Numerical investigation of conical vortex tube with combined axial-tangential injection

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Abstract. The present work is a numerical investigation of the combined axial- tangential injection effect on conical Ranque-Hilsch vortex tube performances. An orifice in the center of the conical control valve is created, allowing the injection of compressed air axially. The computational fluid dynamics CFD commercial software ANSYS CFX has been used for the present investigation, and the mean field is determined using the Reynolds-averaged Navier-Stokes equations RANS method. The main results show that the conical configuration causes a considerable drop in temperature of the cold and hot outlet flows up to 3°C on average and a significant increase in the cold mass flow of 25% compared to the cylindrical vortex tube with axial injection.

Abbreviations

H : Total enthalpy [ J/Kg]
F1 : First SST blending function
F2 : Second SST blending function
K : Turbulence kinetic energy
P_k : Shear production of turbulence
P : Pressure [Pa]
R : Universal gas contant [J/mol K]
t : Time [s]
T : Temperature [K]
u, v : Velocity [m/s]
ρ : Density [Kg/m³]
τ : Shear stress [N/m²]
μ : Dynamic viscosity
μ_t : Turbulent viscosity
γ_t : Kinematic turbulent viscosity
ω : Turbulent frequency

1. Introduction

The Ranque–Hilsch vortex tube is a simple cylindrical device capable of dividing the incoming compressed air into two fluxes with different temperatures, cold and hot. This process is known as energy separation or the Ranque effect. Several numerical and experimental studies have been realized in this research field, with the aim of finding a scientific explanation for the temperature separation...
phenomenon. In 1933, the first study was proposed by Ranque [1], affirming that separation is the result of adiabatic compression of peripheral flow. In 1947, Hilsch [2] noticed that the separation phenomenon is due to the spiral motion of the incoming compressed air, which has been divided into two streams. One heated by compression near the wall, and the other cooled by expansion in the middle part of the tube. Many researchers have proposed different theories and hypotheses related to the temperature separation effect. Xue et al [3] presented a study to provide a new explanation for the energy separation inside a vortex tube. Then, exergy analysis is also carried out using current and previous experimental investigations. Li et al [4] have experimentally studied the effect of different working fluids R 134, R744, R32, R227 relative to different inlet pressures (500 Kpa - 850 Kpa), and different inlet temperature (308.15 K - 33.1 K) for a range of cold mass fractions (20% - 97%) on the vortex tube performance and temperature separation. They concluded that the working fluid R134 has great effect on the cold temperature difference and the enthalpy difference comparing with other gases. Furthermore, with the inlet pressure increase, the cold temperature also improves and the hot temperature decreases gradually. Thakare et al [5] numerically studied the influence of the turbulence models, namely one equation Spalart-Allmaras, two equations, standart K-ε and Standart K-ω model on the thermo-physical characteristic relative to different working fluids. Their main results showed that the predicted energy separation using Spalart-Allmaras model on the cold side gives better results. However, the secondary flow occurrence was detected by all turbulence models. Rafiee [6] conducted an experimental investigation on the influential parameters such as the control valve cone length, nozzle number and inlet pressure on the performance of vortex tube. The results show the existence of optimum cone length to reach minimum cold temperature, and the number of inlet nozzles is found to be the main parameter influencing swirling flow inside the tube. Farzaneh-Gord et al [7] experimentally investigated the influence of vortex generator characteristics namely; cold orifice angle, cold orifice diameter and nozzle area using a natural gas as working fluid under inlet pressure of 4 bars. The main