Numerical study of a double inlet chamber counter flow vortex tube with insulation

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Abstract. The present study intends to improve the performance of the Ranque-Hilsch counter flow vortex tube, analysed using computational fluid dynamics. In the axisymmetric 3-D, steady-state, compressible, and turbulent flow vortex tube, the air has been used as the working fluid. The ANSYS17.1 FLUENT software has been used with the standard $k$-$\varepsilon$ turbulent model for different mass fraction of cold fluid and inlet pressure in the numerical simulation and validated with the experimental results. It is observed from the study that as the inlet chambers number increases from 1 to 2, there is a decrease of 7.8 % in the cold exit temperature of the vortex tube. However, insulating the double chamber vortex tube leads to a further reduction of 4.2% in the cold exit temperature. Therefore, it indicates that the overall decline in the cold exit temperature from one chamber non-insulated vortex tube to double chamber insulated vortex tube is 9.6%. In terms of cold exit temperature, it can be concluded that using a double inlet chamber vortex tube with insulation yields the optimum results.

1. Introduction
The vortex tube (VT), which has no moving mechanical part, is a flow separator device. The working fluid enters tangentially to increase the swirling effect and at a higher pressure through inlet nozzles. It breaks up the inlet fluid into two natures of the fluid, one near to centreline is at low temperature and obtained through the cold exit, and the other near the periphery, which is at a higher temperature than inlet fluid obtained through the hot exit. The mass flow rate in the VT is controlled by adjusting the throttle valve at the hot exit. The vortex tube has a wide variety of applications in refrigeration, manufacturing, spot cooling of brazed and soldered components, cooling of electronic components, and gas liquefaction. It has high potential in future with modifications to replace the conventional refrigeration and air conditioning system.

The VT was first discovered by Ranque[1]. Many efforts were made ever since by researchers to improve its efficiency. Skye et al. [2] numerically and experimentally investigated the vortex tube and found that the maximum performance of VT obtained for 0.65 of the mass fraction of cold fluid. Liu and Liu [3] investigated a 3D VT model to compare the different turbulence models by considering the strong swirling effect and fluid compressibility characteristic. They observed that standard turbulence models and RNG $k$-$\varepsilon$ are promising in the computational study. Moraveji et al. [4] have investigated a 3-dimensional vortex tube model computationally for the inlet, cold orifice diameter, and different L/D ratios. They concluded that the rate of flow increased as diameter increases from the cold exit. By increments in the VT length, there is a slight increase in the rate of flow from the cold and hot outlet. Manimaran [5] uses CFD to determine how the inlet nozzles number affects the flow behaviour in the VT and concluded that by increasing the nozzle number, the vorticity and turbulent kinetic energy between the core and peripheral fluid decreases. Acar et al. [6] have studied the vortex tube and proposed a new electrical dryer design (RHVTAD) Ranque-Hilsch vortex tube aided drying. They
explain the efficiency and economic feasibility of the new design in comparison to the present drying system under the same working condition. Kaya et al. [7] has experimented on the two vortex tube with parallel connections, the different number of the nozzle and three different nozzle materials namely aluminium, polyamide plastic, and brass. They suggested that six inlets nozzle and aluminium as the working material shows the optimum performance in terms of cold outlet temperature. Baghdad et al. [8] have accessed the effect of the conductive and non-conductive wall and the kinetic energy of the vortex tube computationally. They proposed the general correlation for the vortex tube design. Xue et al. [9] examined the flow behaviour of fluid inside the vortex tube experimentally by Particle Image Velocimetry (PIV) measurement technique. Guo et al. [10] performed the experimental studies on the transparent counter flow VT with 2D laser Doppler velocimetry (LVD) technique to investigate the mean flow field, radial and axial velocities with different cold mass fraction and mean tube length and proposed the optimum design of VT. Qyyum et al. [11] proposed the optimization of VT considering the throttle valve shape, geometry, and different inlet pressure. As per the literature, the computational result of the truncated shape throttle value with optimized geometry performs better. Li et al. [12] experimentally investigated how the mass fraction of cold fluid affects the other parameters in the VT with the help of a five-hole probe. The static temperature, pressure, total temperature, and axial velocity variation has been explained with different value of the mass fraction of cold fluid. Aghagoli et al. [13] explained the effect of CO2 when used as working fluid in VT computationally. The performance of the VT in terms of hot and cold exit temperature, the performance characteristics curve has been explained in detail at different cold mass fractions and inlet pressure. Yan et al. [14] considered the valveless convergent VT and a straight vortex tube at the same working condition in terms of energy separation and temperature variation. They found that the valveless convergent vortex tube is more efficient than the straight vortex tube. Rafiee et al. [15] experimental studies show the performance enhancement of the VT by placing a device known as a fluid navigator in the inlet chamber, which guides the fluid inside the main pipe. They observed that by varying the angle of the navigator, there is an improvement in the cold and hot exit temperature value. Thakare et al. [16] explained the vortex tube (VT) performance experimentally using five different materials, namely Copper, Brass, Aluminum, Mild steel, and PA6, for the main tube of the VT. They conclude that PA6 material shows a promising result by 10.47% increment in terms of cooling energy than aluminium and holds the consistency in the result even at a higher pressure and different working condition. Kumar et al. [17] has examined the relative humidity (RH) separation and cold exit temperature in an insulated and non-insulated vortex tube experimentally. They concluded that with insulation, there is an increase of 2% in the relative humidity and the cold exit temperature separation increases slightly in the case of the insulated vortex tube. AlSaghir et al. [18] studied the temperature and velocity field numerically for the VT and reported the location of the free and forced vortex, respectively. The study reveals the flow parameter dependency on the mass fraction of cold fluid rate.

