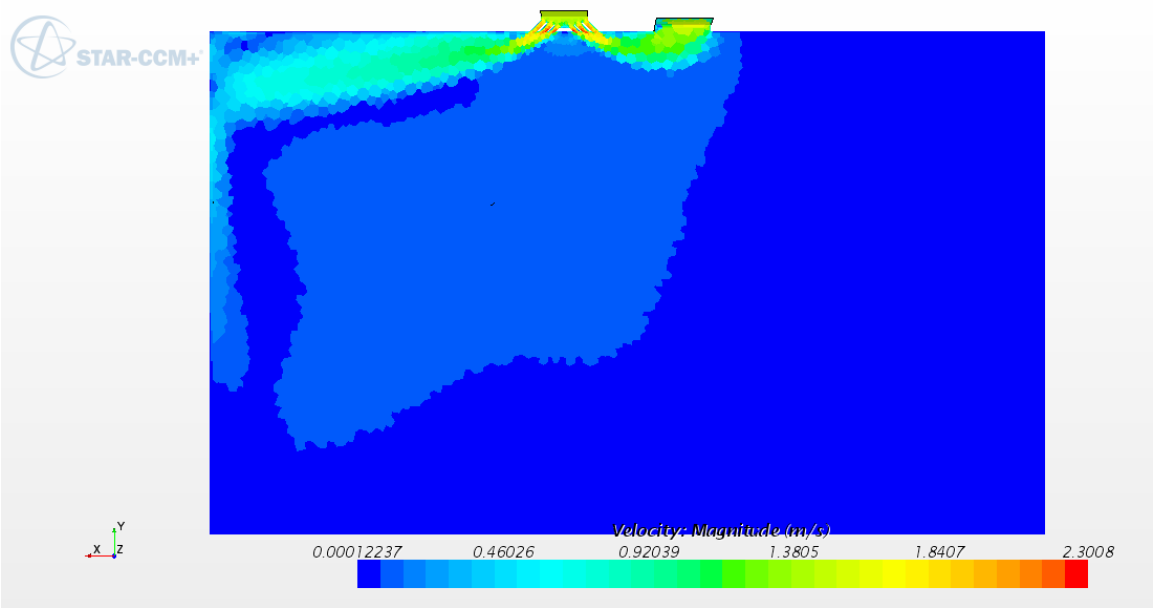

Figure 27: Two Dimensional Planar Section of the office; 12 in – 200cfm

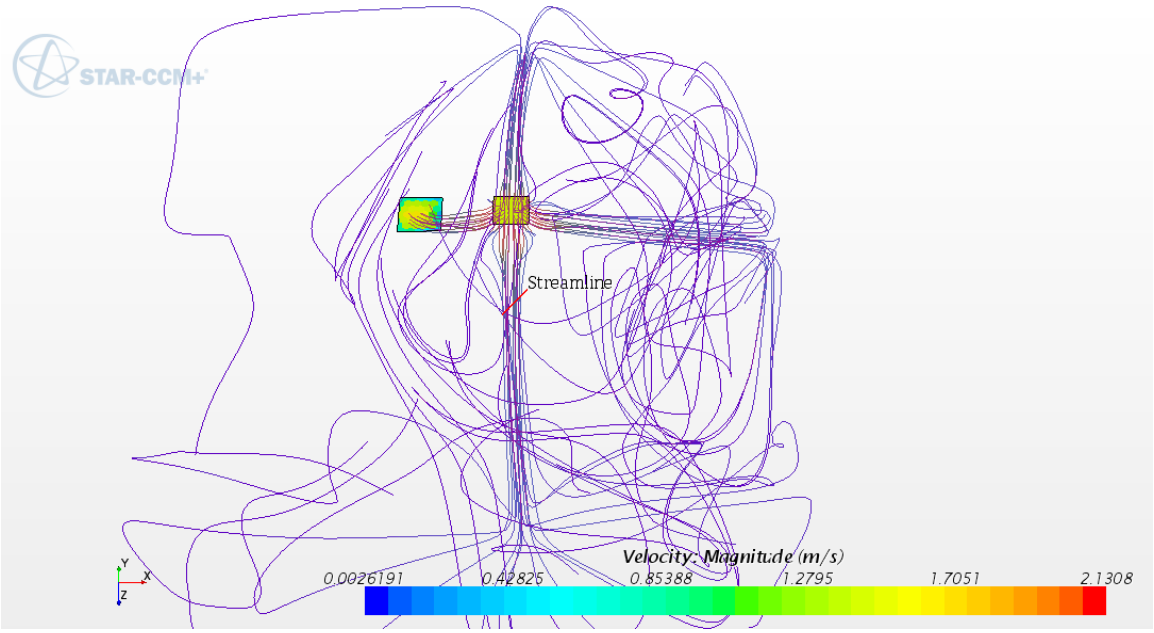

Figure 28: Three dimensional model of air circulation in the zone; 12 in – 200cfm

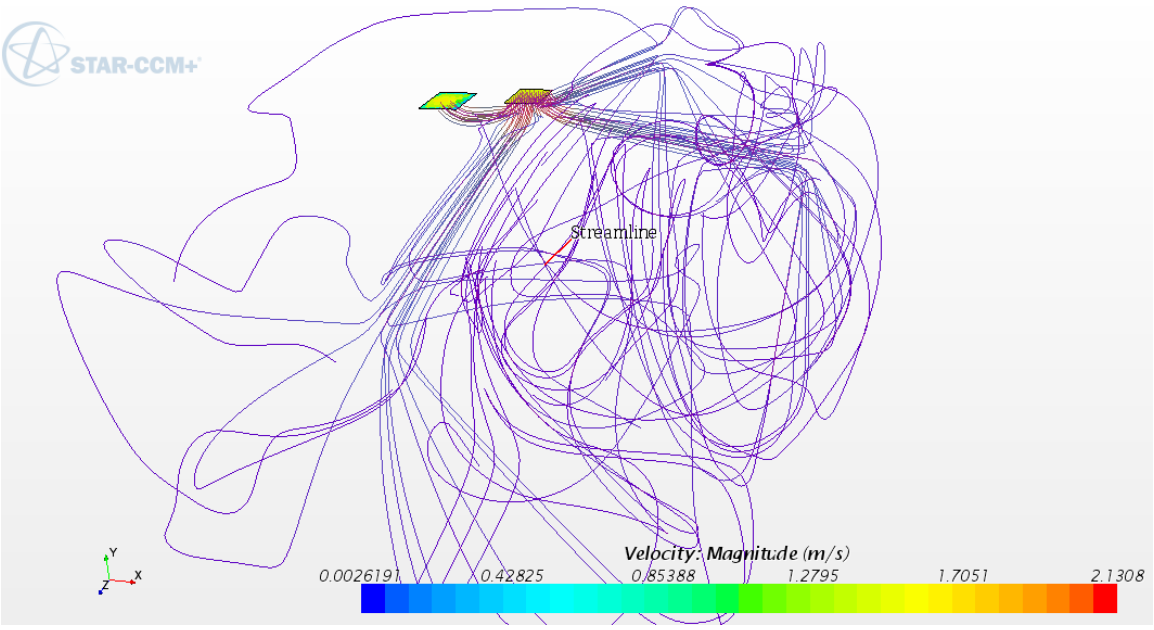

Figure 29: Three dimensional model of air circulation in the zone; 12 in – 200cfm

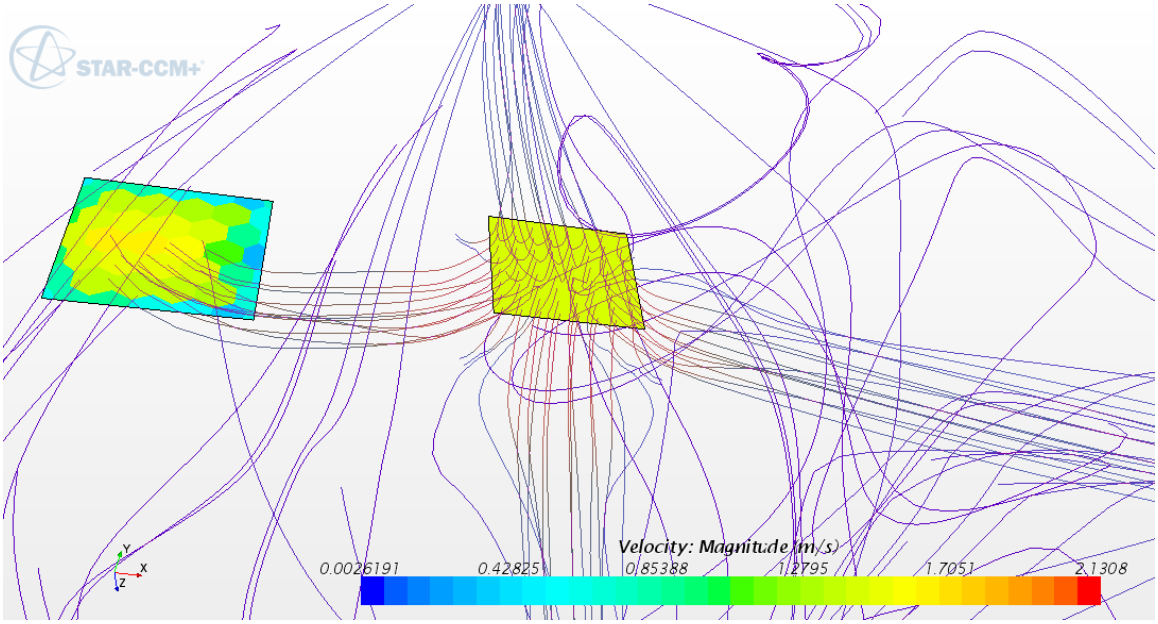

Figure 30: Air directly bypassing the system; 12 in – 200cfm

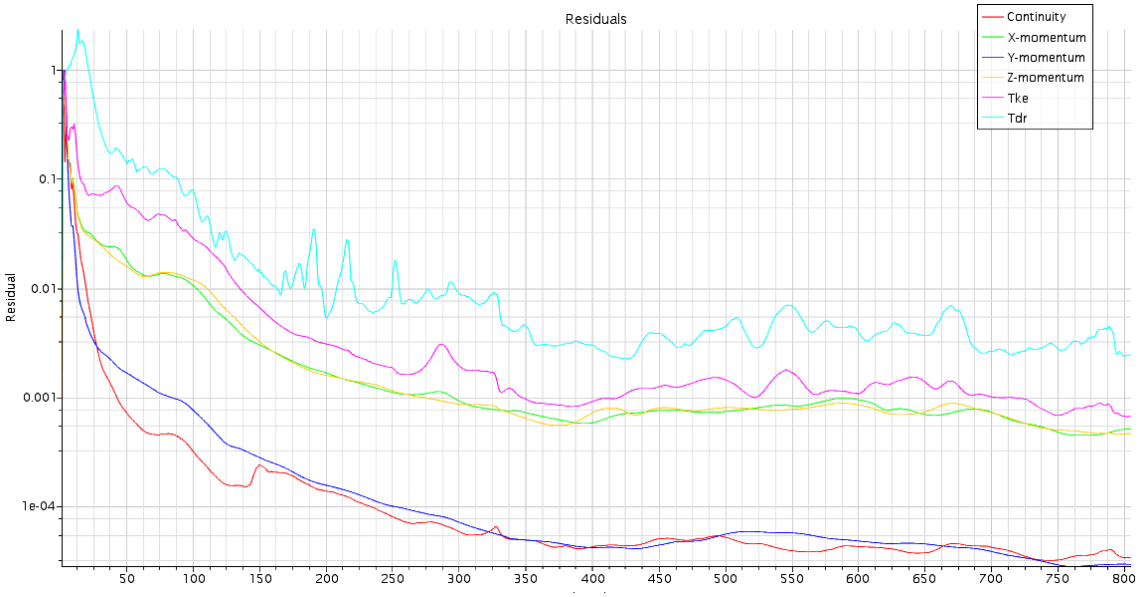

Figure 31: Plot Residuals; 12 in – 200cfm

Square Diffuser: 12 inches away from the return; 350 cfm

The results obtained from figures 32, 33, 34 and 35 show a little higher by-pass rate than the simulation of 12 inches and 200 cfm. Approximately 15 of the 64 stream lines, 23.4%, that were diffused in the room by-passed the system and did not circulate in the room.

A plot of the simulation residuals is shown in Figure 36. The plot shows that all 6 residuals were relatively converged at or below the 0.005 level which show how precise the results of the simulation are.

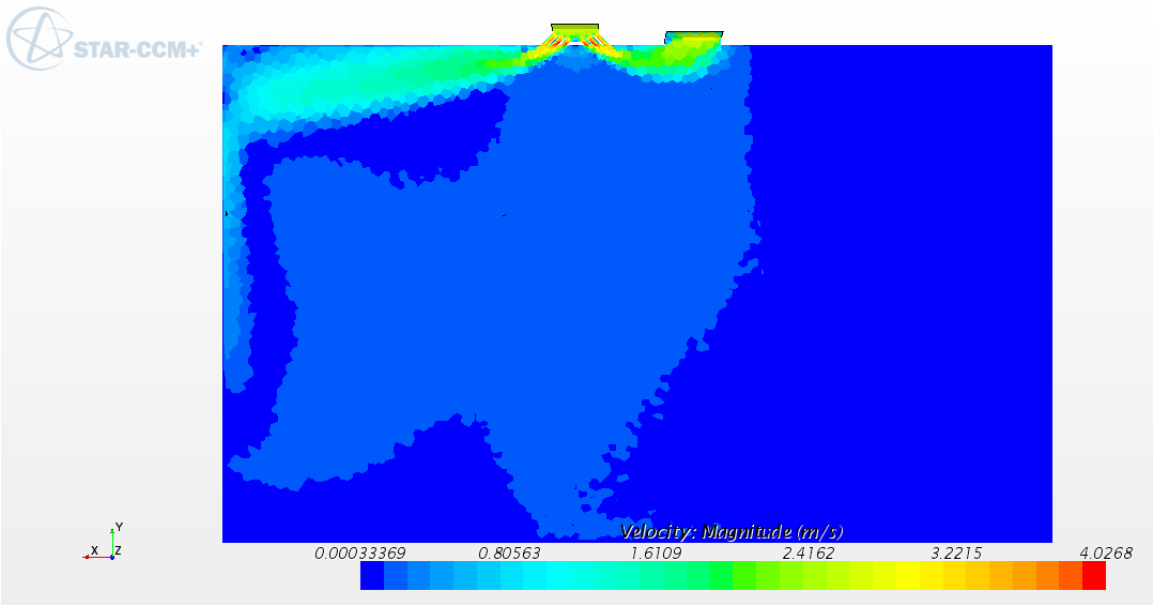

Figure 32: Two Dimensional Planar Section of the office; 12 in – 350cfm

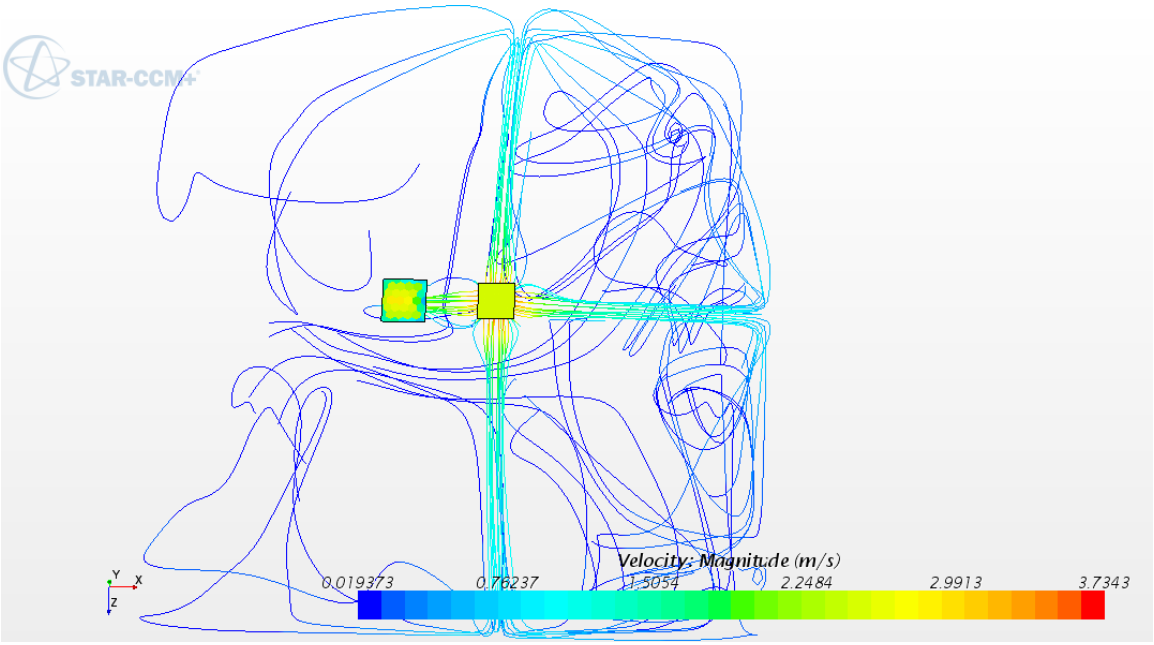

Figure 33: Three dimensional model of air circulation in the zone; 12 in – 350cfm

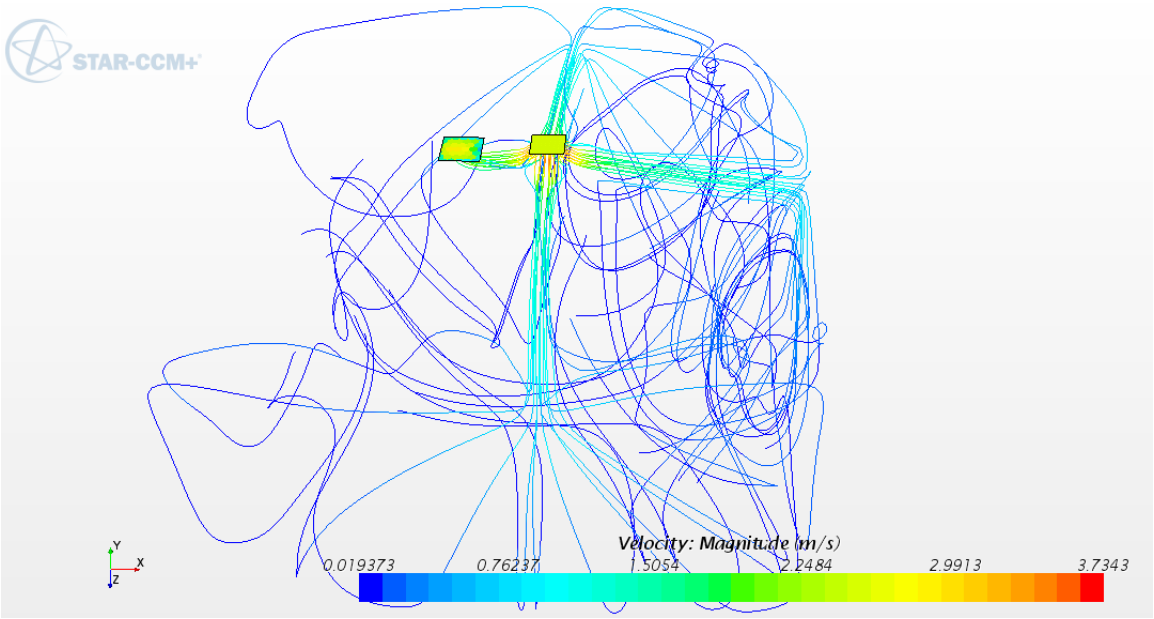

Figure 34: Three dimensional model of air circulation in the zone; 12 in – 350cfm

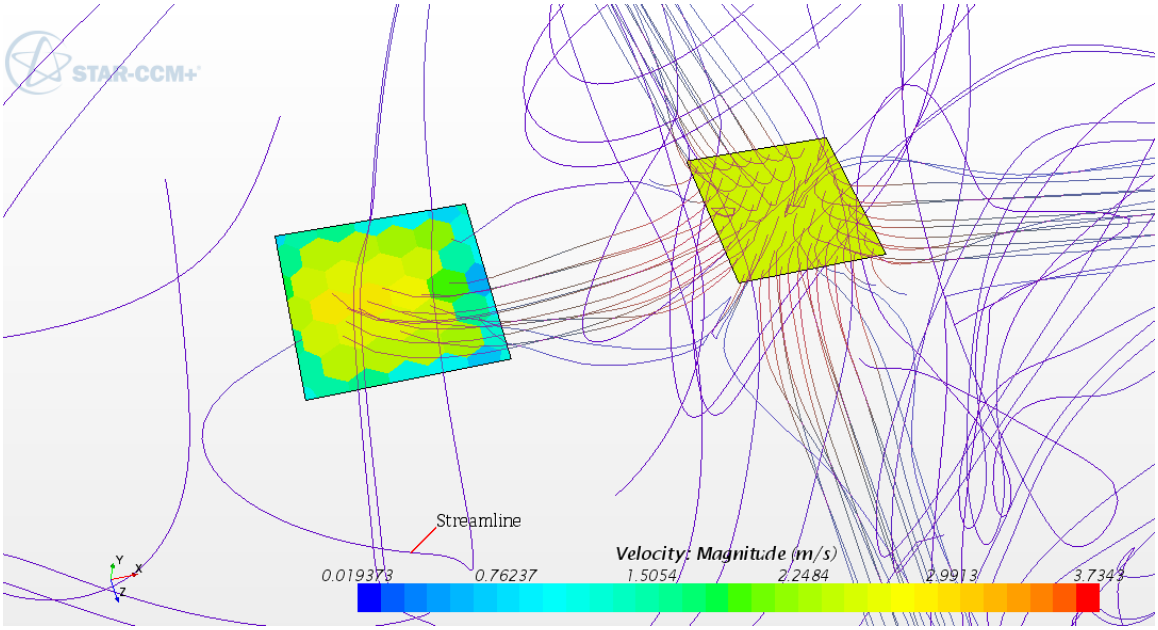

Figure 35: Three dimensional model of air circulation in the zone; 12 in – 350cfm

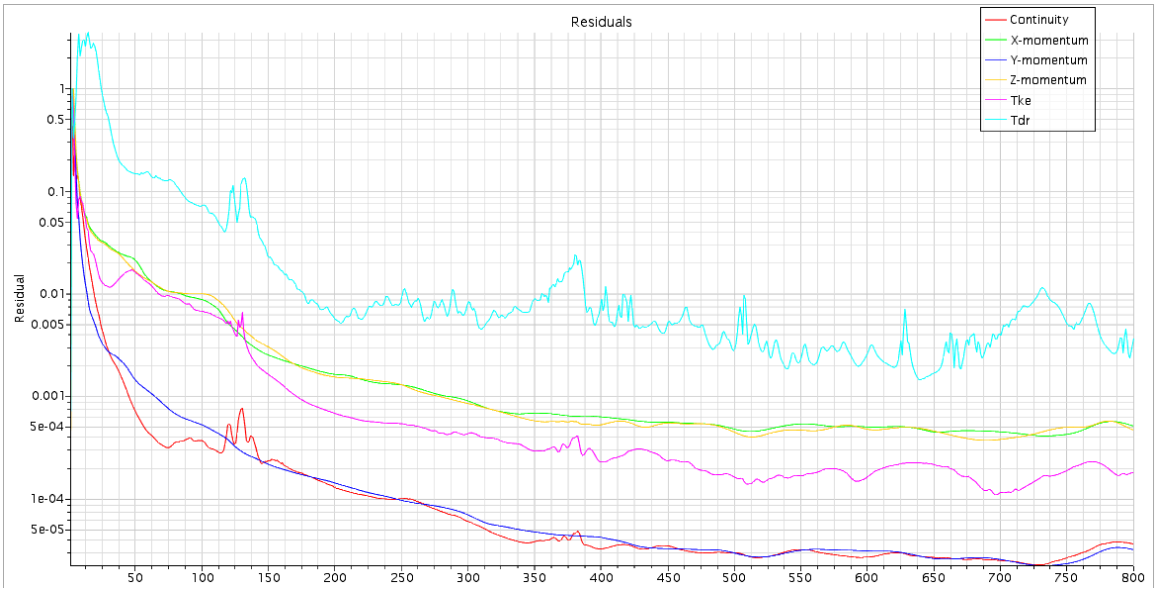

Figure 36: Plot Residuals; 12 in – 350cfm

Square Diffuser: 12 inches away from the return; 500 cfm

The results obtained from figures 37, 38 and 39 show a lower by-pass rate that that of the 350 cfm but higher than that of the 200 cfm. Approximately 14 of the 64 stream lines, 21.9%, that were diffused in the room by-passed the system and did not circulate in the room.

A plot of the simulation residuals is shown in Figure 40. The plot shows that all 6 residuals were relatively converged at or below the 0.001 level which show how precise the results of the simulation are.

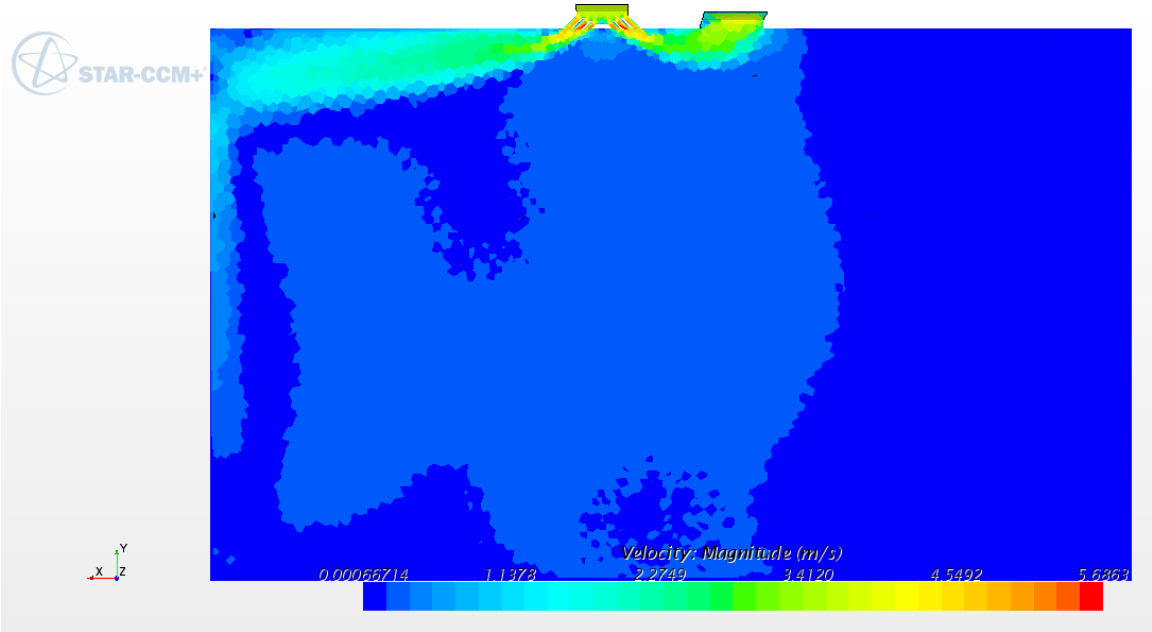

Figure 37: Two Dimensional Planar Section of the office; 12 in – 500cfm

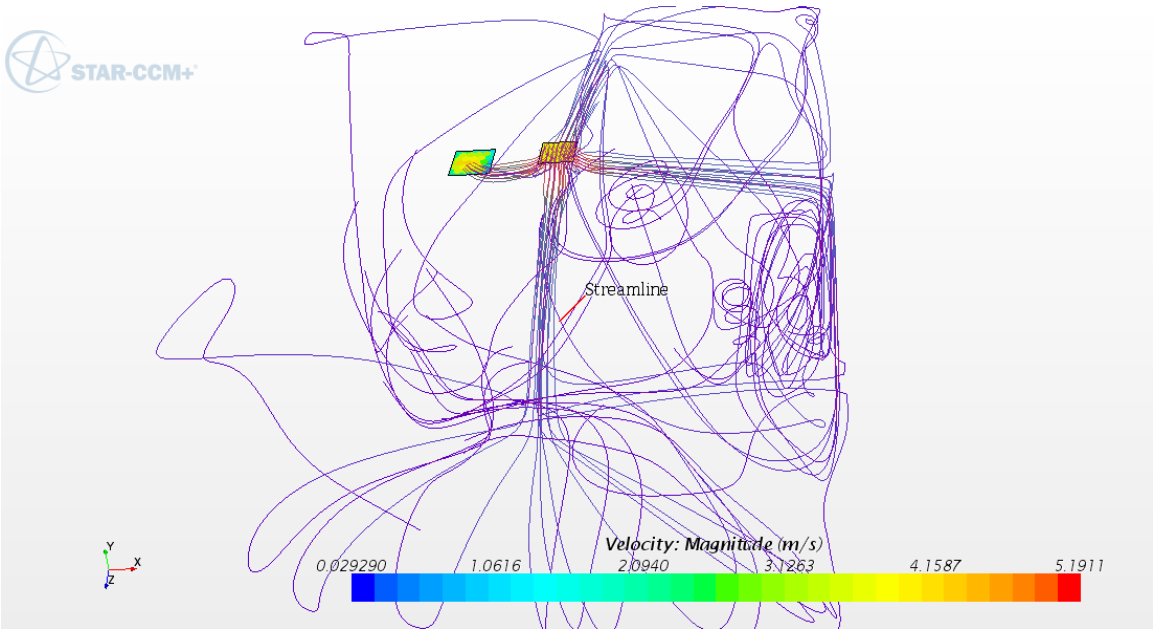

Figure 38: Three dimensional model of air circulation in the zone; 12 in – 500cfm

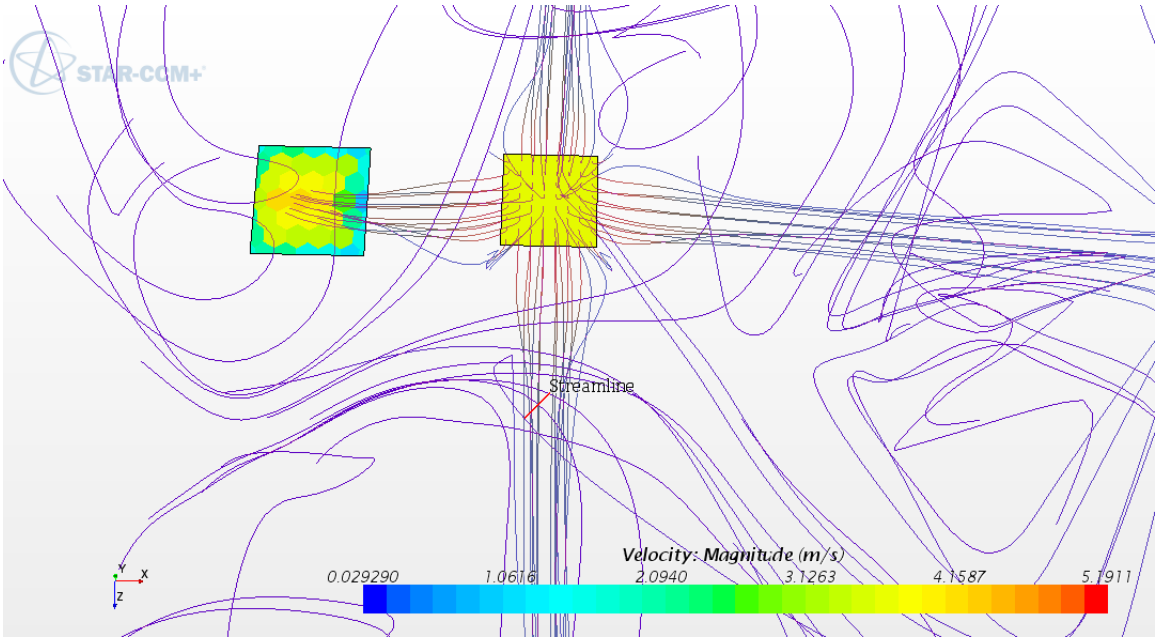

Figure 39: Three dimensional model of air circulation in the zone; 12 in – 500cfm

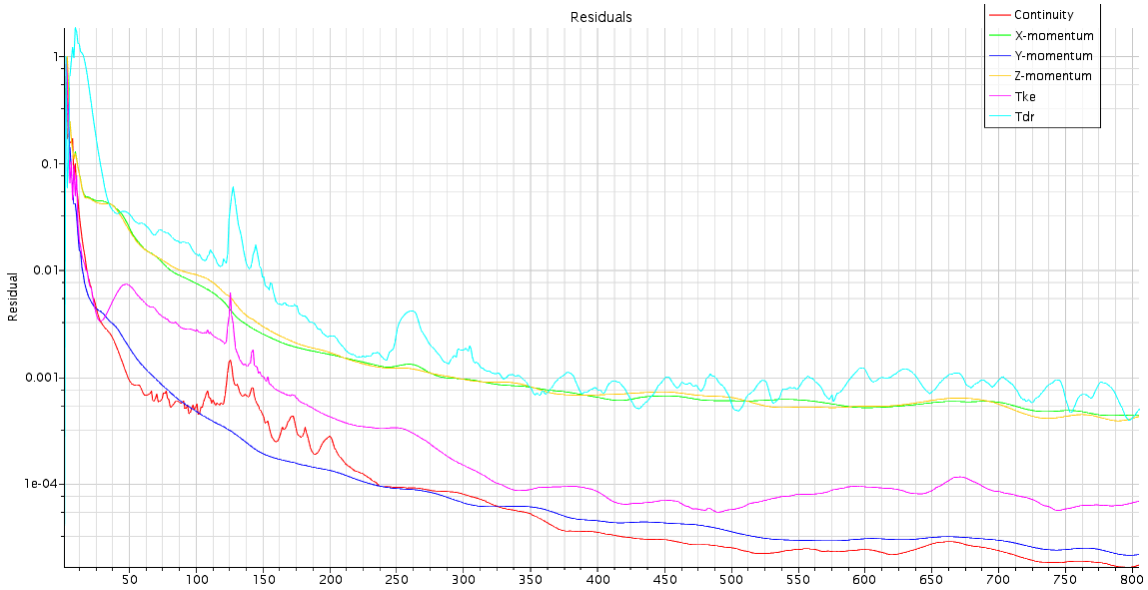

Figure 40: Plot Residuals; 12 in – 500cfm

Square Diffuser: 24 inches away from the return; 200 cfm

The results obtained from figures 41, 42 and 43 show a lower rate of air by-pass and more circulation than that of the 12 inches away from the return duct. This is due to the fact that the diffuser is further away from the return grille which allowed for more air to circulate. All figures show that some flow continued to circulate past the return grille and onto the room. Approximately 13 of the 64 stream lines, 20.3%, that were diffused in the room by-passed the system and did not circulate in the room.

A plot of the simulation residuals is shown in Figure 44. The plot shows that all 6 residuals were relatively converged at or below the 0.001 level which show how precise the results of the simulation are.