results indicated that the cold orifice angle of 4.1°, cold orifice ratio of 0.64 and nozzle area ratio of 0.14 give the best vortex tube efficiency. Rahbar et al [8] presented a numerical investigation on the flow pattern of micro scale vortex tube. Two models were performed for 2D axisymmetric and 3D using a commercial software Fluent 6.3.26. The main results were in good agreement with experimental data of the previous study. They concluded that the 3D simulation is more efficient than the 2D numerical investigation. The maximum cold temperature drop and maximum efficiency were achieved at cold mass fraction 0.58. The maximum cold energy occurred at a cold mass fraction 0.65. Due to the friction of fluid layers, the tangential velocity decreases along the tube after achieving the maximum value near the inlet entrance. A. Vladimir et al [9] performed a numerical study of 3D model of a double-circuit vortex tube by using commercial software ANSYS- CFX 14.5. Six turbulence models have been tested such as K-ε , - RNG - K-ε, SAS-SST, RSM-LRR and LES. The most results showed that the LES turbulence model described the flow behaviour inside vortex tube more accurately. In addition, the LES model indicates a very high radial velocity, which is the most influential parameter on the turbulent intensity and mass transfer. H.R. Thakare et al [10] carried out a numerical investigation of a 2D axisymmetric model of a counter flow vortex tube. The K-ε turbulence model was used to predict the flow behaviour inside the tube. The result indicates the existence of recirculation secondary flow near the entrance. Furthermore, the velocity vectors showed the appearance of backflow near the cold exit, which diminishes with the increase of cold mass flow fraction. G. M. P. Yadav et al [11] experimentally investigated the performance of forced-vortex flows positioned at the end of the cold exit. This new design of vortex tube is a Dual forced flow vortex tube (DFFVT). The experiment was conducted under different conditions such as inlet pressure, variable cold mass fraction and cold end diameter I and II, which have been shown to have a great effect on flow pattern and temperature drop. The maximum temperature difference is achieved with larger diameters at high cold mass fractions. In low fractions, the smaller cold diameter II attains a high temperature drop. The coefficient of performance COP achieve its maximum value of 0.057 with an orifice diameter of 4mm.V.Nikitin et al [12] presented an overview of different numerical works for simulating the energy separation and flow pattern inside vortex tube, relative to 2D axisymmetric and 3D numerical models using a different turbulence models. This analysis is focused on recent works which are in good agreement with experimental results. S. E. Rafiee et al [13] numerically studied the
influence of rounding-off edge radius inside the entrance of vortex chamber in 3D vortex tube mode with the increasing of total pressure, and tangential velocity. These conditions have a major effect on the energy separation, and improve the cold temperature drop. The highest cold temperature drop is achieved at edge radius of 1.5 mm. S. N. JadHAV el al [14] presented an experimental study on the effect of working and geometrical parameter of vortex tube constructed with UPVC (Unplasticised Polyvinyl Chloride) material, which has a very low thermal conductivity. The maximum cold and hot temperature difference were reached at L/D = 50. The hot temperature increases with a decrease in the cone angle of the control valve. Furthermore, the cold temperature drop is reached with increasing inlet pressure. K. D. Divade et al [15] conducted an experimental investigation on the effect of geometrical parameters of vortex tube such as L/D ratio, exit control valve angles, number of nozzle and tube divergence angle, and using air as working fluid composed of only nitrogen, oxygen and water vapour. Operational parameters have also been investigated like inlet pressure, cold mass fraction, and Mach number (subsonic, sonic, and supersonic). The obtained results confirmed that all these parameters have