In the present study, the focus is to attain the following objectives:

a) To obtain the 3-dimensional vortex tube with solid works software for the numerical simulation with the ANSYS17.1 FLUENT software using the standard k-ε turbulent model.
b) Study of vortex tube with single inlet chamber, double inlet chamber, and double inlet chamber with insulation.
c) To obtain the best possible vortex tube between a single inlet chamber, double inlet chamber, and double inlet chamber with insulation in terms of cold exit temperature, coefficient of performance, and isentropic efficiency.

2. Mathematical modelling and formulation

The parameters used to assess the VT performance are as follows:

Cold exit temperature gradient (ΔTc) is the difference between the temperature of the fluid at the inlet and the temperature of the fluid at the cold exit,

\[ \Delta T_c = T_{in} - T_c \]  

The cold mass fraction (μc): It is the ratio of cold stream mass to inlet mass of fluid, and expressed as,
\[ \mu_c = \frac{\dot{m}_c}{\dot{m}_{in}} \]  

Total temperature separation (\(\Delta T\)) is the difference between the temperature at the hot exit and cold exit,
\[ \Delta T = T_h - T_c \]  

Hot end temperature gradient (\(\Delta T_h\)) is the difference of temperature between hot exit and inlet fluid of VT,
\[ \Delta T_h = T_h - T_{in} \]  

Isentropic efficiency (\(\eta_{isen}\)):

For the cooling efficiency calculation of the VT, the ideal gas adiabatic expansion principle is used. As the fluid flows into the VT, the isentropic expansion of fluid is taking place and, it is given as
\[ \eta_{isen} = \frac{T_{inlet} - T_{cold}}{T_{inlet} \left(1 - \frac{P_{atm}}{P_{inlet}}\right)} \]

Where \(\eta_{isen}, P_{inlet}, \Delta T_{isen}, \gamma\) and, \(P_{atm}\) are the isentropic efficiency, pressure inlet, isentropic temperature gradient, specific heat ratio, and atmospheric pressure, respectively.

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2.1. Governing equations

The flow of fluid in the vortex tube is turbulent and compressible in nature. The governing equations solved by the solver of FLUENT are momentum, continuity, energy, and ideal gas. The transport equations associated with turbulent kinetic energy (\(k\)) and turbulent dissipation (\(\varepsilon\)) in the standard realizable \(k - \varepsilon\) turbulence model together with energy are as follows.

\[ \frac{\partial}{\partial x_i}(\rho k \mu) + \frac{\partial}{\partial x_j}\left[\mu + \frac{\mu_t}{\sigma_k} \frac{k^2}{\varepsilon}\right] + G_i + G_k - \rho c_{\rho} + Y_{\mu} \]

\[ \frac{\partial}{\partial x_i}(\rho \varepsilon) + \frac{\partial}{\partial x_j}\left[\mu + \frac{\mu_t}{\sigma_\varepsilon} \frac{\varepsilon^2}{\varepsilon^{2/3}}\right] - \rho c_{\rho} + C_{\rho} \frac{\varepsilon^2}{k + \sqrt{\varepsilon}} + C_{\varepsilon} \frac{\varepsilon}{k^2} (G_{i} + C_{\varepsilon} G_{i}) \]

The residuals convergence criteria are taken as 10\(^{-6}\) for energy and 10\(^{-3}\) for other quantizes. The values of constants in standard realizable \(k - \varepsilon\) turbulence model are given as \(\sigma_k = 0.9, \sigma_\varepsilon = 1.0, \gamma = 2.0, \sigma_{\rho} = 1.44,\) and \(C_{\rho} = 1.9\).