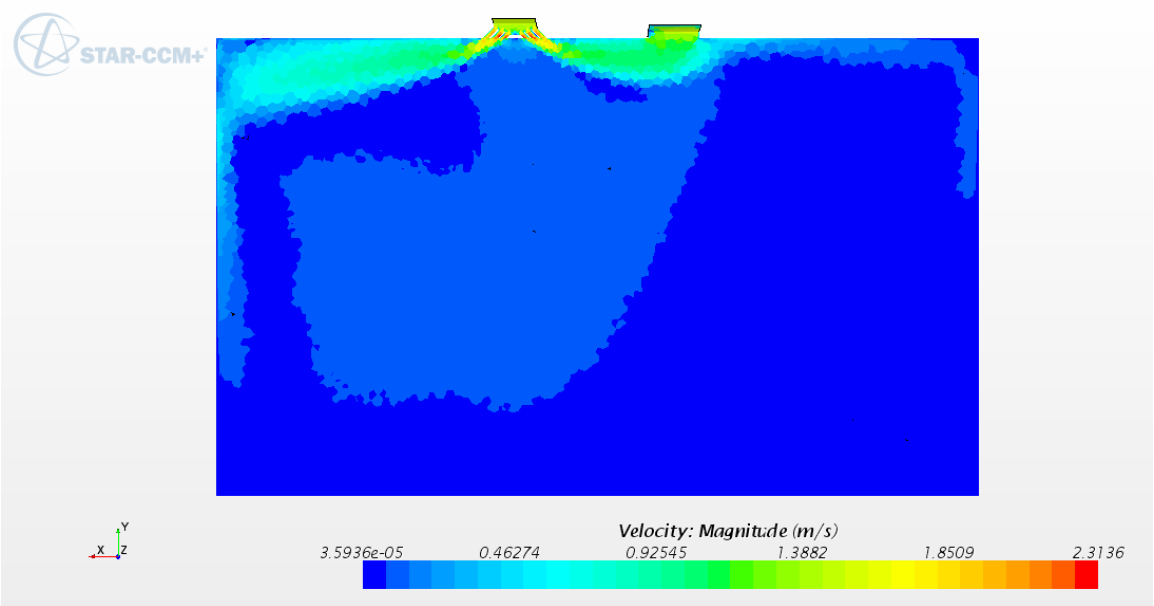

Figure 41: Two Dimensional Planar Section of the office; 24 in – 200cfm

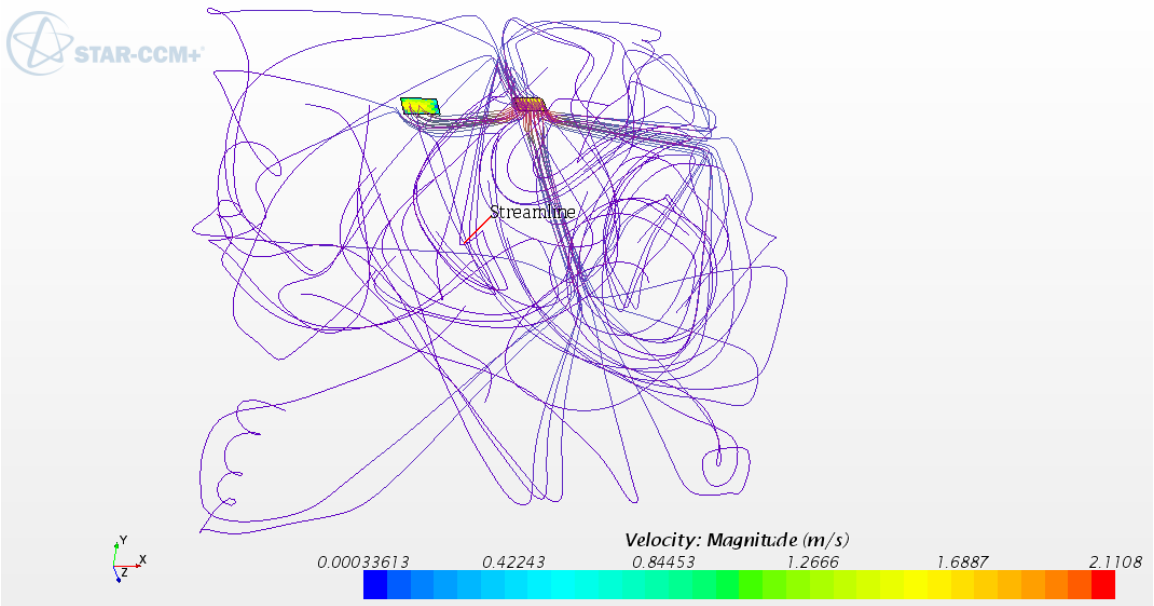

Figure 42: Three dimensional model of air circulation in the zone; 24 in – 200cfm

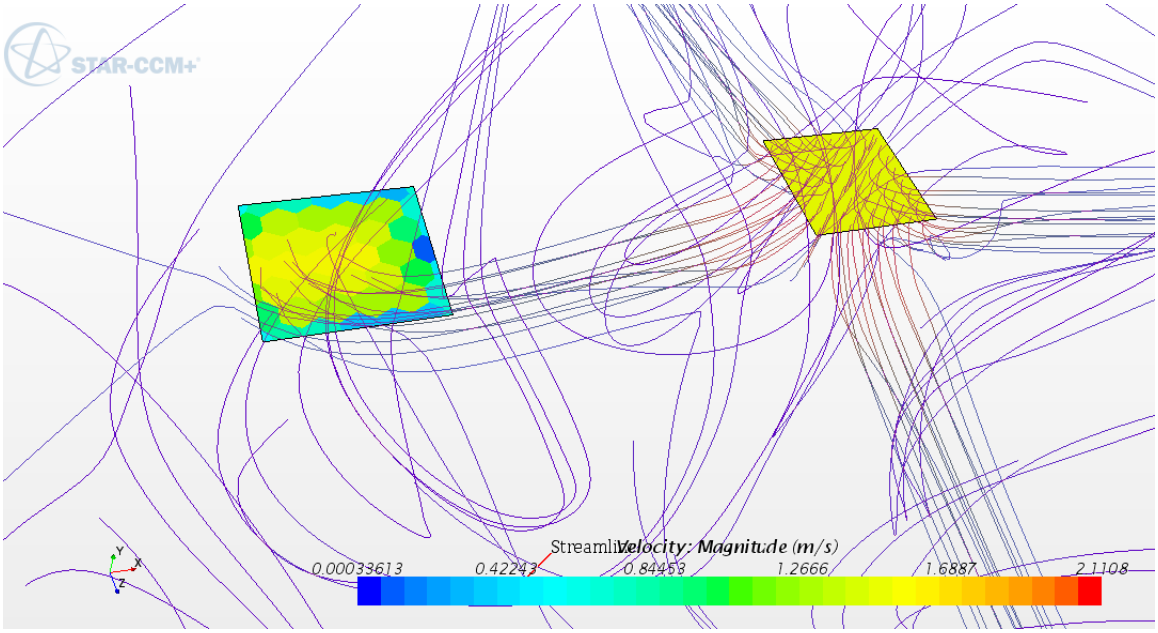

Figure 43: Three dimensional model of air circulation in the zone; 24 in – 200cfm

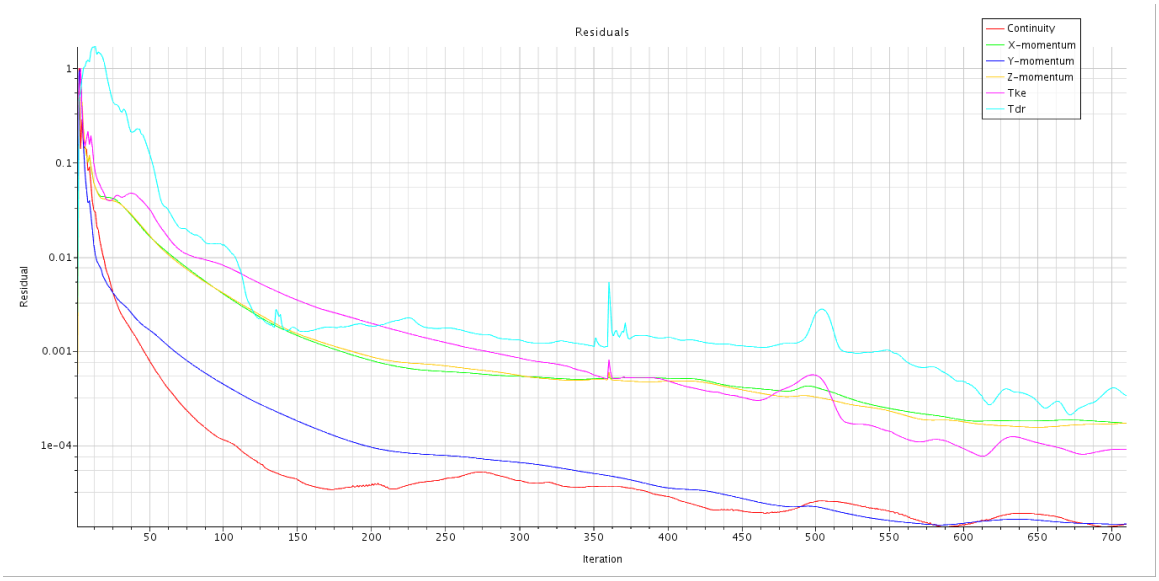

Figure 44: Plot Residuals; 24 in – 200cfm

Square Diffuser: 24 inches away from the return; 350 cfm

At 24 inches and 350 cfm, figures 45, 46 and 47 show a higher rate of by-pass than other simulations at the same placement, 24 inches away. Figure 45 shows that some flow continued to circulate past the return grille and onto the room. Approximately 14 of the 64 stream lines, 21.9%, that were diffused in the room by-passed the system and did not circulate in the room.

A plot of the simulation residuals is shown in Figure 48. The plot shows that all 6 residuals were relatively converged at or below the 0.005 level which show how precise the results of the simulation are.

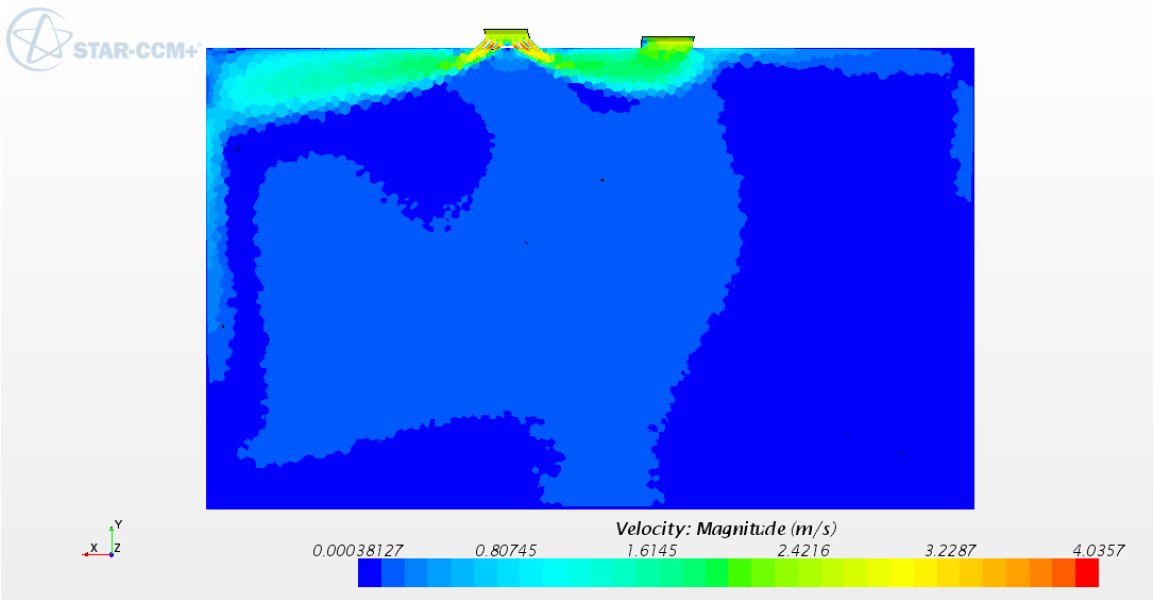

Figure 45: Two Dimensional Planar Section of the office; 24 in – 350cfm

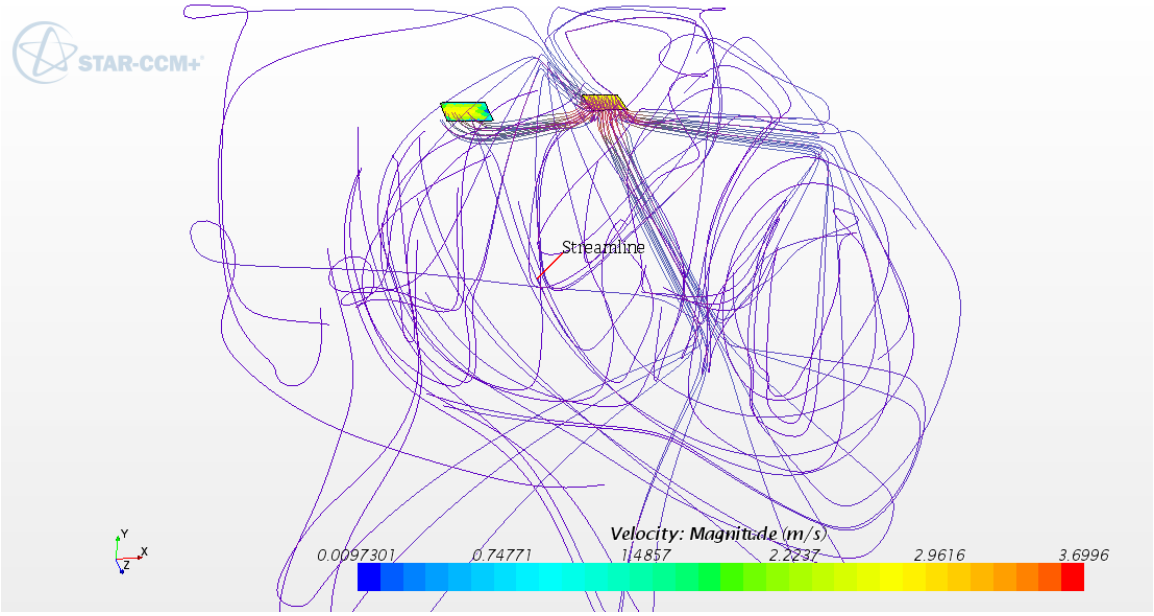

Figure 46: Three dimensional model of air circulation in the zone; 24 in – 350cfm

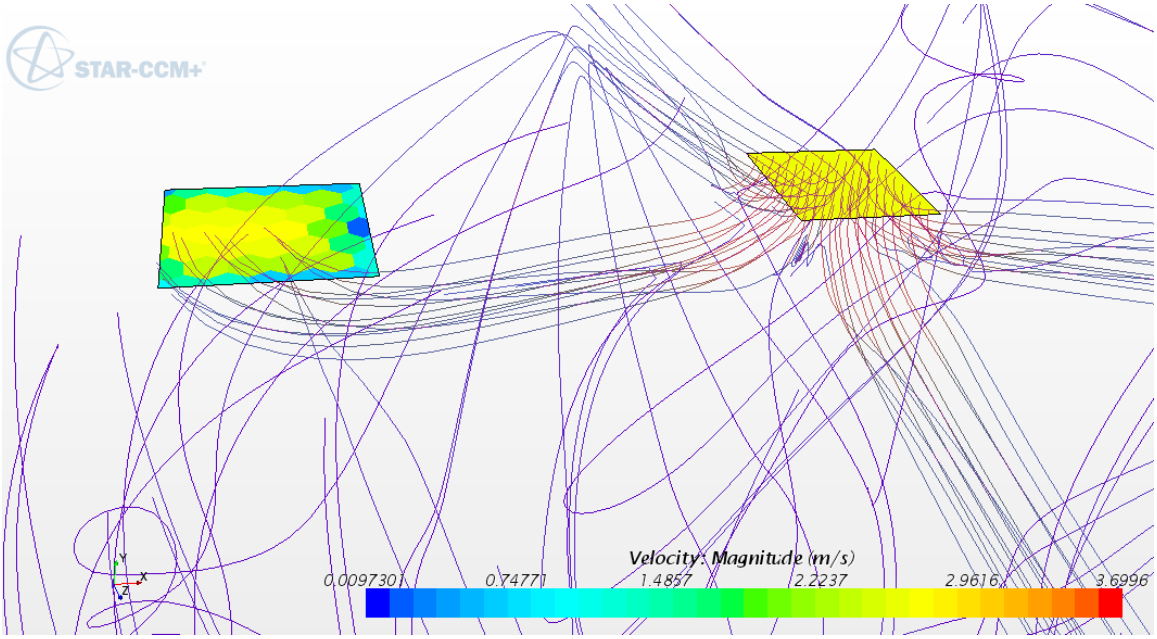

Figure 47: Three dimensional model of air circulation in the zone; 24 in – 350cfm

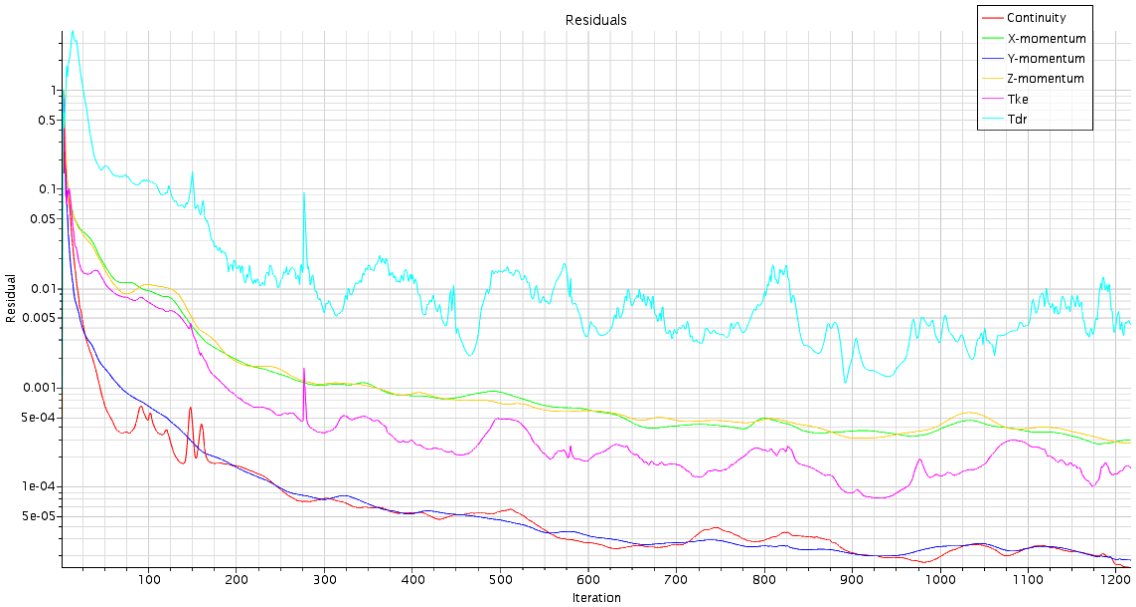

Figure 48: Plot Residuals; 24 in – 350cfm

Square Diffuser: 24 inches away from the return; 500 cfm

The results obtained from figures 49, 50 and 51 show similar behavior of lower rate of air by-pass and more circulation than the simulations with a square diffuser at 12 inches away from the return grille. Figure 49 shows that some flow continued to circulate past the return grille and onto the room. Approximately 13 of the 64 stream lines, 18.8%, that were diffused in the room by-passed the system and did not circulate in the room.

A plot of the simulation residuals is shown in Figure 52. The plot shows that all 6 residuals were relatively converged at or below the 0.0001 level which show how precise the results of the simulation are.

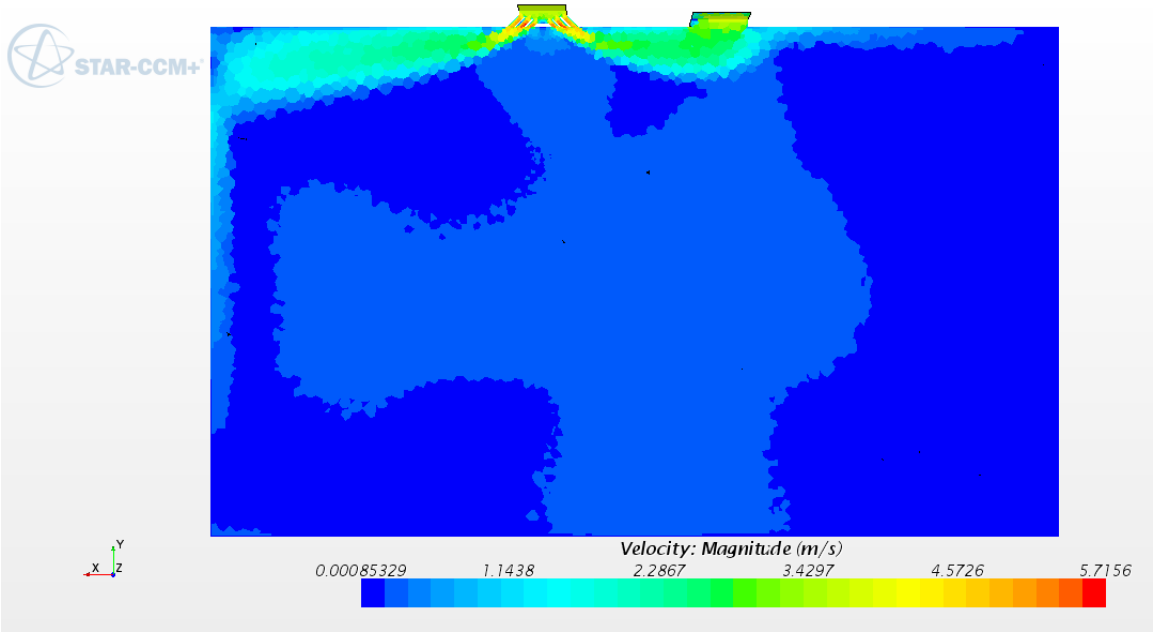

Figure 49: Two Dimensional Planar Section of the office; 24 in – 500cfm

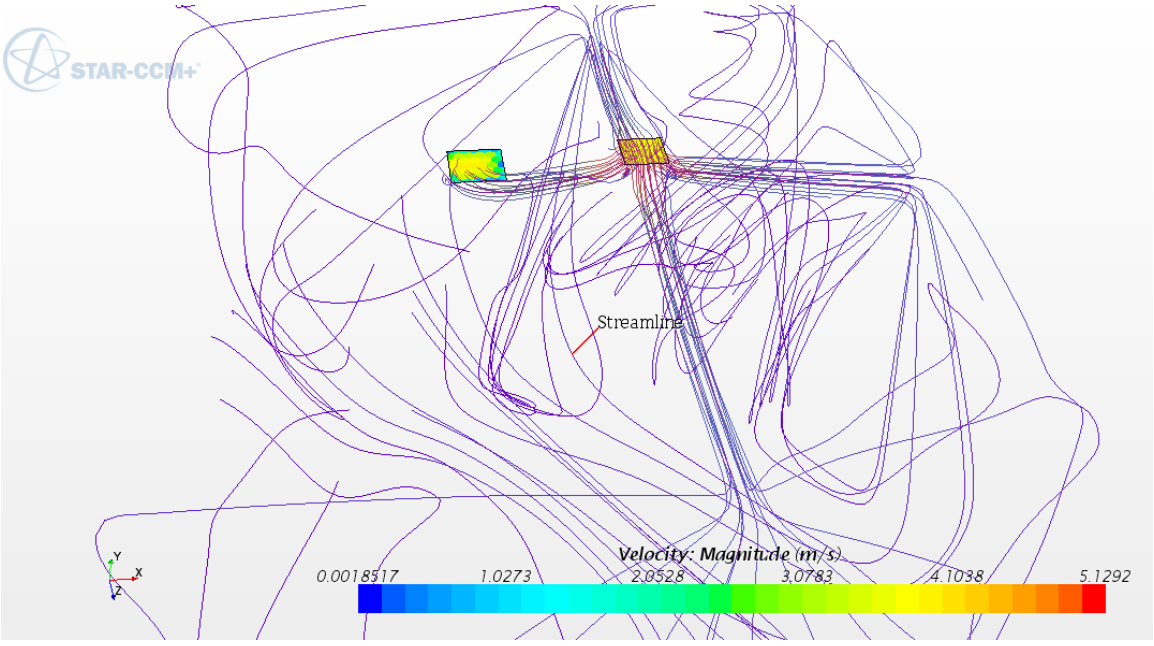

Figure 50: Three dimensional model of air circulation in the zone; 24 in – 500cfm

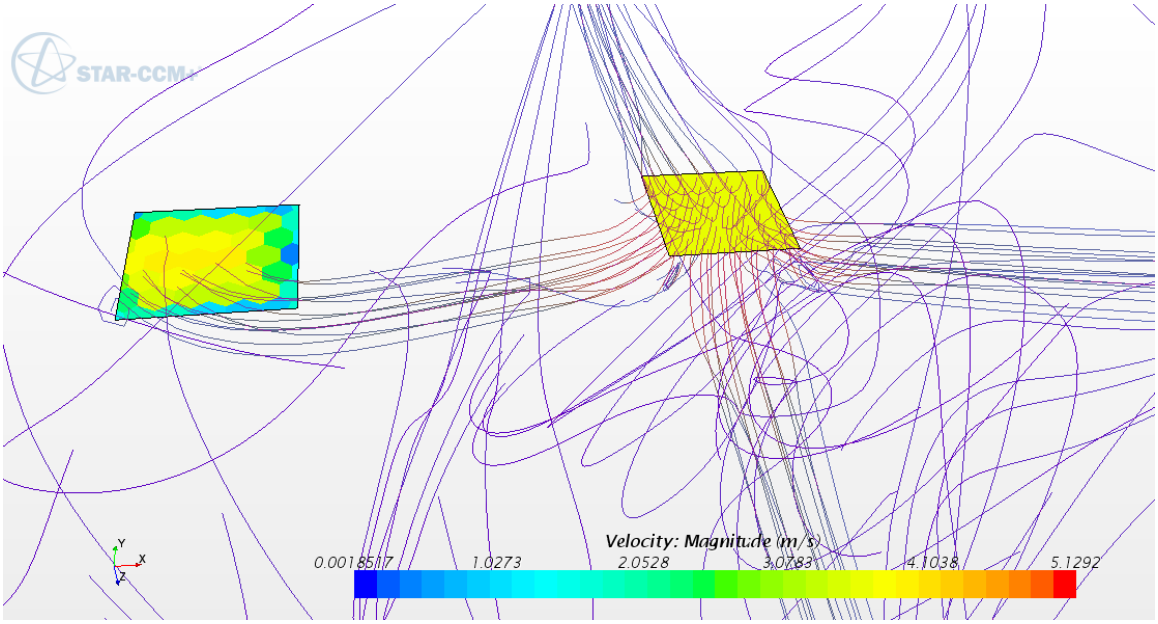

Figure 51: Three dimensional model of air circulation in the zone; 24 in – 500cfm

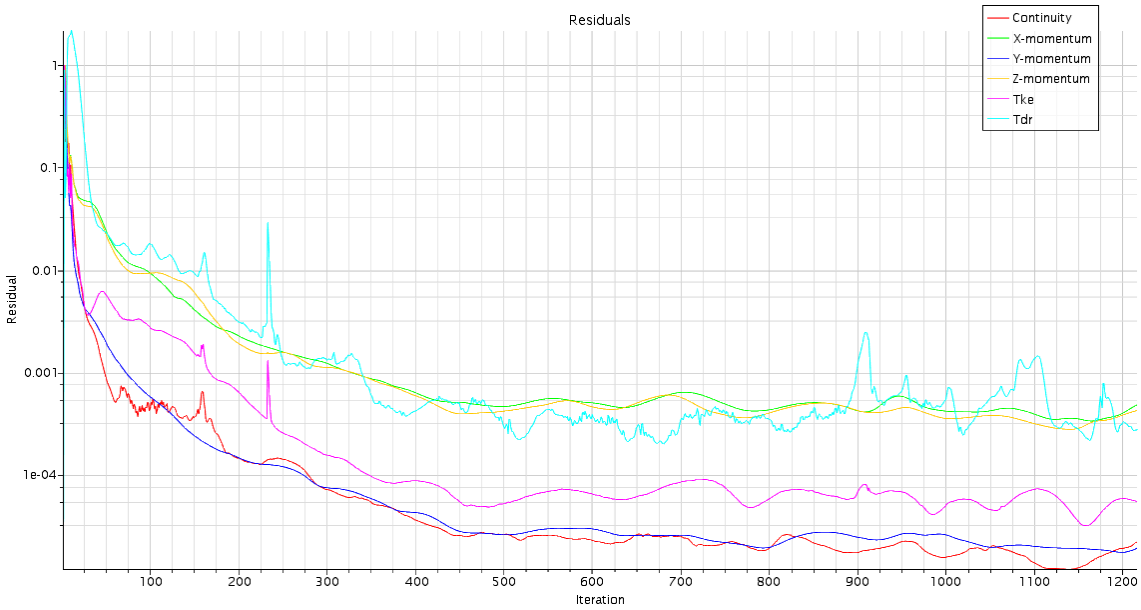

Figure 52: Plot Residuals; 24 in – 500cfm

Square Diffuser: 36 inches away from the return; 200 cfm

The results obtained from figures 53, 54 and 55 show a big drop in the amount of air by-pass; only 5 out of the 64 stream lines by-passed the system while the rest of the air flow continued to circulate in the room. Figure 53 has a clear visible image of the air pass under the return grille and on onto the rest of the room.

A plot of the simulation residuals is shown in Figure 56. The plot shows that all 6 residuals were relatively converged at or below the 0.005 level which show how precise the results of the simulation are.

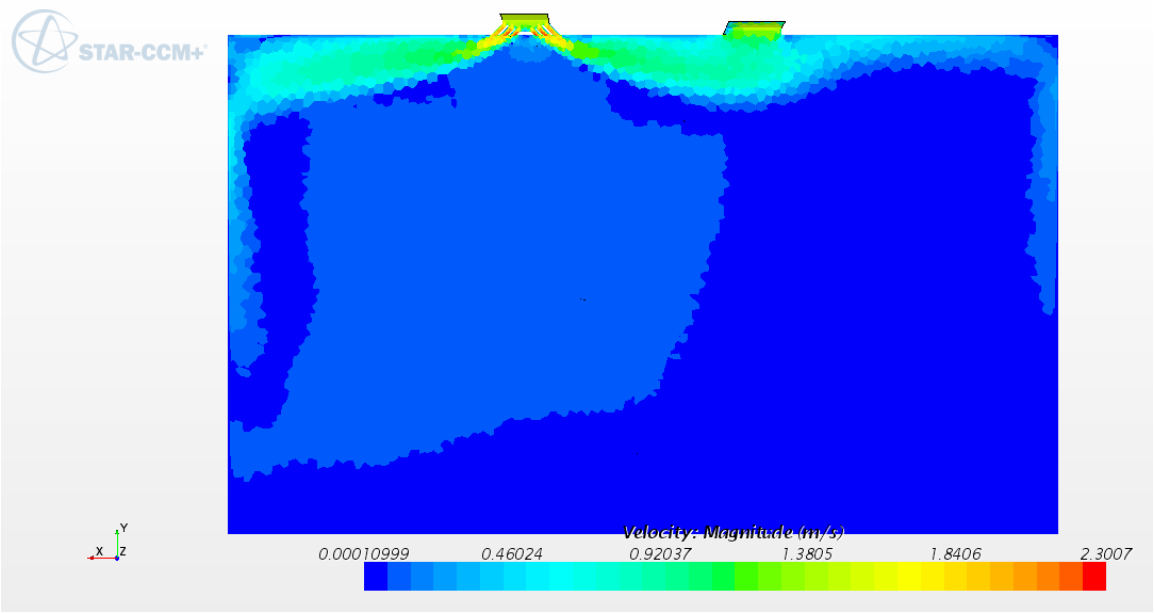

Figure 53: Two Dimensional Planar Section of the office; 36 in – 200cfm

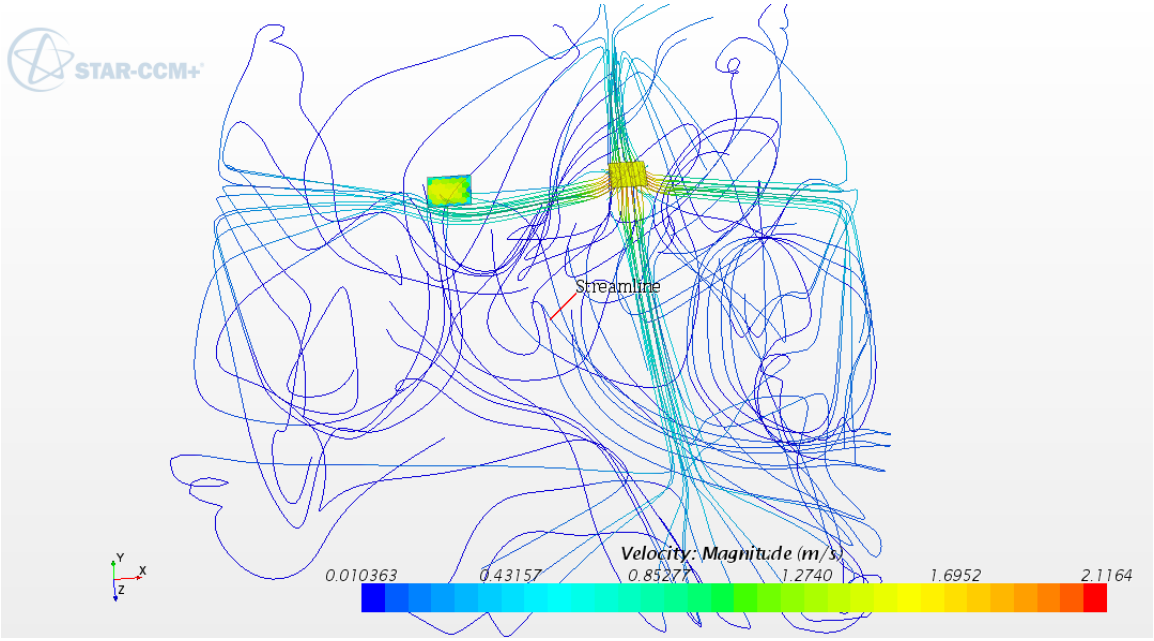

Figure 54: Three dimensional model of air circulation in the zone; 36 in – 200cfm

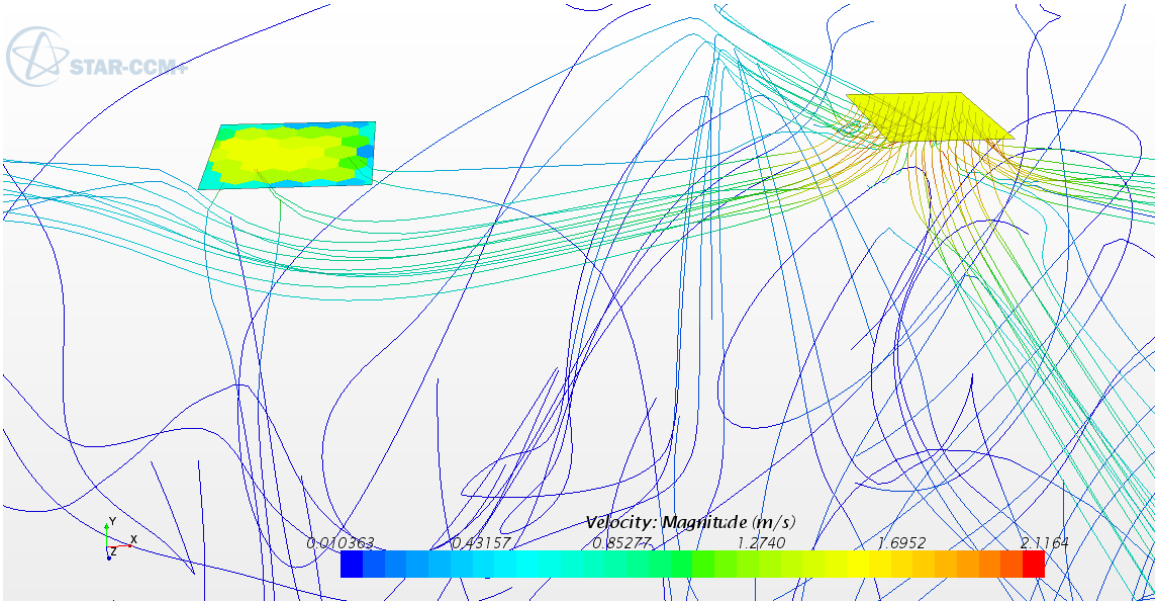

Figure 55: Three dimensional model of air circulation in the zone; 36 in – 200cfm

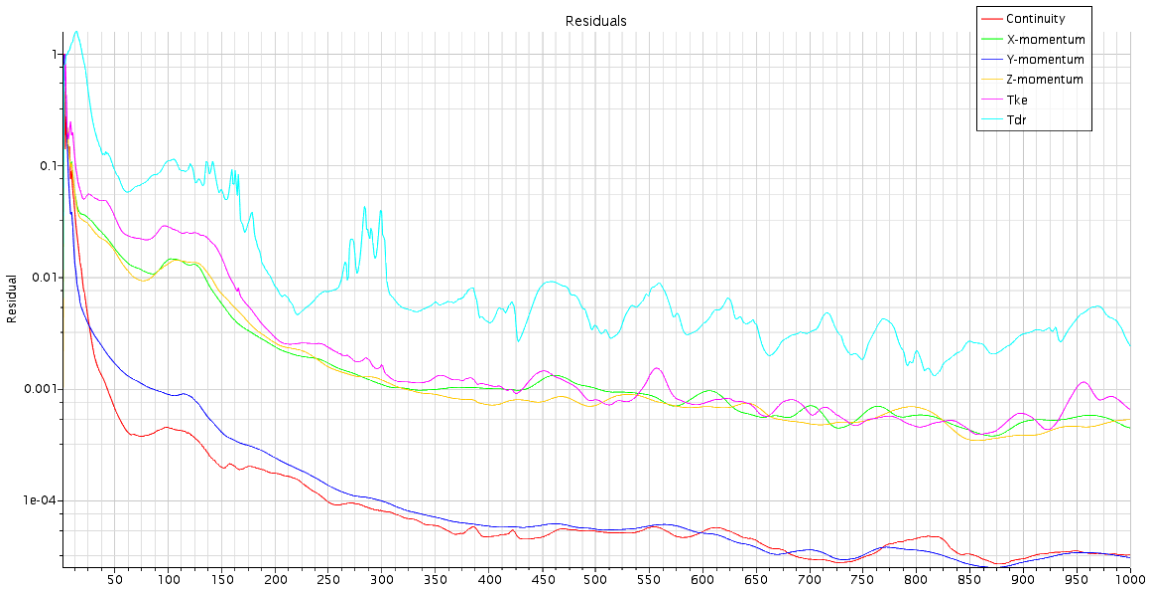

Figure 56: Plot Residuals; 36 in – 200cfm

Square Diffuser: 36 inches away from the return; 350 cfm

The results obtained from figures 57, 58 and 59 show a difference in the efficiency of the system at 350 cfm versus 200 cfm; 11 out of the 64 stream lines by-passed the system. That is approximately 17.2% by-pass rate compared to 7.8% when the flow was 200 cfm. The reason for the big difference is the higher air flow rate which forces more air towards the return grille thus extra short-circuiting. Figure 57 shows that much of the air that is going towards the return grille by-passes the system.

A plot of the simulation residuals is shown in Figure 60. The plot shows that all 6 residuals were relatively converged at or below the 0.01 level which show how precise the results of the simulation are.

