Volume: 9 Issue: VIII Month of publication: August 2021

DOI: https://doi.org/10.22214/ijraset.2021.37780

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.429
Volume 9 Issue VIII Aug 2021- Available at www.ijraset.com

CFD Analysis of the Optimization of Length of Capillary Tube for a Vapor Compression Refrigeration System

Ms. K. P. Bhangle[1], Mr. M. M. Deshmukh[2]
[1]PG Student, [2]Professor, Mechanical Engineering, Government College of Engineering, Amrawati

Abstract: The capillary tube is commonly employed in refrigerant flow control systems. As a result, the capillary tube's performance is optimal for good refrigerant flow. Many scholars concluded performance utilising experimental, theoretical, and analysis-based methods. This paper examines the flow analysis of a refrigerant within a capillary tube under adiabatic flow circumstances. For a given mass flow rate, the suggested model can predict flow characteristics in adiabatic capillary tubes. In the current work, R-134a refrigerant has been replaced by R600a refrigerant as a working fluid inside the capillary tube, and the capillary tube design has been modified by altering length and diameter, which were obtained from reputable literature. The analysis is carried out using the ANSYS CFX 16.2 software. The results show thatutilising a small diameter and a long length (R-600a refrigerant flow) is superior to the present helical capillary tube. The most appropriate helical coiled design with a diameter of 0.8 mm and a length of 3 m is proposed.
Keywords: Capillary Tube, Condenser, Refrigeration effect, CFD.

I. INTRODUCTION

Capillary tubes have been studied extensively for decades. In small refrigeration and air-conditioning systems, a capillary tube is a popular expansion device. A capillary tube is a constant area expansion device positioned between the condenser and the evaporator in a vapor-compression refrigerationsystem, and its role is to lower the high pressure in the condenser to low pressure in the evaporator. Capillary tube expansion devices are commonly employed in refrigeration equipment, particularly in compact unitslike refrigerators, freezers, and air conditioners. The most significant reason to keep utilising it instead ofother extension devices is its simplicity. Capillaries are a cheaper and more complicated alternative to thermostatic valves. Capillary tubes, for example, are employed in certain complicated cooling systems for particle detectors. Other reasons for their use in highly specialized cooling circuits can be found, though.In reality, the flow through the capillary tube is adiabatic rather than isenthalpic. The refrigerant expands from high pressure to low pressure in an adiabatic capillary tube with no heat exchange with theenvironment. In many cases, the refrigerant enters the capillary as a sub-cooled liquid. The pressure lowerslinearly as the liquid refrigerant passes through the capillary owing to friction, but the temperature remainsconstant. A percentage of the liquid refrigerant flashes into vapour as the refrigerant pressure falls below the saturation pressure. The fluid velocity increases as the refrigerant's density decreases owing to evaporation. As a result, the whole length of the capillary tube seems to be split into two separate areas.

The liquid phase occupies the region near the entry, while the two-phase liquid vapour region occupies theother.

In this study, an attempt is made to examine the flow of refrigerant within a capillary tube of variouslengths and diameters under adiabatic flow circumstances. For a given mass flow rate, the suggested modelcan predict flow characteristics in adiabatic capillary tubes. In this study, R-600a was utilised as the workingfluid within the capillary tube, and the flow properties of the refrigerant were studied using the same modelin ANSYS CFX software. It is proposed to investigate the effects and applications of various capillary tubelengths and diameters in order to determine the most effective combination.

II. LITERATURE REVIEW

Abhinav Kulmitra et al.[1] has studied numerically the use of capillary tube in the mostly in the refrigerant flow control devices. As a result, the capillary tube's performance is optimal for good refrigerant flow. Analyze the flow of refrigerant within a capillary tube for adiabatic flow conditions in his study. Fora given mass flow rate, the author's suggested model can predict flow characteristics in adiabatic capillarytubes. In his study, R-22 has been replaced with ammonia as the working fluid inside the capillary tube, and the capillary tube architecture has been modified from straight to coiled. ANSYS CFX 16.2 software is used for the analysis. The results show that using a helical capillary tube improves the dryness fraction over using a straight and existing helical capillary tube.

Raja Y et al. [2] presents in investigation for attempt to analyze the flow Analysis of the refrigerantinside a straight capillary tube and coiled capillary tube for adiabatic flow conditions. For a given mass flow rate, the provided model predicts flow characteristics in adiabatic capillary tubes. In this study, R-22 was utilised as a working fluid within a straight capillary tube and a coiled capillary tube with a diameter of 1.27 mm, and the flow properties of refrigerant were studied using the same model in ANSYS CFX software. The results showed that the helical capillary tube performed better than the straight capillary tube in terms of dryness fraction. The best helical coiled design is recommended.

Chetan Kumar et al. [3] were worked to improve the Coefficient of Performance, It was to be required that the Refrigerating Effect be increased while the Compressor Work be reduced. The findings of an experimental investigation of a vapour compression refrigeration (VCR) system with ammonia refrigerant were described. The impacts of key performance factors such as superheating on the refrigerating effect, power required to run the compressor at various evaporation temperatures, refrigerant mass flow, % increase in COP, and coefficient of performance on the refrigerating effect (COP). The findings from a vapour compression refrigerant plant were examined, with factors such as compressor suction pressure, delivery pressure, evaporator and condenser temperatures documented and the coefficientof performance computed. CFD simulation was used to verify the results obtained. A diffuser has been added between the compressor and the condenser, reducing the power input to the compressor and therefore increasing the COP. CFD simulation was used to improve the findings; modelling and meshing will be donein ICEMCFD, analysis in CFX, and post-processing in CFD POST.

Buddha Chouhan et al [4] has used Low-capacity refrigeration devices, such as household refrigerators and window air conditioners, use capillary tubes as an expansion mechanism. The benefits ofthe capillary tube over alternative expansion devices are that it is simple, cheap, and allows the compressor to start at low torque during the off-cycle when the pressure across the tube equalises. The flow propertiesof refrigerants via capillary tubes have been widely investigated both experimentally and analytically over the last six decades, with the majority of these studies focusing on straight capillary tubes. The impact of important factors on the flow characteristics of R134a and R-22 running through adiabatic helical tubes areexperimentally investigated in this thesis. The diameter of the capillary tubes, coil diameter, and flow- related factors such as intake pressures and degree of subcooling were the main parameters examined. Theheat transfer rate, pressure drop, velocity, mass flow rate, and heat transfer coefficient for the fluids R134Aand R-22 with various tube and coil sizes are determined using CFD analysis in this thesis. The purpose ofthermal analysis for copper and aluminium tube materials is to determine the temperature distribution and heat flux. Pro-engineer is used for 3D modelling, while ANSYS is used for analysis.

Narendra Yadav et al. [5] Capillary tubes are commonly employed in refrigerant flow control systems. As a result, the capillary tube's performance should be enough for smooth refrigerant flow. Manyresearchers investigated these topics both experimentally and analytically. The current study examines theflow of refrigerant inside a capillary tube under adiabatic flow conditions. For a given mass flow rate, the suggested model has predicted flow characteristics in adiabatic capillary tubes. In the current study, R-22 has been replaced by ammonia as the working fluid within the capillary tube, and the capillary tube designhas been modified from straight to coiled, as per reputable literature. ANSYS CFX 16.2 software is used for the analysis. The results show that utilising a helical capillary tube (Ammonia refrigerant flow) improves the dryness fraction over using a straight and existing helix capillary tube (R22 refrigerant flow). The besthelical coiled design is recommended.