3. Boundary condition

ANSYS 17.1 has been used to perform the CFD simulation based on the experimental result of Skye et al. [2] in the present study. The CFD simulation is carried out under steady-state conditions, and a pressure-based implicit solver has been used. The standard realizable \(k - \varepsilon\) turbulence model is considered for flow modelling of VT. The ideal gas assumption is used for the compressible flow inside the VT. The gravity effect has been neglected. Under relaxation has been used for the dependent variable. SIMPLE algorithm with pressure velocity coupling has been used in the present study. The inlet of the VT has been considered as pressure inlet, and its value varies from \(P_{in} = 3\) bar to \(P_{in} = 6.5\) bar, and inlet stagnation temperature is fixed as 294.15 K. The hot exit has been assigned as a pressure outlet which also governs the mass flow rate in the vortex tube and the cold exit is open to atmosphere. At the wall, adiabatic and no-slip boundary condition has been considered. The
geometrical measurement and the thermophysical properties of the working fluid have been mentioned in Table 1 and Table 2, respectively. The explicit schematic view of the VT has been depicted in Figure 1. The polyurethane is taken as insulating material to insulate the main tube of the VT in the present study.

**Table 1.** Dimension of VT taken into consideration.

| Parameters                          | Dimension   |
|-------------------------------------|-------------|
| vortex tube diameter(D)             | 11.4 mm     |
| Working tube length of vortex tube  | 106 mm      |
| Nozzle number (N)                   | 5           |
| Cold exit diameter, d_c             | 5.25 mm     |

**Table 2.** Working fluid thermophysical property.

| Working Fluid                  | Air                     |
|--------------------------------|-------------------------|
| Thermal Conductivity(K)        | 0.0242 W/m-k            |
| Density(\(\rho\))             | Ideal gas equation      |
| Specific heat(\(C\))          | 1.00643 kJ/kg-k         |

**Figure 1.** schematic 3D view of VT.

4. **Grid independence study**

Meshing is the most crucial part of flow analysis through computational techniques. The accuracy and convergence of the solution process mainly depend upon the type and quality of the mesh. In the present model, the unstructured tetrahedral mesh has been used, as depicted in Figure 2.

In the present study, the unstructured tetrahedral mesh has been used. It can be seen from Figure 2. The simulation has carried out for different unit cells from 19587 to 134840 on the VT in order to nullify the error which occurs due to unmapped grids. The mass fraction of cold fluid maintained as \(\mu_c=0.35\), and an L/D ratio of 9.298 considered. It is noted from Figure 3 that there is no notable change in the cold outlet temperature after 134840 unit cell. Thus, further study on VT by CFD simulation is carried out by using 134840 grid sizes to reduce the extra computational time.

**Figure 2.** View of unstructured tetrahedral mesh.
Figure 3. Effect of grid size on the cold outlet temperature.

5. Validation
The FLUENT software has been used to derive and solve the Navier Stokes equation in the compressible form accompanying the standard $k$-$\varepsilon$ turbulence model. The quick schemes and second order upwind are implemented to discretize the derivative terms of turbulence, momentum, and energy equations. Skye et al. [2] experimental results have been compared with the present temperature separation result for validation. Figures 4 and 5 show the cold and hot temperature variation with the cold mass fraction. It is noticed from Figure 4, the results of cold temperature gradient ($\Delta T_c$) obtained by the present three-dimensional model of VT are in accord with Skye et al. [2] experimental results. The results obtained for the cold temperature gradient ($\Delta T_c$) is lies between experimental and computational results but closer to Skye et al. [2] experimental study. However, the CFD results obtained for hot temperature gradient ($\Delta T_h$) are in close proximity to the experimental result, as shown in Figure 5.