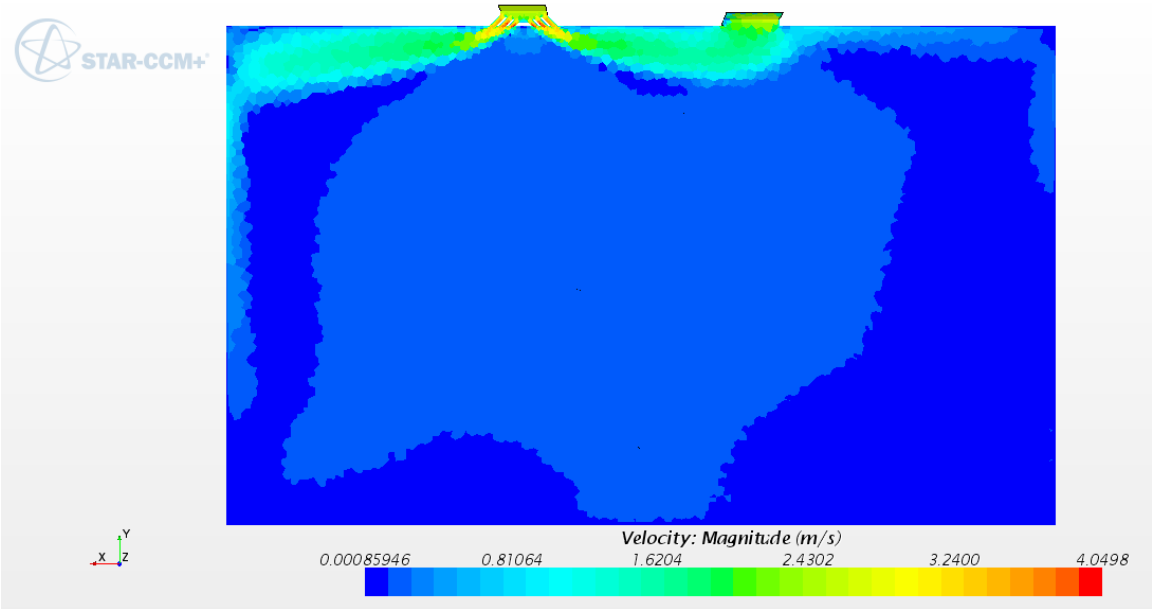

Figure 57: Two Dimensional Planar Section of the office; 36 in – 350cfm

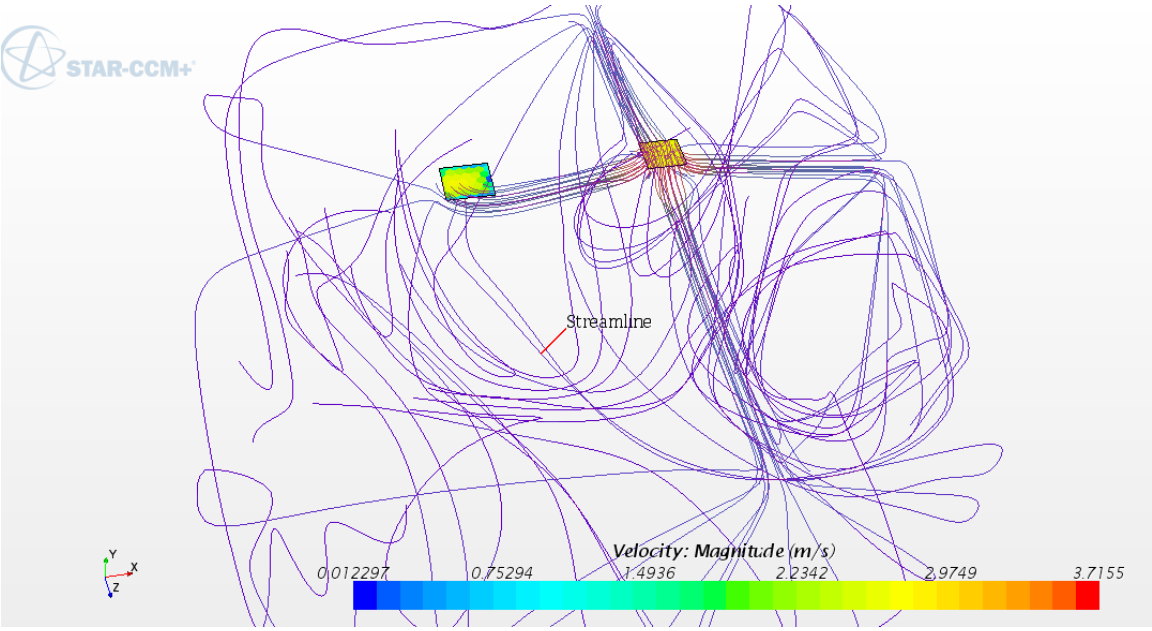

Figure 58: Three dimensional model of air circulation in the zone; 36 in – 350cfm

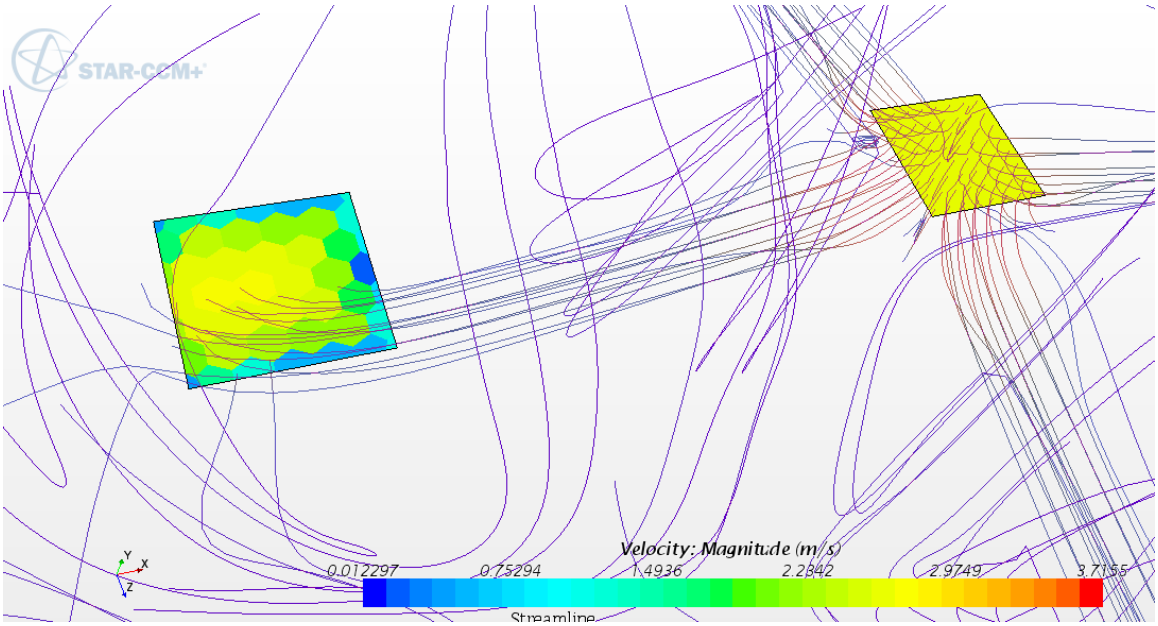

Figure 59: Three dimensional model of air circulation in the zone; 36 in –350cfm

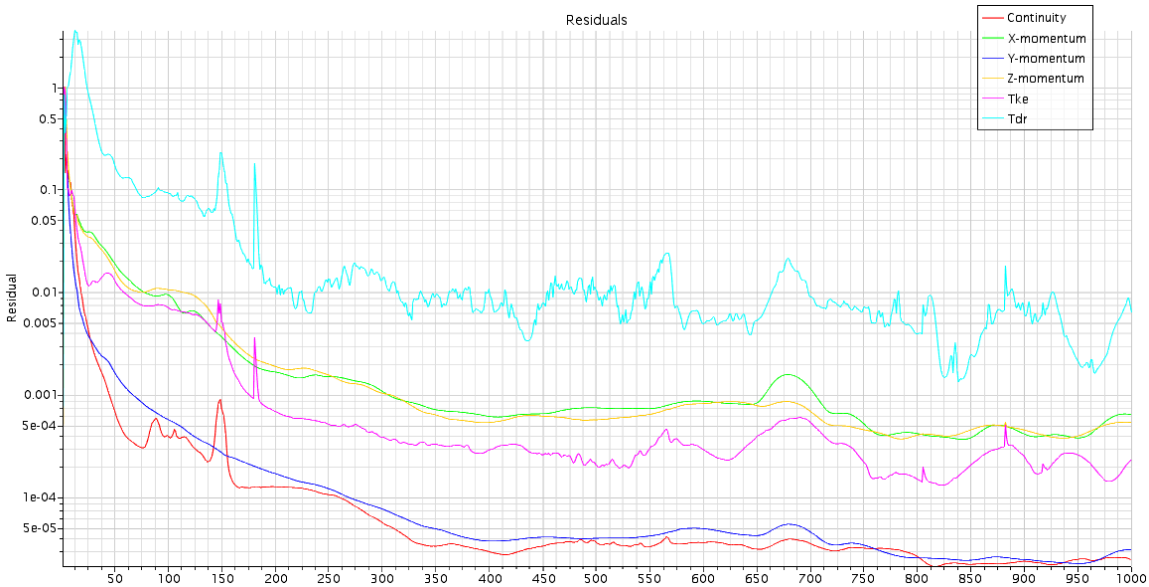

Figure 60: Plot Residuals; 36 in – 350cfm

Square Diffuser: 36 inches away from the return; 500 cfm

The results obtained from figures 61, 62 and 63 show a difference in the efficiency of the system at 500 cfm versus 200 cfm and 350 cfm; 8 out of the 64 stream lines by-passed the system. That is approximately 12.5% by-pass rate compared to 7.8% at 200 cfm and 17.2% at 350 cfm. At 500 cfm, the flow was strong enough to force the air past the return grille and continue circulating.

A plot of the simulation residuals is shown in Figure 64. The plot shows that all 6 residuals were relatively converged at or below the 0.001 level which show how precise the results of the simulation are.

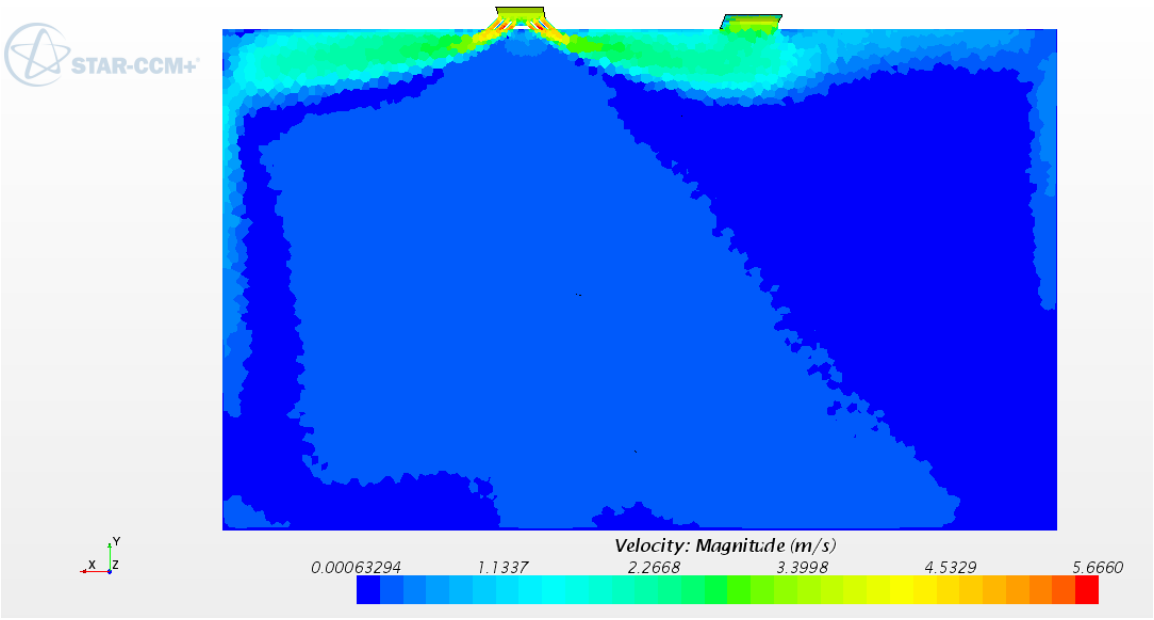

Figure 61: Two Dimensional Planar Section of the office; 36 in – 500cfm

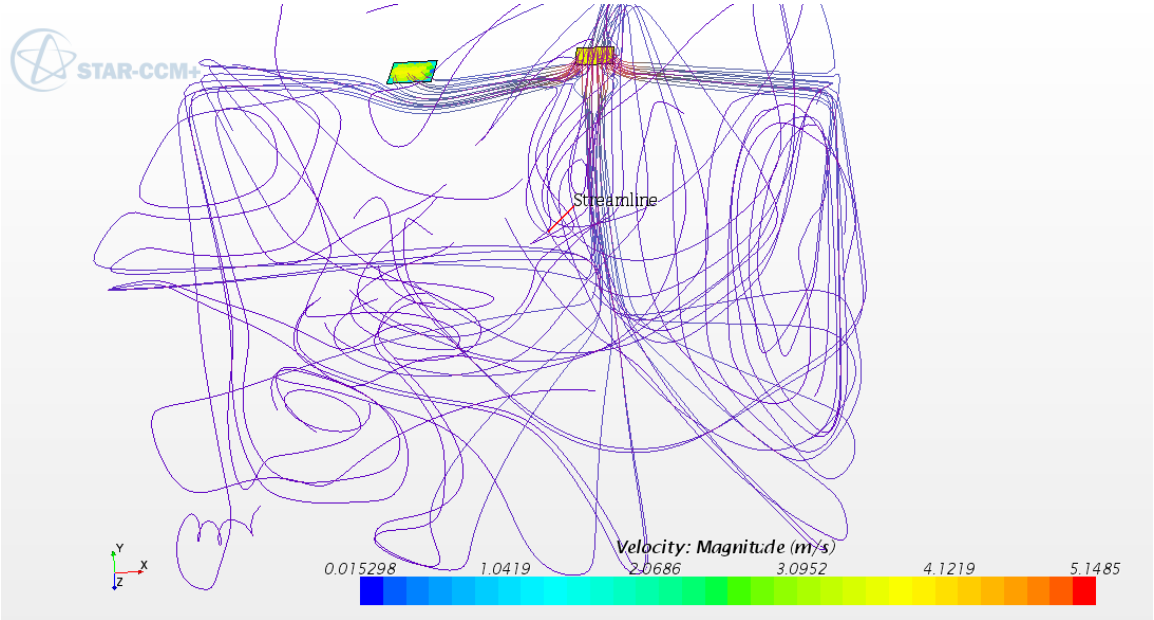

Figure 62: Three dimensional model of air circulation in the zone; 36 in –500cfm

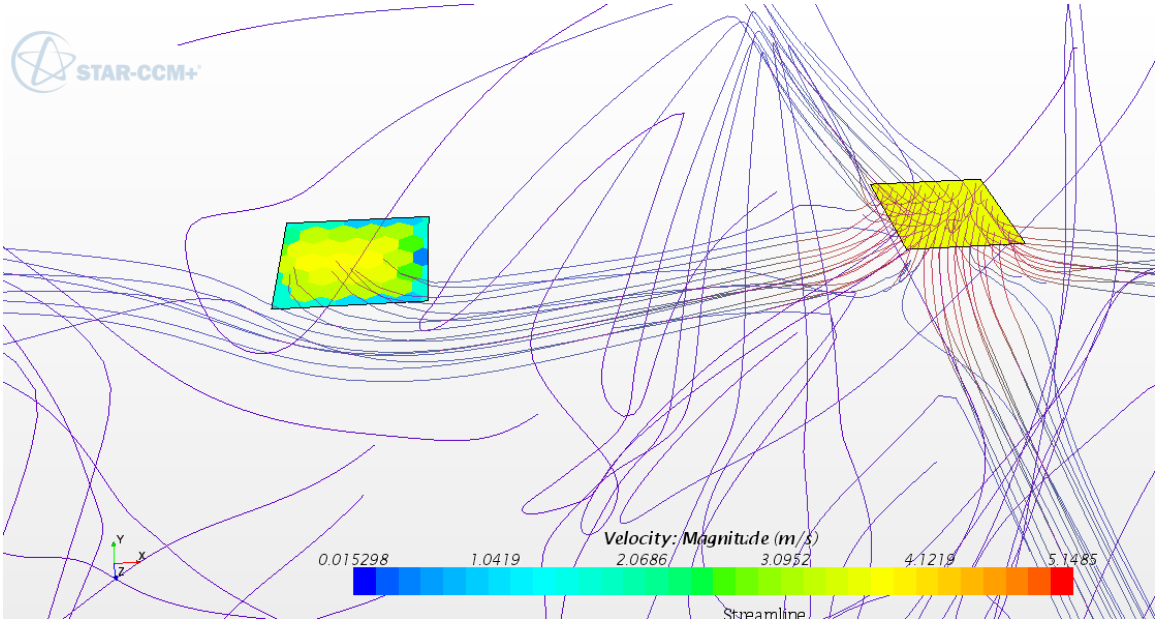

Figure 63: Three dimensional model of air circulation in the zone; 36 in –500cfm

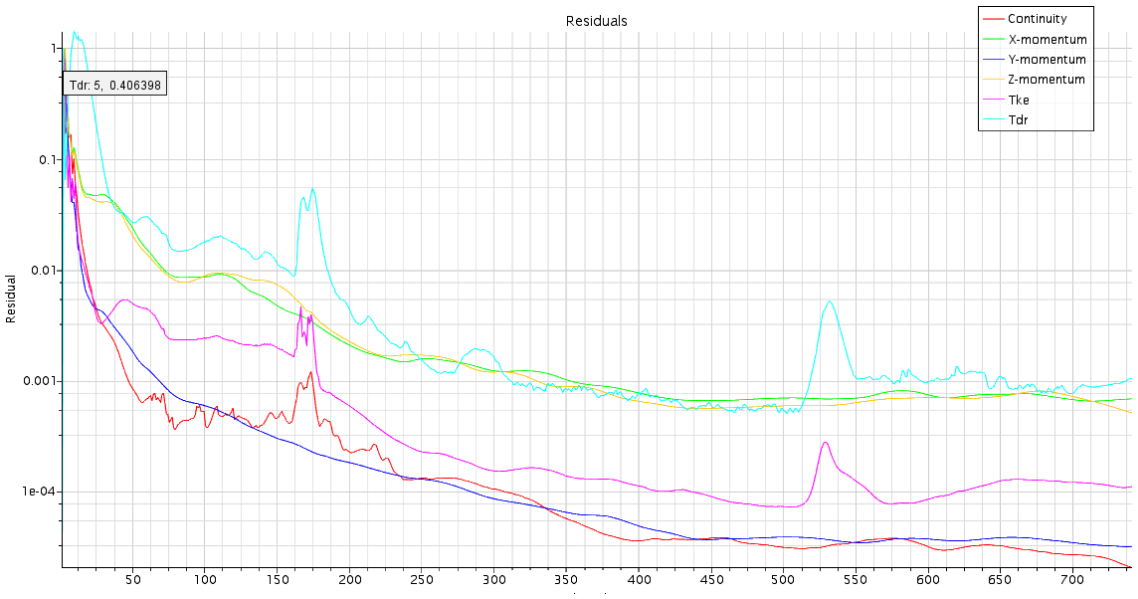

Figure 64: Plot Residuals; 36 in – 500cfm

Square Diffuser: 48 inches away from the return; 200 cfm

The results obtained from figures 65, 66 and 67 show a by-pass rate of 6.3%; only 4 out of the 64 stream lines by-passed the system. At lower air flow, air circulation seems to be better as the short-circuiting of the system is greatly reduced.

A plot of the simulation residuals is shown in Figure 68. The plot shows that all 6 residuals were relatively converged at or below the 0.005 level which show how precise the results of the simulation are.

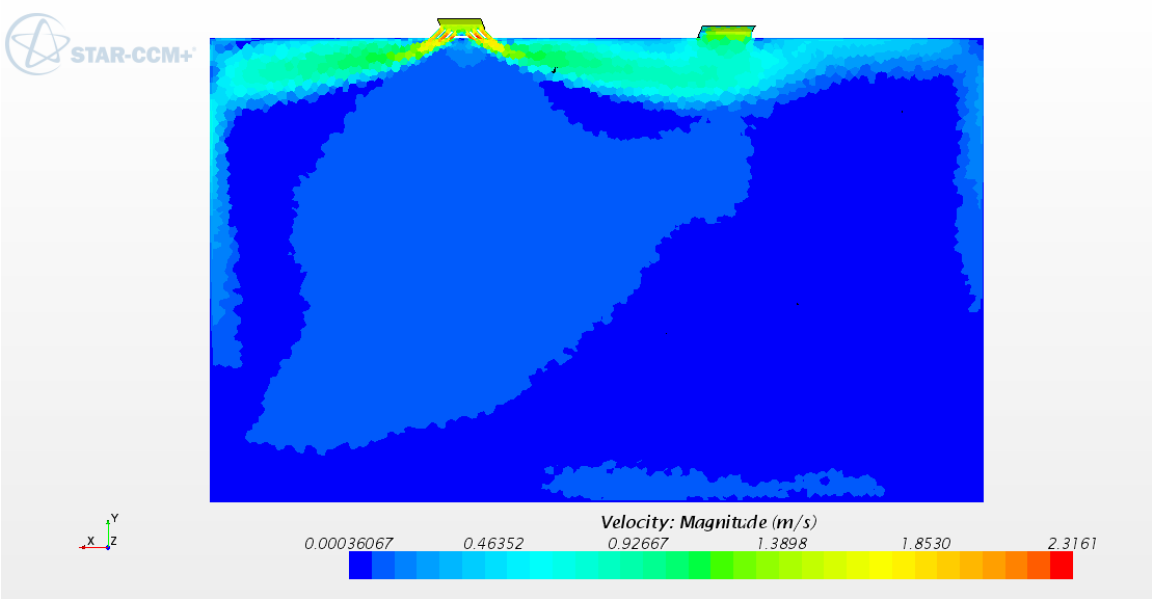

Figure 65: Two Dimensional Planar Section of the office; 48 in – 200cfm

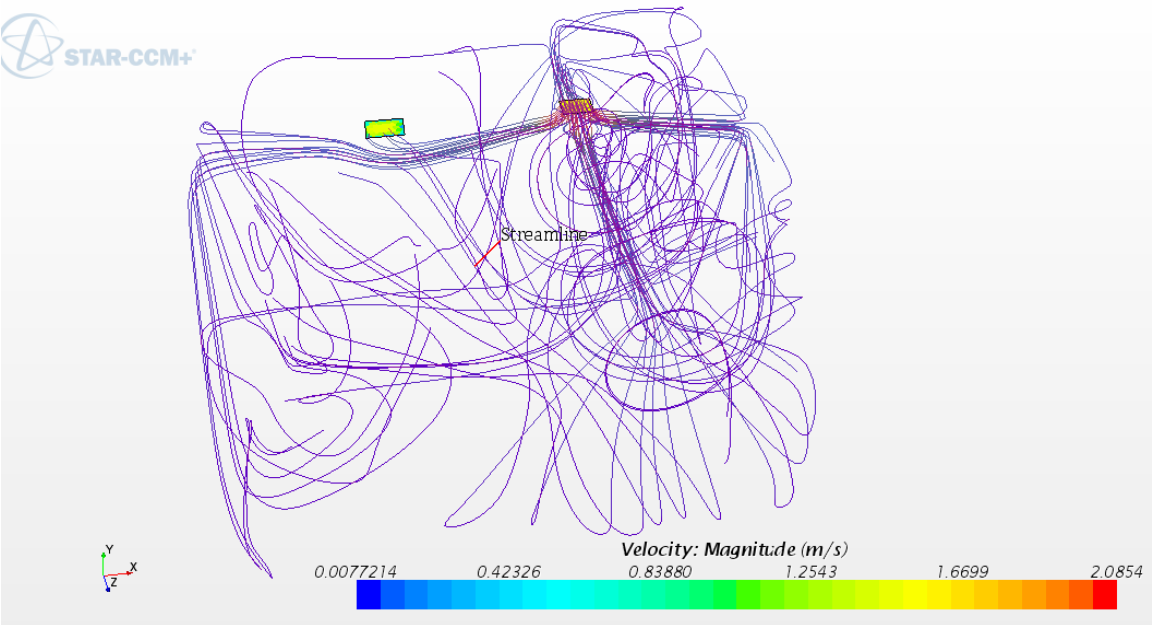

Figure 66: Three dimensional model of air circulation in the zone; 48 in –200cfm

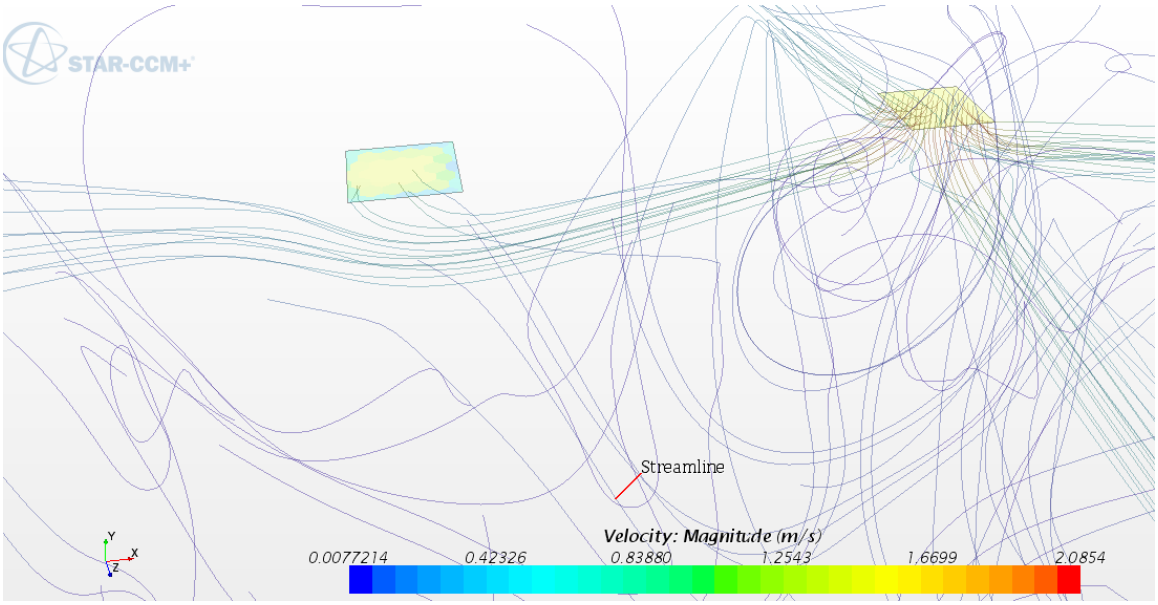

Figure 67: Three dimensional model of air circulation in the zone; 48 in –200cfm

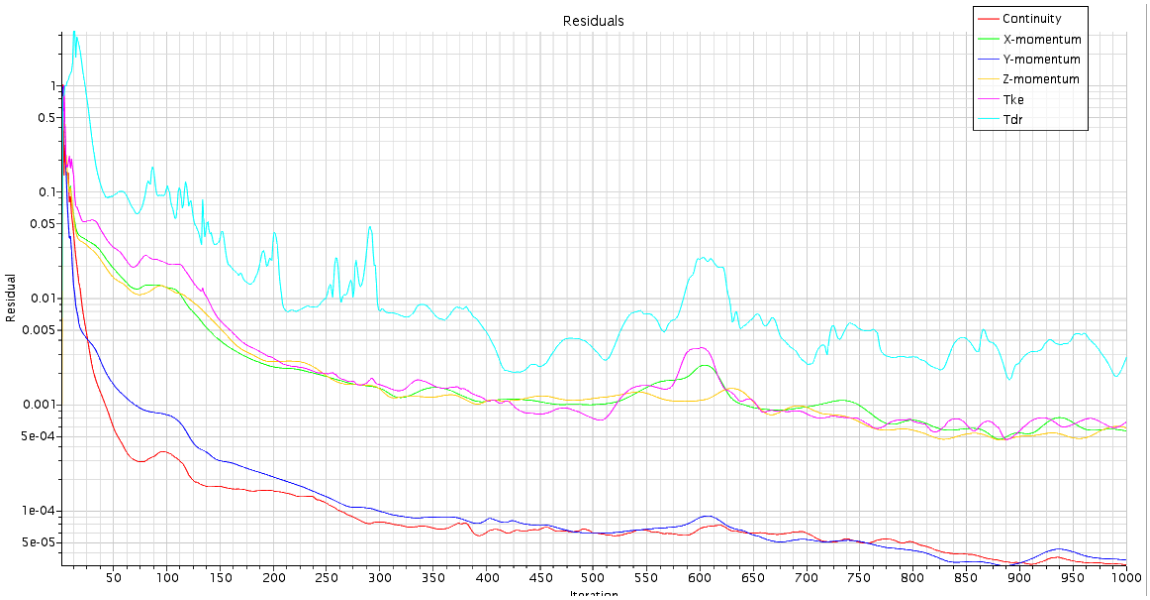

Figure 68: Plot Residuals; 48 in – 200cfm

Square Diffuser: 48 inches away from the return; 350 cfm

The results obtained from figures 69, 70 and 71 show a by-pass rate of 10.9%; 7 out of the 64 stream lines by-passed the system. At 350 cfm, the efficiency of the system tends to be lower than the efficiency of the system at 200 cfm and 500 cfm; at 200 cfm, low air velocity and long distance reduce the by-pass rate. On the other hand, at 500 cfm, high air velocity pushes air past the return grille which reduces the by-pass rate.

A plot of the simulation residuals is shown in Figure 72. The plot shows that all 6 residuals were relatively converged at or below the 0.002 level which show how precise the results of the simulation are.

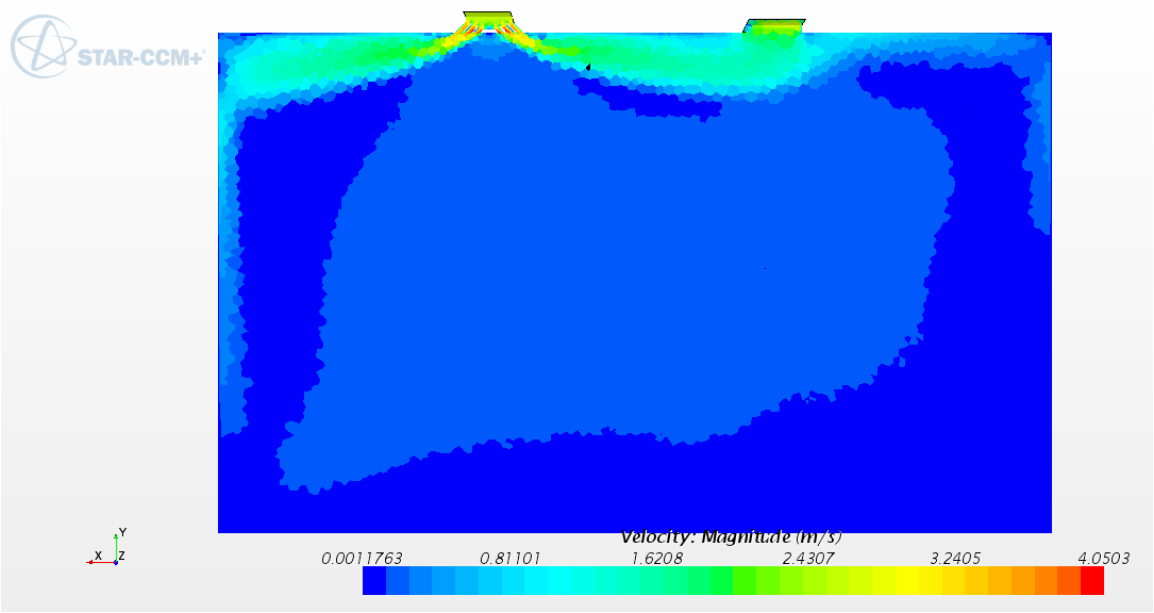

Figure 69: Two Dimensional Planar Section of the office; 48 in – 350cfm

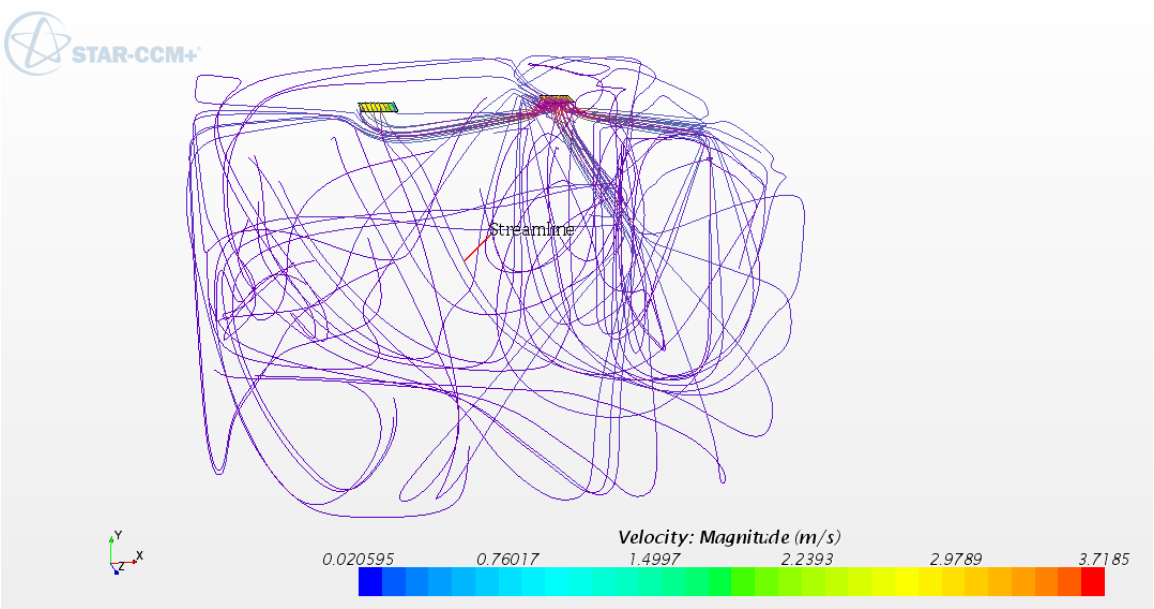

Figure 70: Three dimensional model of air circulation in the zone; 48 in – 350cfm

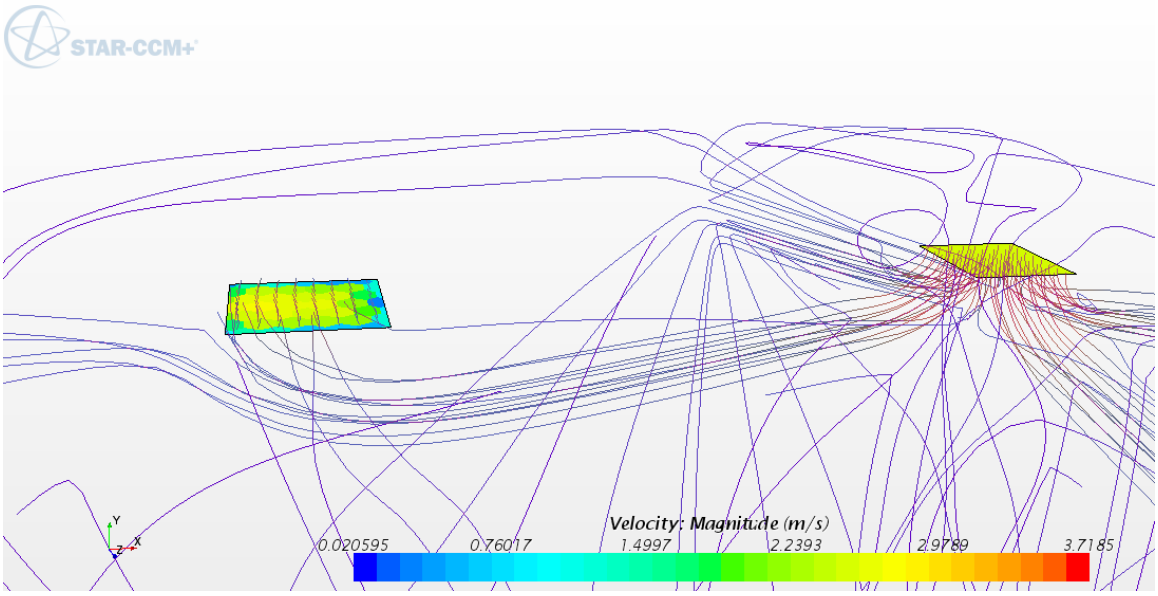

Figure 71: Three dimensional model of air circulation in the zone; 48 in – 350cfm

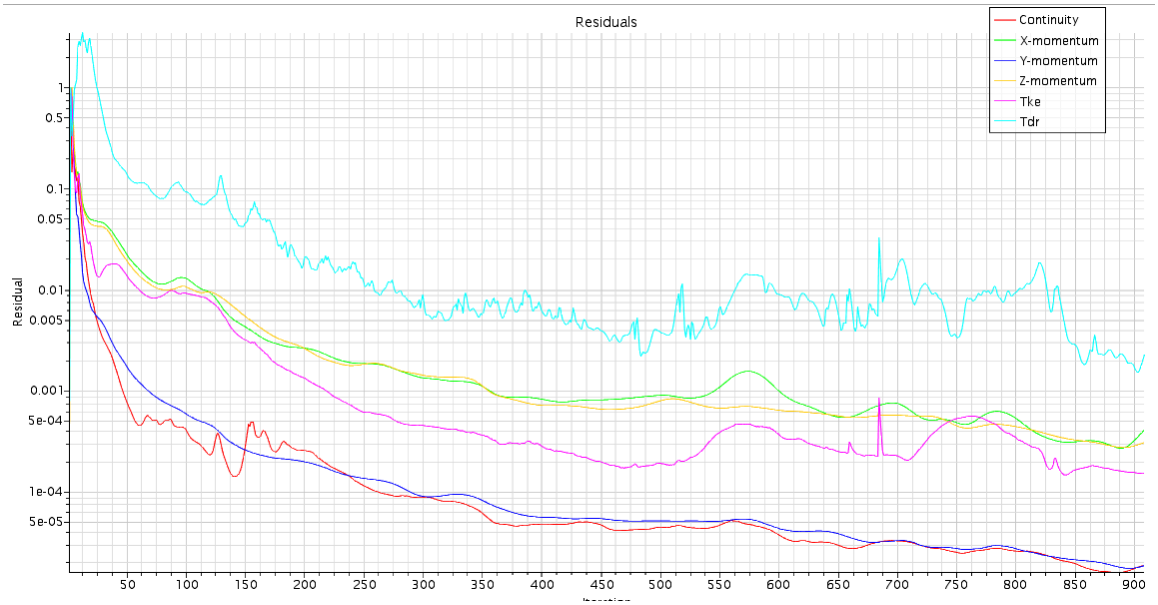

Figure 72: Plot Residuals; 48 in – 350cfm

Square Diffuser: 48 inches away from the return; 500 cfm

The results obtained from figures 73, 74 and 75 show a by-pass rate of 7.8%; 5 out of the 64 stream lines by-passed the system. The efficiency of the system at 500 cfm is better than that of the 350 cfm but lower than that of the 200 cfm.

A plot of the simulation residuals is shown in Figure 76. The plot shows that all 6 residuals were relatively converged at or below the 0.001 level which show how precise the results of the simulation are.