Oyedepo et. al. [6] has conducted an experimental comparison of R600A and LPG performance ina home refrigerator with variable refrigerant charge (wr) and capillary tube length (L). REFPROP software was used to calculate the enthalpy of the refrigerants R600A and LPG for each data set under the experimental circumstances (version 9.0). The results demonstrate that with a refrigerant charge of 60 g of LPG and a 1.5 m capillary tube length, the design temperature and pull-down time for a compact refrigeratordefined by ISO may be attained earlier. The greatest COP (4.8) was achieved with a 60-g LPG charge anda 1.5-m L. The average COP achieved with LPG was 1.14 percent greater than with R600A. R600A has the lowest power usage based on the results of the electric power consumption test. When compared to the LPG in the system, the compressor used 20% less electricity. In terms of COP and cooling capacity, LPG was the best, whereas R600A was the best in terms of power consumption.

Reddy et al. [7] present work the experimental examination of a domestic refrigerator was computed by varying the refrigerant charge (mr) and capillary tube length at the same time (Lc). Due to the growing necessity and relevance of the climate change consequences faced by humanity, a recent literatureanalysis reveals consistent development in the field of novel refrigerants for use in refrigeration and air conditioning systems. CFCs have been gradually replaced by HFCs, and then by a combination of HFCs, in recent decades. However, HFCs' high GWP allowed for the creation of novel refrigerants based on hydrocarbons or hydrocarbon blends.

The experimentation on the household refrigerator employing two distinct hydrocarbon mixtures, namely R550/50(R290/R600a) and R436A with mass fraction of 56/44 of R290 and R600a, is presented in this work. The tests are carried out by varying the refrigerant charge (mr) and capillary tube length (LC) at various evaporator temperatures (Te). Under steady state conditions, the refrigerator is subjected to continuous sequential and cycle testing. The pressure at the compressor's input and exit is recorded and analysed, and temperature readings are taken at different points throughout the refrigerator. In addition, the draw down time for the three distinct refrigerants is recorded for altering refrigerant charge and capillary length, and graphs are displayed. The findings reveal that with 70g of R436A with L=4.5M, a design temperature of -150C (according to IS1476 part1), and a pull-down time of90 minutes can be obtained for CARE, and for R134a, a pull-down time of 100 minutes can be reached with 70g and L=5m. The performance of the R 436A refrigerant has demonstrated that it may be a good alternative for R134a.

Sunu et al. [8] The performance of a water chiller air conditioning simulation with a thermostatic expansion valve (TEV) is compared to that of a capillary tube. The same charge of refrigerant was used inthe water chiller system. Comparative evaluations were conducted using the refrigeration system's coefficient of performance (COP) and performance parameters at a medium cooling load level, with an ambient temperature of 29-31oC, constant compressor speed, and a fixed chilled water volume flowrate of15 lpm. It was discovered that the TEV system consumed less energy than the capillary tube system. Whencomparing the thermostatic expansion valve system to the capillary tube system, the thermostatic expansion valve system had a better coefficient of performance (21.4%).

Rocha et al. [9] An algebraic solution was provided for studying adiabatic coiled capillary tubes inthe transcritical CO2 cycle. With a total of nine possibilities, this solution tested three distinct friction factors and three possible k factors (connected to the specific volume). A correlation was also developed for estimating sand grain roughness. Experimental tests were performed with two distinct capillary tubes in a total of 60 locations to confirm the algebraic solution and to be beneficial for future investigations. The findings revealed that 95 percent and 98.3 percent of the projected mass flow rates, respectively, fell within 10% and 15% error ranges, with average deviation (AD) and absolute average deviation (AAD) of 0.1 percent and 4.4 percent, respectively. The C-M&N (M&N modified) friction factor fared better overall, but the Schmidt friction factor was the only one that could accurately anticipate 100% of data within a 12% error range. For the coiled shape, an algebraic equation and a dimensionless correlation for straight capillary tubes were also modified, and the results were adequate. It is possible to infer that the offered solution is an adequate tool for this application. An algebraic solution and experimental data for coiled capillary tubes in the trans-critical CO2 cycle are provided for the first time, to the best of the authors' knowledge.

Zareh et al. [10] offers a drift flux model for steady state refrigerant flow via horizontal lateral straight capillary tube-suction line heat exchangers using R134a as the working fluid. A capillary tube's geometry is separated into two parts: an entry length and a lateral heat exchanger length. In this study, the metastable flow phenomena is ignored. Three modes were examined and the fundamental equations were solved for all three modes, then the results were evaluated, in order to better understand fluid behaviour in non- adiabatic capillary tubes, according to the refrigerant condition at the entry of the heat exchanger area. The current results, which were obtained using a drift flux model, were compared to prior numerical work using a homogeneous model and found to be in reasonable accord. Because the slip ratio effects in the drift flux model were taken into account, this model was able to predict refrigerant flow behaviour under non- adiabatic flow conditions through a capillary tube more precisely than previous two-phase flow model.

III. NUMERICAL METHOD

The new results were compared to previous numerical work employing a homogeneous model and found to be in reasonable agreement. This model was able to predict refrigerant flow behaviour under non- adiabatic flow circumstances through a capillary tube more precisely than prior two-phase flow models because the slip ratio effects in the drift flux model were taken into consideration.

A. Governing Equations

The new results were compared to prior numerical work using a homogeneous model, and they were found to be reasonably consistent. Because the slip ratio effects in the drift flux model were taken into account, this model was able to predict refrigerant flow behaviour under non-adiabatic flow conditions through a capillary tube more precisely than previous two-phase flow models.

The natural convection flow under investigation was modeled by a set of partial differential equations describing the conservation of mass, momentum and energy in three dimensional Cartesian coordinate system.

1) Conservation of Mass

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0 \qquad \text{…Equation 1}$$

2) Conservation of Momentum

$$\frac{\partial(\rho u^2)}{\partial x}+\frac{\partial(\rho uv)}{\partial y}+\frac{\partial(\rho uw)}{\partial z}=-\frac{\partial(p_0)}{\partial x}+\rho\nu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right) \qquad \text{…Equation 1}$$

$$\frac{\partial(\rho uv)}{\partial x}+\frac{\partial(\rho v^2)}{\partial y}+\frac{\partial(\rho uw)}{\partial z}=-\frac{\partial(p_0)}{\partial y}+\rho\nu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)+g(\rho-\rho_0) \qquad \text{…Equation 3}$$

$$\frac{\partial(\rho uw)}{\partial x}+\frac{\partial(\rho vw)}{\partial y}+\frac{\partial(\rho w^2)}{\partial z}=-\frac{\partial(p_0)}{\partial x}+\rho\nu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \qquad \text{…Equation 2}$$

3) Conservation of Energy

$$\frac{\partial(\rho uT)}{\partial x}+\frac{\partial(\rho vT)}{\partial y}+\frac{\partial(\rho wT)}{\partial z}=\frac{k}{\rho C_p}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \qquad \text{…Equation 3}$$

Where, ρ is the variable density, ρ_0 the reference density, P_0 the operating pressure and ν is the kinematic viscosity.
The numerical model is based on a control volume formulation. The above equations are integrated over each control volume to obtain a set of discretized linear algebraic equations of the form:

$$a_p\phi_p=\sum a_{nb}\phi_{nb}+b \qquad \text{… Equation 4}$$

4) Standard k-ε Model

The standard k-ε model is a two-equation turbulence model proposed by Launder and Spalding. It is a high-Reynolds number model. The two terms on which the model is based are the turbulent kinetic energy (k) and dissipation rate of turbulence kinetic energy (ε). The turbulent kinetic energy (k) is derived from an exact equation by assuming that the effect of the molecular viscosity of the flow is negligible while the turbulence dissipation rate (ε) is obtained by employing physical reasoning. The transport equations, turbulent viscosity and model constants for the standard models.