considerable influence on tube performance and temperature separation. Particularly in low cold mass fraction values and subsonic Mach number, the vortex tube achieves its optimum performance. K. K. KumarRao et al [16] experimentally studied the influence of geometrical parameters on the performance of vortex tubes, using a wood tube build with two kinds of wood, rosewood and sapota wood. The main results showed that the rose wood provides slightly better performance compared to sapota wood. H. Pouraria et al [17] presented work based on Artificial Neural Networks (A.N.N) model to predict the effect of geometrical and operational parameters such as ratio Length to diameter of the tube, ratio of cold exit diameter to tube diameter inlet pressure cold mass fraction, of a vortex tube on its cooling performance. The generic algorithms help to optimize the ANN model. The results of their study show that ANN model is considered to be an adequate tool for predicting the thermodynamics parameter of the vortex tube with good precision. The vortex tube achieves optimal performance at a ratio of cold exit diameter to tube diameter d/D=0.5. Additionally, the results show the increase in length to the diameter ratio led to an increase in the cooling performance of vortex tube. K. D. Divade et al [18] conducted an experimental investigation on vortex tube geometrical and operational parameters effect such as L/D ratio, number of nozzles, nozzle geometry, valve angles and cold orifice diameters under different inlet pressures, their results showed that the L/D = 17 provide a high COP and cold temperature drop relative to cold mass fraction equal to 0.78. A. Haydari et al [19] presented a new method to investigate the effect of fin installation 1, 3 and 5 on the hot side in order to improve the cooling performance of the vortex tube. Their results denote that the 5 triangular fins and 3 square fins provide the highest COP, isentropic efficiency and lowest cold temperature. Y. Xue [20] presented work based on flow visualization using PIV measurement on transparent vortex tube in order to understanding of the internal flow mechanism to explain the separating processes inside the vortex tube and improve the tube performance their results showed a good agreement between the observed flow and previous published studies. A. Bazgir et al [21] carried out a study in order to investigate the impact of different turbulence models on the energy separation. Ten turbulence models have been tested in steady and transient state. The result reveals that the RNG- K-ε turbulence model provides a good agreement with an experimental results. The outcomes prove that the separation energy result from the pressure loss inside the vortex tube. A. Bazgir et al [22] conducted a numerical investigation on the effect of fins installation on the cold tube including triangle, square, rectangle, circle, parallelogram and trapezium. The outcomes indicate that the parallelogram and rectangular fins have the lower cold temperature additionally the maximum values of COP and isentropic efficiency were reached. The present study focuses on the performance of a new configuration of conical vortex tube with a cone angle of 10° and axial injection characterized by the installation of a second inlet arranged axially on the hot outlet side in the conical control valve Figure.1. This orifice is geometrically identical to the tangential orifice, it is expected from this new conical vortex tube configuration that the axial injection of compressed air will accelerate the central flow inside the vortex tube, which causes a considerable increase in the cold mass flow and significantly a decreasing the temperature of both hot and cold flows. The present study was performed numerically, using the commercial software ANSYS-CFX. The dimensions of conical vortex tube are identical to the commercial vortex tube used in the experimental investigation of Ref
The main results showed that a gain of 25% has been reached in the flow of the cold flow in addition considerable drop of temperature of the two sides warm and cold estimated at 3°C compared with Ref [24].