The validation of the Present study by CFD is performed with Skye et al. [2]. It is observed from Figure 4 and Figure 5 that the results obtained from the current CFD study for hot and cold exit temperature separation and reconcilable with the CFD study of Skye et al. [2]. It indicates that appropriate methodology has been followed during the simulation of the present study. Figure 4 illustrates that with an increase in the mass percent of cold fluid, there is an increase in a cold temperature gradient, reaches the maxima and then shows a decreasing trend. Figure 5 shows how the cold mass fraction of fluid affects the fluctuation of a hot temperature gradient. It is noted from Figure 5 that when the mass fraction of cold fluid rises, the hot temperature gradient rises with it. The deviation for cold temperature gradient for the different mass fractions of cold fluid between the Skye et al. [2] and the present study is less than 7% up to a mass fraction of cold fluid 0.6 and 8-10% beyond a mass fraction of cold fluid 0.6. The average deviation for the entire mass fraction of cold fluid is 6.5% which is justifiable, and this present study follows a similar trend as Skye et al. [2].

Table 3. Details of the study performed by different authors on the vortex tube.

| S.No | Parameter → | Number of Inlet Chamber | Cold exit temperature ($T_c$) (K) | Vortex tube with insulation | Number of Inlet nozzle (N) | Vortex tube (L/D) (mm) | Diameter of cold exit $d_c$ (mm) |
|------|-------------|--------------------------|-----------------------------------|-----------------------------|----------------------------|--------------------------|-------------------------------|
| 1.   | Rafiee et al. | 1                        | 266.34                            | No                          | 2,3,4,5,6                 | 13.89                    | 9                             |
| 2.   | Skye et al.  | 1                        | 264.2                             | No                          | 2                         | 9.29                     | 6.2                           |
| 3.   | Present study | 2                        | 241.053                           | Yes                         | 5                         | 9.29                     | 5.25                          |
Figure 4. Similarity relations were obtained between previous results and the present study.

Figure 5. Comparison of the present study with other similarity relations obtained.

Studies performed by the various authors have been summarized in Table 3. It has been noted that using the two inlet chambers with insulation in the VT shows a significant impact on the cold outlet temperature of VT instead of using a single inlet chamber.

6. Results and discussion

The working fluid, mainly compressed air at the high pressure of 6.5 bar, enters tangentially through the inlet nozzles with high swirling velocity. There are two outlets one is for the hot gases, which is in the direction of flow connected with a throttle which can regulate the inlet mass flow rate of fluid and the other in the opposite direction of the fluid flow for the cold gases, which is near the inlet chamber. The fluid enters in the vortex tube flow towards the hot outlet, and due to the throttling effect at the hot outlet, the flow reversal takes place, and one secondary flow generated at the core and flows in the opposite direction of the peripheral flow towards the cold exit which is depicted in Figure 8. The peripheral fluid has relatively had a higher temperature than the core fluid. The temperature separation phenomenon occurs due to kinetic energy transfer and momentum diffusivity between the peripheral and core layers at the hot exit in the vortex tube.

The temperature contours of a single inlet chamber, double inlet chamber, double inlet chamber with insulation has been shown in Figure 6 (a), (b), and (c). It is noted from the temperature contours that the cold exit temperature is minimum for the vortex tube having a double inlet chamber with insulation shown in Figure 6 (c). As the number of inlet chambers increases from one to two, the mass flow rate of fluid increases and the cold exit of the second chamber mixed with the cold exit of the first chamber, which tends to decrease by 7.8 % in the cold exit temperature of the vortex tube. The non-insulating vortex tube release heat through the main tube, which disturb the temperature separation phenomenon, which mainly occurs due to kinetic energy transfer between the core and peripheral fluid layer.

However, insulating the double chamber vortex tube leads to a further reduction of 4.2% in the cold exit temperature. Therefore, it indicates that the overall decline in the cold exit temperature from one chamber non-insulated vortex tube to double chamber insulated vortex tube is 9.6%.

The velocity streamlines in the form of stream surface have shown in Figure 7(a), (b), and (c) of a single inlet chamber, double inlet chamber, double inlet chamber with insulation. It is noted that as the number of inlet chambers increases, the zone of vortices is shifted left, and it is further shifted leftward in the double chamber with insulation. It depicts that the flow reversal takes place earlier in the double chamber with insulation vortex tube in comparison to other vortex tubes used in the study.
Figure 6. Temperature contour of a vortex tube with (a) single inlet chamber, (b) Double inlet chamber, and (c) Double inlet chamber with insulation.

Figure 8, indicates that there is two vortex flow that exists in the vortex tube (VT). The peripheral fluid is a free vortex that comes out from the hot exit of the VT. The core fluid which represents the reverse flow is a forced vortex flow that comes out from the cold exit of the VT.