Turbulence kinetic energy equation:

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)*\frac{\partial k}{\partial x_j}\right]+G_k+G_b-\rho\varepsilon \qquad \text{….Equation 7}$$

Dissipation rate of turbulence kinetic energy equation:

$$\frac{\partial}{\partial t}(\rho\varepsilon)+\frac{\partial}{\partial x_i}(\rho\varepsilon u_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)*\frac{\partial\varepsilon}{\partial x_j}\right]+C_{1\varepsilon}\frac{\varepsilon}{k}*(G_k+C_{3\varepsilon}G_b)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \qquad \text{…Equation 8}$$

where $G_k$is the generation of turbulence kinetic energy due to the mean velocity gradients; G_b is the generation of turbulence kinetic energy due to buoyancy; $C_{1\varepsilon}$, $C_{2\varepsilon}$ and $C_{3\varepsilon}$ are model constants; the quantities σ_k and σ_ε are the turbulent Prandtl numbers for k and **ε** respectively.

Turbulent viscosity is given as:

$$\mu_t=\rho C_\mu\frac{k^2}{\varepsilon} \qquad \text{….Equation 9}$$

The setup was modeled for only entire fluid domain was used. It is logical to assume part of fluid domain instead of whole domain that saves computation time. Domain has to build by considering inside fluid volume only.

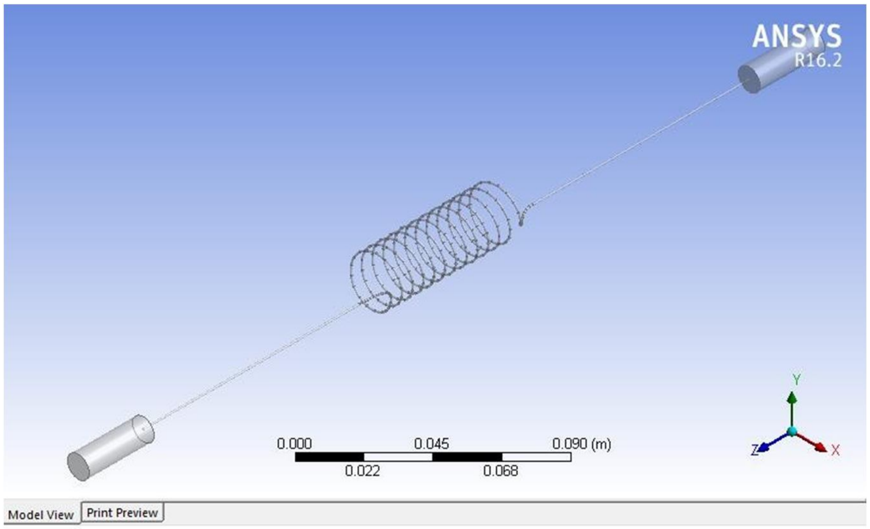


Figure 1 Geometry used for computational analysis

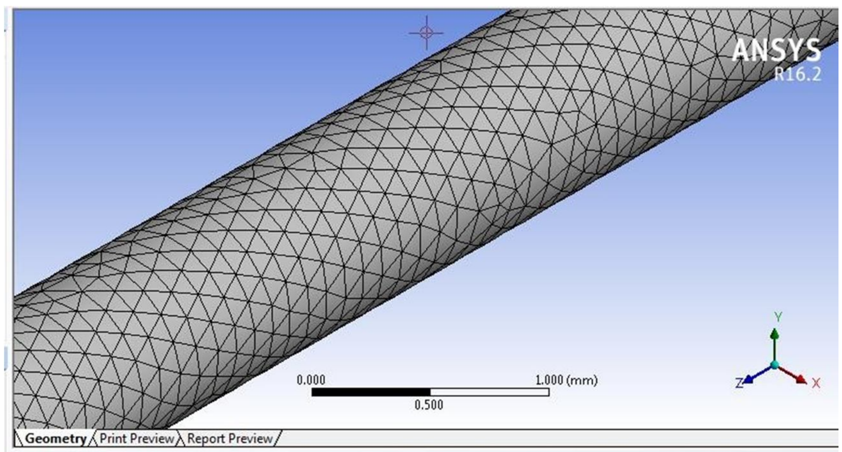


Figure 2 Meshing

Table 1: Mesh Statistic

Parameter	0.8mm * 1 m	0.8mm * 2 m	0.8mm * 3 m	1mm * 1 m	1mm * 2 m	1mm * 3 m	1.2mm * 1 m	1.2mm * 2 m	1.2mm * 3 m
Relevence Center	Fine	Fine	Fine	Fine	Fine	Fine	Fine	Fine	Fine
Type of Mesh	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral	Tetrahedral
No. of Node	422906	832635	113451	342319	674079	112039	282142	564865	120399
No. of Element	1907303	3766408	5372920	1544493	3043221	5402932	1281742	2548654	5628291

IV. BOUNDARY CONDITIONS

The inclusion of additional source and/or sink terms in the finite volume formulation for computational cells at the boundaries implements all boundary conditions. There is experimental velocity in force convection flows. The temperature field causes the velocity field to evolve, which in turn impactsthe temperature field, because the governing equations are invariably connected. The inlet is given a temperature value; this temperature value is determined via a laboratory experiment. All of the walls in this study are considered to be in no-slip condition.

Table 2: Boundary condition

Model	Domain	Boundary Condition	Parameter
Viscous (K-Epsilon)& Energy	R600a	Inlet	Inlet Pressure and Temperature (8 bar, 313K)
		Wall Surface	Heat Flux (0 W/m^2)
		Outlet	Pressure Outlet (Atmosphere)

Following boundary condition applied to model:

1) The inlet condition of tube is constant for all the tubes
2) Heat flux applied is zero as the condition is adiabatic.
3) All remaining wall are maintained at no slip condition.

V. RESULTS AND DISCUSSION

CFD analysis is completed at different diameter and length and fixed values of heat flux. These results are given in below. Temperature counters over the domain surface give a clear idea of the drop in temperature after flow through the tube. Pressure counter over the domain also gives information about the drop in pressure over the length. The refrigeration characteristics are presented in below section by variationin length of capillary tube and below section presents the effect of variation in diameter on COP, Refrigeration effect, temperature and pressure outlet.