![Figure 1. Schematic of Conical Vortex Tube with Axial Injection](image)

**2. Governing equation**

In order to compute the flow field inside the vortex tube, some assumptions are considered:
- Compressible fluid,
- Steady and turbulent flow
- Physical proprieties expressed as temperature functions.

The governing equations including conservation of mass, momentum, energy and state equation were discretized using high-resolution schemes based on finite volume approach. These equations are:

**Continuity equation**

\[
\frac{\partial p}{\partial t} + \nabla (\rho u) = 0 \tag{1}
\]

**Momentum equation**

\[
\frac{\partial (\rho u)}{\partial t} + \nabla (\rho u u) = -\nabla p + \nabla (\tau) \tag{2}
\]
**Energy conservation equation**

\[ \nabla(\rho \cdot v \cdot H) = \nabla \left( \frac{k}{c_p} \nabla H \right) - \nabla (\tau \cdot u) \]  

(3)

**State equation of ideal gas**

\[ p = \rho \cdot R \cdot T \]  

(4)

3. **Computational domain**

The 3D numerical domain shown in Fig 2 conserves the same dimensions reported in Ref [25], and the only modification was made to the conical valve by creating a hole in the middle. The rotational periodicity is also applied in the present study.

**Figure 2.** Numerical domain
4. Turbulence model

The flow behaviour inside vortex tube is assumed to be highly turbulent, therefore only the mean field is determined by using RANS method. Due to its ability to resolve several numerical problems related to the convergence, the K-ω SST turbulence model is applied in the present study. The transport equations are given as [25].

Turbulence kinetic energy

\[
\begin{align*}
\frac{\partial}{\partial t}(\rho, k) + \frac{\partial}{\partial x_j}(\rho U_j, k) - \frac{\partial}{\partial x_j}\left(\mu + \frac{\mu_t}{\sigma_t} \frac{\partial k}{\partial x_j}\right) &= \tau_{ij} \frac{\partial U_i}{\partial x_j} - \beta^* \rho k \omega \\
\frac{\partial}{\partial t}(\rho, \omega) + \frac{\partial}{\partial x_j}(\rho U_j, k) - \frac{\partial}{\partial x_j}\left(\mu + \frac{\mu_t}{\sigma_t} \frac{\partial \omega}{\partial x_j}\right) &= \gamma \frac{\tau_{ij}}{\nu_i} \frac{\partial U_i}{\partial x_j} - \beta \rho \omega^2 + 2 \rho (1 - F_i) \frac{1}{\sigma_{\omega}, \omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j}
\end{align*}
\]

The constant values are given as

\[
\begin{align*}
\sigma &= F_1 \sigma_1 + (1 - F_1) \sigma_2 \\
F_1 &= \tanh(\arg_1^2) \\
F_2 &= \tanh(\arg_2^2)
\end{align*}
\]

With

\[
\begin{align*}
\arg_1 &= \min\left[ \max\left( \sqrt{\frac{k}{0.09 \omega}}, \frac{500 \mu}{\nu^2 \omega} \right) \frac{4 \rho k}{CD_{k\omega} \gamma^2 \sigma_{\omega}} \right] \\
\arg_2 &= \min\left[ \max\left( 2 \sqrt{\frac{k}{0.09 \omega}}, \frac{500 \mu}{\nu^2 \omega} \right) \frac{4 \rho k}{CD_{k\omega} \gamma^2 \sigma_{\omega}} \right] \\
CD_{k\omega} &= \max\left( 2 \rho \frac{1}{\sigma_{\omega}, \omega} \frac{\partial k}{\partial x_j} \cdot 10^{-20} \right)
\end{align*}
\]

The constant values are given as

\[
\begin{align*}
\beta_1 &= 0.0750, \quad \gamma_1 = 0.5530 \quad \sigma_{k_1} = 1.176 \quad \sigma_{\omega_1} = 2.000 \quad \beta_2 &= 0.0828 \quad \beta^* = 0.090 \quad \gamma_2 = 0.4404 \\
\sigma_{k_2} &= 1.000 \quad \sigma_{\omega_2} = 1.170 \quad \alpha_i = 0.31
\end{align*}
\]

5. The performance parameters

The cold mass fraction parameter is the most studied in several vortex tube studies. The majority of performance curves are drawn function of this parameter. The cold mass fraction can be expressed as

\[
\varepsilon = \frac{m_c}{m_i}
\]

1-\varepsilon \quad \text{Hot mass fraction}

Hot and cold temperature differences are given by the relationships

\[
\begin{align*}
\Delta T_c &= T_i - T_c \\
\Delta T_h &= T_h - T_i
\end{align*}
\]

Using some assumptions such as neglecting the kinetic energy of the gas and considering no heat exchange to the vortex tube from its surrounding the isentropic efficiency can be expressed as [25]
\[ \eta_{is} = \frac{T_{in} - T_c}{T_{in} - T_s} \]

Knowing that
\[ T_s = T_{in} \left( \frac{P_c}{P_{in}} \right)^{\gamma - 1} \]