It is evident from Figure 9(a) to (c) as the mass fraction of cold fluid increases, there is an increase in the cold temperature gradient between 0.2<µ<0.3 and then decreases thereafter. The study has been performed at a higher inlet pressure of $P_{in}=6.5$ bar. However, the hot temperature gradient follows the increasing trends for all the mass fractions of cold fluid values. However, the cold temperature gradient increases from a single inlet chamber vortex tube to a double inlet chamber vortex tube with insulation Figure 9(a) to 9(c). As $Z/L$ value increases, i.e., moving far from the cold exit where $Z$ is the distance along z-axis of vortex tube and $L$ is the vortex tube length, the cold temperature shows a decreasing trend in all the types of vortex tube used in the study because at the farthest point from the cold exit, i.e. near the hot exit, there is an intermingling of peripheral (hot) and core fluid (cold) is more which disturb the temperature separation in the vortex tube.
Figure 7. Velocity streamline distribution along the length of the VT of (a) single inlet chamber, (b) Double inlet chamber, and (c) Double inlet chamber with insulation.

Figures 9 (c) depicts the cold temperature gradient as maximum among all the types used for the study. Therefore, the double inlet chamber with insulation vortex is proposed to obtain the maximum cold temperature gradient.

Figure 8. Velocity stream of a vortex tube with double inlet chamber with insulation.
It is noted from Figure 10 that the isentropic efficiency of VT is highest for the entire cold mass fraction considered for a double inlet chamber with insulation vortex tube as compared to a single inlet chamber and double inlet chamber VT. The isentropic efficiency shows a decreasing trend with increases in the cold mass fraction. It is highest between the cold mass fraction value 0.2 to 0.3. The result depicted in Figure 10 follows a similar trend as concluded by Rafiee et al. [15].

The coefficient of performance (COP) is the ratio of refrigeration effect to the rate of work required by the device as per A. Kumar et al. [17] and is given by

$$COP = \frac{Q_c}{W} = \frac{\mu C_p \Delta T_c}{\gamma RT_e \left( \frac{P_e}{P_{sen}} \right)^{\gamma/(\gamma-1)} - 1}$$

Where $W$ is the work required, $Q_c$ is the refrigeration effect, $R$ is the specific gas constant, and $C_p$ is the specific heat at constant pressure. It can be noted from Figure 11. There is an increase in the coefficient of performance (COP) with the cold mass fraction for all the vortex tubes, i.e. single inlet chamber, double inlet chamber, and double inlet chamber with insulation up to the mass fraction of cold fluid of 0.7 and decreases thereafter. As the cold mass fraction ($\mu$) increases, the cold temperature gradient ($\Delta T_c$) decrease or increase in cold outlet temperature ($T_c$) due to the intermingling of hot and
cold fluid in the vortex tube, which disturbs the temperature separation phenomenon. It can be seen from Figure 9(a), (b), and (c) and a similar trend is also observed by Thakare et al.[16], Skye et al.[2] and Manimaran et al. [5]. The parameters for the calculation of COP from equation 8 are fixed. The only variable quantities are in the numerator, i.e. , as the value of increases, the value decreases, which helps in increasing the COP up to from equation 8. The rise in value is more dominating than the decrease in the value up to , after that the reduction in the value of value dominant over increase in value, which tends to decrease in the COP value. It is highest for the double chamber with an insulation vortex tube.

Figure 10. Isentropic efficiency variation with a mass fraction of cold fluid the different types of vortex tubes used in the study

Figure 11. Coefficient of performance variation with a cold mass fraction of the different types of vortex tube used in the study

7. Conclusions

1. The study shows that the phenomenon of temperature gradient in VT is observed as a result of diffusion activity of average kinetic energy, which leads to high swirling motion in the VT. It is observed that the temperature gradient is mostly due to the kinetic energy transfer and the interaction of angular momentum transition.

2. It is concluded from the study that the vortex tube performance increases as the number of inlet chambers increase from one to two by 7.8% in terms of cold exit temperature. However, insulating the double chamber vortex tube leads to a further reduction of 4.2% in the cold exit temperature. Therefore, it indicates that the overall decline in the cold exit temperature from one chamber non-insulated vortex tube to double chamber insulated vortex tube is 9.6%.

3. The double inlet chamber with insulation shows a promising result in terms of cold exit temperature, coefficient of performance, and isentropic efficiency in comparison to a single inlet chamber and double inlet chamber without insulation.

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