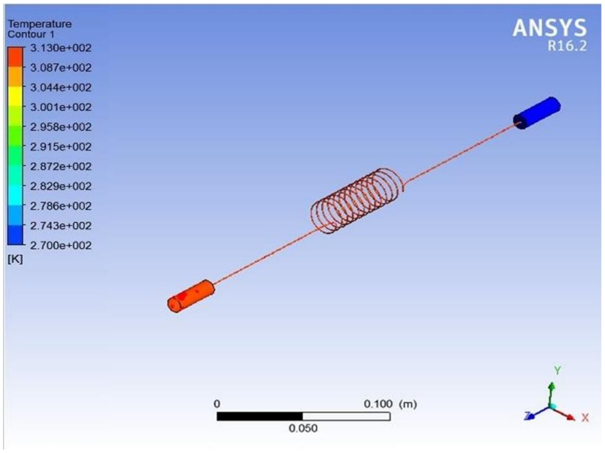
By considering the experimental results it has been observed that the inlet temperature of capillarytube is 40^0C and pressure if 8bar and tube was adiabatic. By keeping same condition as a boundary conditionfor the simulation of the capillary tube the analysis has done for the varying diameter and length so as to optimize the dimensions of capillary tube and gives maximum output. Following is the observation table from the computational results coming from the analsyis.

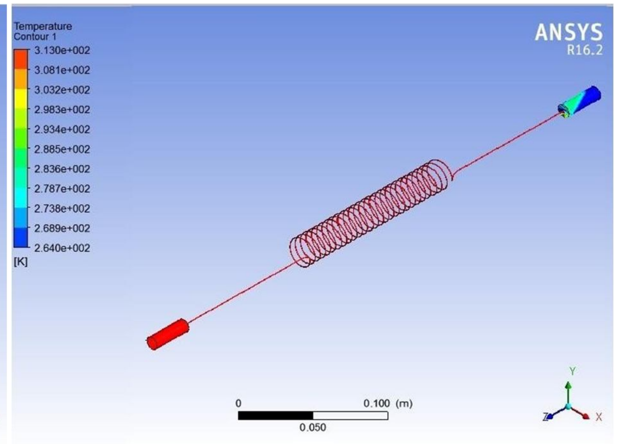
Table 3 Observation Table of Computational results

Expmental	CASE I	CASE II	CASE III	CASE IV	CASE V	CASE VI	CASE VII	CASE VIII	CASE IX
Dia	0.8	0.8	0.8	1	1	1	1.2	1.2	1.2
Length	1	2	3	1	2	3	1	2	3
Pin	8	8	8	8	8	8	8	8	8
Tcapin	313	313	313	313	313	313	313	313	313
Pout(bar)	-0.00617	-0.05003	-0.1198	0.0224	-0.0342	-0.0846	0.0326	-0.0284	-0.0459
Tcapout(K)	270	264	260	270	268	263	271	269	264
Tcin	284	284	284	284	284	284	284	284	284
Tcout	334	334	334	334	334	334	334	334	334
h1	574.5	574.7	575	574.4	574.6	574.9	574.4	574.6	574.7
h2	638.2	638.2	638.2	638.2	638.2	638.2	638.2	638.2	638.2
h4	195.1	186	171.2	195	192.1	184.6	196.5	193.7	186
RE	379.3	388.7	403.8	379.3	382.5	390.2	377.8	380.9	388.7
Wcom P	63.68	63.47	63.17	63.78	63.54	63.32	63.83	63.56	63.48
COP	5.9572	6.1244	6.3925	5.9482	6.0198	6.1628	5.9188	5.9931	6.1235

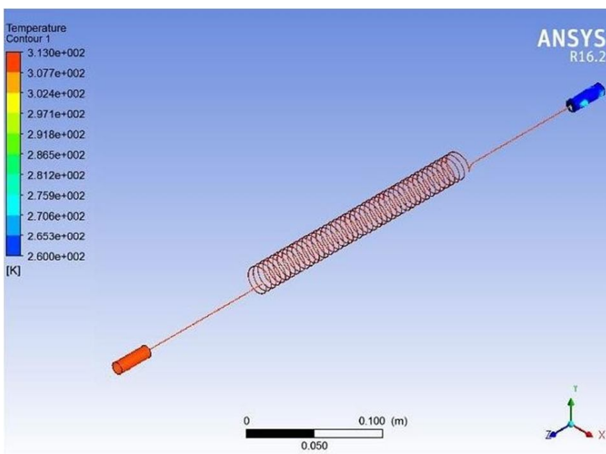
A. *Effect of Temperature*

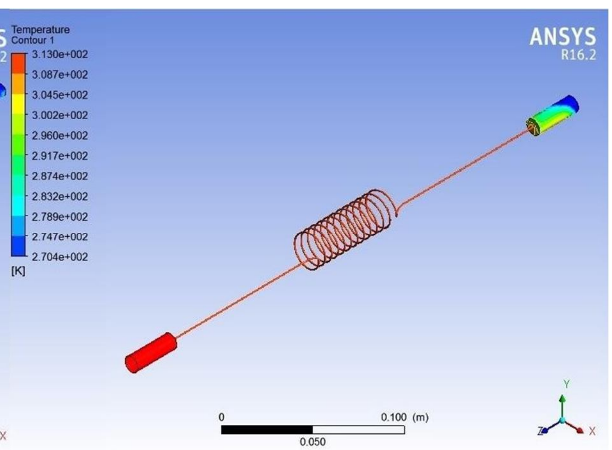
Temperature contour on outlet wallare shown in figure 3 for condenser tube for diameter of 0.8mm, 1mm and 1.2mm and length 1m, 2m and 3m. Maximum temperature drop of 49K obtain for the diameter 0.8mm and 3m length. Percentage improvement in temperature drop as compared previous is 22%.



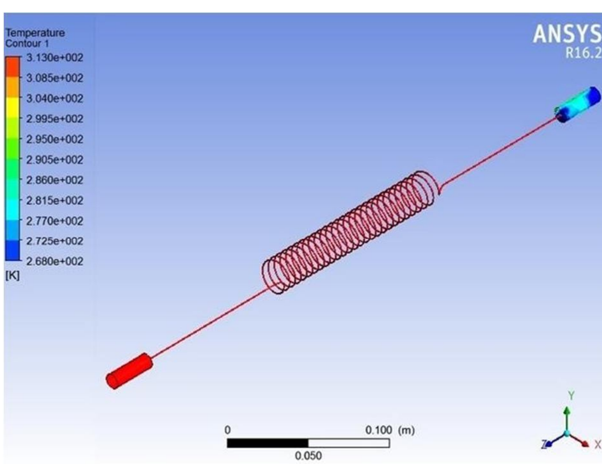


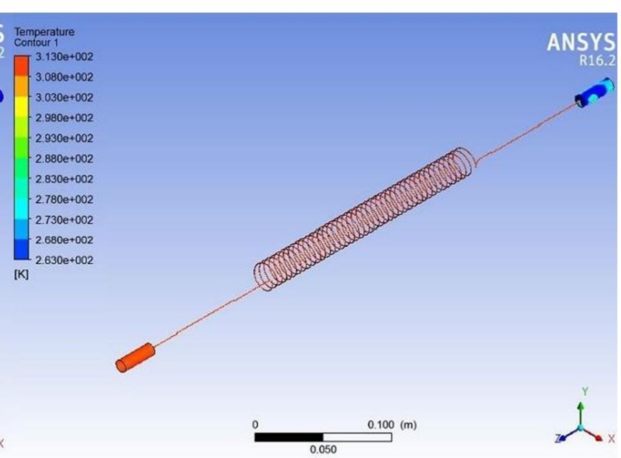
a) b)