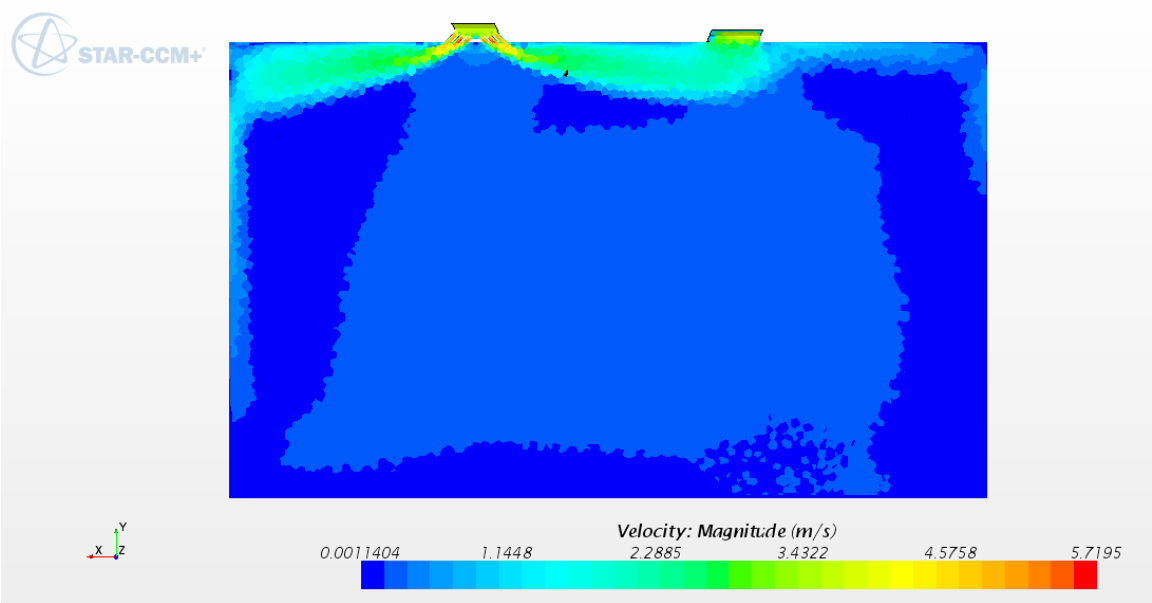

Figure 73: Two Dimensional Planar Section of the office; 48 in – 500cfm

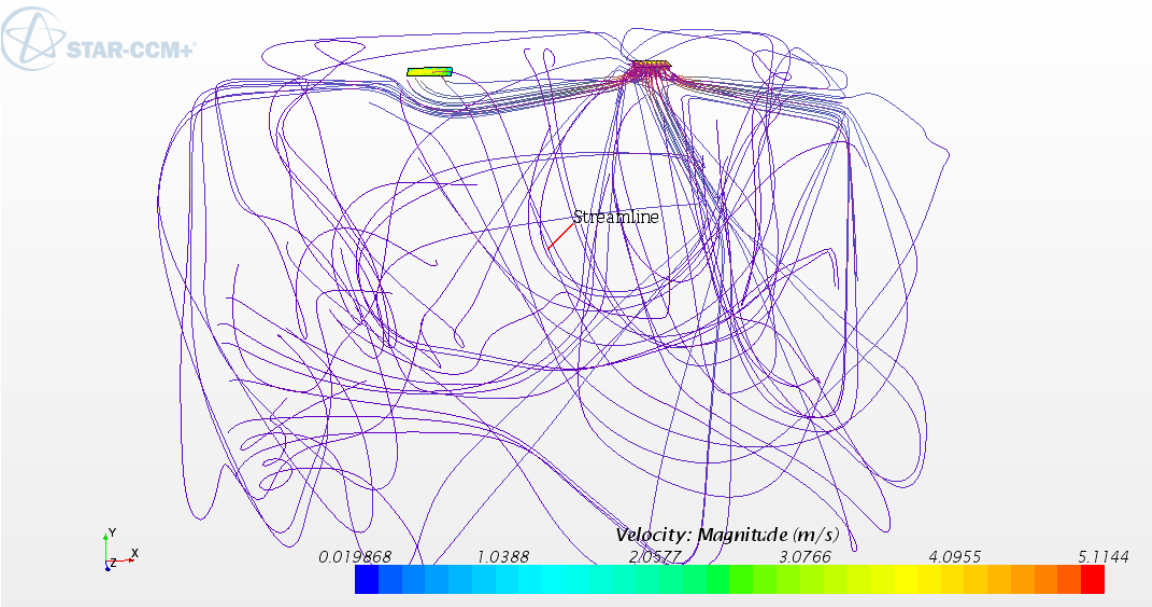

Figure 74: Three dimensional model of air circulation in the zone; 48 in – 500cfm

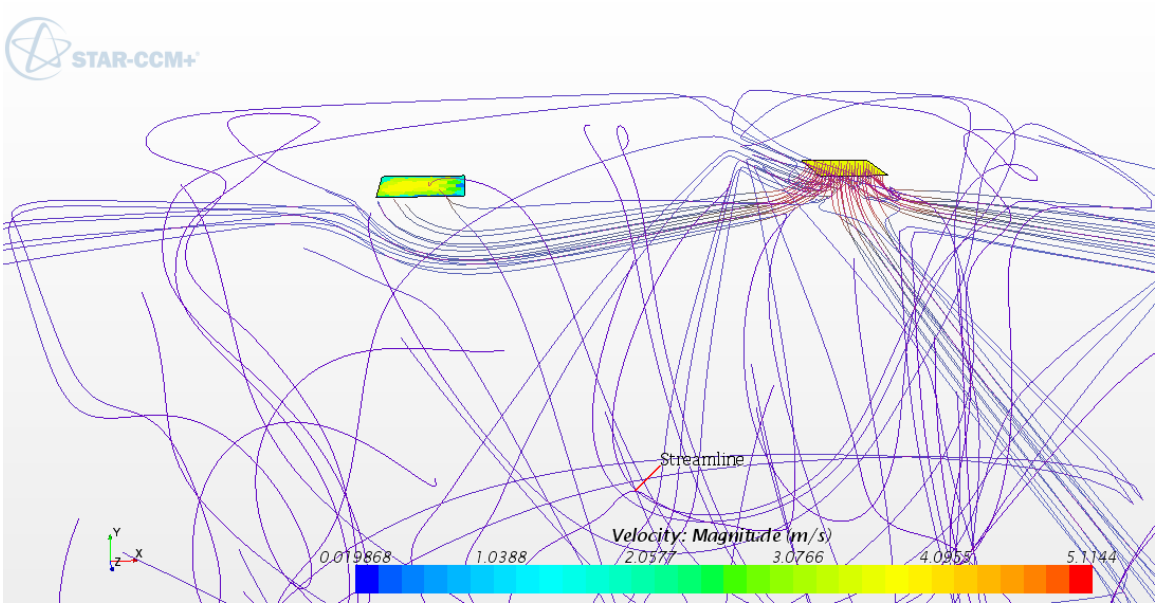

Figure 75: Three dimensional model of air circulation in the zone; 48 in – 500cfm

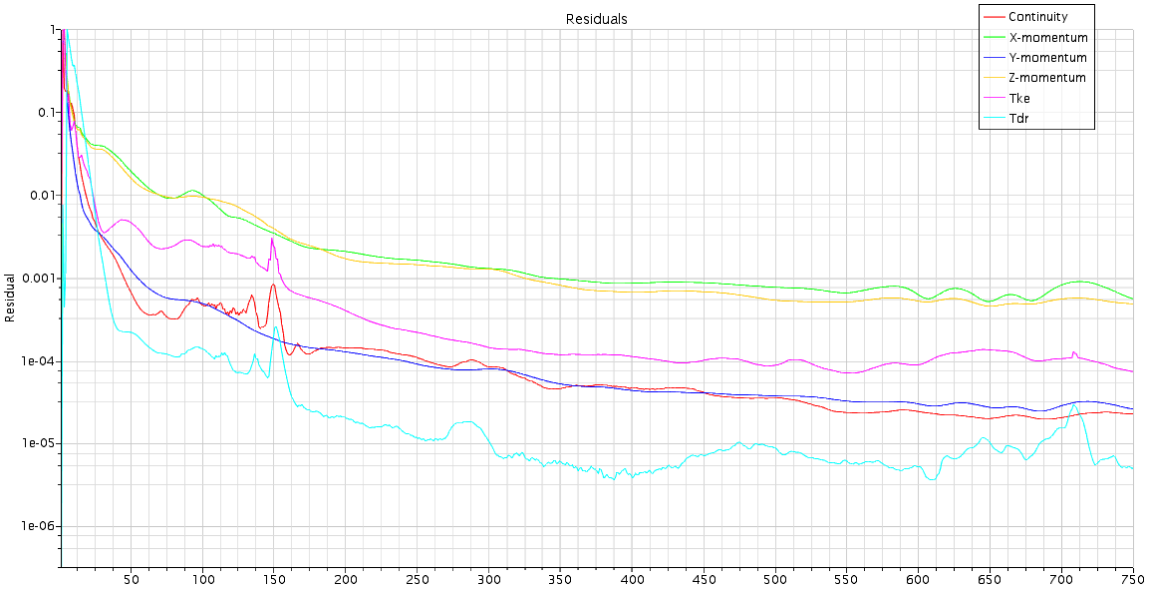

Figure 76: Plot Residuals; 48 in – 500cfm

Square Diffuser: 60 inches away from the return; 200 cfm

The results obtained from figures 77, 78 and 79 show a by-pass rate of 4.1%; 8 out of the 196 streamlines by-passed the system. 196 streamlines were used in this simulation for better accuracy that is needed at higher efficiency systems. The difference between the efficiencies of the 48 and 60 inches systems couldn't be observed when 64 stream lines were used as they both have only few lines of by-passed air out of the total sum.

A plot of the simulation residuals is shown in Figure 80. The plot shows that all 6 residuals were relatively converged at or below the 0.005 level which show how precise the results of the simulation are.

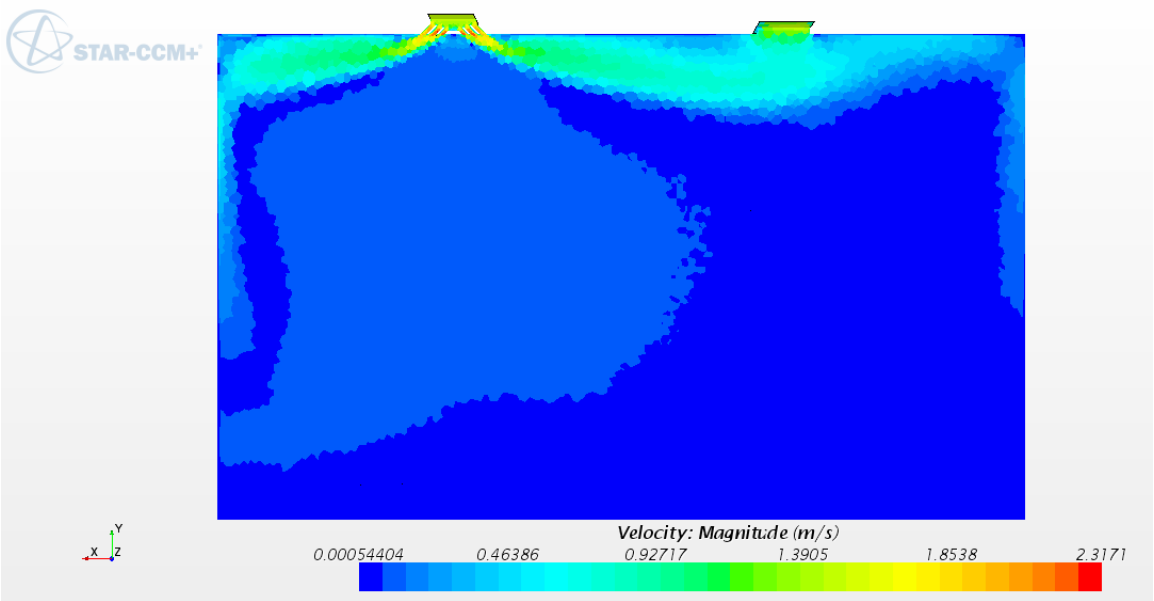

Figure 77: Two Dimensional Planar Section of the office; 60 in – 200cfm

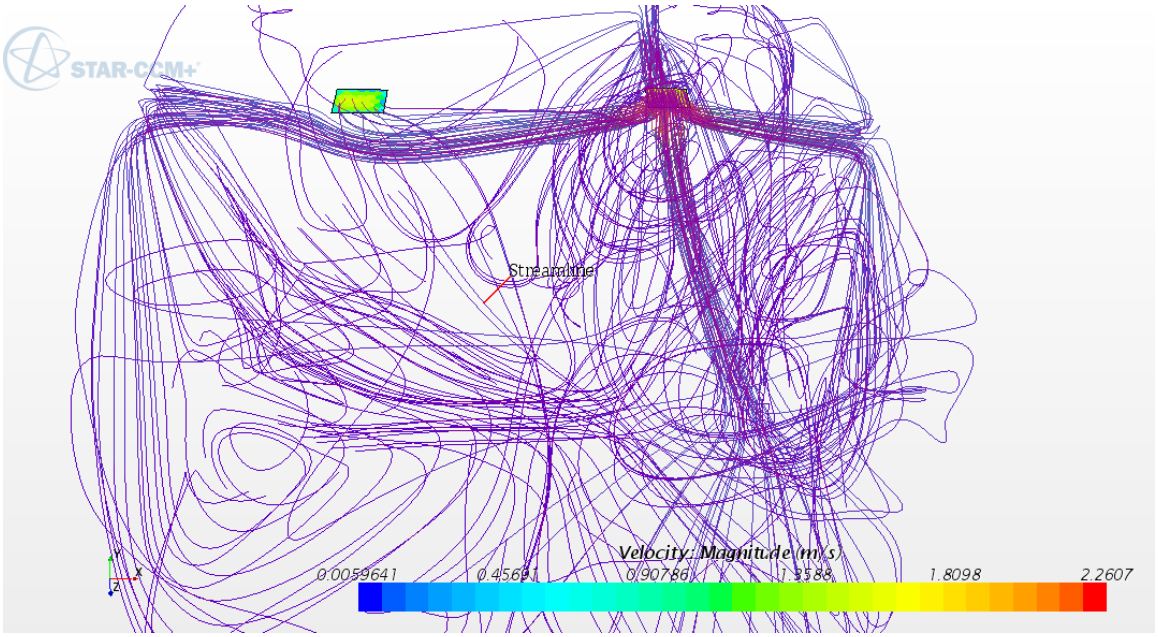

Figure 78: Three dimensional model of air circulation in the zone; 60 in – 200cfm

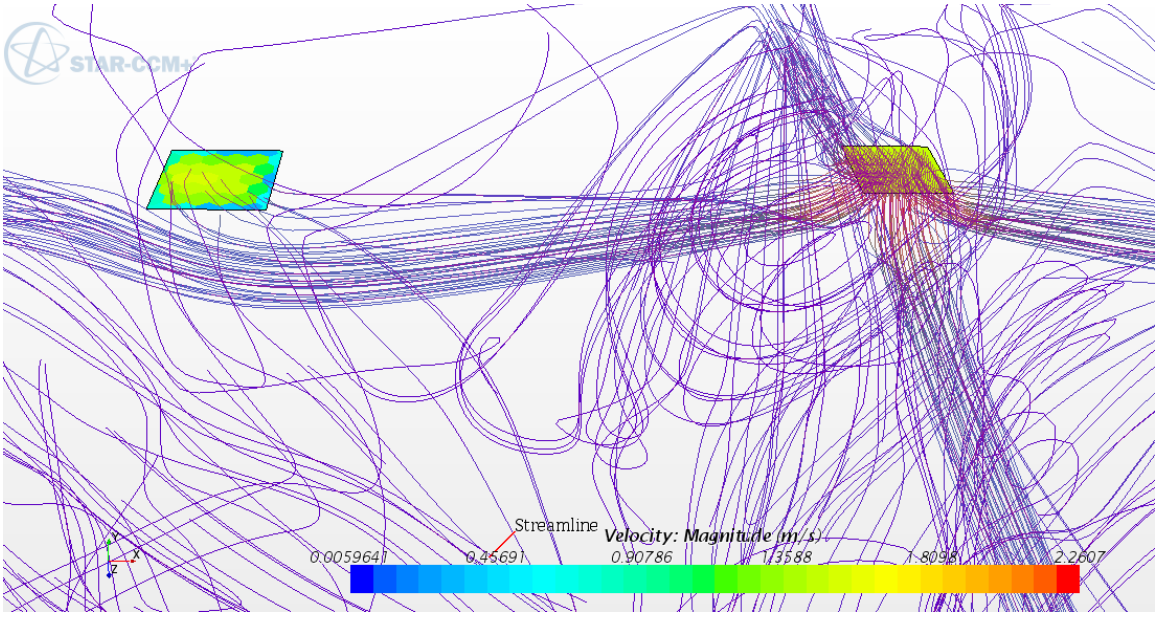

Figure 79: Three dimensional model of air circulation in the zone; 60 in – 200cfm

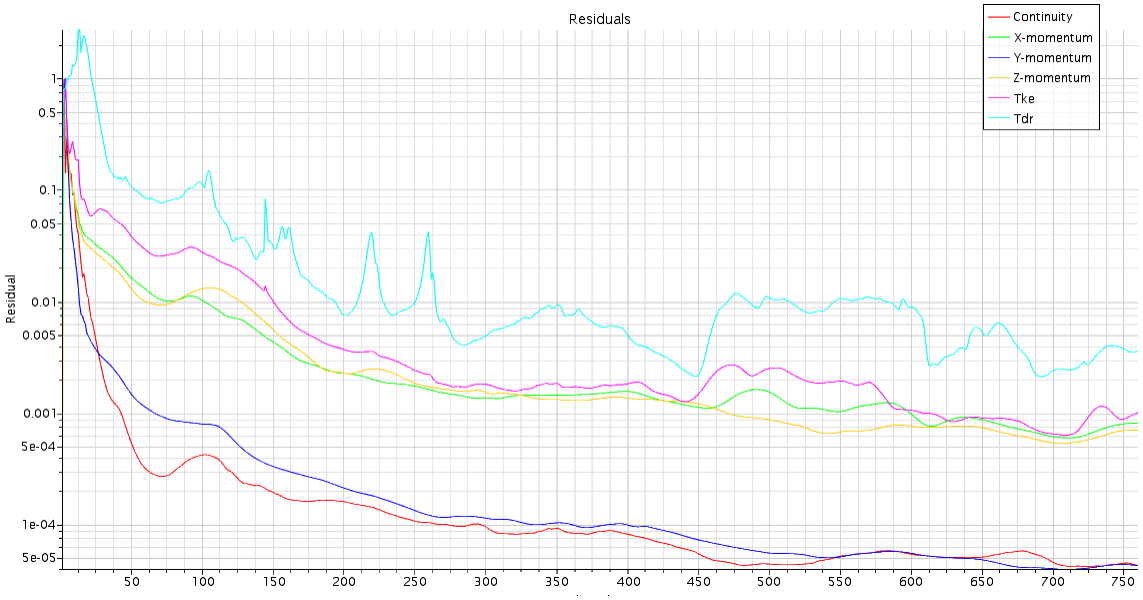

Figure 80: Plot Residuals; 60 in – 200cfm

Square Diffuser: 60 inches away from the return; 350 cfm

The results obtained from figures 81, 82 and 83 show a by-pass rate of 5.6%; 11 out of the 196 stream lines by-passed the system.

A plot of the simulation residuals is shown in Figure 84. The plot shows that all 6 residuals were relatively converged at or below the 0.01 level which show how precise the results of the simulation are.

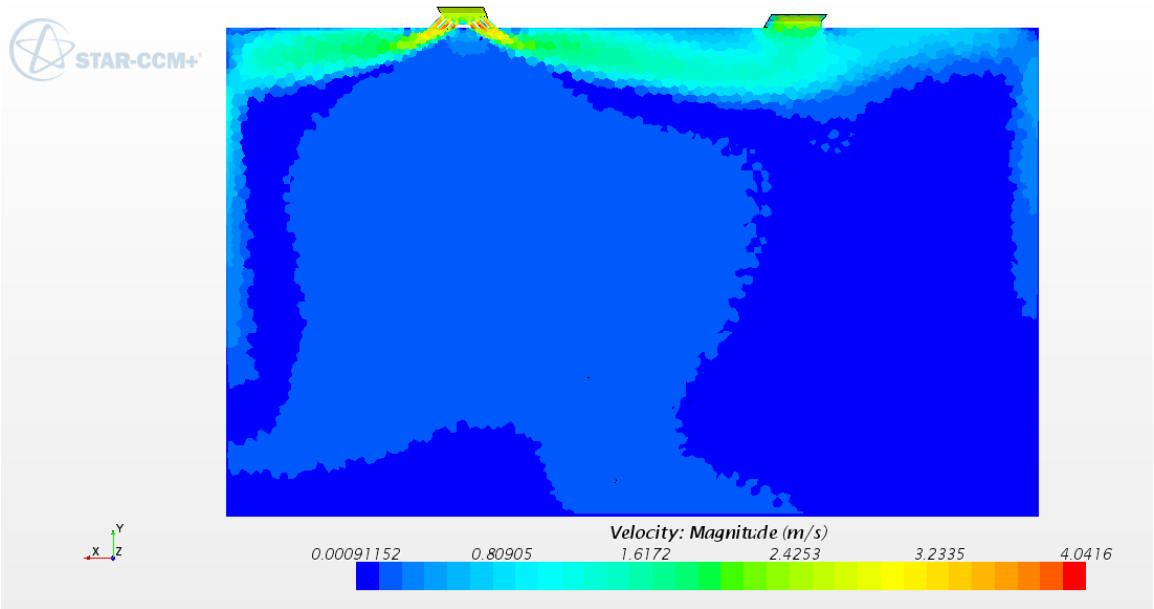

Figure 81: Two Dimensional Planar Section of the office; 60 in – 350cfm

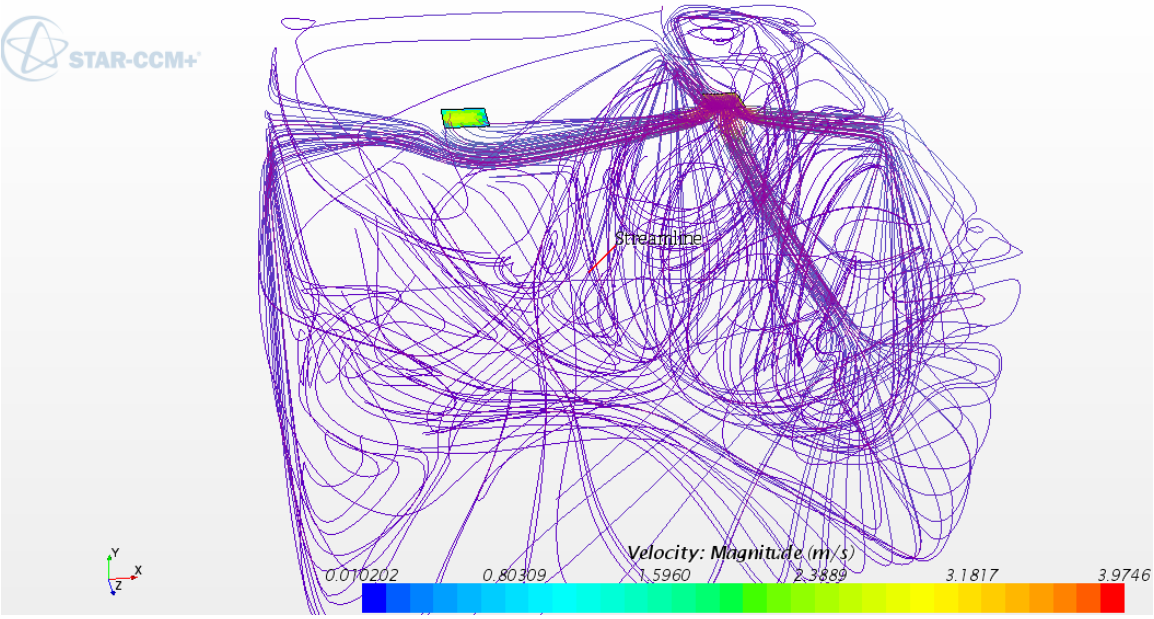

Figure 82: Three dimensional model of air circulation in the zone; 60 in – 350cfm

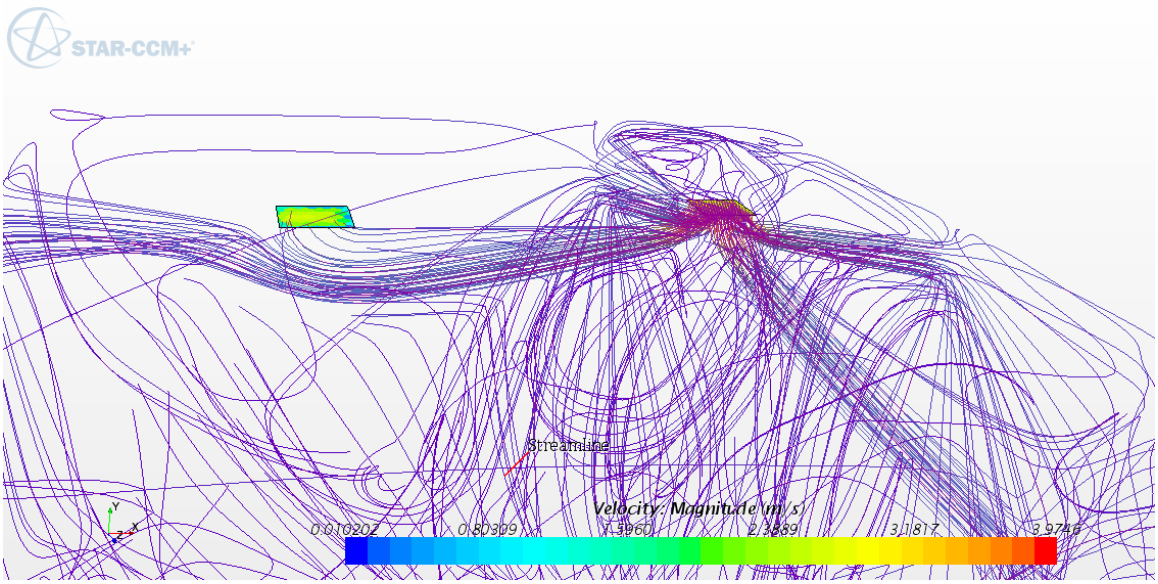

Figure 83: Three dimensional model of air circulation in the zone; 60 in – 350cfm

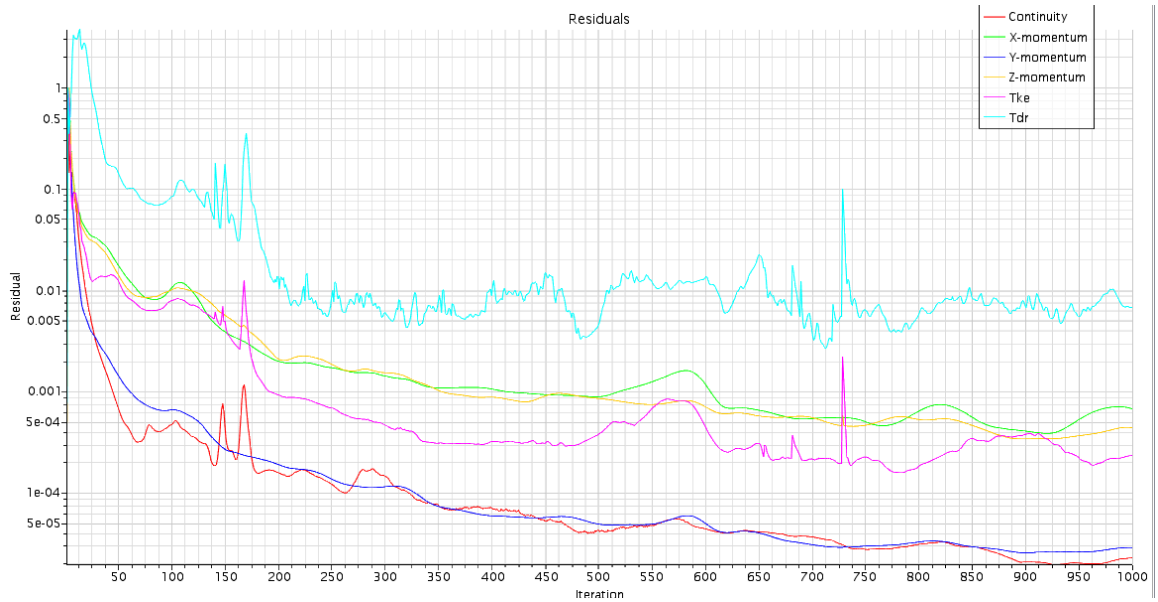

Figure 84: Plot Residuals; 60 in – 350cfm

Square Diffuser: 60 inches away from the return; 500 cfm

The results obtained from figures 85, 86 and 87 show a by-pass rate of 4.6%; 9 out of the 196 streamlines by-passed the system. 196 streamlines were used in this simulation for better accuracy that's needed at higher efficiency systems. The difference between the efficiencies of the 48 and 60 inches systems couldn't be observed when 64 stream lines were used as they both have only few lines by-passing out of the total sum.

A plot of the simulation residuals is shown in Figure 88. The plot shows that all 6 residuals were relatively converged at or below the 0.01 level which show how precise the results of the simulation are.

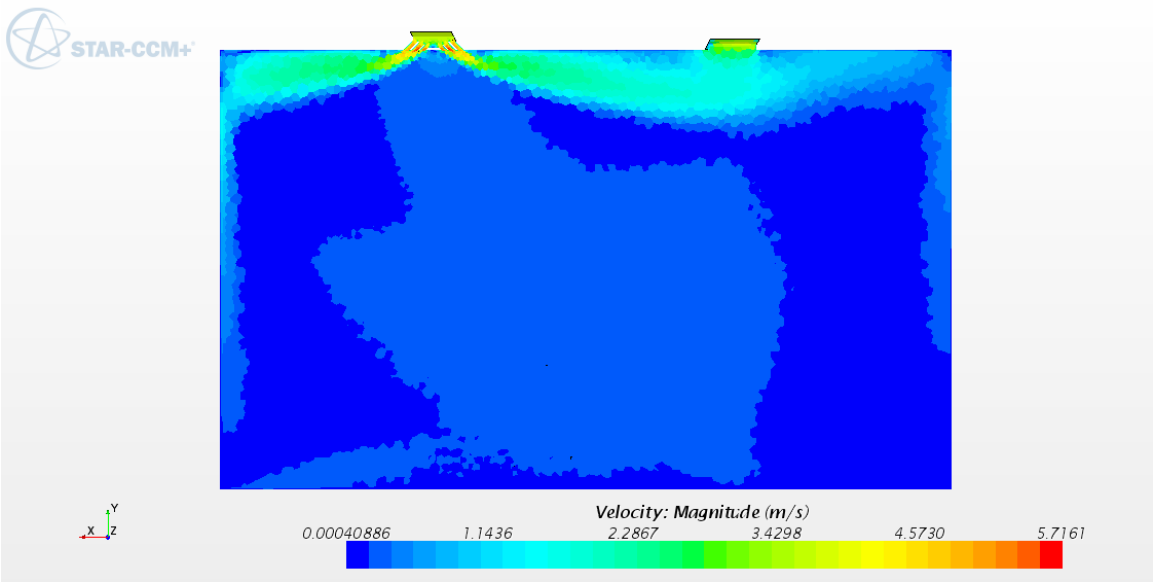

Figure 85: Two Dimensional Planar Section of the office; 60 in – 500cfm

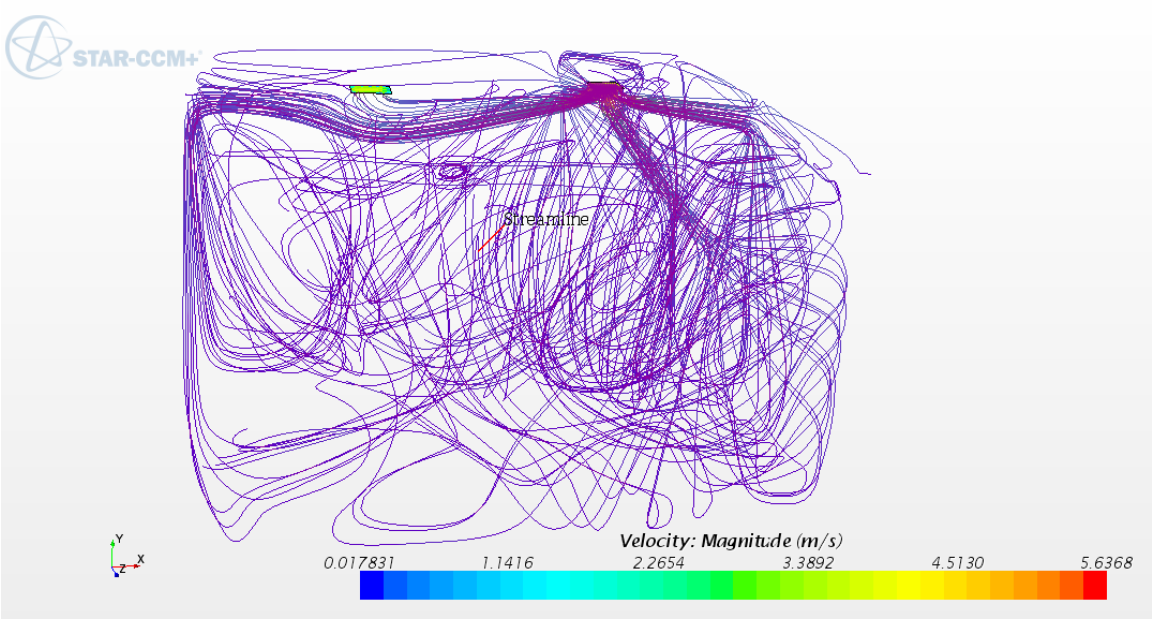

Figure 86: Three dimensional model of air circulation in the zone; 60 in – 500cfm

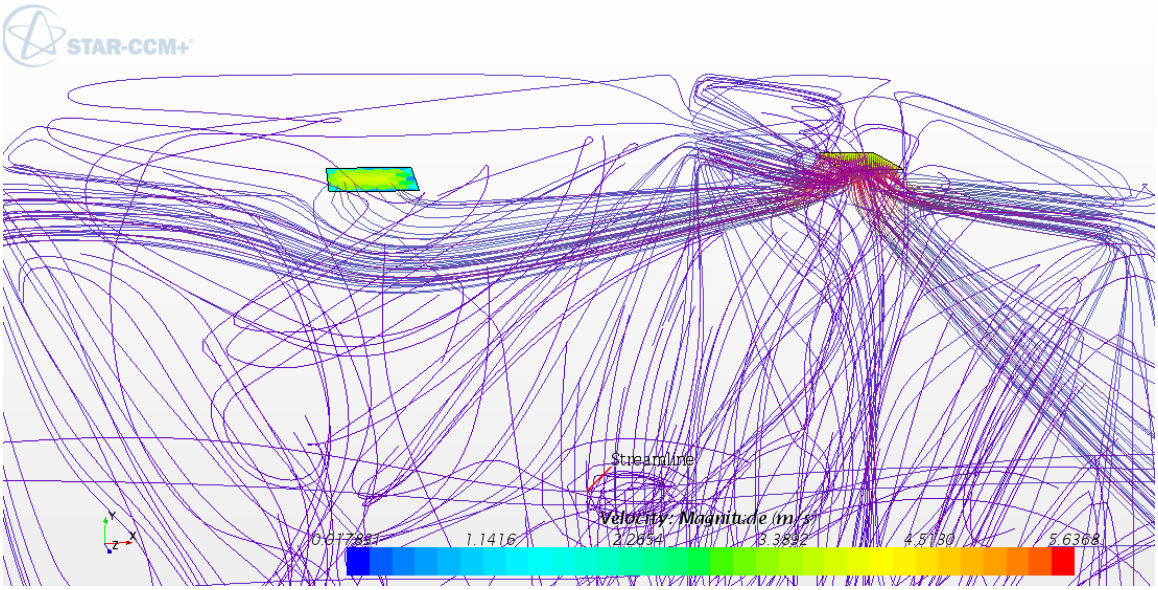

Figure 87: Three dimensional model of air circulation in the zone; 60 in – 500cfm

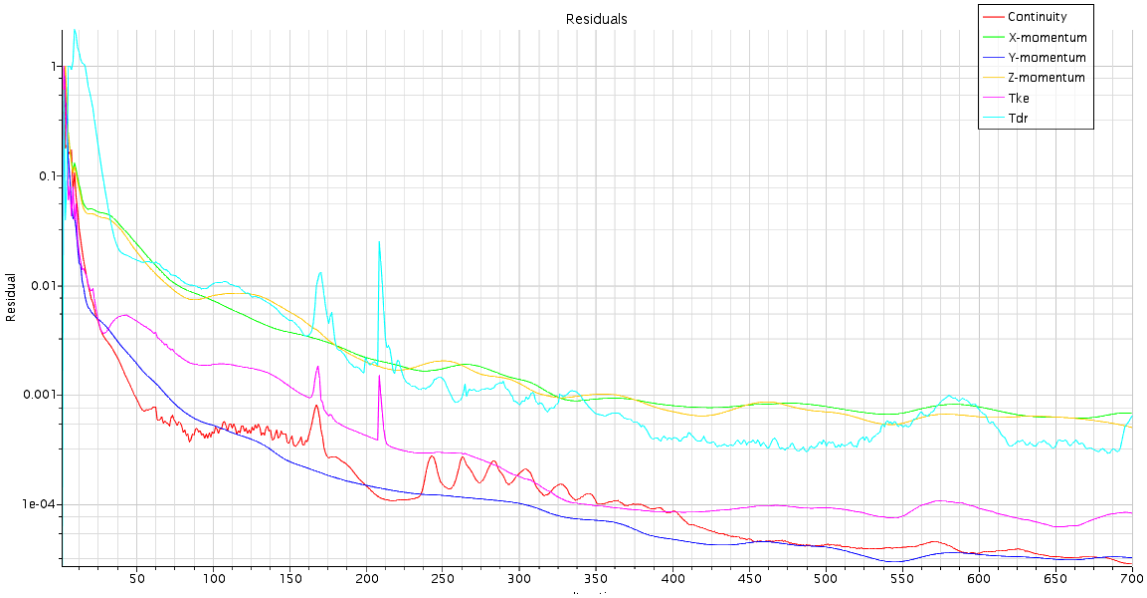

Figure 88: Plot Residuals; 60 in – 500cfm

Part Two: Round Diffusers

The simulations that were run using round diffusers at different supply to return distances and different flow rates proved that round diffusers are more efficient than square diffusers when placed 60 inches or less away from the return grille. Air by-pass rates are lower using round diffusers in every case compared to the square diffuser at the same set parameters.

Round Diffuser: 12 inches away from the return; 200 cfm

As can be seen in Figure 89, the air that's diffused from the right side of the diffuser is not circulating past the return grille which means that it is by-passing the zone; it is going directly back into the return grille. Figures 90 and 91 show the three dimensional streamlines that represent the actual circulation of air; the figures show that approximately 8 of the 64 stream lines, 12.5%, that were diffused in the room by-passed the system and did not circulate in the room.

A plot of the simulation residuals is shown in Figure 92. The plot shows that all 6 residuals were relatively converged at or below the 0.005 level which show how precise the results of the simulation are; residuals are the difference between the iterated value from the software and the exact solution.