By replacing the two last equations the isentropic efficiency expression becomes
\[ \eta_{is} = \frac{T_{in} - T_c}{T_{in} \left[ 1 - \left( \frac{P_{atm}}{P_{in}} \right)^{\gamma - 1} \right]} \]

6. Mesh sensitivity
The studied geometry meshing is carried out by using ICEM software. To assure greater precision, the domain is meshed by adopting hexahedral elements which insufficiently refined near the wall to represent correctly the flow behaviour in the boundary layer. In the present study, five grid cases are tested namely \( N = 57240 \) cells, \( N = 114480 \) cells, \( N = 130251 \) cells, \( N = 190254 \) cells, Fig. 3. As it is shown in the Table 1, the total temperature difference between cold and hot exits relative to different cases shows that the solution remains unchanged after \( N = 170145 \) cells, therefore the mesh of \( N = 190254 \) cells is adopted for the following computations.

| Cells   | 114480 | 228960 | 260000 | 280000 | 300000 |
|---------|--------|--------|--------|--------|--------|
| \( \Delta T \) | 22.1   | 18.86  | 15.8   | 133.1  | 13.3   |

Table 1. Total temperature difference for different grids.
In the present work, the cold and hot exits sections are considered constant at atmospheric pressure. Therefore to modify the cold mass fraction, it is necessary to increment the hot exit pressures with 0.05 bars in each step of numerical calculation and keep the cold exit pressure constant. The mass flow rate at the tangential entrance is fixed at 8.35 g/s with circumferential angle of 42°. On the other hand, the mass flow at the axial entrance is maintained at 1.75 g/s with a section of 6.15 mm². The total temperature at the tangential entrance is 294.2 K.

**8. Numerical results**

All calculations in the present study have been performed by using the commercial software ANSYS-CFX. The governing equations are discretized by using high resolution schemes. The convergence is achieved when the residues reach $10^{-7}$ for the continuity and $10^{-6}$ for the remaining equations. Figure 4 show the streamlines inside the vortex tube in the central plane. The exam of figure 4 reveals that the central flow undergoes a slight deceleration, followed by acceleration. It is clear that the existence of secondary flow in neighbouring of the cold side, and two others vortices near the hot exit, which contribute to the contraction of central streamline resulting in central flow acceleration. The figure 4 gives an exemple of the spiral flow inside the vortex tube where it is remarked the acceleration of the central flow which starts at one-fifth of the length (1/6) of vortex tube from the control valve. Therefore an noticeable increase in the cold flow rate is expected in comparison to the case of classical vortex.
Figure 4. Flow field inside Vortex Tube
Figures 5 and 6 represent the temperature separation predicted by the CFD investigation. As seen in figure 5, it is showed a significant improvement in cold temperature comparing with ordinary vortex tube. Indeed, an important drop in the cold temperature, about 3°C, is observed. However, in figure 6, it is remarked that the warm outflow through the hot exit was reduced. Compared to classical vortex tubes, this effect is due to the conical shape of the tube.
Figure 7. Cold mass flow rate

Figure 7 and figure 8 shows the cold mass flow rate predicted numerically for the two cases conical and classical vortex tube. It can be seen that there is a significant gap between the two curves. It is to note that the conical shape of the vortex tube has a significant effect on the increases of the mass flow rate of the cold side. This indicates that a part of the warm flow crosses the cold exit. This is effectively confirmed by figure 8, which shows the evolution of hot mass flow rate.

Figure 8. Hot mass flow rate

9. Conclusion
The numerical investigation of conical vortex tube with axial injection has revealed many interesting results compared to those of cylindrical vortex tube. The main findings are cited below:
- An improvement in cold temperature separation compared with ref [24]
- A significant drop in the hot temperature separation regarding to ref [24]
- An important gain of 25% in cold mass flow rate
The present results have to be checked by carrying out an experiment series on the conical configuration of vortex tube with axial injection. Furthermore, it would be of interest to study the variation of some vortex tube thermodynamics parameters performance.

10. References

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