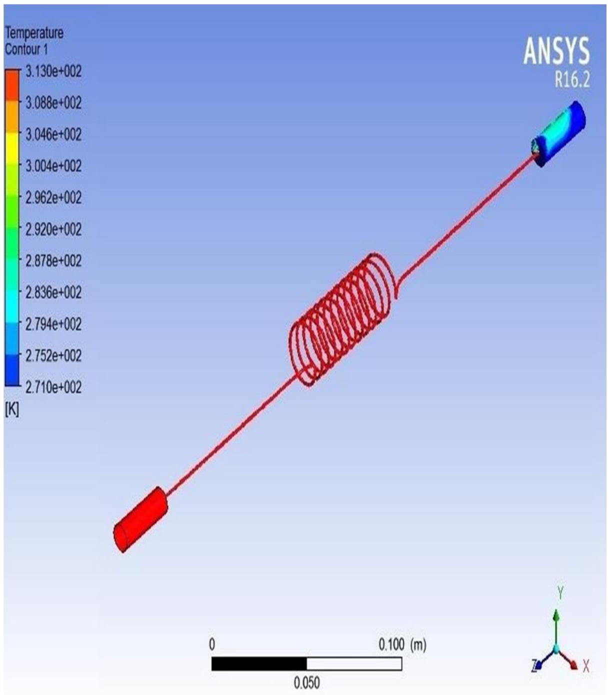


c) d)

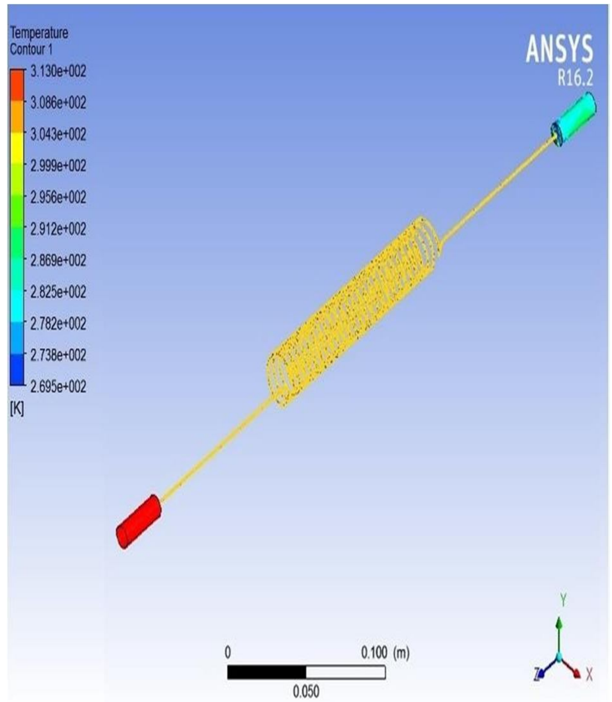




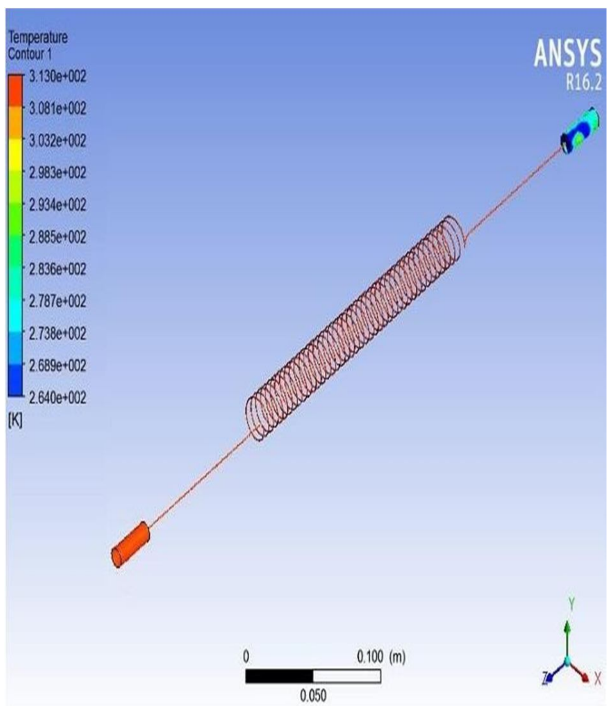
e) f)



(g)



(h)

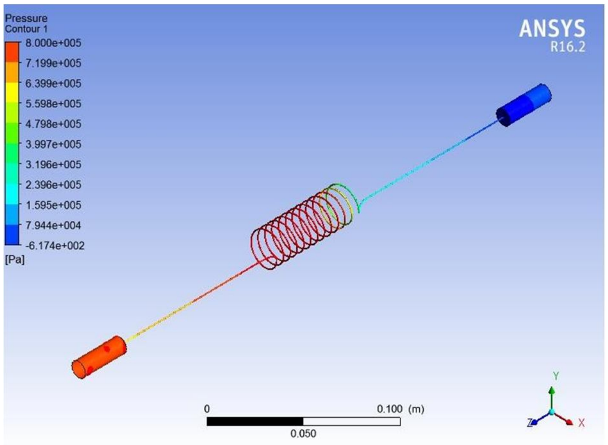


(i)

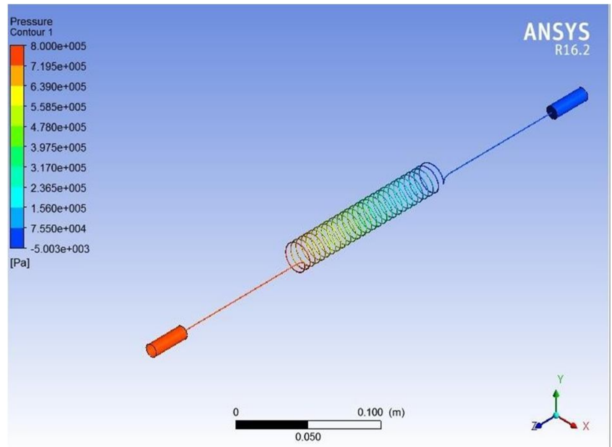
Figure 3 Temperature contour at profile (a) The diameter of 0.8mm and length 1 meter (b) Thediameter of 0.8mm and length 2 meter (c) The diameter of 0.8mm and length 3 meter (d) Thediameter of 1mm and length 1 meter (e) The diameter of 1mm and length 2 meter (f) The diameter of 1mm and length 3 meter (g) The diameter of 1.2mm and length 1 meter (h) The diameter of 1.2mm and length 2 meter (i) The diameter of 1.2mm and length 3 meter

B. Pressure Profile

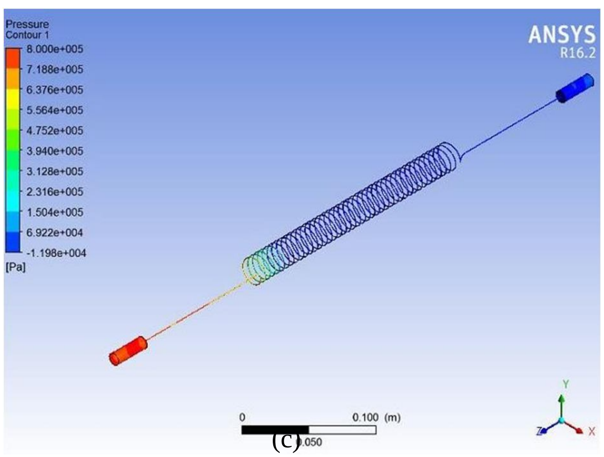
Pressure contour on outlet wall are shown in figure 4 for condenser tube for diameter of 0.8mm, 1mm and 1.2mm and length 1m, 2m and 3m. Minimum pressure obtain for -6231 Pa obtain for the diameter0.8mm and 3m length.