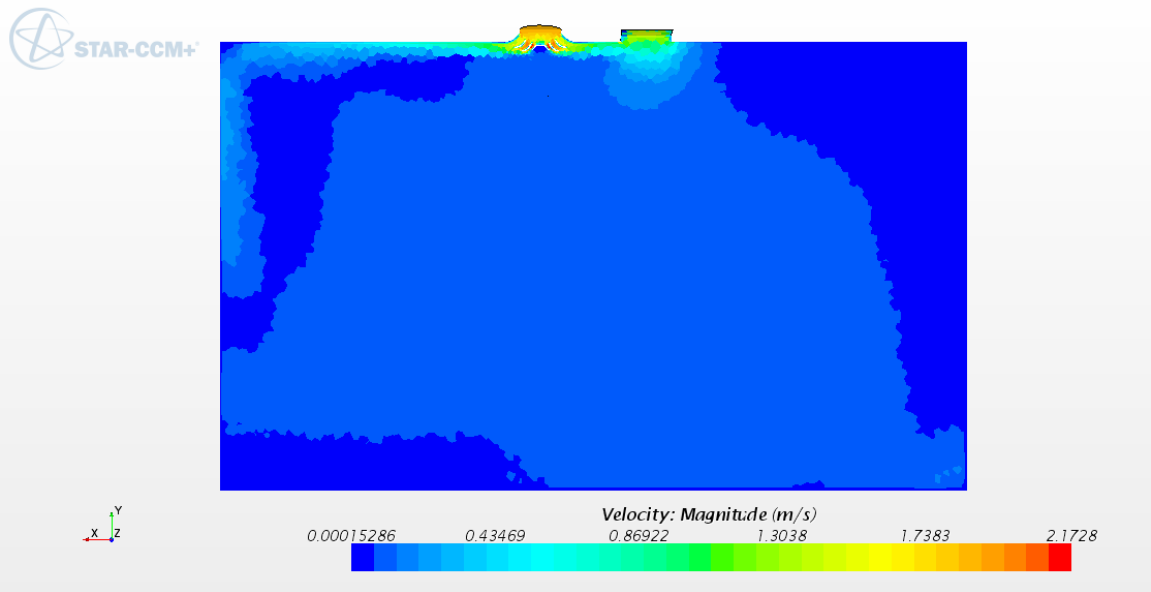

Figure 89: Two Dimensional Planar Section of the office; 12 in – 200cfm

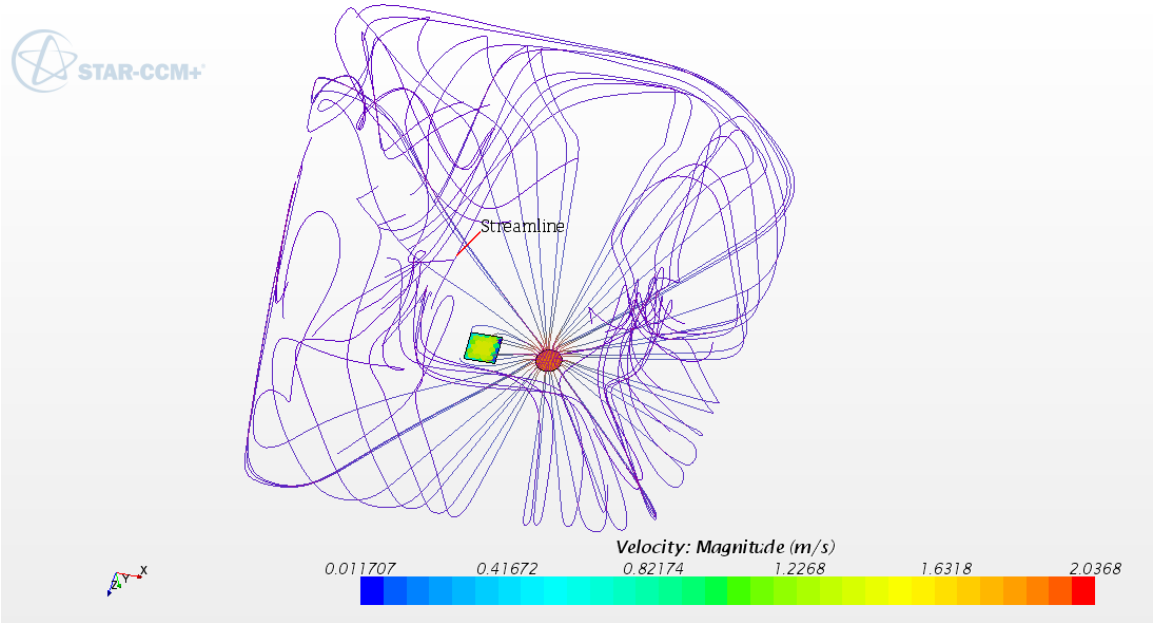

Figure 90: Three dimensional model of air circulation in the zone; 12 in – 200cfm

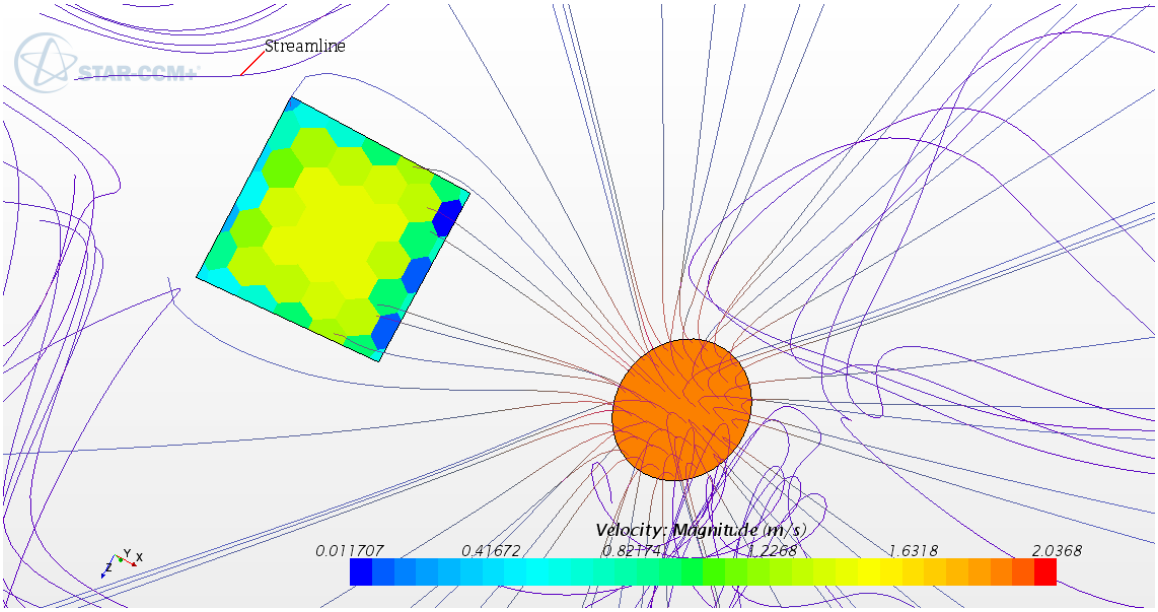

Figure 91: Three dimensional model of air circulation in the zone; 12 in – 200cfm

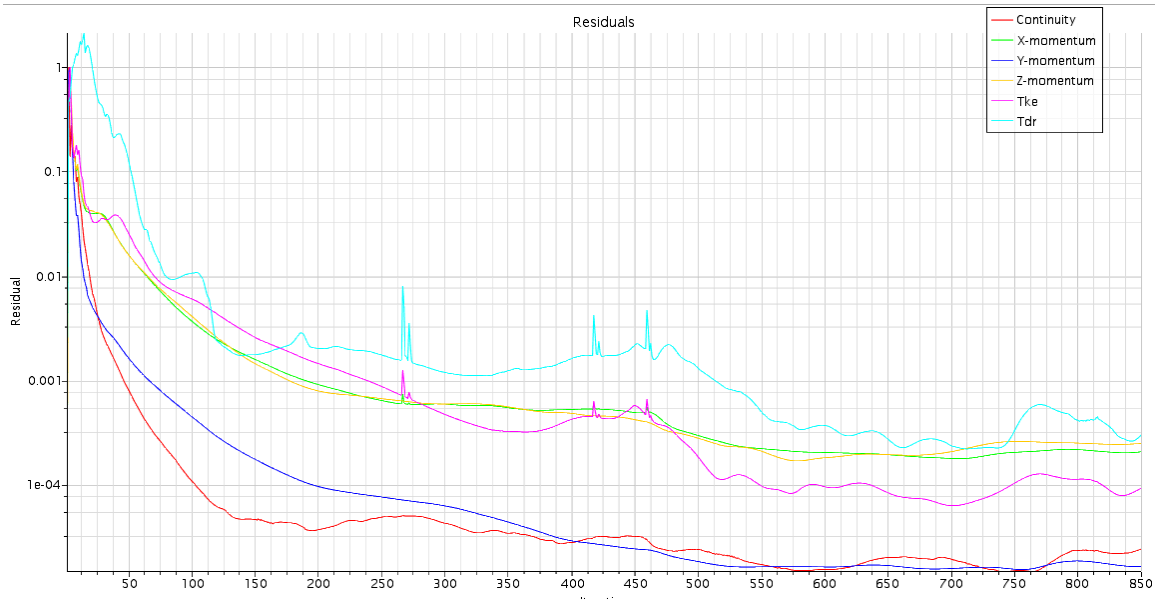

Figure 92: Plot Residuals; 12 in – 200cfm

Round Diffuser: 12 inches away from the return; 350 cfm

At supply to return distance of 12 inches and 350 cfm, figures 93, 94 and 95 below show a by-pass rate of 10.9%.

A plot of the simulation residuals is shown in Figure 96. The plot shows that all 6 residuals were relatively converged at or below the 0.0001 level which show how precise the results of the simulation are.

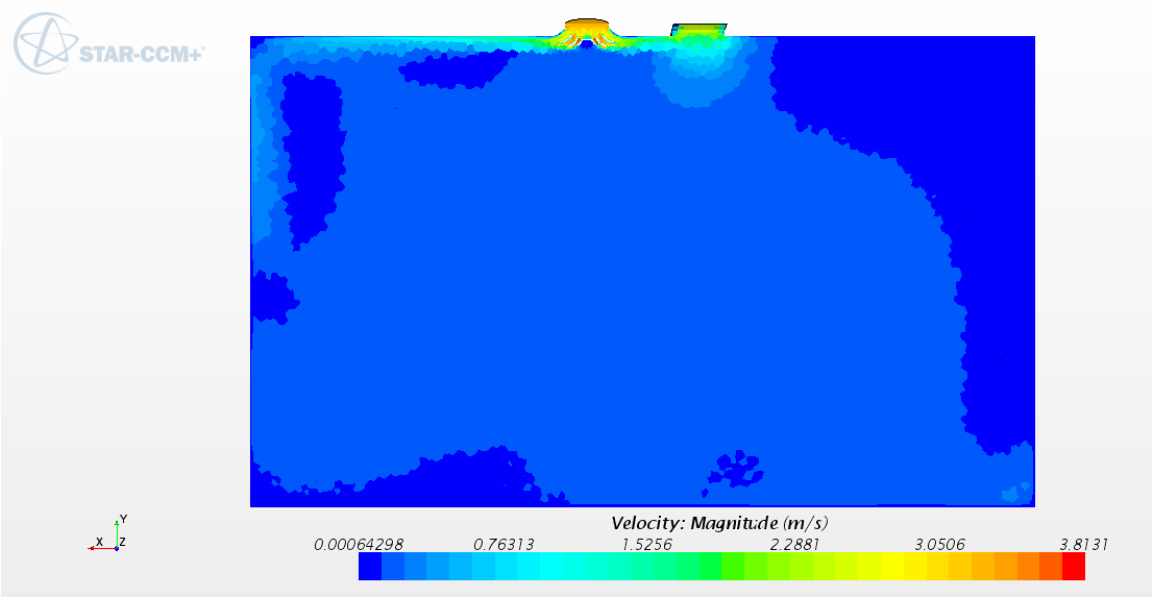

Figure 93: Two Dimensional Planar Section of the office; 12 in – 350cfm

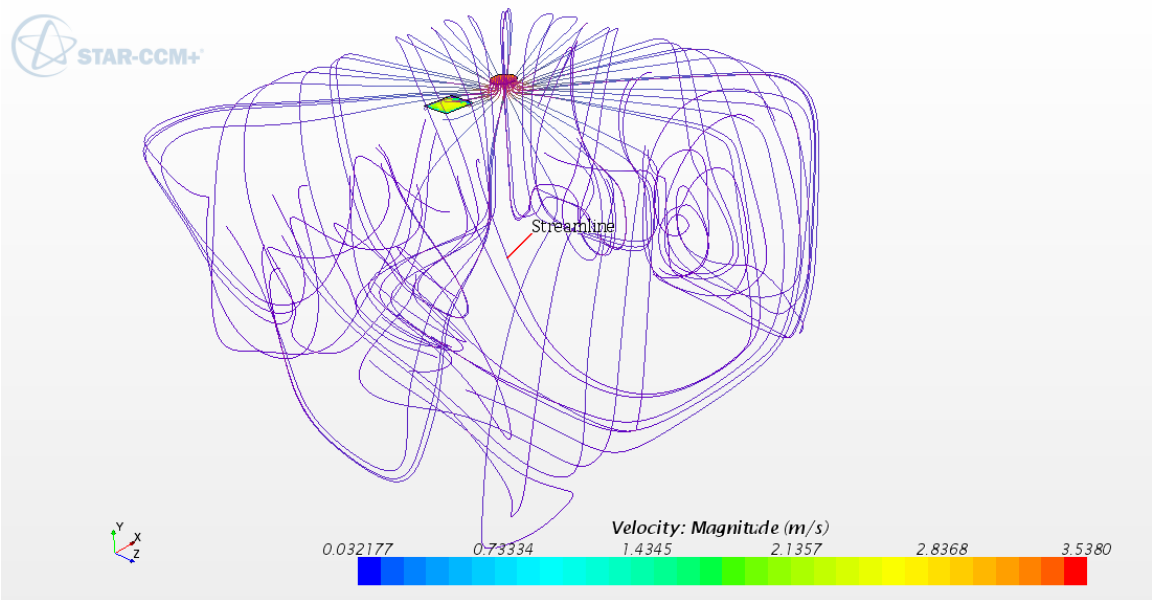

Figure 94: Three dimensional model of air circulation in the zone; 12 in – 350cfm

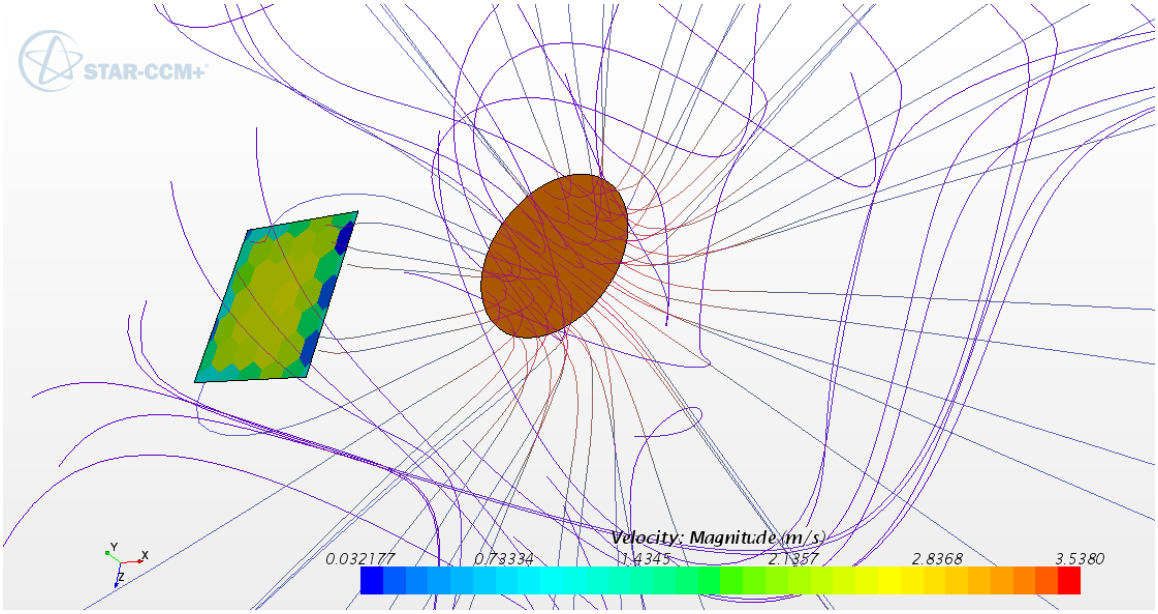

Figure 95: Three dimensional model of air circulation in the zone; 12 in – 350cfm

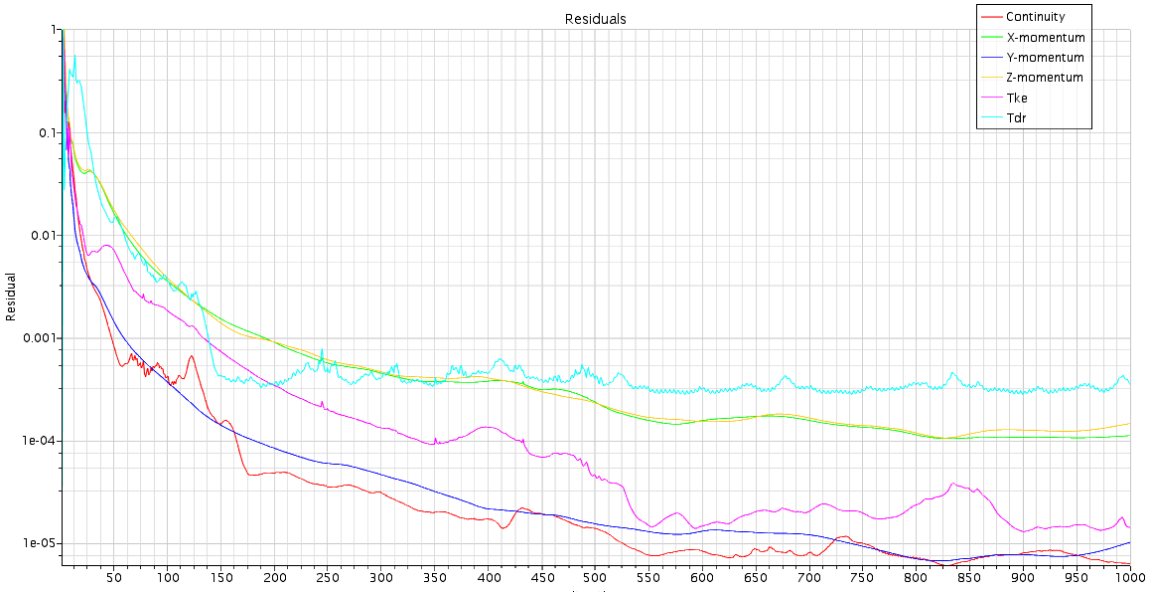

Figure 96: Plot Residuals; 12 in – 350cfm

Round Diffuser: 12 inches away from the return; 500 cfm

At supply to return distance of 12 inches and 500 cfm, figures 97, 98 and 99 below show a by-pass rate of 12.5%; this is similar to the results obtained when the flow was 200 cfm but higher than when the flow was 350 cfm.

A plot of the simulation residuals is shown in Figure 100. The plot shows that all 6 residuals were relatively converged at or below the 0.0001 level which show how precise the results of the simulation are.

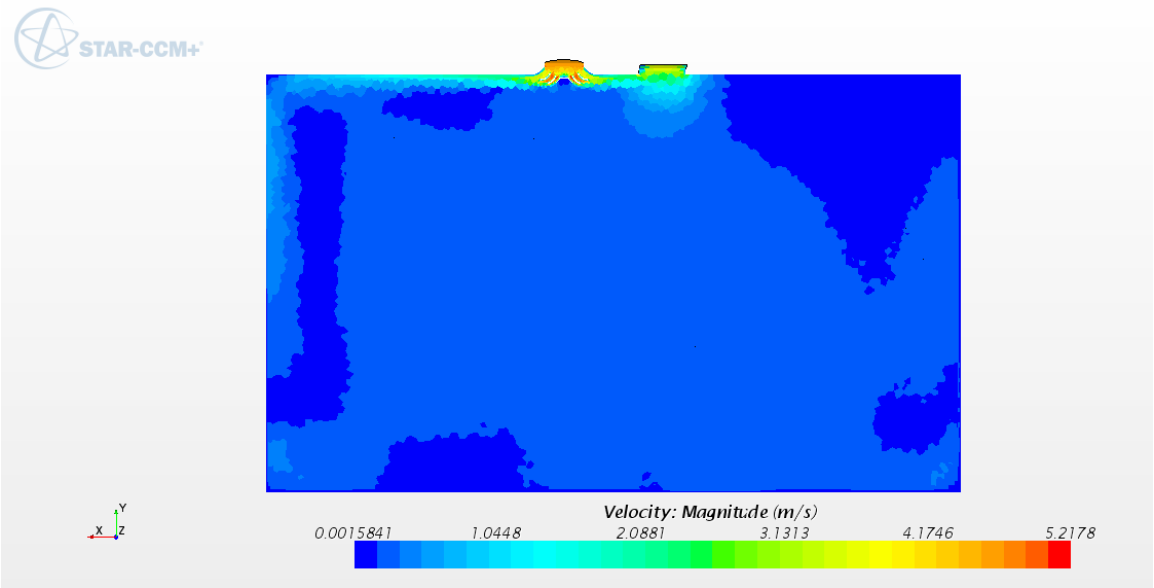

Figure 97: Two Dimensional Planar Section of the office; 12 in – 500cfm

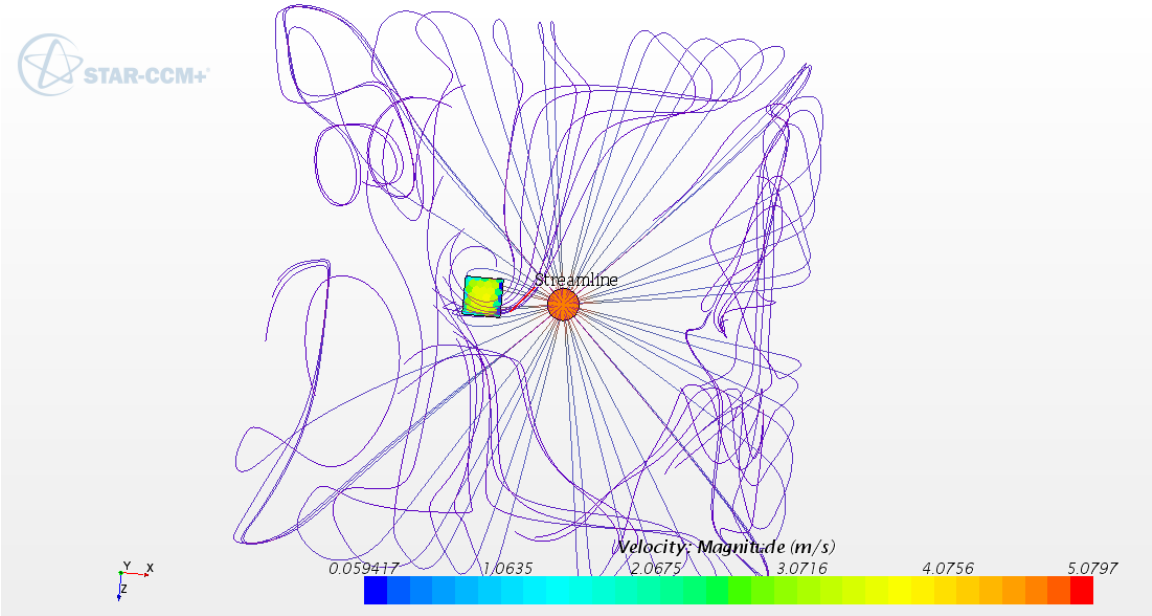

Figure 98: Three dimensional model of air circulation in the zone; 12 in – 500cfm

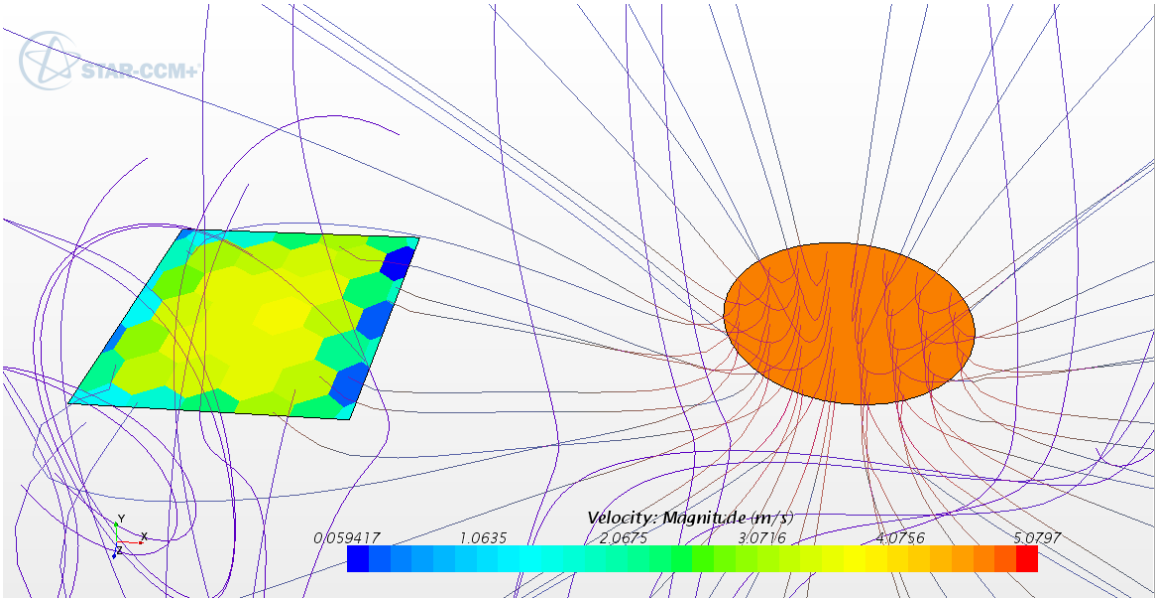

Figure 99: Three dimensional model of air circulation in the zone; 12 in – 500cfm

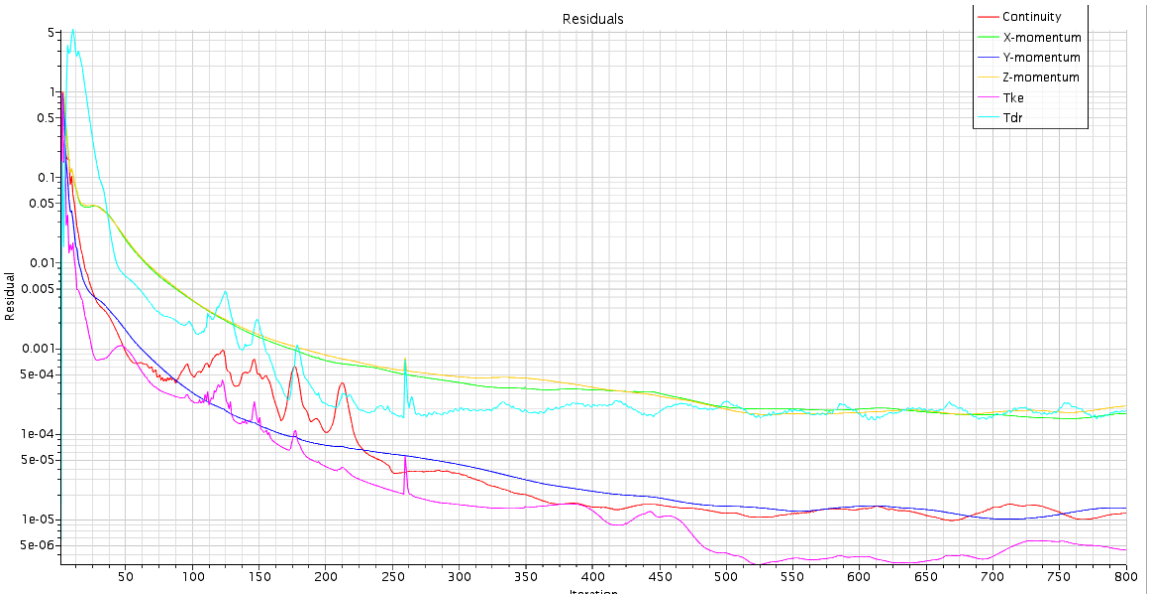

Figure 100: Plot Residuals; 12 in – 500cfm

Round Diffuser: 24 inches away from the return; 200 cfm

At supply to return distance of 24 inches and 200 cfm, figures 101, 102 and 103 below show a by-pass rate of 9.4% as 6 out of 64 lines by-passed the system; this meets the expectations as the by-pass rate decreased when the distance between supply and return apertures increased.

A plot of the simulation residuals is shown in Figure 104. The plot shows that all 6 residuals were relatively converged at or below the 0.0001 level which show how precise the results of the simulation are.

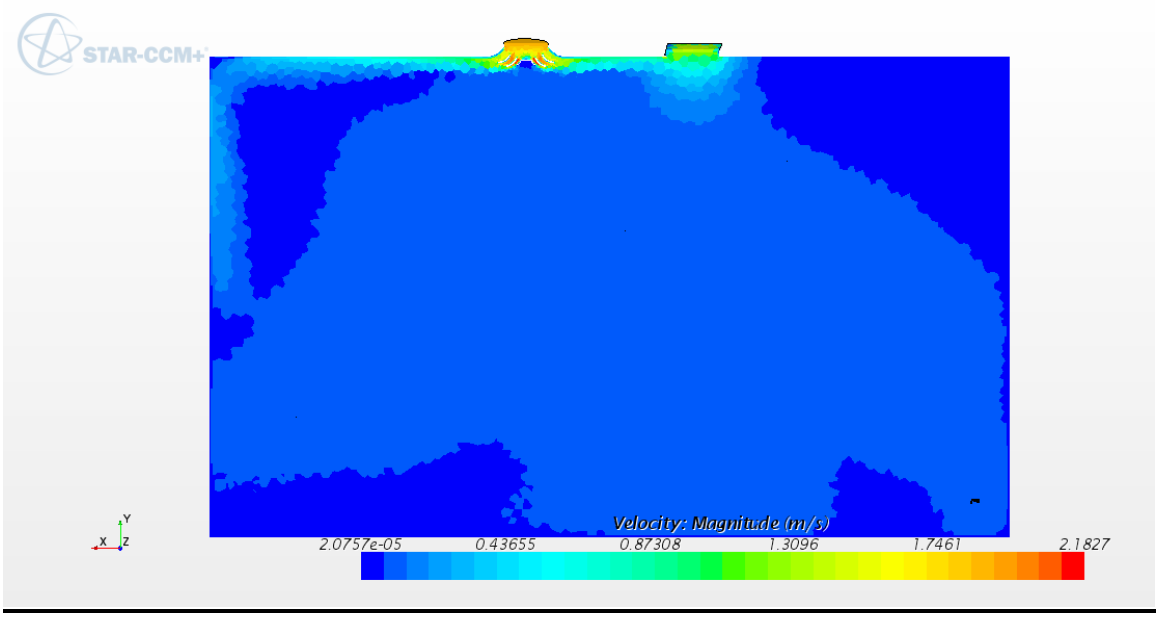

Figure 101: Two Dimensional Planar Section of the office; 24 in – 200cfm

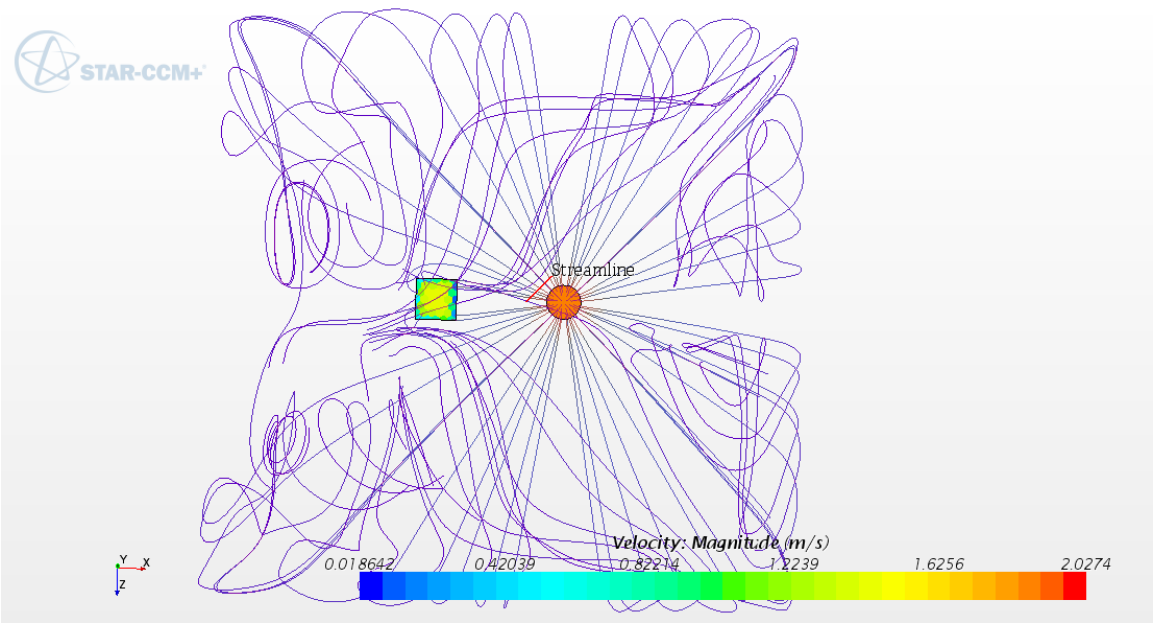

Figure 102: Three dimensional model of air circulation in the zone; 24 in – 200cfm

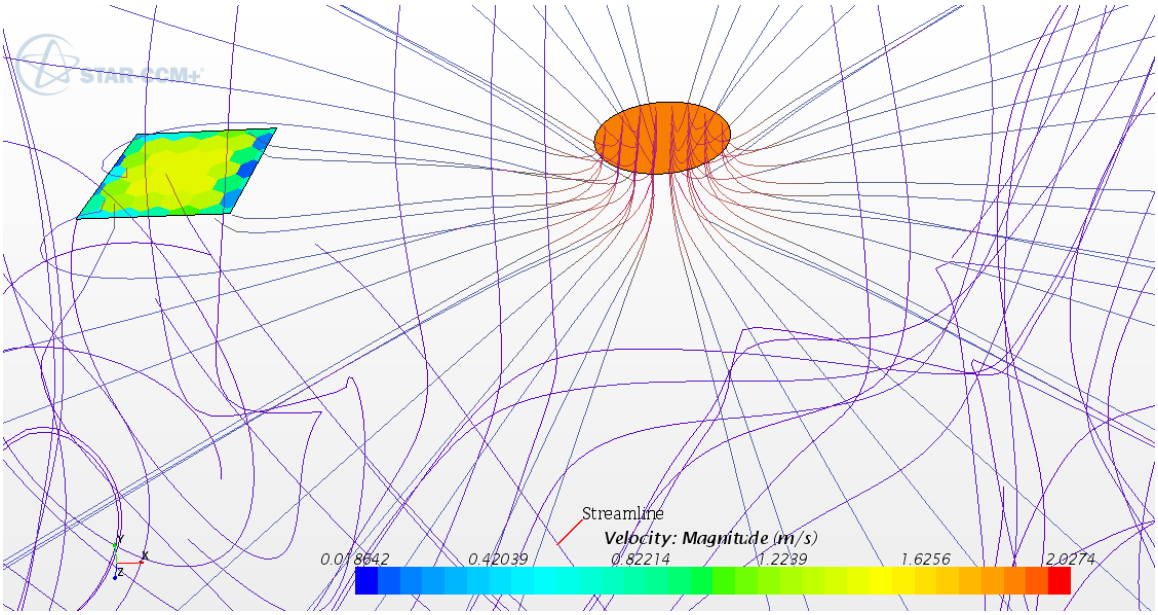

Figure 103: Three dimensional model of air circulation in the zone; 24 in – 200cfm

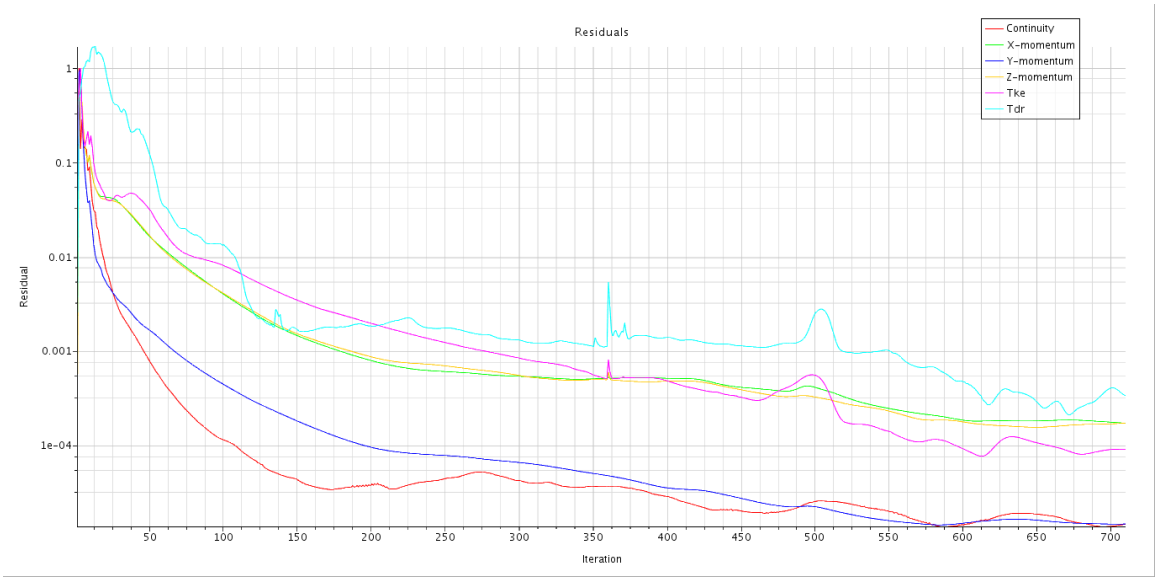

Figure 104: Plot Residuals; 24 in – 200cfm

Round Diffuser: 24 inches away from the return; 350 cfm

At supply to return distance of 24 inches and 350 cfm, figures 105, 106 and 107 below show a by-pass rate of 9.4% as 6 out of 64 lines by-passed the system; the results of this simulation is similar to the simulation of 24 inches and 200 cfm. The effect of air flow rate on the efficiency of HVAC systems with round diffuser seem to be lower than that of systems with square diffusers.

A plot of the simulation residuals is shown in Figure 108. The plot shows that all 6 residuals were relatively converged at or below the 0.005 level which show how precise the results of the simulation are.