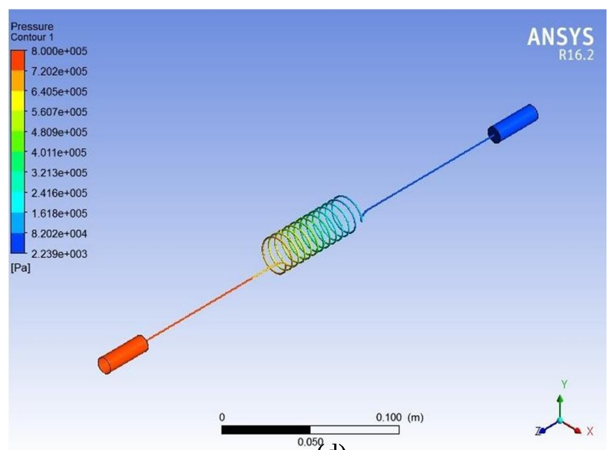
(a)



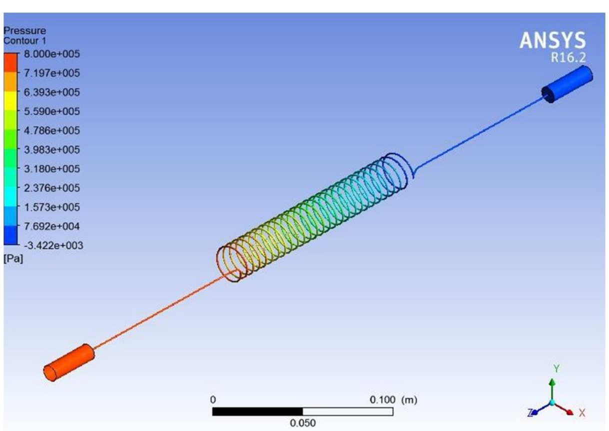
(b)



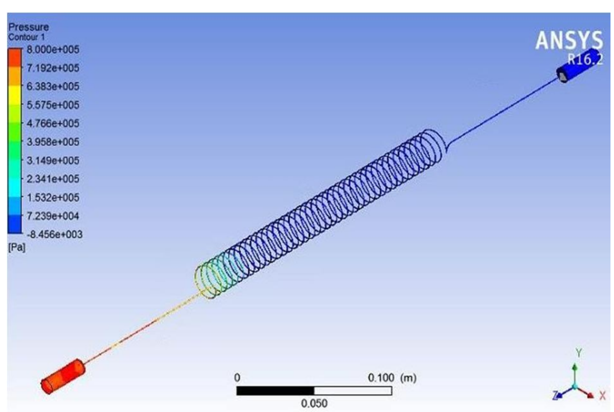
(c)



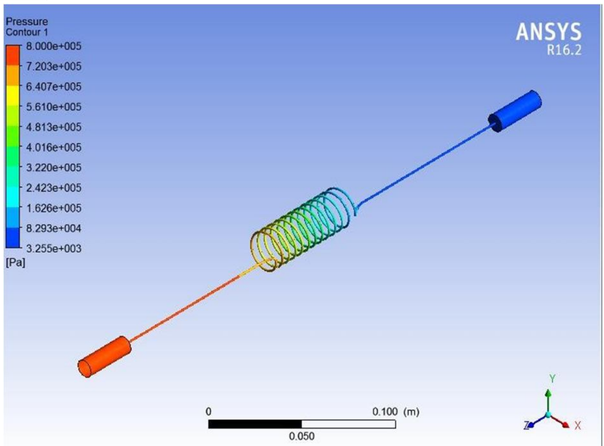
(d)

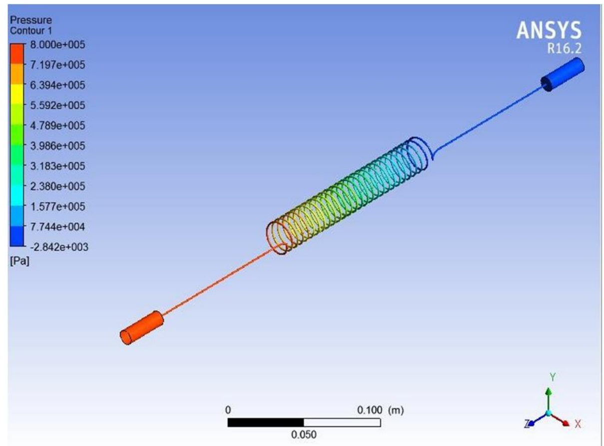


(e)

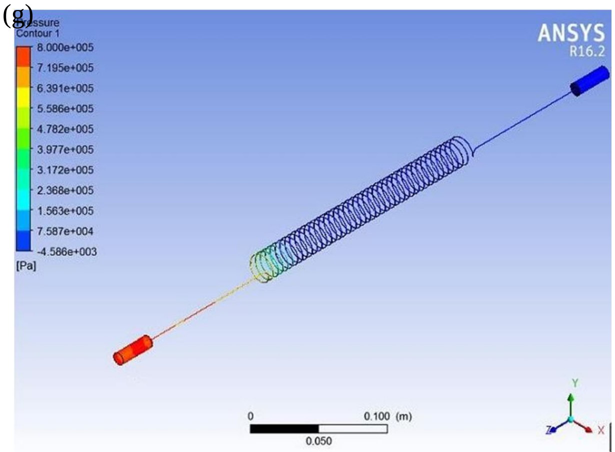


(f)





(g) (h)



(i)

Figure 4 Pressure contour at profile (a) The diameter of 0.8mm and length 1 meter (b) The diameter of 0.8mm and length 2 meter (c) The diameter of 0.8mm and length 3 meter (d) Thediameter of 1mm and length 1 meter (e) The diameter of 1mm and length 2 meter (f) The diameter of 1mm and length 3 meter (g) The diameter of 1.2mm and length 1 meter (h) The diameter of 1.2mm and length 2 meter (i) The diameter of 1.2mm and length 3 meter.

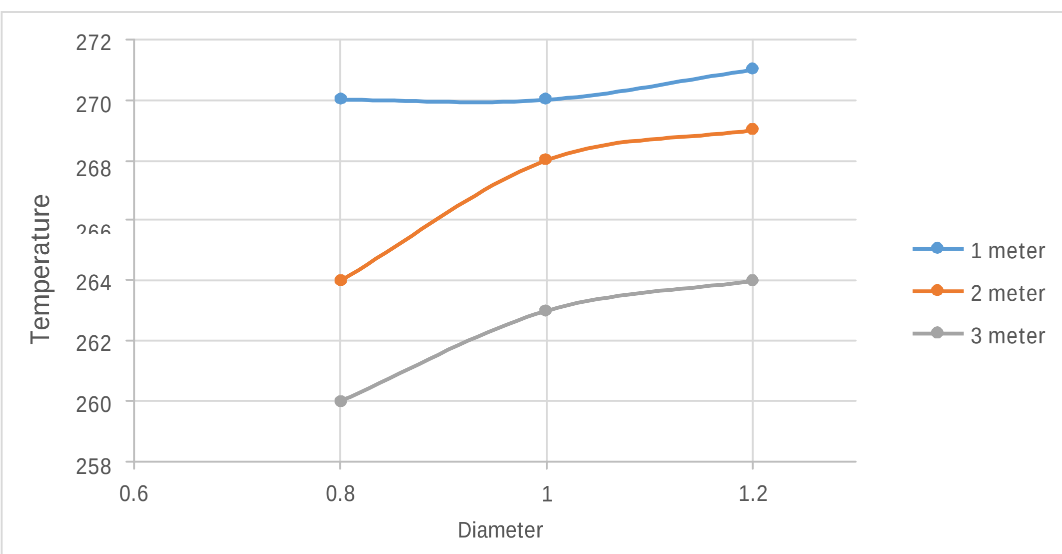


Figure 5 Effect on outlet Temperature of Capillary tube by varying diameter and length

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.429
Volume 9 Issue VIII Aug 2021- Available at www.ijraset.com

The analysis are performed with different length and diameter of capillary tube. As shown in abovefigure the temperature of capillary tube are studied. It has been found that temperature of R600a is increasing as diameter increases and length decreases of capillary tube. Minimum output temperature can be found in a R600a for combination of 0.8 mm diameter and 3 meter length.

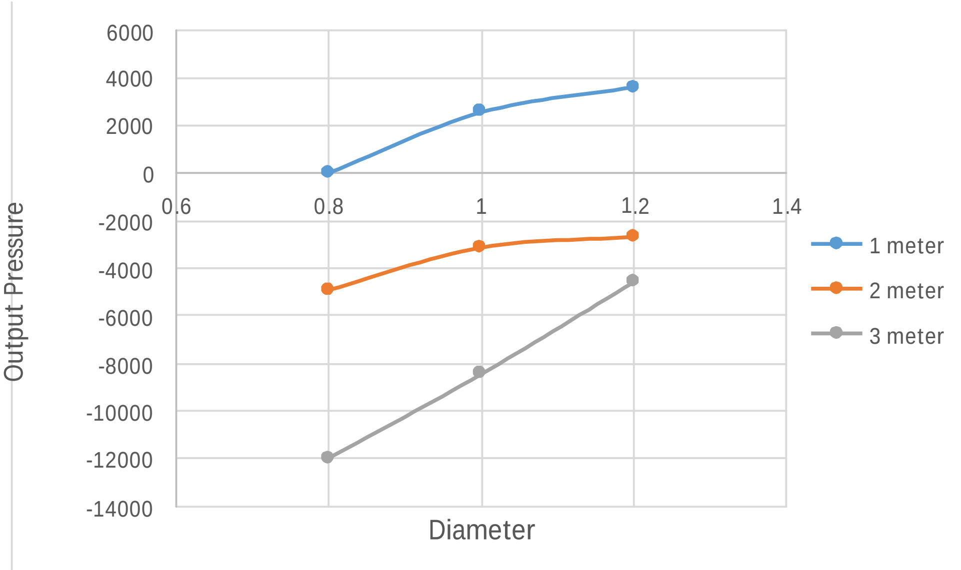


Figure 6 Effect on outlet Pressure of Capillary tube by varying diameter and length

The analysis are performed with different length and diameter of capillary tube. As shown in above figure the Output pressure of capillary tube are studied. It has been found that output pressure of R600a isincreasing as diameter increases and length decreases of capillary tube. Minimum output pressure can be found in a R600a for combination of 0.8 mm diameter and 3 meter length.

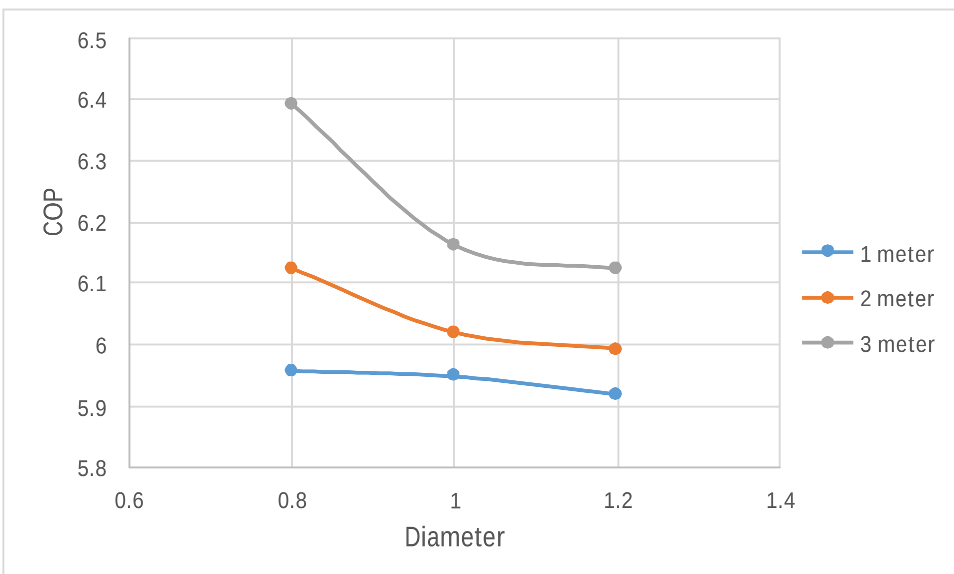


Figure 7 Effect on outlet Temperature of Capillary tube by varying diameter and length

The analysis are performed with different length and diameter of capillary tube. As shown in abovefigure theCOP of capillary tubeare studied. It has been found that COP of R600a is decreasing as diameterincreases and length decreases of capillary tube. Maximum COP can be found in a R600a for combinationof 0.8 mm diameter and 3 meter length.

VI. CONCLUSION

Numerical analysis of capillary tube was carried out with varying diameter and length. The following conclusions may be drawn from the present investigation. The effect of dimensions of the capillary tube on the refrigeration system can be seen in the simulation. Average temperature drop from the tube increased with decreasing diameter of tube by keeping length as a constant parameter. Average temperature drop from the tube increased with increasing length of tube by keeping diameter as a constant parameter. From the overall results it has been conclude for the temperature parameter is the minimum value of diameter and maximum length gives the maximum temperature drop in this case the values are 0.8mm dia and 3 m length. Average pressure drop from the tube increased with decreasing diameter of tube by keeping length as a constant parameter. Average pressure drop from the tube increased with increasing length of tube by keeping diameter as a constant parameter. From the overall results it has been conclude for the pressure parameter is the minimum value of diameter and maximum length gives the maximum pressure drop in this case again the values are 0.8mm dia and 3 m length. The maximum refrigeration effect obtain i.e. 403.82 watts for the same combination. The Maximum COP obtain i.e. 6.39 for the domestic refrigeration system for the same combination.