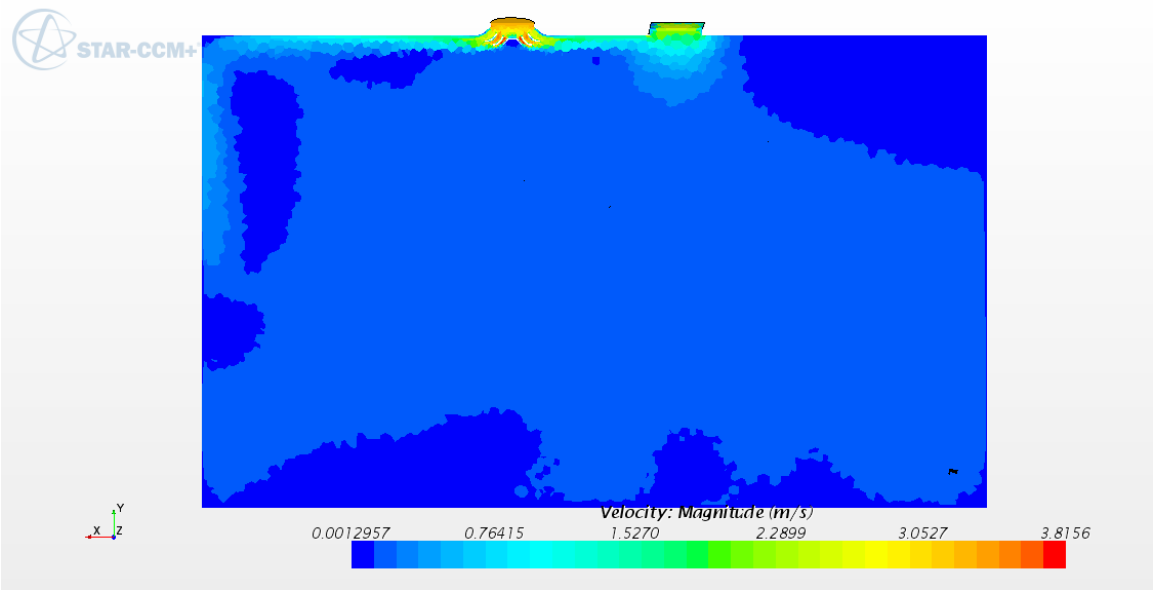

Figure 105: Two Dimensional Planar Section of the office; 24 in – 350cfm

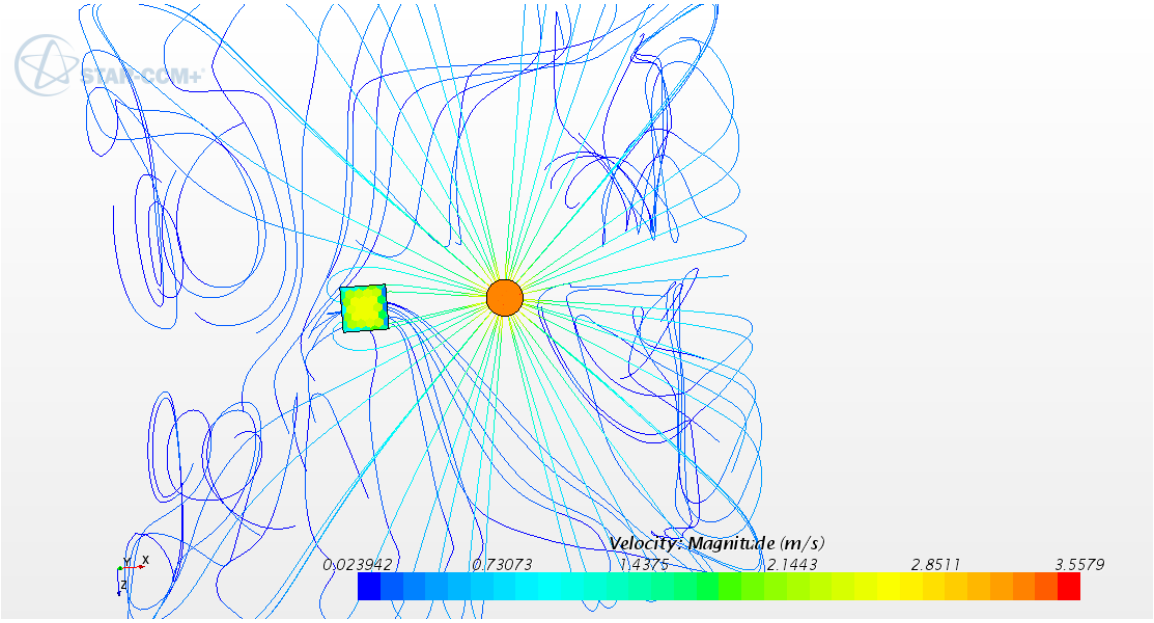

Figure 106: Three dimensional model of air circulation in the zone; 24 in – 350cfm

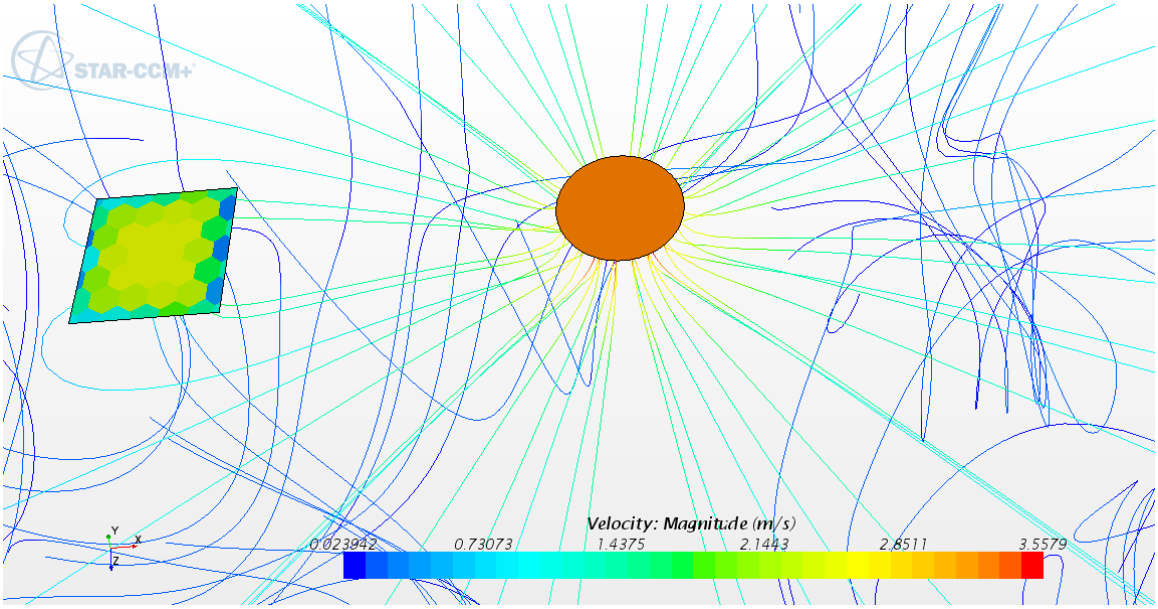

Figure 107: Three dimensional model of air circulation in the zone; 24 in – 350cfm

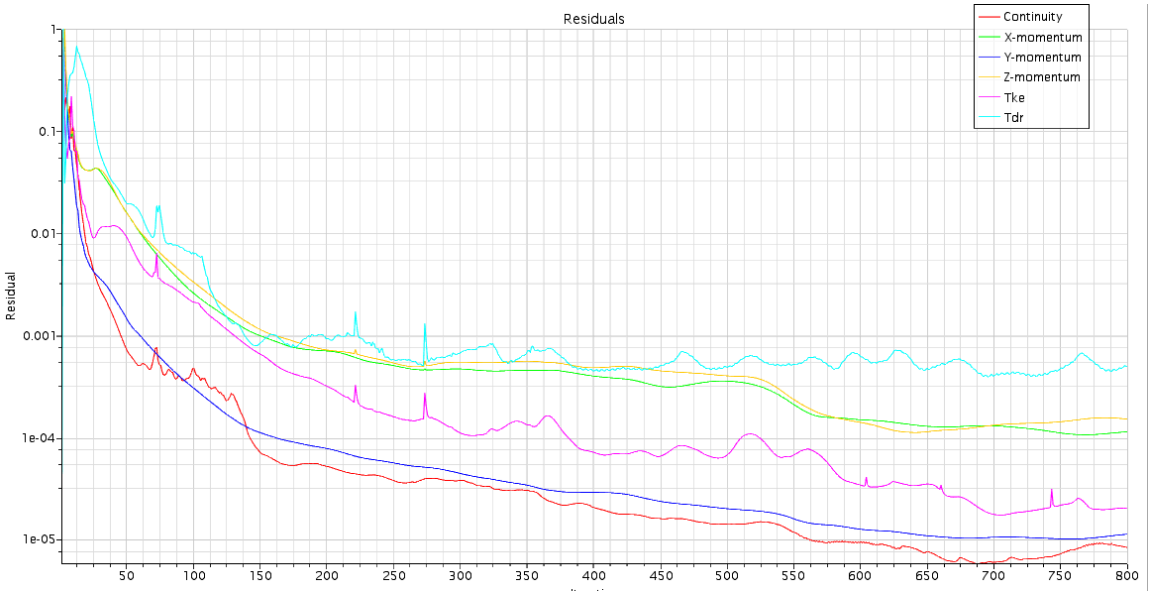

Figure 108: Plot Residuals; 24 in – 350cfm

Round Diffuser: 24 inches away from the return; 500 cfm

At supply to return distance of 24 inches and 500 cfm, figures 109, 110 and 111 below show a by-pass rate of 7.8% as 5 out of 64 lines by-passed the system; By-passing at 500 cfm seem to be lower than that at 200 and 350 cfm.

A plot of the simulation residuals is shown in Figure 112. The plot shows that all 6 residuals were relatively converged at or below the 0.0001 level which show how precise the results of the simulation are.

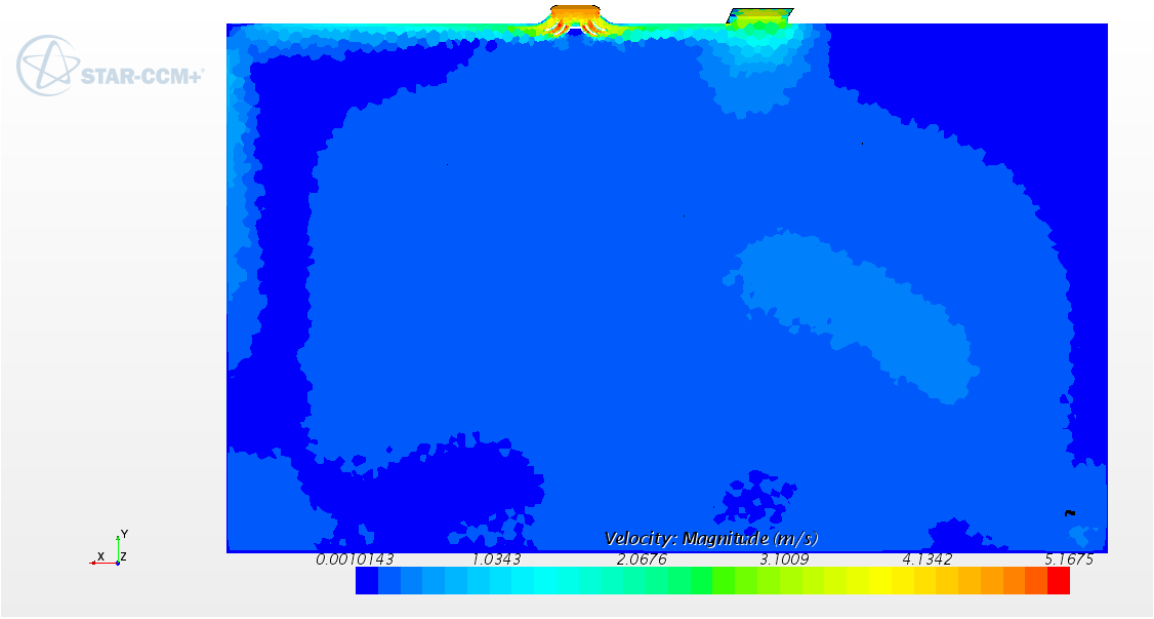

Figure 109: Two Dimensional Planar Section of the office; 24 in – 500cfm

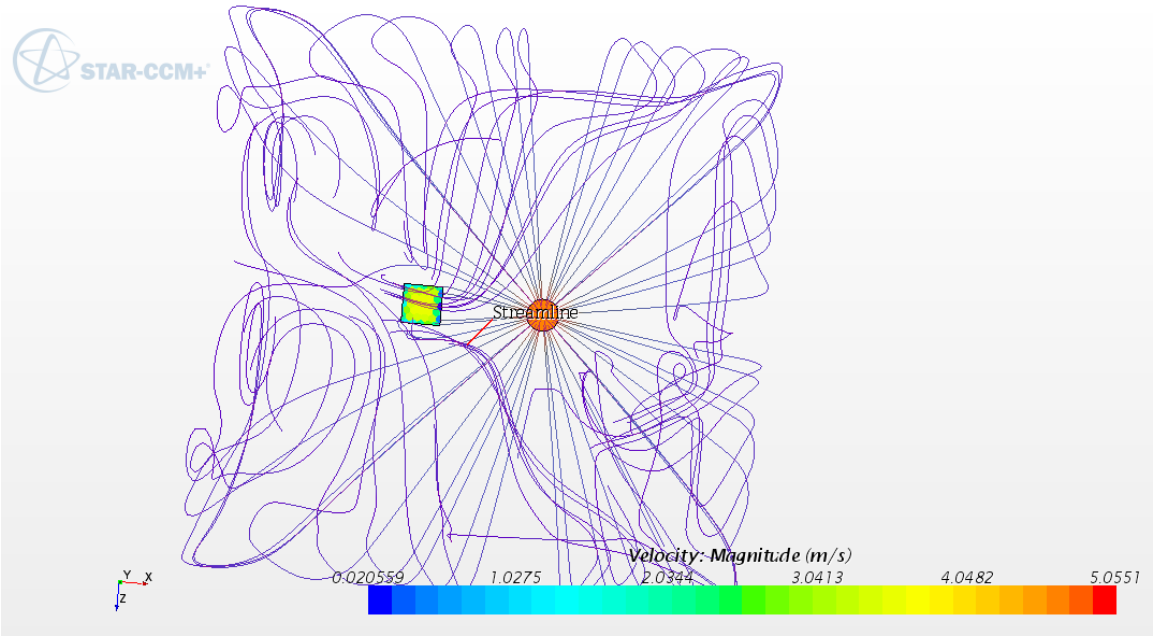

Figure 110: Three dimensional model of air circulation in the zone; 24 in –500cfm

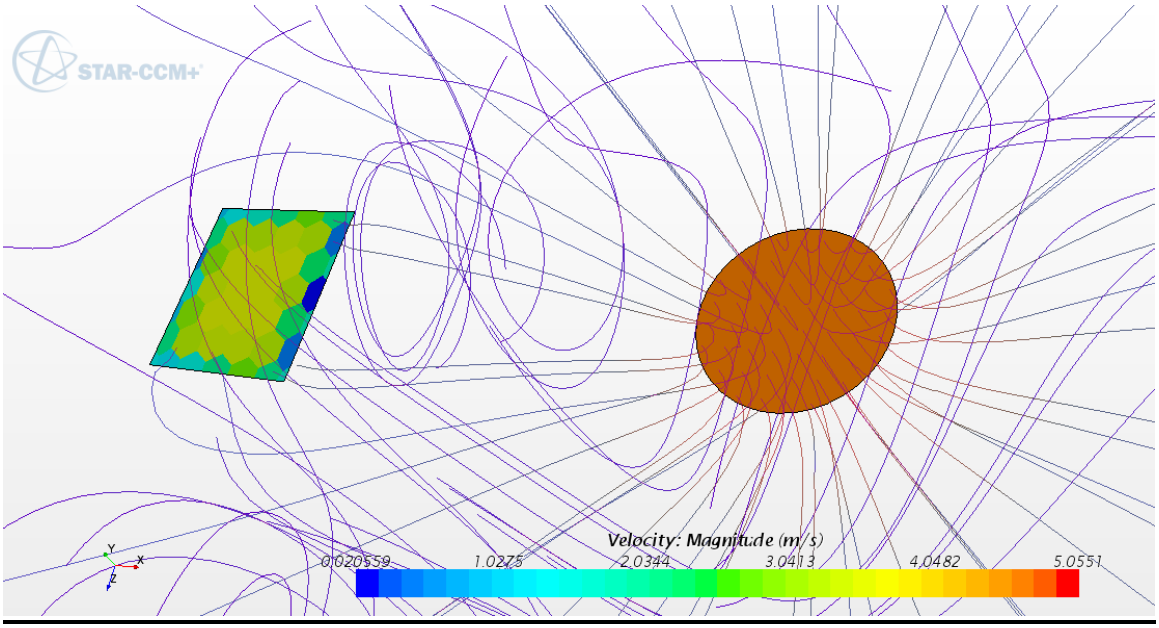

Figure 111: Three dimensional model of air circulation in the zone; 24 in – 500cfm

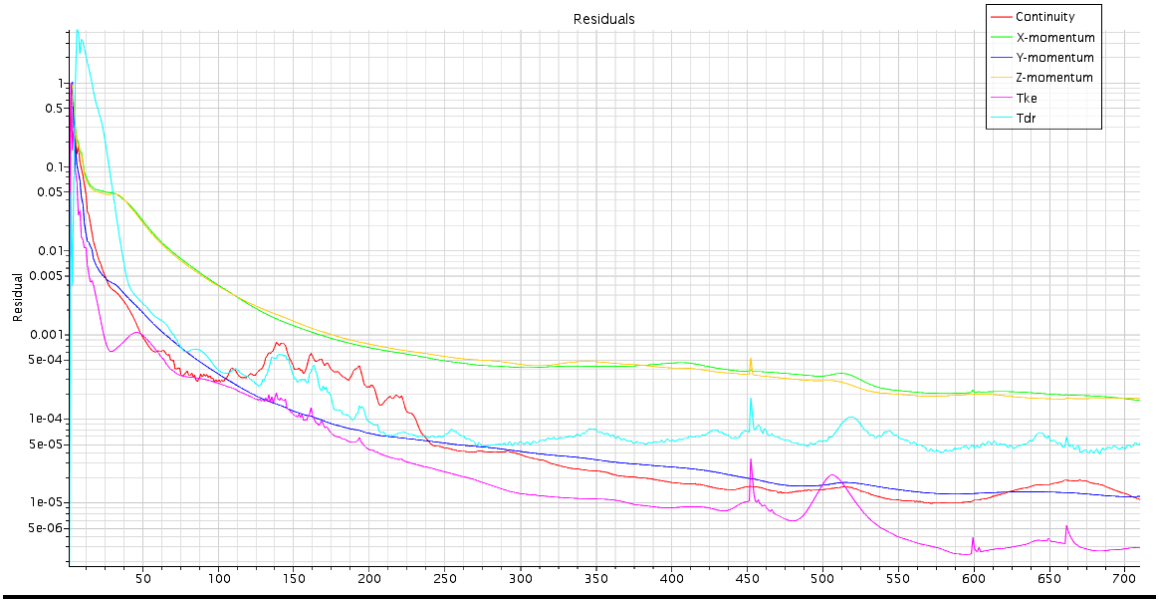

Figure 112: Plot Residuals; 24 in – 500cfm

Round Diffuser: 36 inches away from the return; 200 cfm

At supply to return distance of 36 inches and 200 cfm, figures 113, 114 and 115 below show a by-pass rate of 7.5% as 5 out of 64 lines by-passed the system; this meets the expectations as the by-pass rate decreased when the distance between supply and return increased.

A plot of the simulation residuals is shown in Figure 116. The plot shows that all 6 residuals were relatively converged at or below the 0.002 level which show how precise the results of the simulation are.

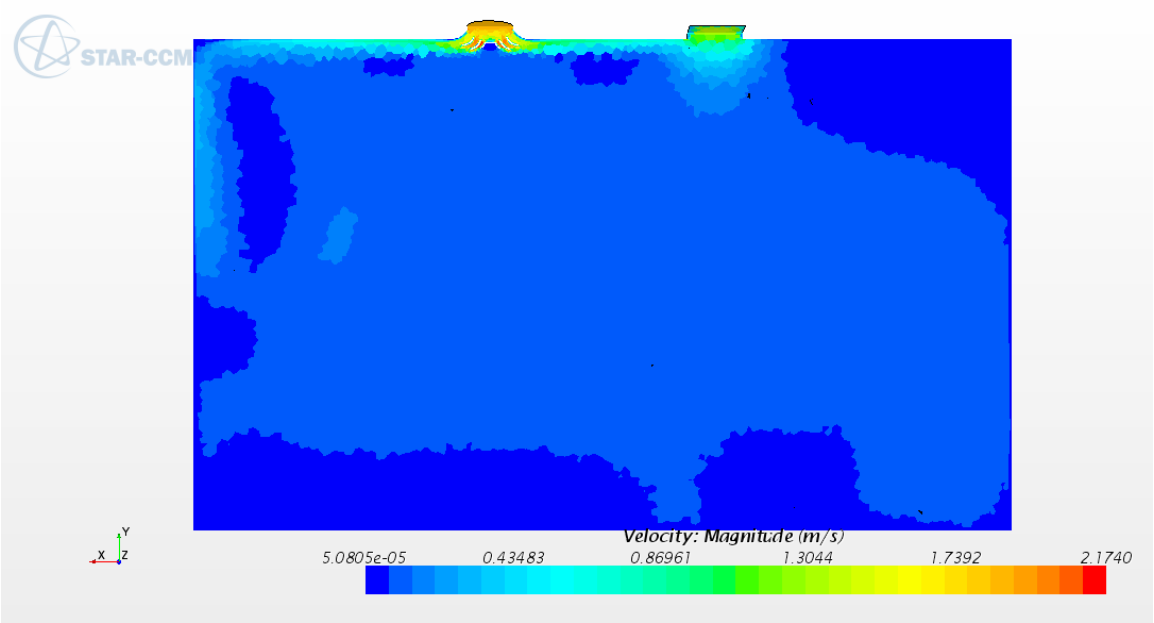

Figure 113: Two Dimensional Planar Section of the office; 36in – 200cfm

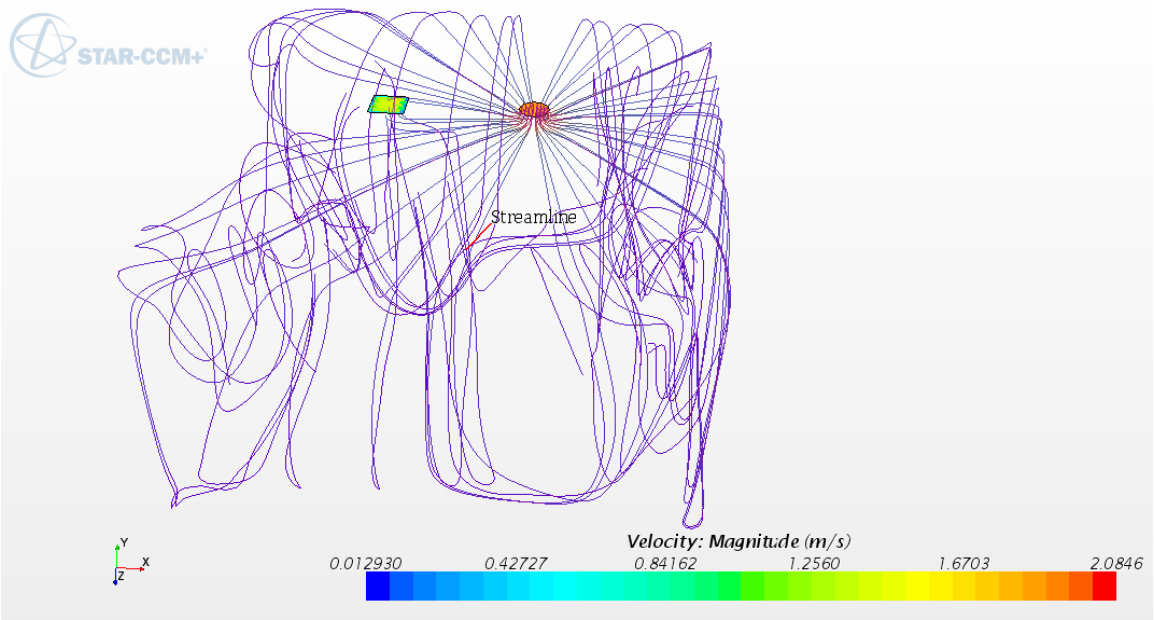

Figure 114: Three dimensional model of air circulation in the zone; 36 in – 200cfm

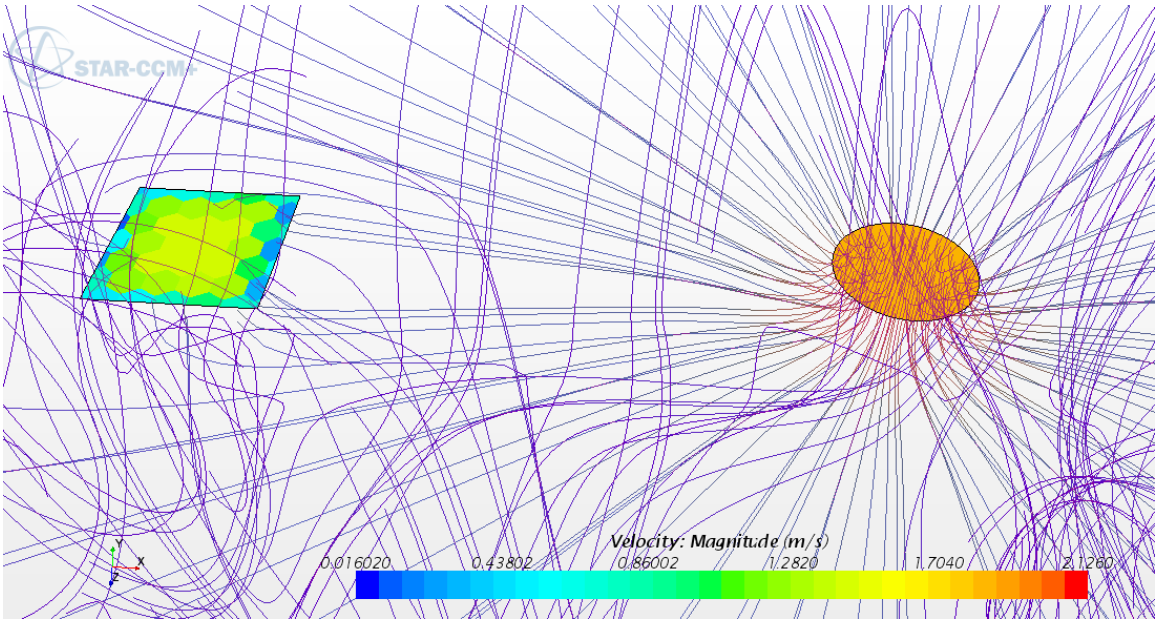

Figure 115: Three dimensional model of air circulation in the zone; 36 in – 200cfm

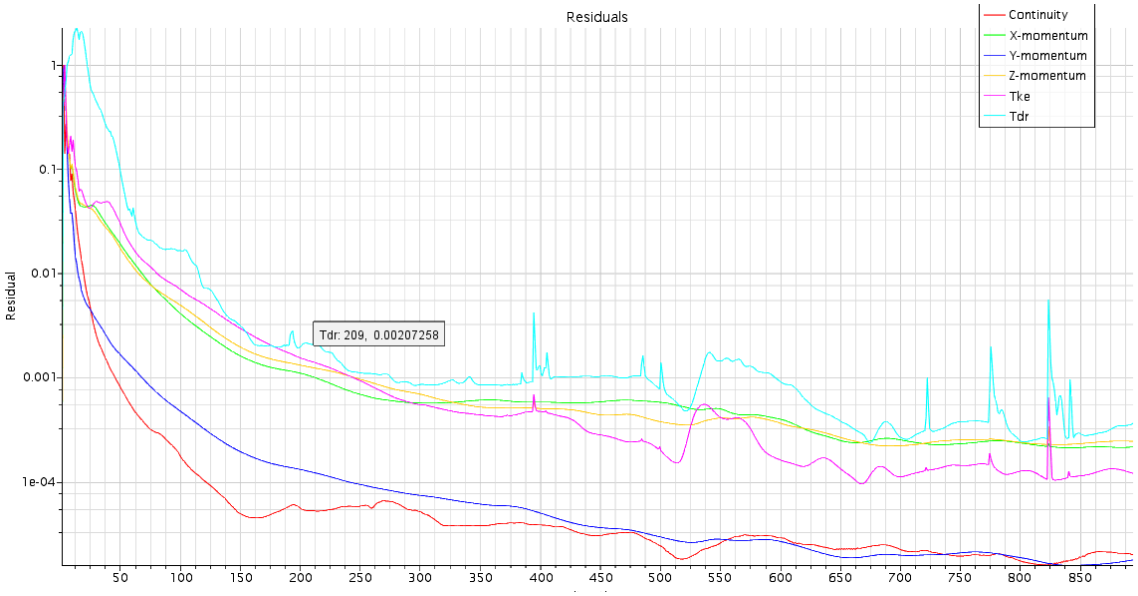

Figure 116: Plot Residuals; 36 in – 200cfm

Round Diffuser: 36 inches away from the return; 350 cfm

At supply to return distance of 36 inches and 350 cfm, figures 117, 118 and 119 below show a by-pass rate of 7.5% as 5 out of 64 lines by-passed the system; the results of this simulation is similar to the simulation at 200 cfm. The effect of air flow rate on the efficiency of HVAC systems with round diffuser seem to be lower than that of systems with square diffusers.

A plot of the simulation residuals is shown in Figure 120. The plot shows that all 6 residuals were relatively converged at or below the 0.005 level which show how precise the results of the simulation are.

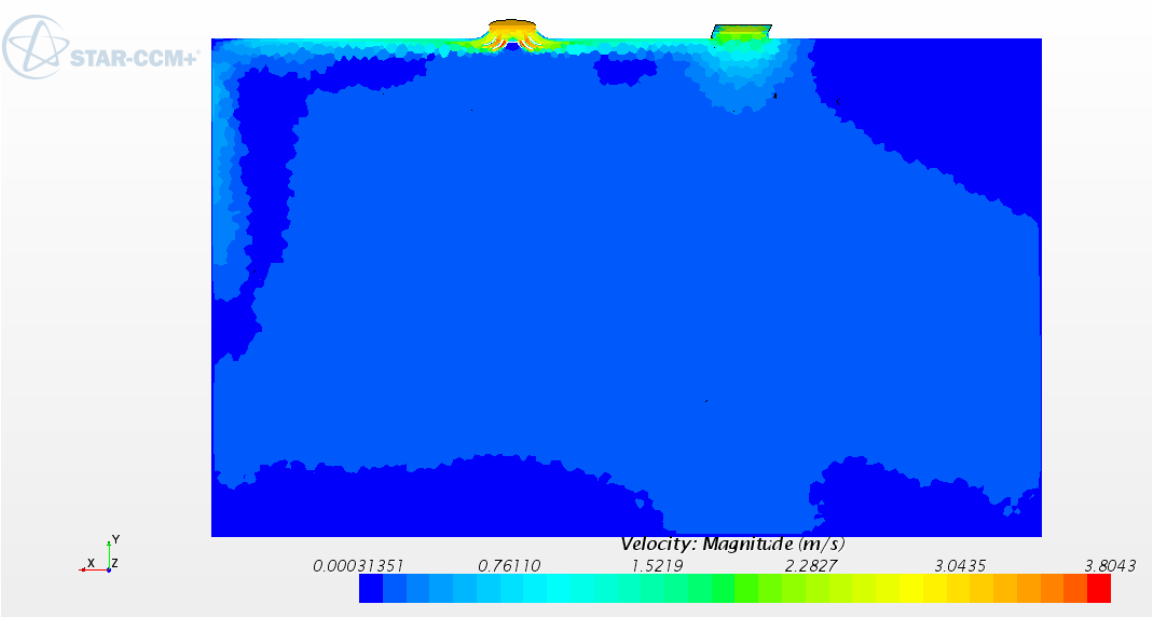

Figure 117: Two Dimensional Planar Section of the office; 36 in – 350cfm

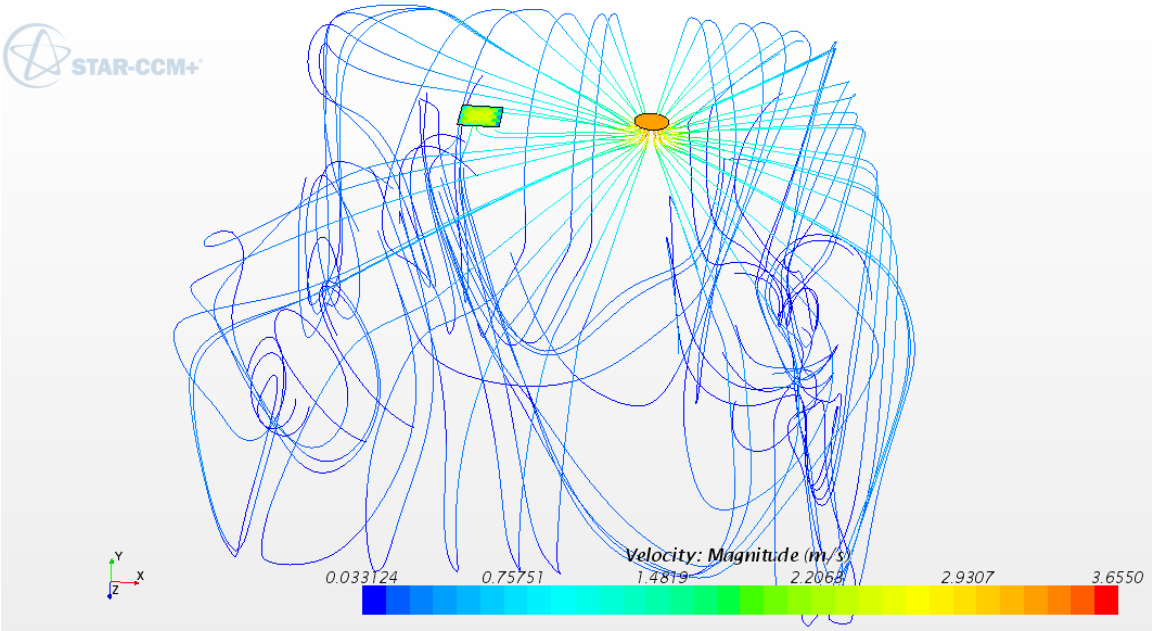

Figure 118: Three dimensional model of air circulation in the zone; 36 in – 350cfm

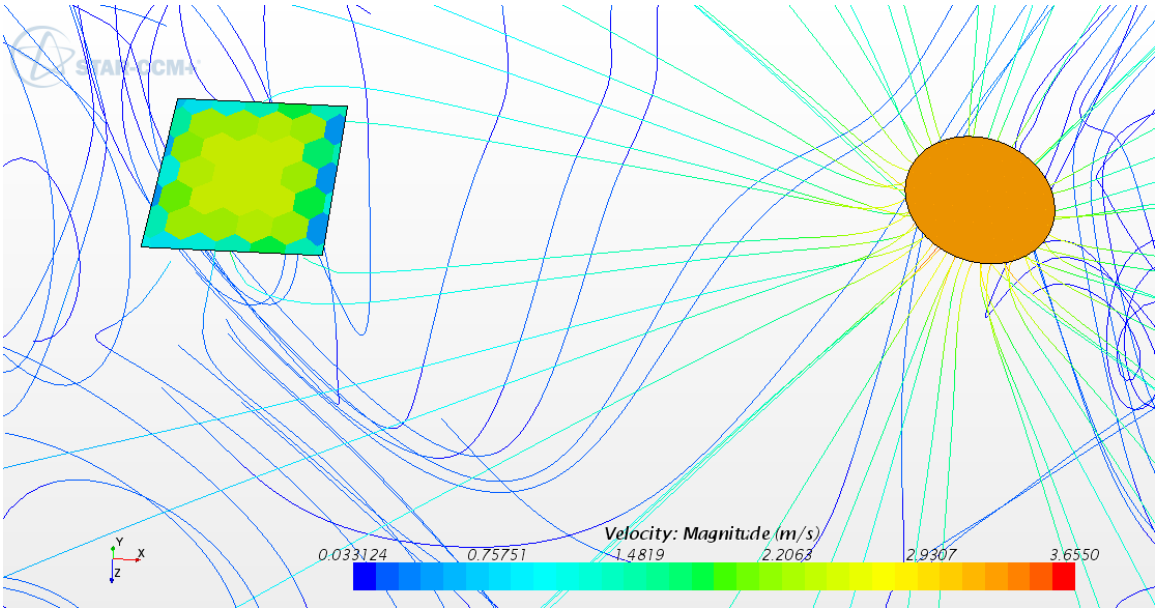

Figure 119: Three dimensional model of air circulation in the zone; 36 in – 350cfm

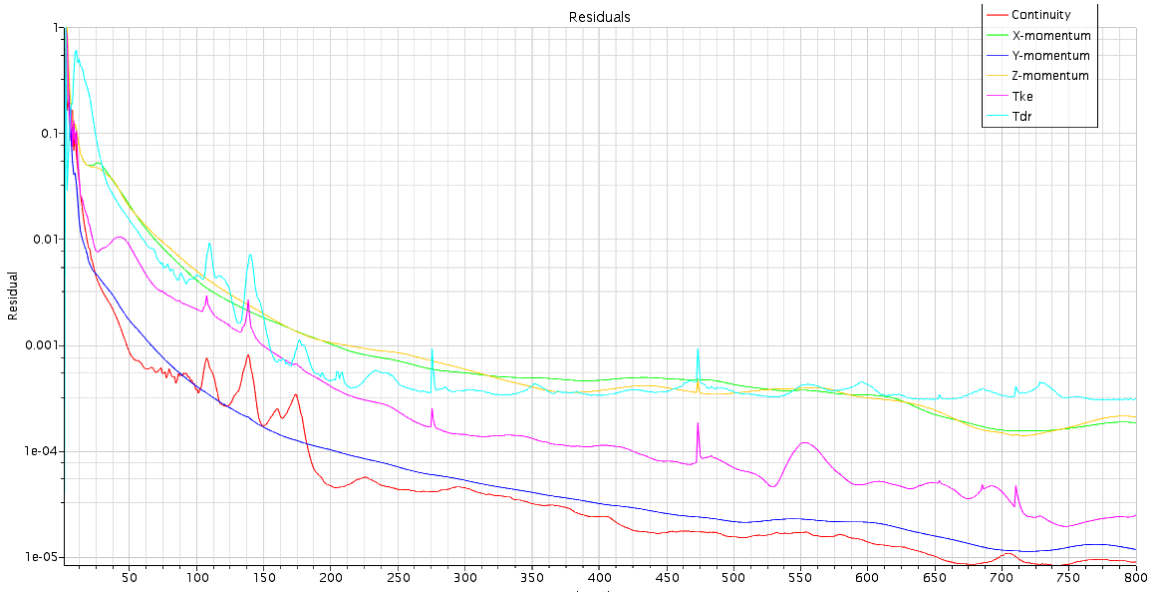

Figure 120: Plot Residuals; 36 in – 350cfm

Round Diffuser: 36 inches away from the return; 500 cfm

At supply to return distance of 36 inches and 500 cfm, figures 121, 122 and 123 below show a by-pass rate of 6.6% as 4 out of 64 lines by-passed the system; By-passing at 500 cfm seem to be lower than that at 200 and 350 cfm.

A plot of the simulation residuals is shown in Figure 124. The plot shows that all 6 residuals were relatively converged at or below the 0.005 level which show how precise the results of the simulation are.

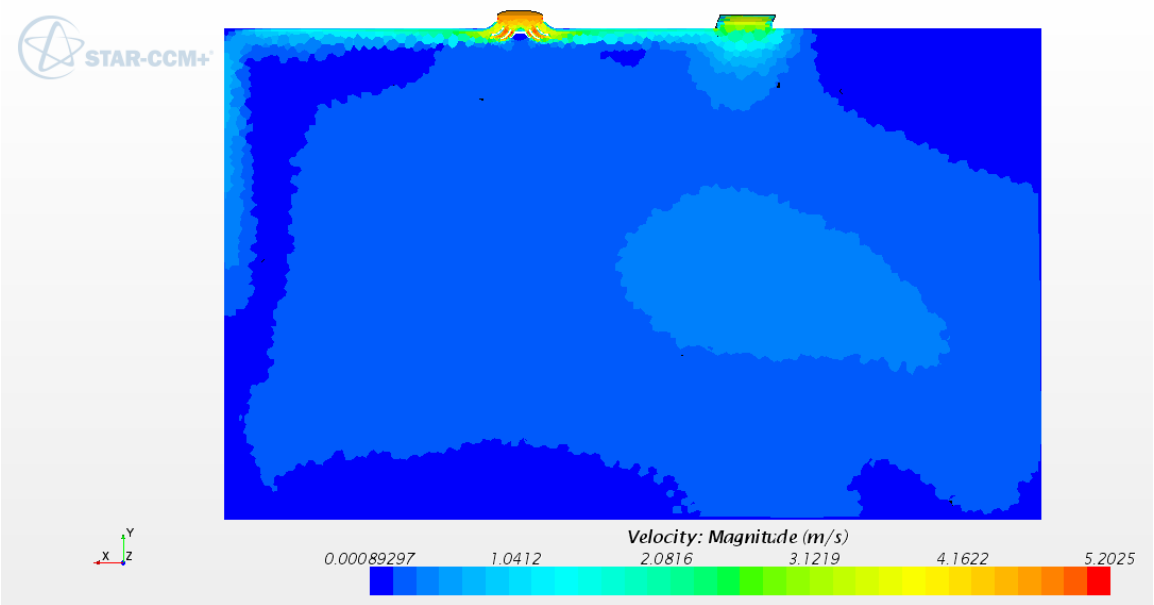

Figure 121: Two Dimensional Planar Section of the office; 36 in – 500cfm

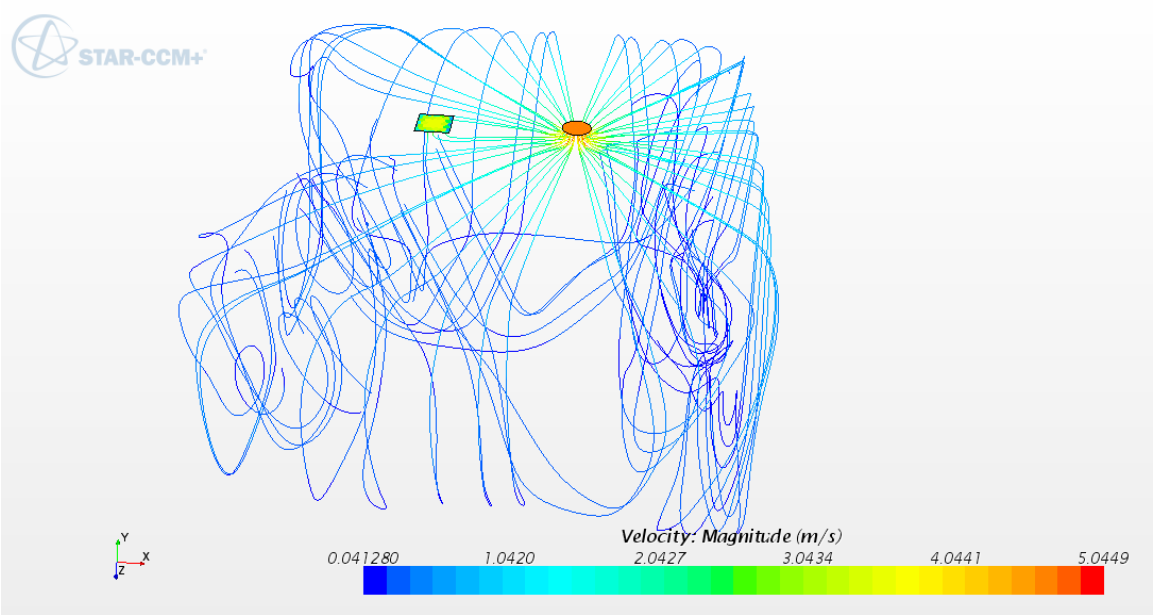

Figure 122: Three dimensional model of air circulation in the zone; 36 in – 500cfm

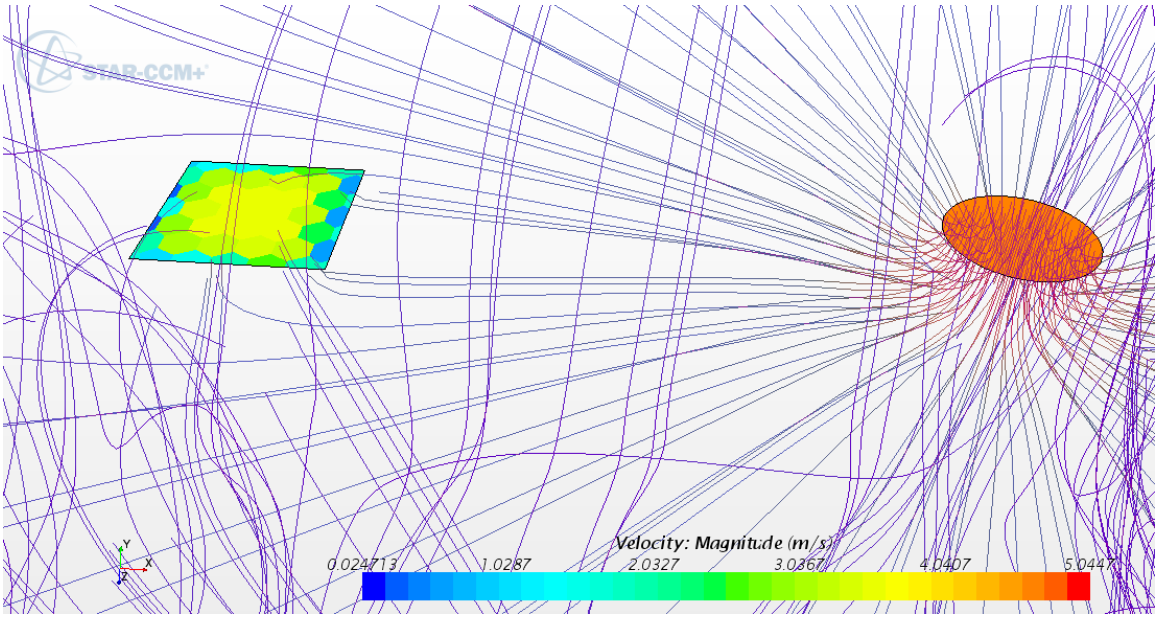

Figure 123: Three dimensional model of air circulation in the zone; 36 in – 500cfm

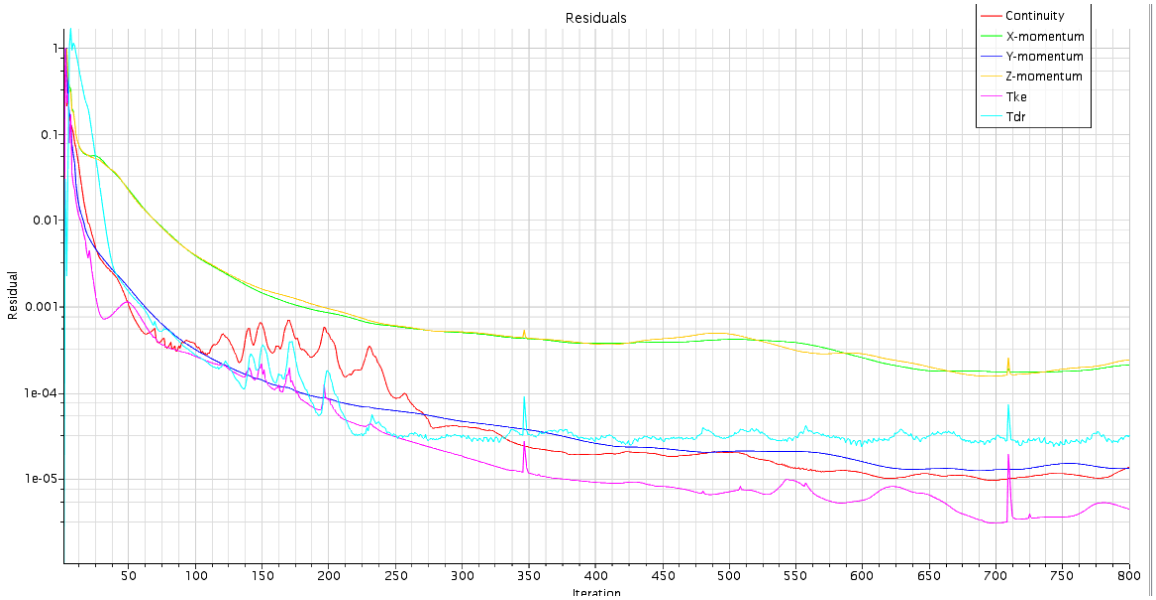

Figure 124: Plot Residuals; 36 in – 500cfm

Round Diffuser: 48 inches away from the return; 200 cfm

At supply to return distance of 48 inches and 200 cfm, figures 125, 126 and 127 below show a by-pass rate of 6.6% as 5 out of 64 lines by-passed the system; this meets the expectations as the by-pass rate decreased when the distance between supply and return increased.

A plot of the simulation residuals is shown in Figure 128. The plot shows that all 6 residuals were relatively converged at or below the 0.0001 level which show how precise the results of the simulation are.

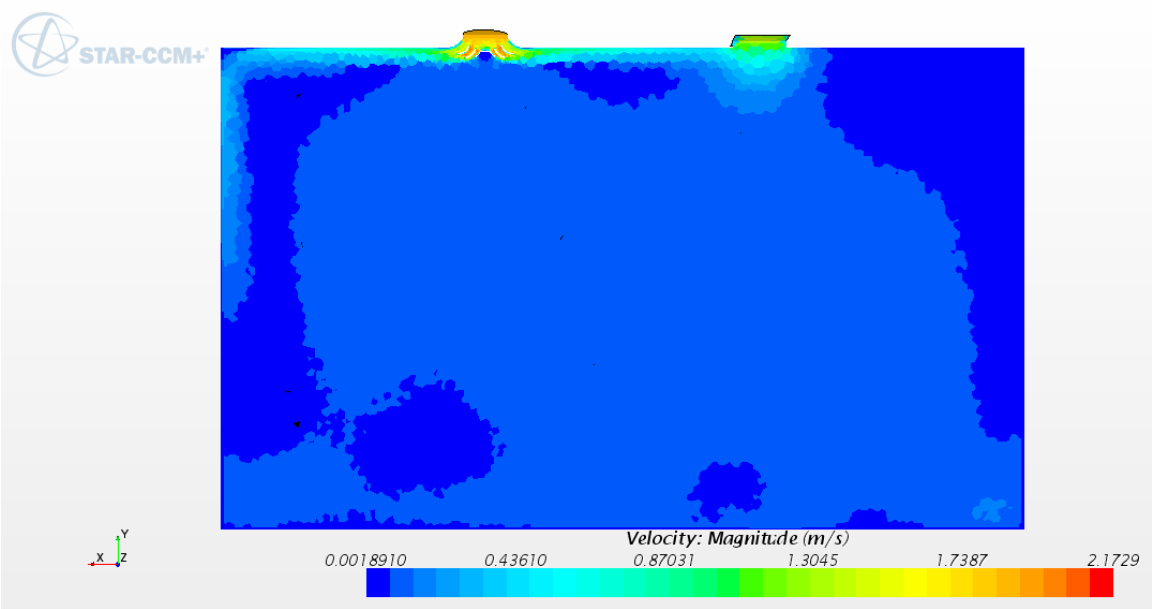

Figure 125: Two Dimensional Planar Section of the office; 48 in – 200cfm

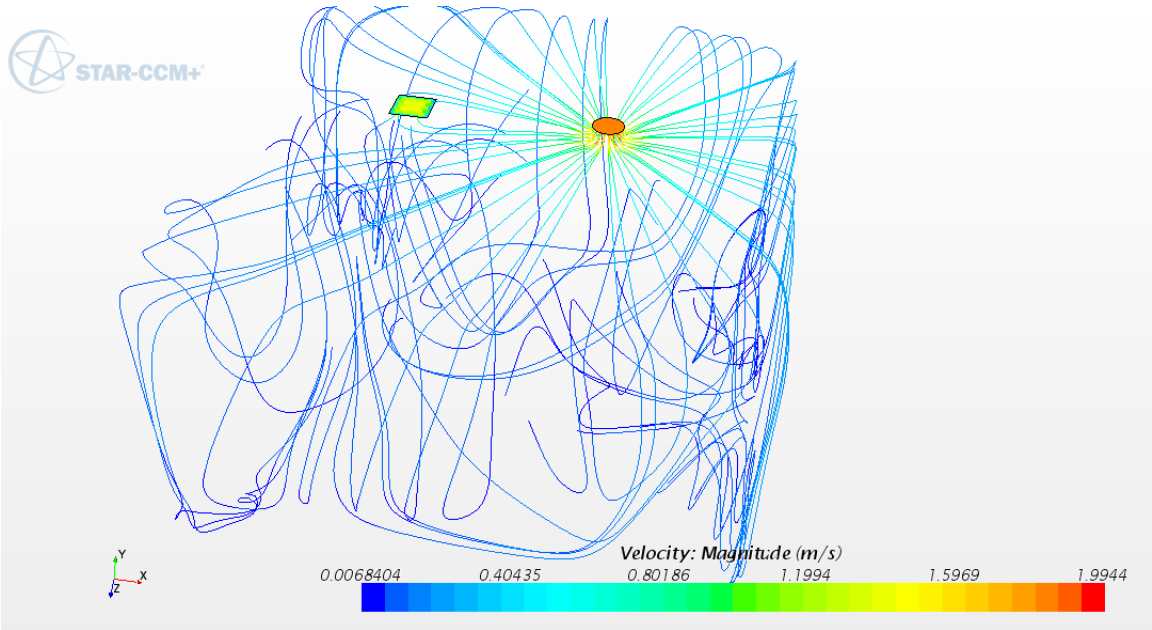

Figure 126: Three dimensional model of air circulation in the zone; 48 in – 200cfm

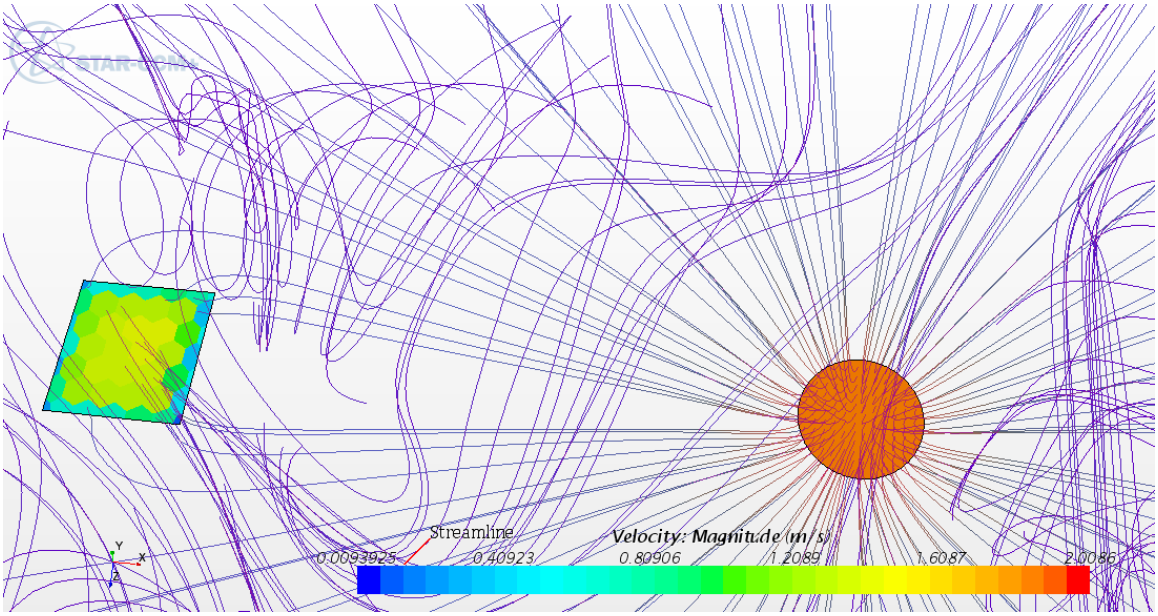

Figure 127: Three dimensional model of air circulation in the zone; 48 in – 200cfm

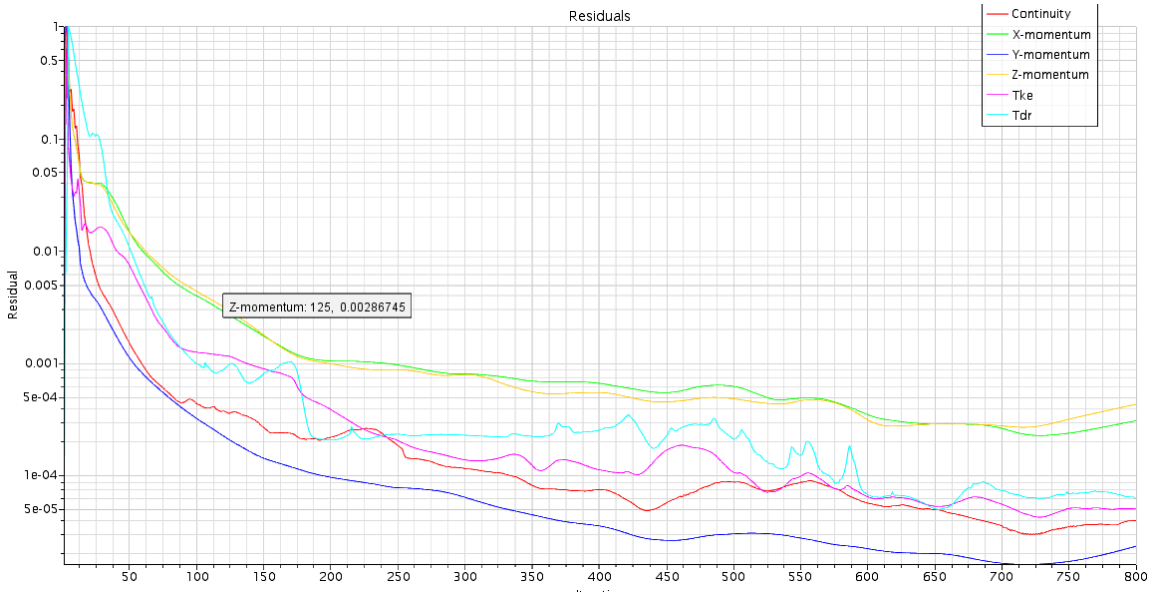

Figure 128: Plot Residuals; 48 in – 200cfm

Round Diffuser: 48 inches away from the return; 350 cfm

At supply to return distance of 48 inches and 350 cfm, figures 129, 130 and 131 below show a by-pass rate of 6.6% as 4 out of 64 lines by-passed the system; the results of this simulation is similar to the simulation at 200 and 500 cfm. The effect of air flow rate on the efficiency of HVAC systems with round diffuser seem to be lower than that of systems with square diffusers.

A plot of the simulation residuals is shown in Figure 132. The plot shows that all 6 residuals were relatively converged at or below the 0.0001 level which show how precise the results of the simulation are.

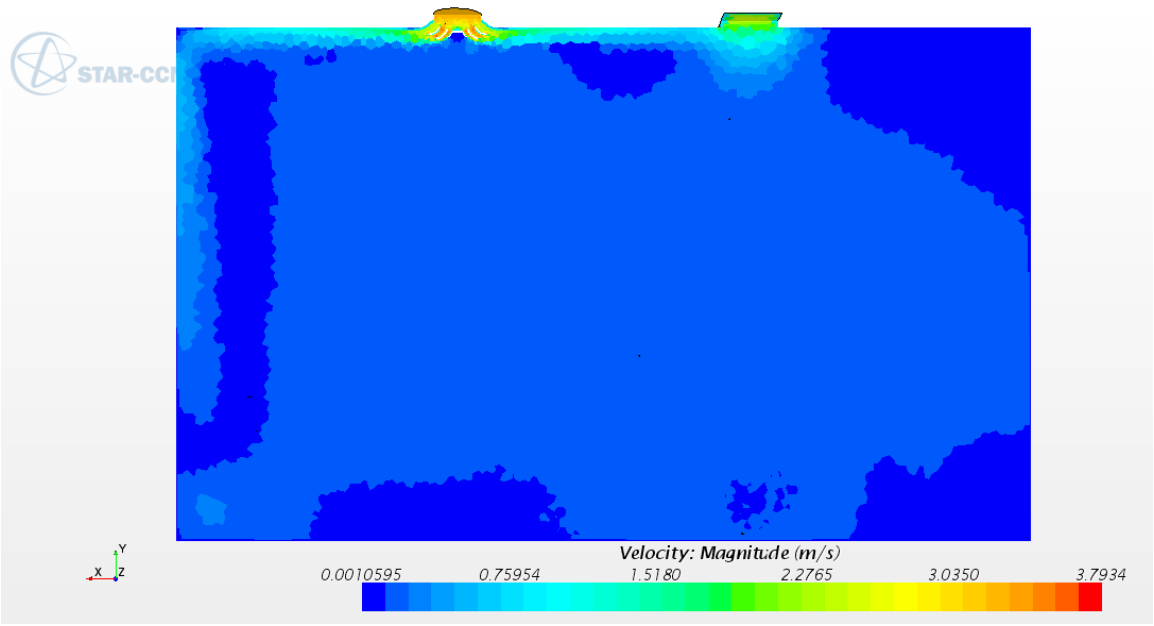

Figure 129: Two Dimensional Planar Section of the office; 48in – 350cfm

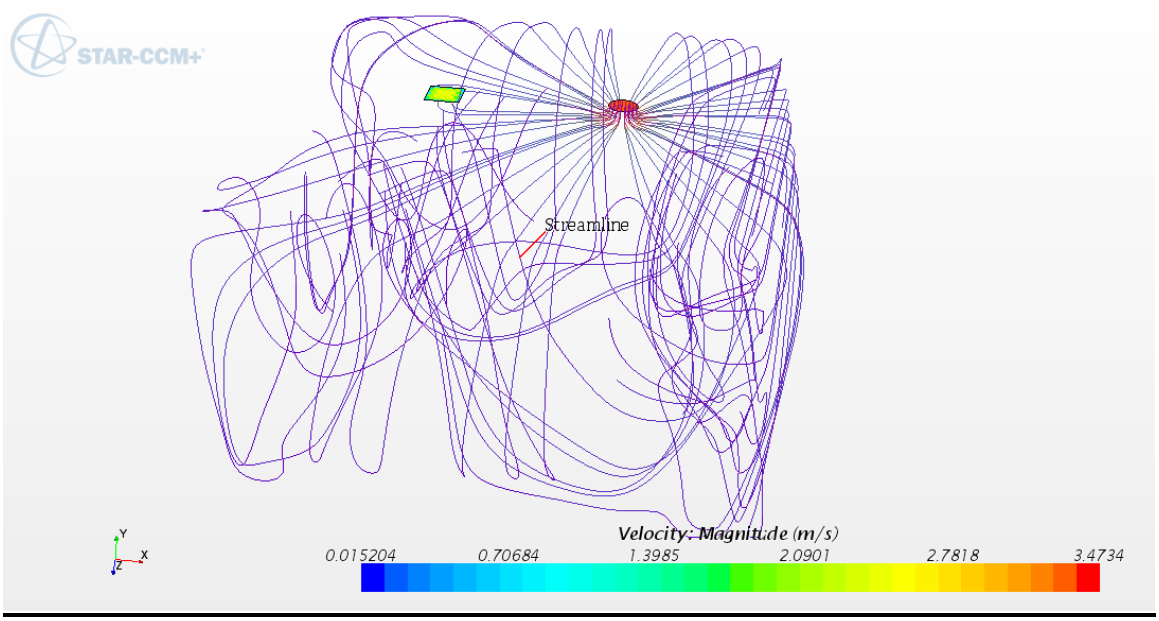

Figure 130: Three dimensional model of air circulation in the zone; 48 in – 350cfm

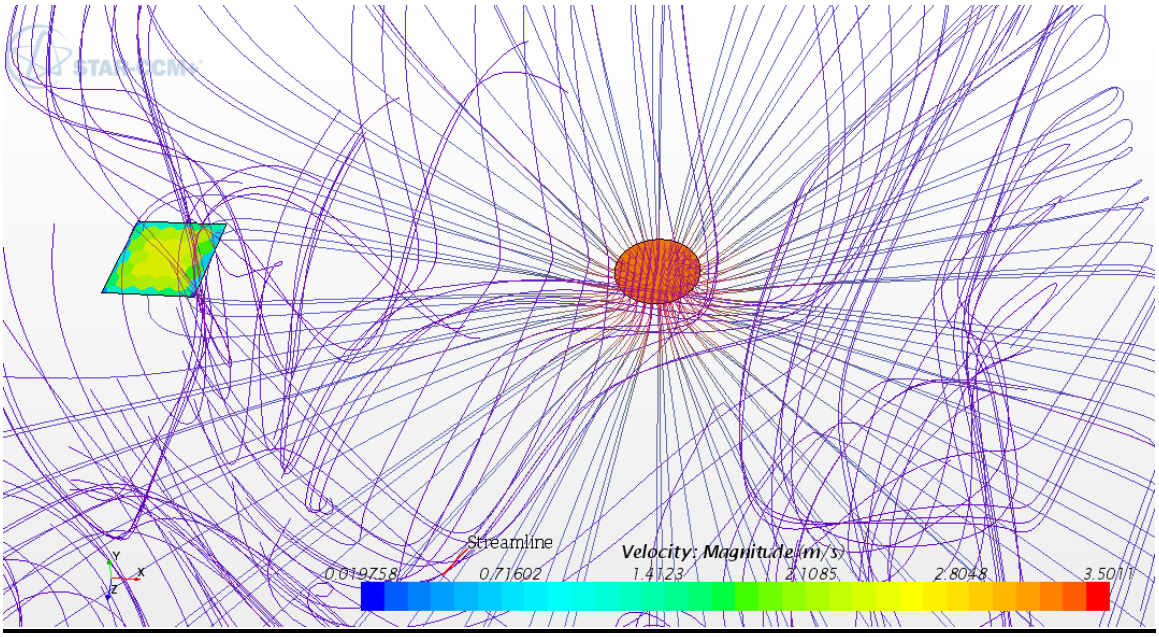

Figure 131: Three dimensional model of air circulation in the zone; 48 in – 350cfm

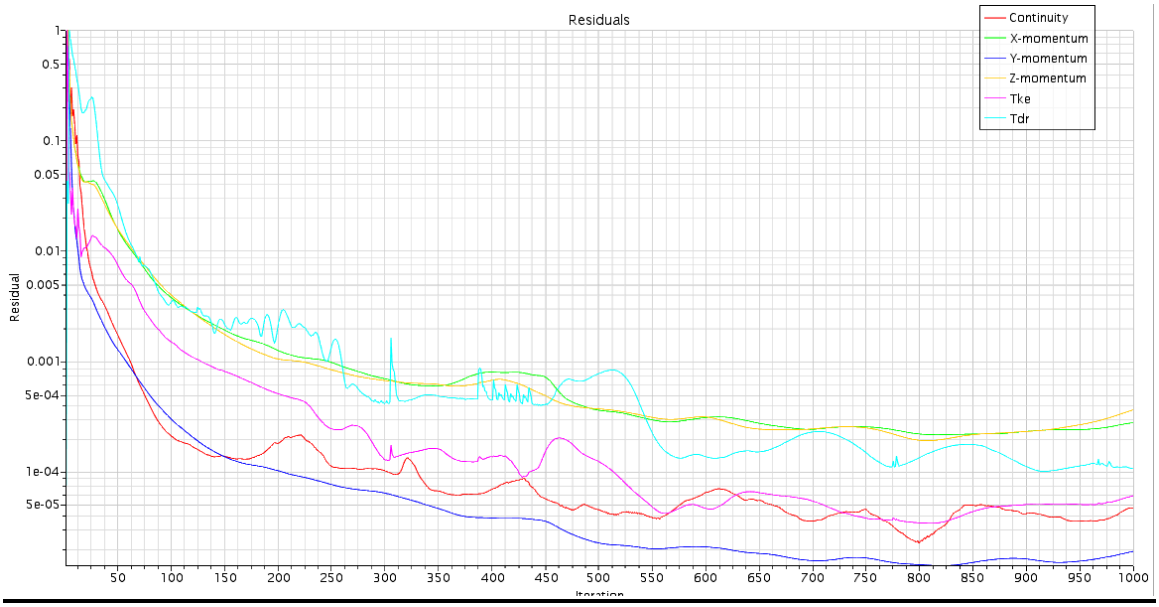

Figure 132: Plot Residuals; 48 in – 350cfm

Round Diffuser: 48 inches away from the return; 500 cfm

At supply to return distance of 48 inches and 500 cfm, figures 133, 134 and 135 below show a by-pass rate of 6.6% as 4 out of 64 lines by-passed the system; by-passing at a supply to return distance of 48 inches was the same for all flow rates.

A plot of the simulation residuals is shown in Figure 136. The plot shows that all 6 residuals were relatively converged at or below the 0.001 level which show how precise the results of the simulation are.

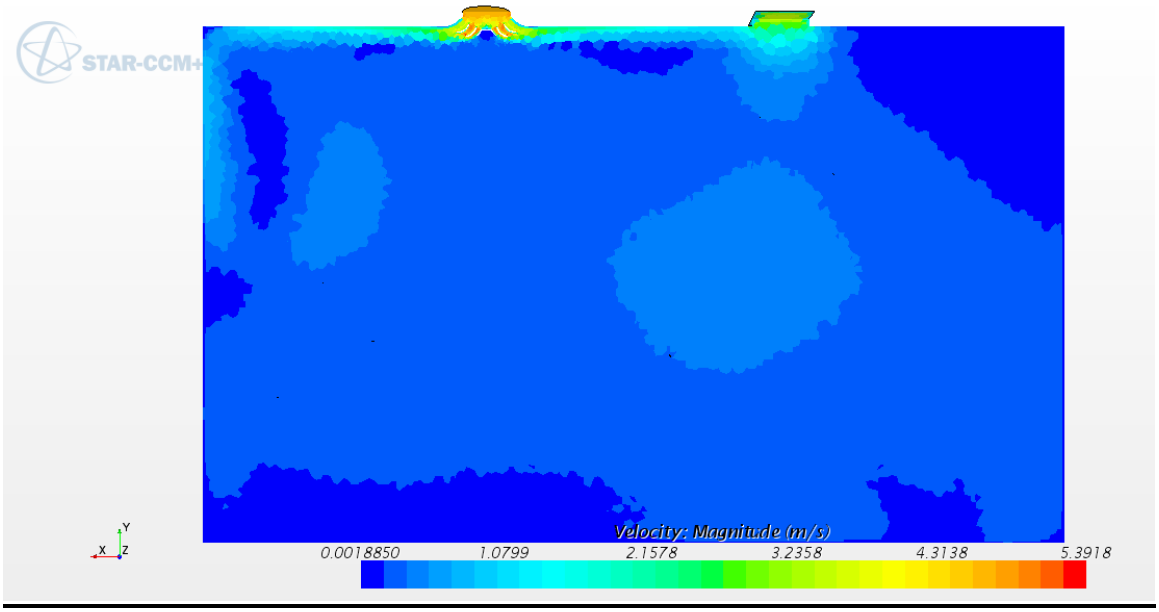

Figure 133: Two Dimensional Planar Section of the office; 48 in – 500cfm

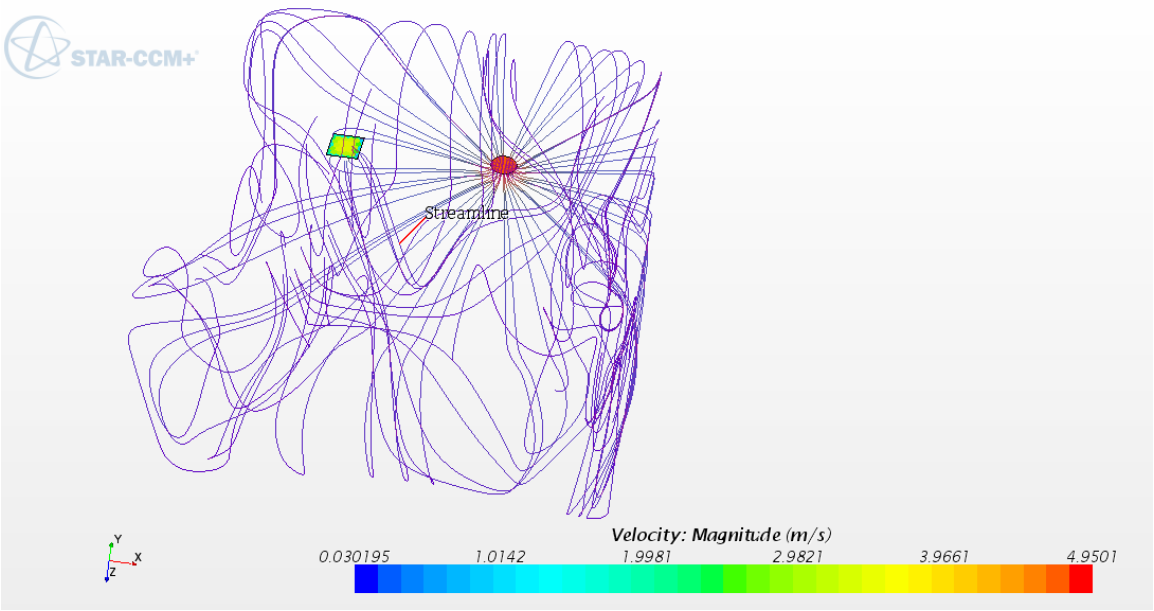

Figure 134: Three dimensional model of air circulation in the zone; 48 in – 500cfm

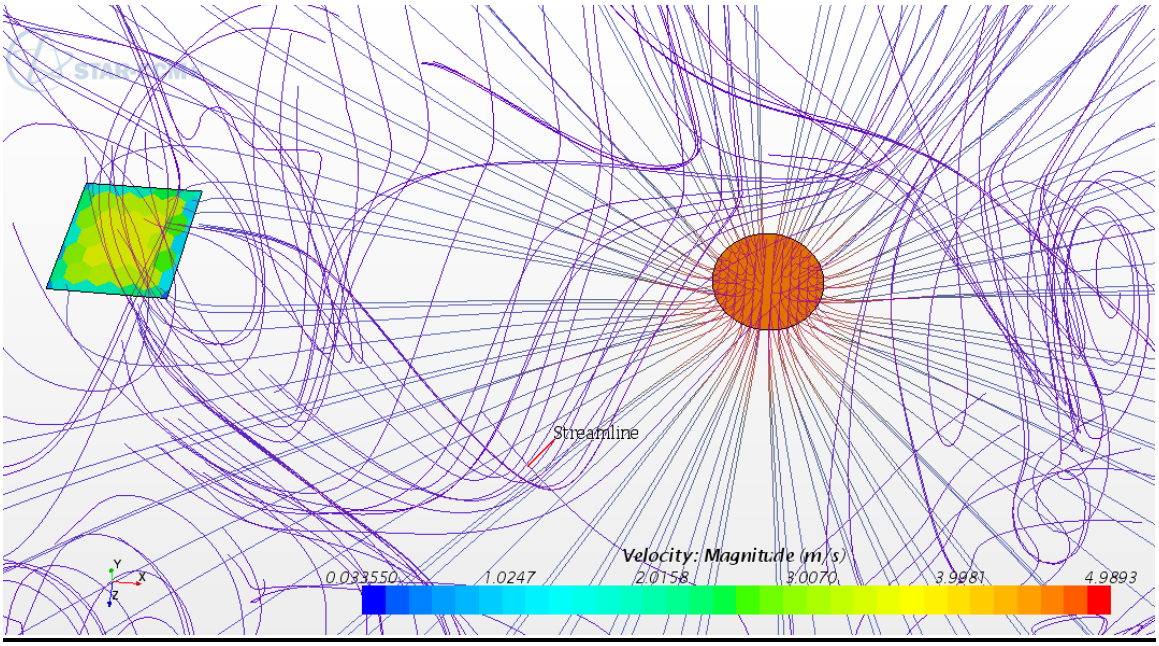

Figure 135 : Three dimensional model of air circulation in the zone; 48 in – 500cfm

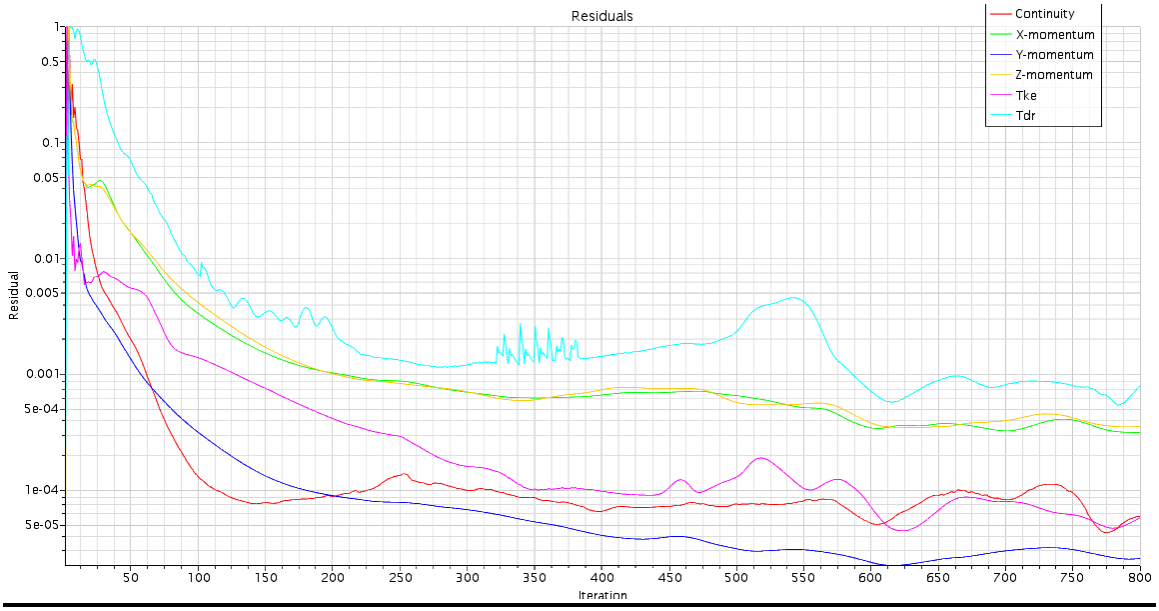

Figure 136: Plot Residuals; 48 in – 500cfm

Round Diffuser: 60 inches away from the return; 200 cfm

At supply to return distance of 60 inches and 200 cfm, figures 137, 138 and 139 below show a by-pass rate of 5.8% as 10 out of 172 lines by-passed the system; the use of extra streamlines was needed in order to get a more accurate reading on the number of by-passing streamlines.

A plot of the simulation residuals is shown in Figure 140. The plot shows that all 6 residuals were relatively converged at or below the 0.0001 level which show how precise the results of the simulation are.

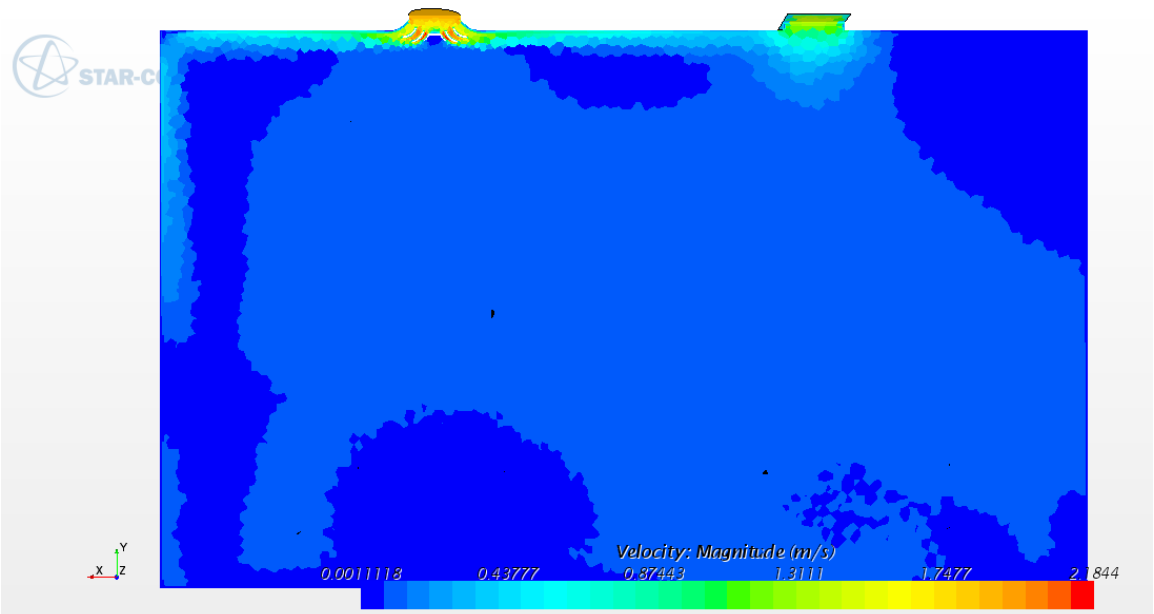

Figure 137: Two Dimensional Planar Section of the office; 60 in – 200cfm

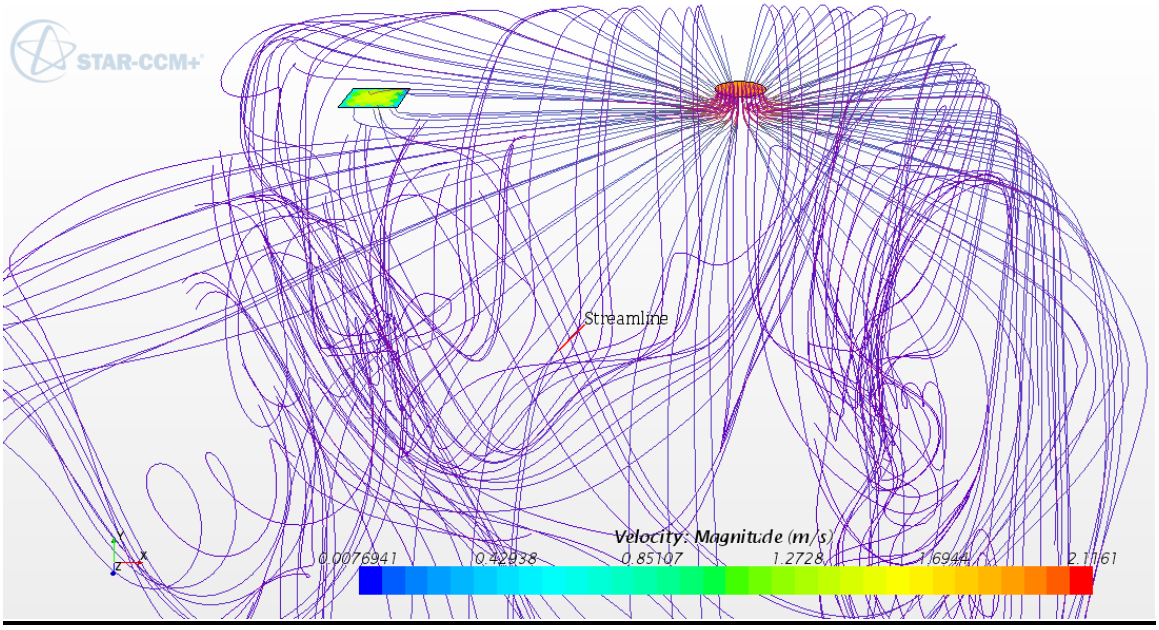

Figure 138: Three dimensional model of air circulation in the zone; 60 in – 200cfm

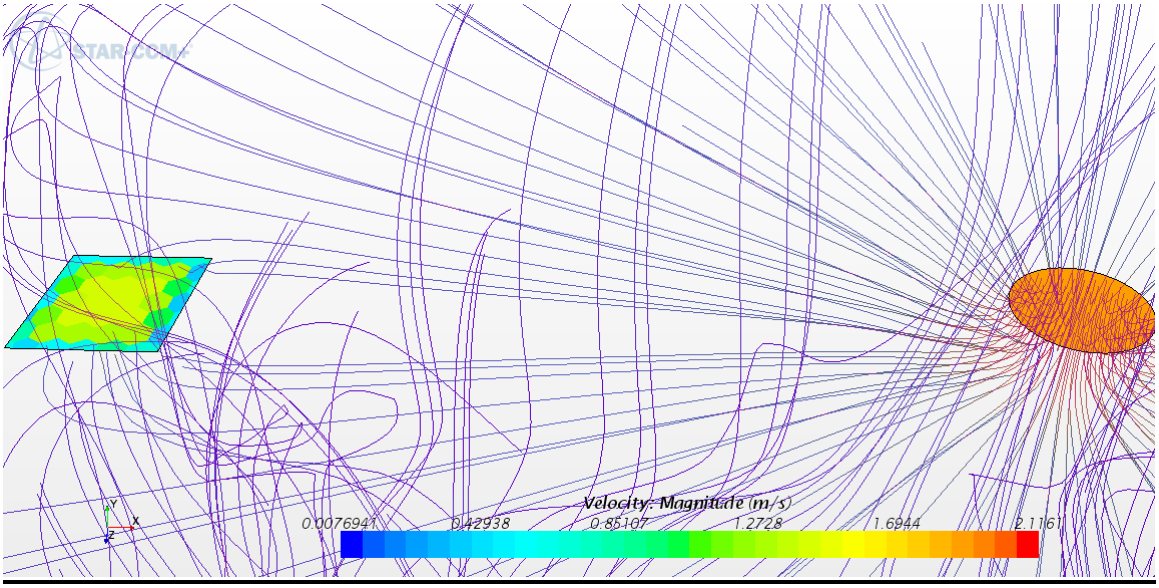

Figure 139: Three dimensional model of air circulation in the zone; 60 in – 200cfm

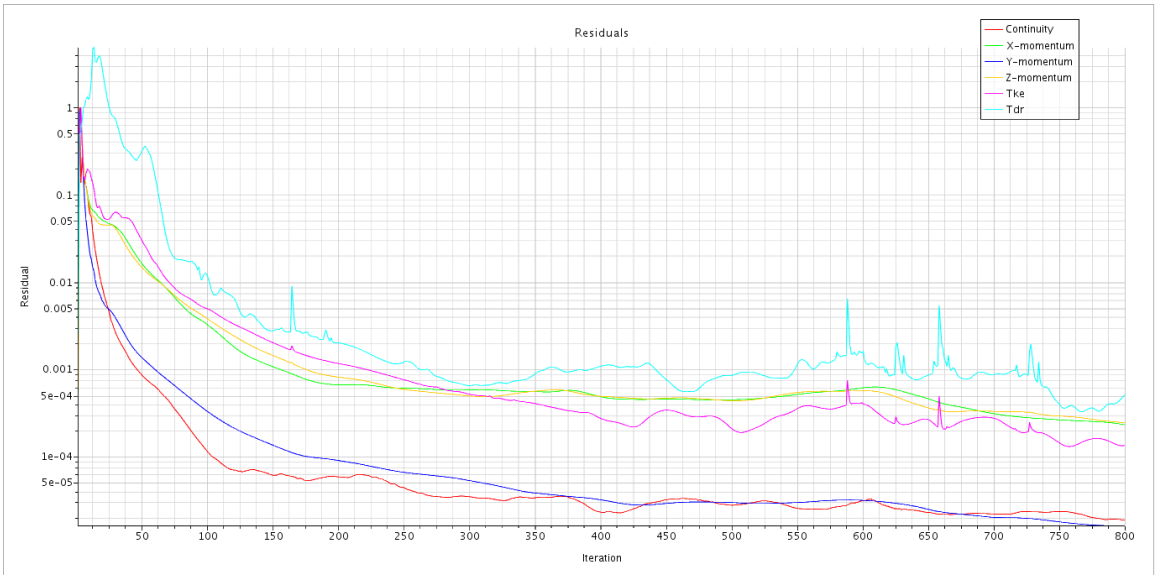

Figure 140: Plot Residuals; 60 in – 200cfm

Round Diffuser: 60 inches away from the return; 350 cfm

At supply to return distance of 60 inches and 350 cfm, figures 141 and 142 below show a by-pass rate of 5.8% as 10 out of 172 lines by-passed the system; the use of extra streamlines was needed in order to get a more accurate reading on the number of by-passing streamlines.

A plot of the simulation residuals is shown in Figure 143. The plot shows that all 6 residuals were relatively converged at or below the 0.005 level which show how precise the results of the simulation are.

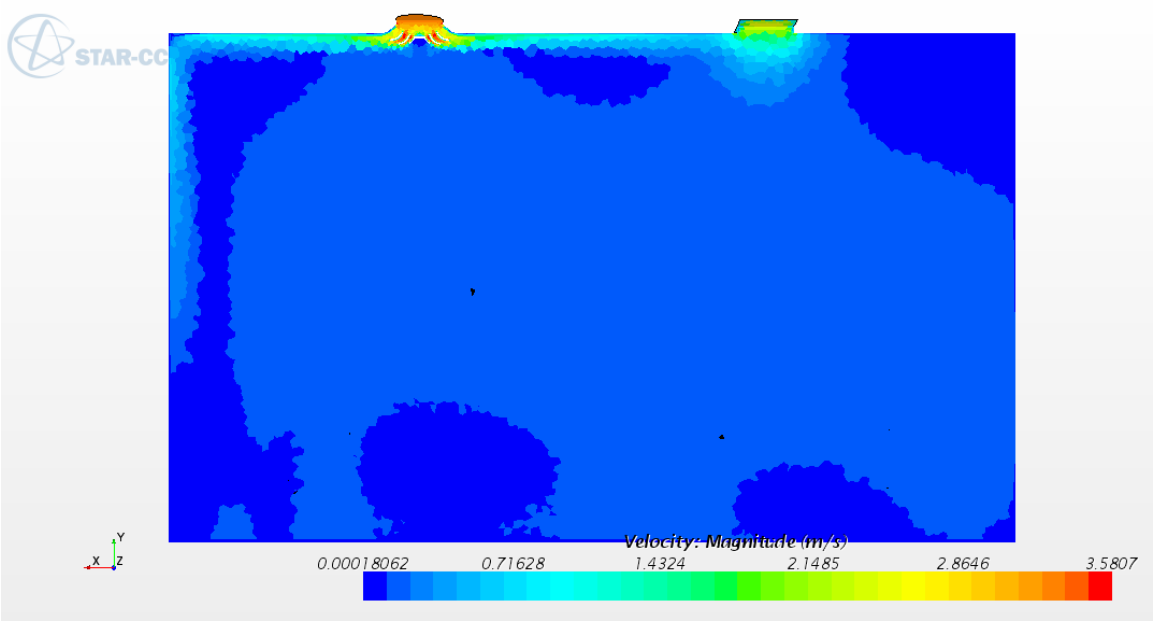

Figure 141: Two Dimensional Planar Section of the office; 60 in – 350cfm

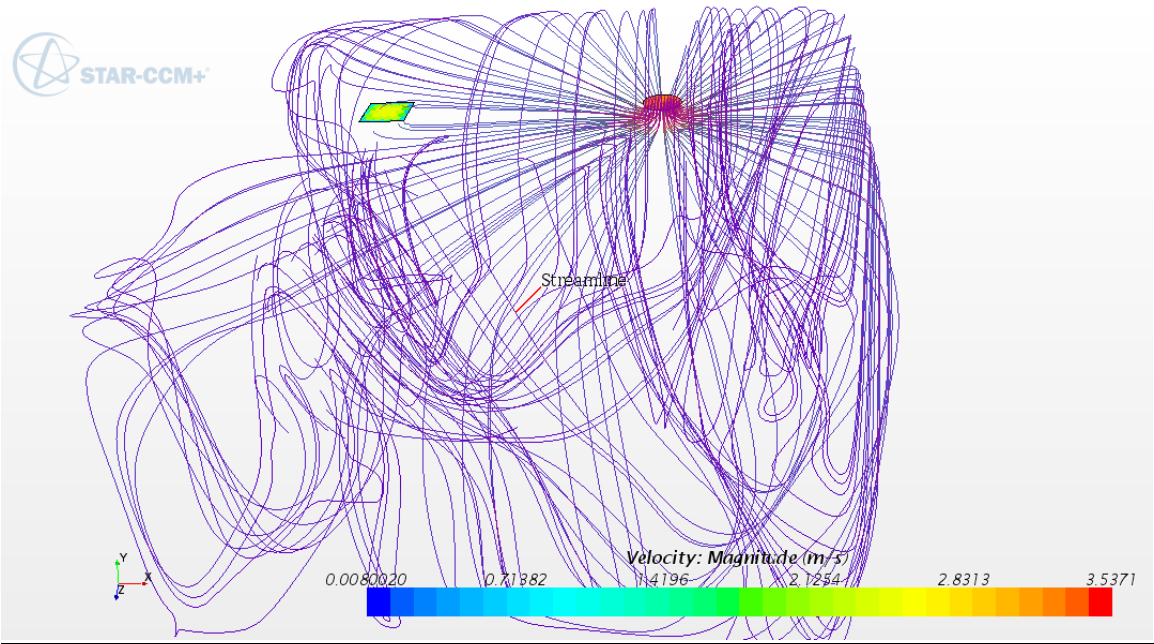

Figure 142: Three dimensional model of air circulation in the zone; 60 in – 350cfm

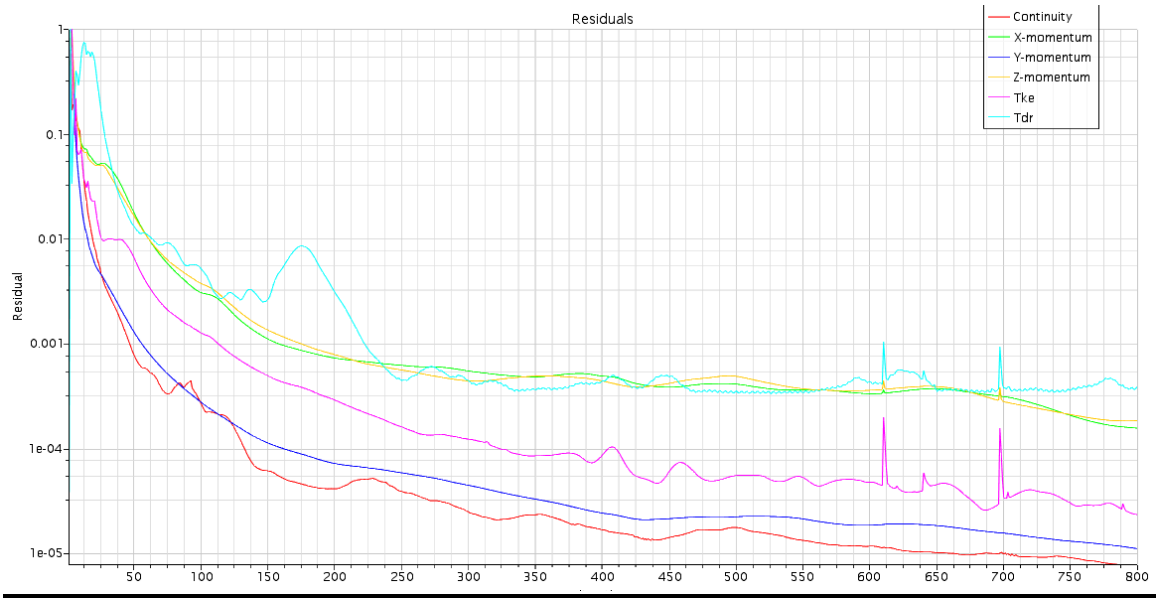

Figure 143: Plot Residuals; 60 in – 350cfm

Round Diffuser: 60 inches away from the return; 500 cfm

At supply to return distance of 60 inches and 500 cfm, figures 144, 145 and 146 below show a by-pass rate of 4.6% as 8 out of 172 lines by-passed the system; the by-pass rate at 500 cfm seem to be lower than that at 200 and 350 cfm when the distance between the supply and return apertures is 60 inches.

A plot of the simulation residuals is shown in Figure 147. The plot shows that all 6 residuals were relatively converged at or below the 0.0001 level which show how precise the results of the simulation are.

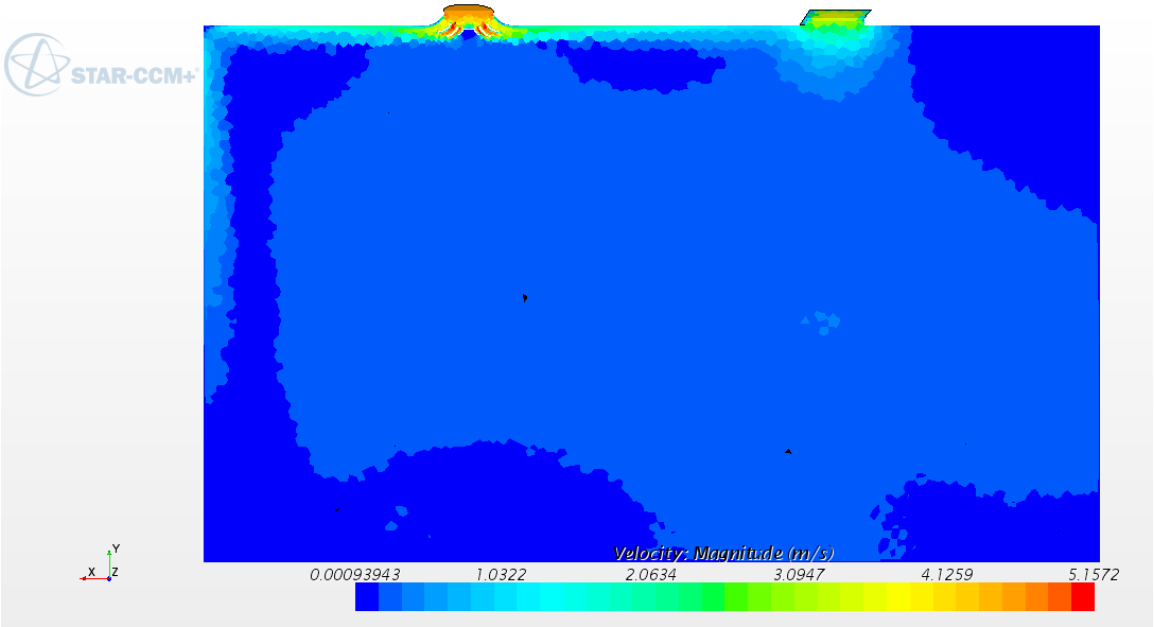

Figure 144: Two Dimensional Planar Section of the office; 60 in – 500cfm

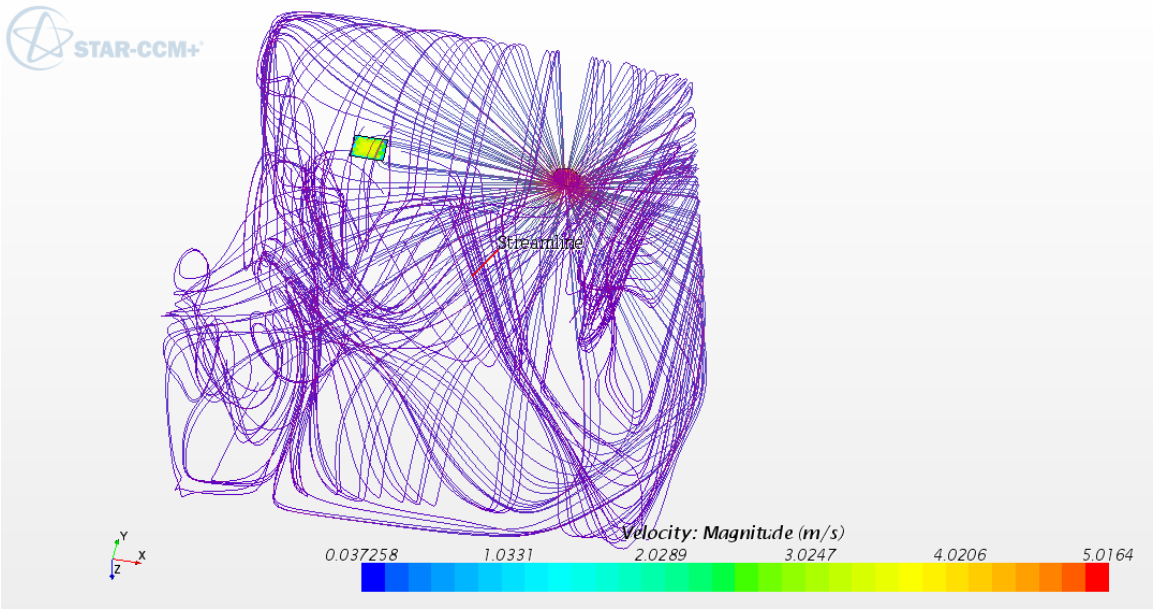

Figure 145: Three dimensional model of air circulation in the zone; 60 in – 500cfm

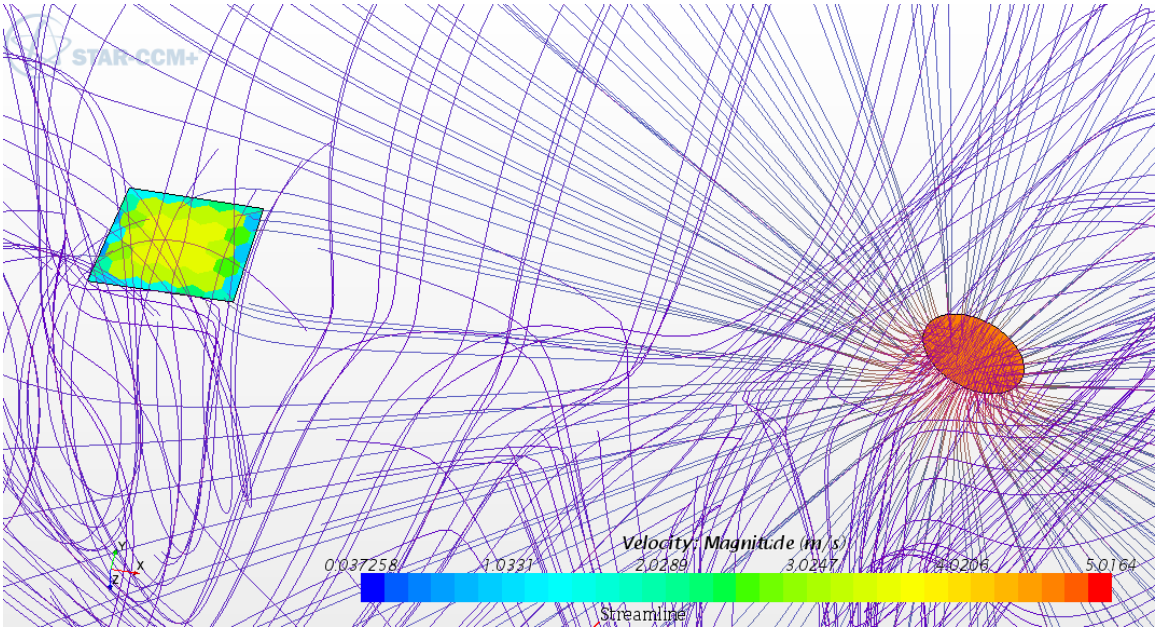

Figure 146: Three dimensional model of air circulation in the zone; 60 in – 500cfm

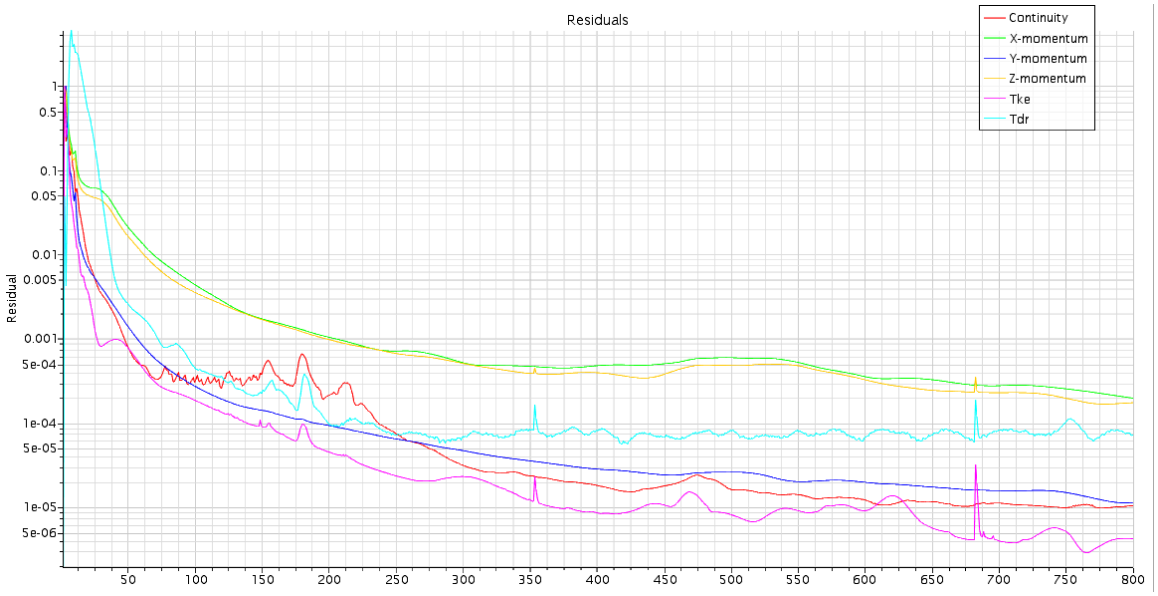

Figure 147: Plot Residuals; 60 in – 500cfm

Results - Summary

Table 3 below shows the percentage of air by-pass for the simulations with the square diffuser at the selected distances, away from the return grille, and different flow rates.

Table 3: Summary of Results: Square

Type	Distance (in)	Flow rate (cfm)	Approximate Short Circuiting (%)
Square	12	200	**18.8**
		350	**23.4**
		500	**21.9**
	24	200	**20.3**
		350	**21.9**
		500	**18.8**
	36	200	**7.8**
		350	**17.2**
		500	**12.5**
	48	200	**6.3**
		350	**10.9**
		500	**7.8**
	60	200	**4.1**
		350	**5.6**
		500	**4.6**

It can be noted that the efficiency of the system increases as the distance between the supply diffuser and the return grille increases. The worst flow rate for efficiency is at 350 cfm; at 500 cfm, the high flow rate pushes air past the return grille. On the other hand, the weak flow rate of 200 cfm causes air to be dissipated following a lower Reynolds number characteristics which overcame by-passing.

Table 4 below shows the percentage of air by-pass for the simulations with the round diffuser at the selected distances, away from the return grille, and different flow rates.

Table 4: Summary of Results: Round

Type	Distance (in)	Flow rate (cfm)	Approximate Short Circuiting (%)
Round	12	200	**12.5**
		350	**10.9**
		500	**12.5**
	24	200	**9.4**
		350	**9.4**
		500	**7.8**
	36	200	**7.5**
		350	**7.5**
		500	**6.6**
	48	200	**6.6**
		350	**6.6**
		500	**6.6**
	60	200	**5.8**
		350	**5.8**
		500	**4.6**

Similar to square diffusers, the efficiency of the system increases as the distance between the supply diffuser and the return grille increases. Lower flow rates seem to cause higher by-pass rates at a fixed distance between the supply diffuser and the return grille. The residuals for the round diffuser simulations are lower and more stable than that of square diffusers.

The percent by-pass for systems with square diffusers, when placed 48 or less inches away from the return grille, almost double that of the round diffusers; the

most by-pass rate for a square diffuser is 23.4%, at 12 inches and 350 cfm, while it is only 12.5%, at 12 inches and 200 or 500 cfm, for systems with the round diffuser. The percent by-pass of air for systems with either the round or square diffusers that are placed 60 or more inches away from the return grille is almost the same for both at around 5% by-pass; this indicates that the type of diffuser does not affect the efficiency of the system when it is placed at a proper distance away from the return grille.

Results - Monetizing the loss in efficiency due to bypass

It is useful to translate the loss off efficiency to a tangible loss. This calculation compares the bypassing air to the energy that it took to deliver that air and therefore the cost of that energy can be quantified.

$The\ cooling\ load\ for\ the\ office\ was\ assumed\ to\ be$:

$$7500\frac{BTU}{hour} = 0.625\ ton\ of\ cooling$$

Table 5 below shows the approximate power inputs per ton of cooling required for the compressor and auxiliaries of an HVAC system. (ASHRAE Application, chapter 57, Table 2). [8]

Table 5: Approximate power inputs

System	Compressor (kW/ton)	Auxiliaries (kW/ton)
Window Unit	1.46	0.32
Through-Wall Units	1.64	0.3
Dwelling Unit, Central Air-Cooled	1.49	0.14
Central, Group or Bldg. Cooling Plants		
(3 to 25 tons) Air Cooled	1.2	0.2
(25 to 100 tons) Air Cooled	1.18	0.21
(25 to 100 tons) Water Cooled	0.94	0.17
(Over 100 tons) Water Cooled	0.79	0.2

From the table, the power inputs are

$$1.2\frac{kW}{ton}\ and\ 0.20\frac{kW}{ton}$$

$$The\ total\ power\ used\ by\ the\ HVAC\ system\ becomes:$$

$$P = 0.625\ ton\left(1.2\frac{kW}{ton} + \ 0.20\frac{kW}{ton}\right)$$

$$P = 0.875\ kW$$

It is reasonable to assume business hours of 45 hours per week at 52 weeks per year which gives

$$P_{yearly} = 0.875\ kW \cdot \frac{45\ hours}{week} \cdot \frac{52\ weeks}{year} = 2047.5\ kWh$$

In real life-applications, the power lost due to by-passing need to be compensated for; the HVAC system adds an equivalent amount of energy equal to energy that was lost to by-pass in order to achieve the temperature and humidity requirements of the system.

If 23.4% of the air by-passed the system, then the amount of power that the HVAC system added in order to compensate for the power lost due to by-pass would be:

$$Power\ replaced = 2047.5\ kWh \cdot 0.234 = 479\ kWh$$

In South Dakota cost of electricity is $0.08 per kWh, so:

$$Cost\ of\ bypass = 479\ kWh \cdot \frac{\$0.08}{kWh} = \$38.4$$

Initially this amount does not seem very significant, however it should be noted that this value was found for one small office. A school or an office building would have much larger offices and many more of them.

For an example, SDSU has 30 buildings and each has 20 offices, which is a total of 600 offices. At $38.4 in lost energy per year

$$Cost_{yearly} = 600\ offices \cdot \frac{\$38.4}{office \cdot year}$$

$$Cost_{yearly} = \$23{,}040$$

At this scale, the cost incurred from wasted energy because of the inappropriate placement of a return air grille becomes significant.

Tables 6 and 7 below show the annual cost of air by-pass.

Table 6: Annual cost of air by-pass; square diffuser at 350 cfm

Distance between Inlet and Outlet	Short Circuiting [%]	Cost per year per office[$]	Cost per year per 600 office[$]
12	23.4	38.4	23,040
24	21.9	35.9	21,540
36	17.2	28.2	16,920
48	10.9	17.9	10,740
60	5.6	9.2	5,520

Table 7: Annual cost of air by-pass; round diffuser at 350 cfm

Distance between Inlet and Outlet	Short Circuiting [%]	Cost per year per office [$]	Cost per year per 600 office[$]
12	10.9	17.9	10,740
24	9.4	15.4	9,420
36	7.5	12.3	7,380
48	6.6	10.8	6,480
60	5.8	9.5	5,700

From tables 6 and 7 above, it is noted that the cost of air by-pass for round diffusers is much lower than that of the square diffusers when they are placed less than 48 inches away from the return air grille; inefficient systems with square diffusers approximately cost twice as much as inefficient systems with round diffusers.

Analogous Support of Model Results

Smoke Test Analogy

Using smoke in an air distribution system allows the flow characteristics of the air to be observed. Figures 148, 149 show the actual flow of air at 250 and 300 cfm respectively. The pattern of air flow is similar to the pattern of air flow that is seen in the CFD simulations which validates the results obtained from the CFD software. Figure 150 shows an advertisement brochure that was published by Lima Company; it states that their diffusers dispense air following a 360-degree pattern which is the most efficient and effective pattern. The study proved that round diffusers are more efficient that square diffusers.

Figure 148: Smoke test showing air flow patterns from diffuser [14]

Figure 149: Smoke test showing air flow patterns from diffuser [15]

Figure 150: Smoke test showing air flow patterns from diffuser [16]

Resampled Volume

Resampled volume scalar views were completed to show validation of the typical two-dimensional scalar velocity views which are built using a plane. Resampled volume views show the volume and velocity of air inside the office space. These

images show that the velocity profiles that are implied from the two-dimensional view are reasonable and verify the streamlines views.

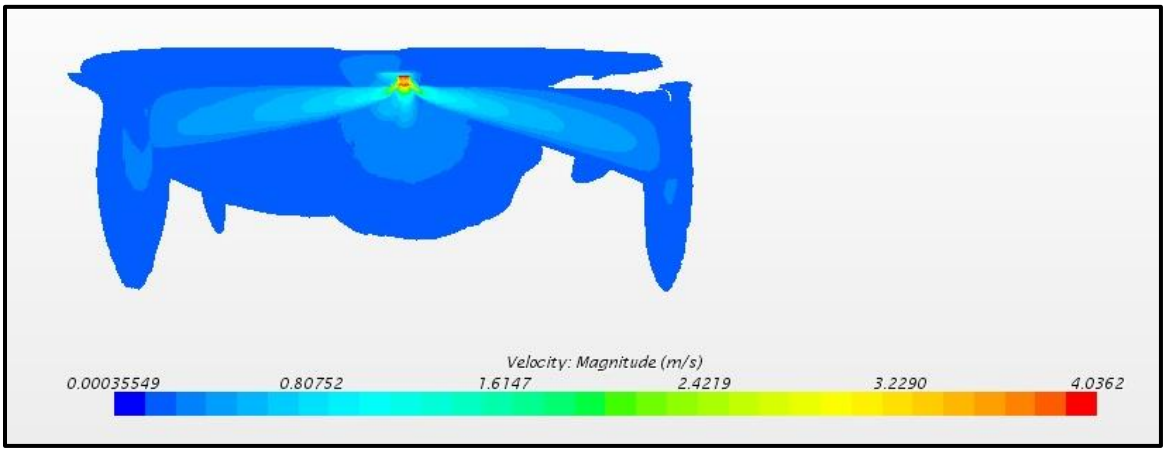

Figure 151: Velocity scalar; resampled volume of room with diffuser; 48 in – 350 cfm

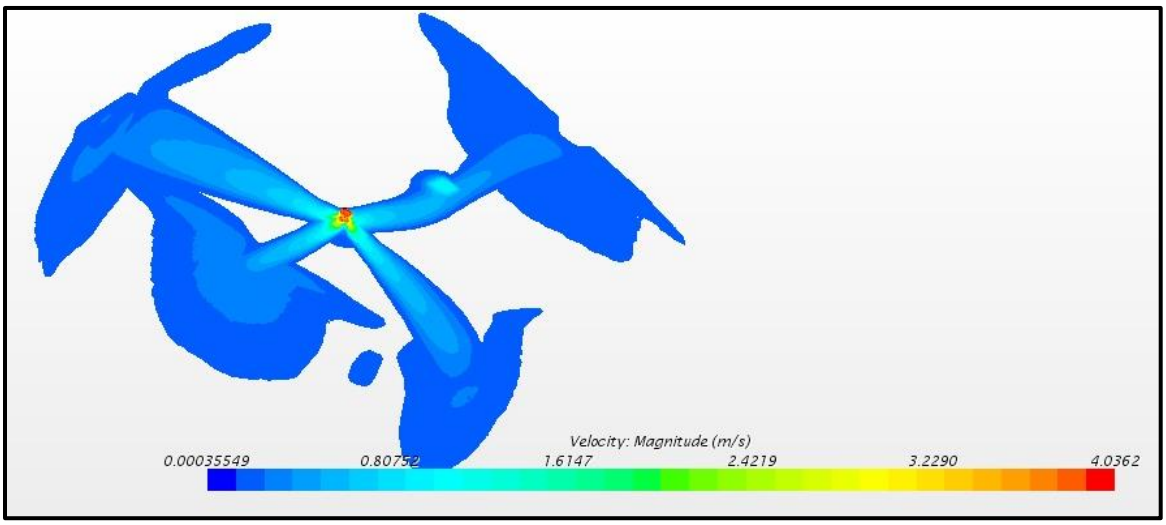

Figure 152: Velocity scalar; resampled volume of room with diffuser; 48 inch – 350 cfm

Conclusion

The scope of this investigation was to demonstrate the relevance of the distance between the supply diffuser and the return grille on the overall efficiency of an HVAC system and to estimate the energy losses due to air by-pass in an office zone. Two types of diffusers, square and round were studied. The diffusers were placed at 5 selected distances of 12, 24, 36, 48 and 60 inches away from the return grille; each diffuser, at every placement, was subjected to a flow rate of 200, 350 and 500 cfm.

As the distance between supply and return apertures decreased, the rate of air by-pass increased; for a square diffuser at 350 cfm, the percentages of air by-pass in the office space were 23.4, 21.9, 17.2, 10.9 and 4.6 respectively at placements of 12, 24, 36, 48 and 60 inches away from the return grille respectively. At 200 cfm, the percentages of air by-pass in the office space were 18.8, 20.3, 7.8, 6.3 and 4.1 respectively placements of 12, 24, 36, 48 and 60 inches respectively. At 500 cfm, the percentages of air by-pass in the office space were 21.9, 18.8, 12.5, 7.8 and 4.6 respectively placements of 12, 24, 36, 48 and 60 inches respectively.

The rates of air by-pass when using a round diffuser were as following: at 350 cfm, the percentages of air by-pass in the office space were 10.9, 9.4, 7.5, 6.6 and 5.8 respectively placements of 12, 24, 36, 48 and 60 inches away from the return grille respectively. At 200 cfm, the percentages of air by-pass in the office space were 12.5, 9.4, 7.5, 6.6 and 5.8 respectively placements of 12, 24, 36, 48

and 60 inches respectively. At 500 cfm, the percentages of air by-pass in the office space were 12.5, 7.8, 6.6, 6.6 and 4.6 respectively placements of 12, 24, 36, 48 and 60 inches respectively.

The rate of air by-pass for the office space with round diffusers was approximately half of that of the rate for the office space with square diffusers when placed 48 or less inches away from the return grille. At 60 inches away from the return grille, the rate of air by-pass was approximately the same for the two types of diffusers.

For an office at 350 cfm, the annual cost of replacing the lost energy due to the air by-pass was estimated to be 17.9, 15.4, 12.3, 10.8 and 9.5 dollars when a round diffuser was used at displacements of 12, 24, 36, 48 and 60 inches away from the return grille respectively. On the other hand, the annual cost of replacing the lost energy due to the air by-pass was estimated to be 38.4, 35.9, 28.2, 17.9 and 9.2 dollars when a square diffuser was used at displacements of 12, 24, 36, 48 and 60 inches away from the return grille respectively.

In conclusion, this investigation found that using round air diffusers is more efficient than using square air diffusers when placed 48 inches or less away from the return air grille. In addition, round diffusers have better air circulation throughout the entire zone, as they distribute air following a 360 degree pattern.

Future research needs to include investigating the relevance of different geometrical locations for the supply and return apertures with respect to each other (that is, not being placed adjacent to each other). Future studies need to

take into account factors like temperature, thermal effects, and zone obstacles. A real life application test of the results is also recommended.

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