REFERENCES

[1] Kulmitra, "Ijesrt International Journal of Engineering Sciences & Research Technology Numerical Investigation of Capillary Tube By Replacing the Inside Refrigerant and Diameter,"Int. J. Eng. Sci., vol. 5, no. 7, pp. 102–114, 2016, [Online]. Available: http://www.ijesrt.com.

[2] Y. R. Kumar and P. U. Sri, "CFD Flow Analysis of a Refrigerant inside Adiabatic CapillaryTube," vol. 2, no. 9, pp. 485–493, 2013.

[3] Kumar, "Enhancement of Coefficient of Performance by Analysis of Flow through Vapour Compression Refrigeration cycle using CFD," vol. 2, no. 6, pp. 66–72, 2015, doi:10.17148/IARJSET.2015.2616.

[4] Chouhan, P. P, and S. Kumar, "Analytical Investigation of R134a Flowing Through AdiabaticHelically Coiled Capillary Tubes," Iarjset, vol. 4, no. 2, pp. 139–144, 2017.

[5] Kulmitra, N. Yadav, A. Tiwari, and A. Dewangan, "Comparative Study of Capillary Tube by Changing Refrigerant and Diameter of Tube by Experimental and CFD Analysis," vol. 5, no. 10,pp. 487–490, 2017.

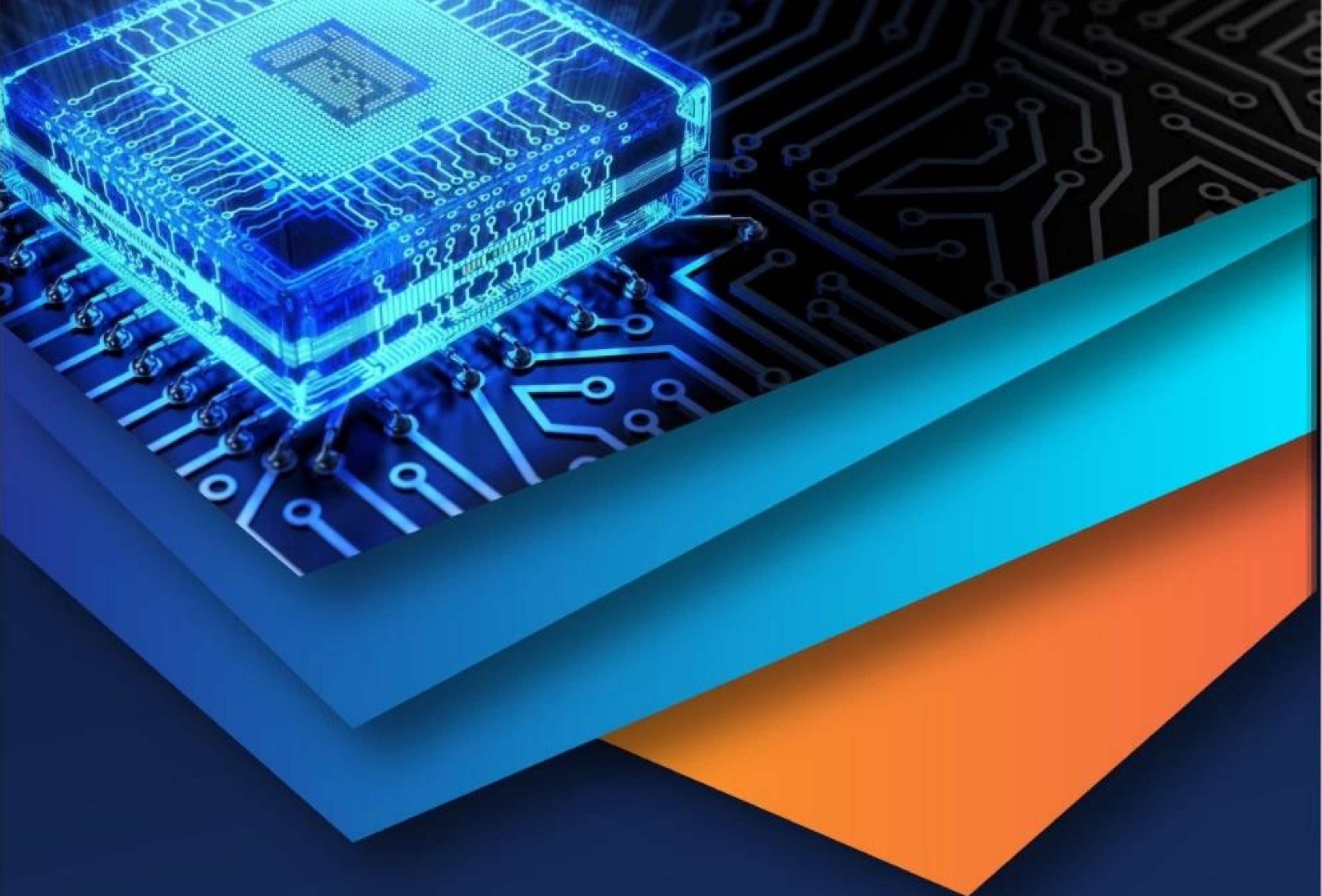
[6] S. O. Oyedepo et al., "A comparative experimental study on performance of domestic refrigeratorusing R600A and LPG with varying refrigerant charge and capillary tube length," 30th Int. Conf. Effic. Cost, Optim. Simul. Environ. Impact Energy Syst. ECOS 2017, vol. 18, no. 4, pp. 287–302, 2017.

[7] V. R. Reddy, P. Bhramara, and K. Govindarajulu, "Experimental Evaluation of the Effect of Refrigerant Charge and Capillary Tube Length on the Performance of Household Refrigerator with Different Configurations of R290 and R600a," Mater. Today Proc., vol. 5, no. 5, pp. 11845–11852, 2018, doi: 10.1016/j.matpr.2018.02.155.

[8] P. W. Sunu, I. Made Rasta, D. S. Anakottapary, I. Made Suarta, and I. D. M. Cipta Santosa, "Capillary Tube and Thermostatic Expansion Valve Comparative Analysis in Water Chiller Air Conditioning," J. Phys. Conf. Ser., vol. 953, no. 1, 2018, doi:10.1088/1742-6596/953/1/012063.

[9] T. Torres Martins Rocha, C. Henrique de Paula, J. Jose Garcia Pabon, T. de Freitas Paulino, and R. Nunes de Oliveira, "Algebraic solution and experimental validation for adiabatic coiled capillary tubes operating in transcritical CO2 cycle," Appl. Therm. Eng., vol. 181, no. July, p.115930, 2020, doi: 10.1016/j.applthermaleng.2020.115930.

[10] M. Zareh, M. Khayat, and H. Fouladi, "Numerical simulation of two phase refrigerant flow through non-adiabatic capillary tubes using drift flux model," J. Mech. Sci. Technol., vol. 32, no.1, pp. 381–389, 2018, doi:10.1007/s12206-017-1238-2.



10.22214/IJRASET

INDEX COPERNICUS 45.98

IMPACT FACTOR: 7.129

IMPACT FACTOR: 7.429

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY