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Numerical Simulation of Turbulent Flow in Stepped-Section Vortex Tube

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Abstract **-** The present work is a numerical investigation of the effect of a stepped section on the performance of a Ranque-Hilsch vortex tube. The tube consists of two sections of different diameters with a tangential orifice. The present study of the stepped vortex tube has been carried out using the standard k- ε turbulence model, the cold, hot temperatures and the flow field is simulated. The results show that the tested configuration generates a considerable temperature drop in the two exit fluxes compared to the classical vortex tube.

Keywords**:** Ranque-Hilsch vortex tube; Stepped section vortex tube; Temperature separation, CFD.

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 warm **Figure.1** This Physical phenomenon is known in several research works as energy separation or temperature separation effect. Much research has been carried out to find a scientific explanation to the energy separation effect [1-7]. Until now, no comprehensive theory has been able to explain in a clear and precise way what is happening inside the vortex tube. **A.Vladimir et al** [8] presented a numerical investigation of double circuit vortex tube six turbulence models were tested such as. They concluded that the LES turbulence model described the flow field inside the vortex tube more accurately.

M. Abdulqyyum et al. [9] numerically studied the flow behavior inside the vortex tube in order to provide a new explanation of the temperature separation. Different conical valve shapes were used relative to different inlet pressures (0.4, 0.5, and 0.6 MPa). The most results showed that the geometrical parameter has important influences on vortex tube performance. **H. Pouriya et al. [10]** performed a numerical investigation to study the influence of the curvature design on the performance of vortex tube. The 3D CFD calculation with a **Baseline Two-Equation** k-ω turbulence model is used to compute the flow pattern inside the tube. The main results showed that the curved shape of vortex tube has noticeable influence on the tube performances with 60 % of the heavy fraction caused by the axial velocity and by swirling effect. **Bazgir et al** [11] conducted a numerical simulation on the effect of fins installation on the cold tube including triangle, square, rectangle, circle, parallelogram and trapezium. The final results show that the parallelogram and rectangular fins have reached the lower cold temperature additionally the maximum values of COP. Seol **Yeon Park et al** [12] performed a numerical simulation in order to explain the principale of energy separation in vortex tube based on the effect of the axial pressure gradient in the near axis region and the influence of the viscous dissipation in the region adjacent to the stagnation point. Guo et al [13] supposed that the energy transfer occurred from the inner vortex region to the outer vortex. J.Burazer et al [14] numerically tested the influence of L/D ratio on the temperature separation inside vortex tube using the OpenFOAM software and two equation and RST as turbulence models. The most results indicated that the lowest value of total temperature is obtained for the L/D = 1.4 with the presence of secondary flow which is responsible of the temperature drop. The present paper presents full numerical investigation on Stepped Section Vortex Tube, in order to improve cooling efficiency of the tube. This new design of the tube is composed of two sections with different diameters. The achieved difference in temperature between the cold and hot output show considerable gain of temperature compared with ref [15].

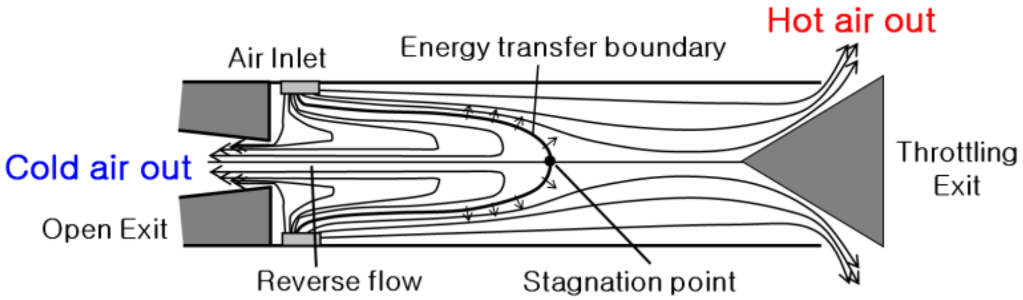


Figure.1. Flow pattern and schematic of vortex tube [18]

2. Thermodynamic equations of vortex tube

By neglecting the heat transfer between the vortex tube and its surroundings the conservation of enthalpy can be written as follow

$$\varepsilon.(h_c - h_i) + (1-\varepsilon).(h_h - h_i) = 0 \qquad (1)$$

The differential of the enthalpy expressed by temperature and pressure is

$$dh = C_p.dT + v(1 - T_\alpha).dp \qquad (2)$$

The flow inside vortex tube is considers as perfect gas and the C_p is independent of temperature and the enthalpy independent of pressure the difference of the enthalpy is as follow

$$\Delta h = h_f - h_i = C_p(T_h - T_i) \qquad (3)$$

Equation (3) in (1) we obtained

$$\varepsilon(T_c - T_i) + (1-\varepsilon).(T_h - T_i) = 0 \qquad (4)$$

Following the thermodynamics principles for ideal gas the differential in entropy function of temperature and pressure is expressed as follow

$$ds = C_p dlnT + RdlnP \qquad (5)$$

In vortex tube case the equation (5) become

$$\Delta s = \varepsilon.\left[C_p.ln\left(\frac{T_c}{T_i}\right) + R.ln\left(\frac{P_i}{P_f}\right)\right] + (1-\varepsilon)\left[C_p.ln\left(\frac{T_h}{T_i}\right)\right] + Rln\left(\frac{P_i}{P_f}\right)] \qquad (6)$$

Equation (4) in equation (6) we obtained

$$\frac{C_p.(\gamma-1)}{R.\gamma} = 1 \qquad (7)$$

If we put that

$$\gamma = \frac{C_p}{C_v} \quad \text{and} \quad x = \frac{T_c}{T_i}$$

$$x^{\varepsilon} = \left[\frac{(1-\varepsilon x)}{(1-\varepsilon)}\right]^{1-\varepsilon} \geq \left(\frac{P_i}{P_f}\right)^{\frac{-(\gamma-1)}{\gamma}} \qquad (8)$$

Due to the reversibility phenomena and using the adiabatic law for the ideal gas led us to write

$$T_c = T_i.\left(\frac{P_f}{P_i}\right)^{\frac{(\gamma-1)}{\gamma}} \tag{9}$$

The coefficient of performance COP is given by the relationship

$$COP = \left[\frac{\left(1-\frac{T_C}{T_i}\right)}{ln\left(\frac{T_i}{T_C}\right)}\right] \tag{10}$$

3. Governing equation

Due to the compressibility of the turbulent flow inside vortex tube the governing equations concerning the conservation of the mass, momentum, energy and the state equation are solved as follow

$$\frac{\partial\rho}{\partial t} + \nabla(\rho u) = 0 \tag{11}$$

$$\frac{\partial(\rho u)}{\partial t} + \nabla(\rho uu) = -\nabla p + \nabla\tau \tag{12}$$

$$\nabla(\rho vH) = \nabla\left(\frac{k}{c_p}.\nabla.H\right) - \nabla(\tau.u) \tag{13}$$

$$p = \rho.R.T \tag{14}$$

4. Turbulence Model

The flow pattern inside vortex tube is considered highly turbulent. Therefore only the RANS method must be used to represent its effects. The transport equations such as the turbulence kinetic energy k and its rate of dissipation ε are given from the following equations [20]

$$\frac{\partial}{\partial t}(\rho.k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_b - \rho\varepsilon - Y_M \tag{15}$$

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x_j}\right] + C_{1\varepsilon}\frac{\varepsilon}{k}(G_k + C_{3\varepsilon}G_b) - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{16}$$

$C_{1\varepsilon}$, $C_{2\varepsilon}$, $C_{3\varepsilon}$ are constants, $\sigma_k, \sigma_\varepsilon$ are the turbulent prandtl numbers relative to k and ε respectively. The combination of k and ε gives the turbulent viscosity as follow

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{17}$$

5. Mesh and Model geometry

The 3D numerical domain and mesh geometry are shown in Fig.2.. As it's shown in the figure the vortex tube is composed of two sections with different diameters only a 1/6 part of the total geometry is considered. The hypothesis of rotational periodicity is considered during simulation.

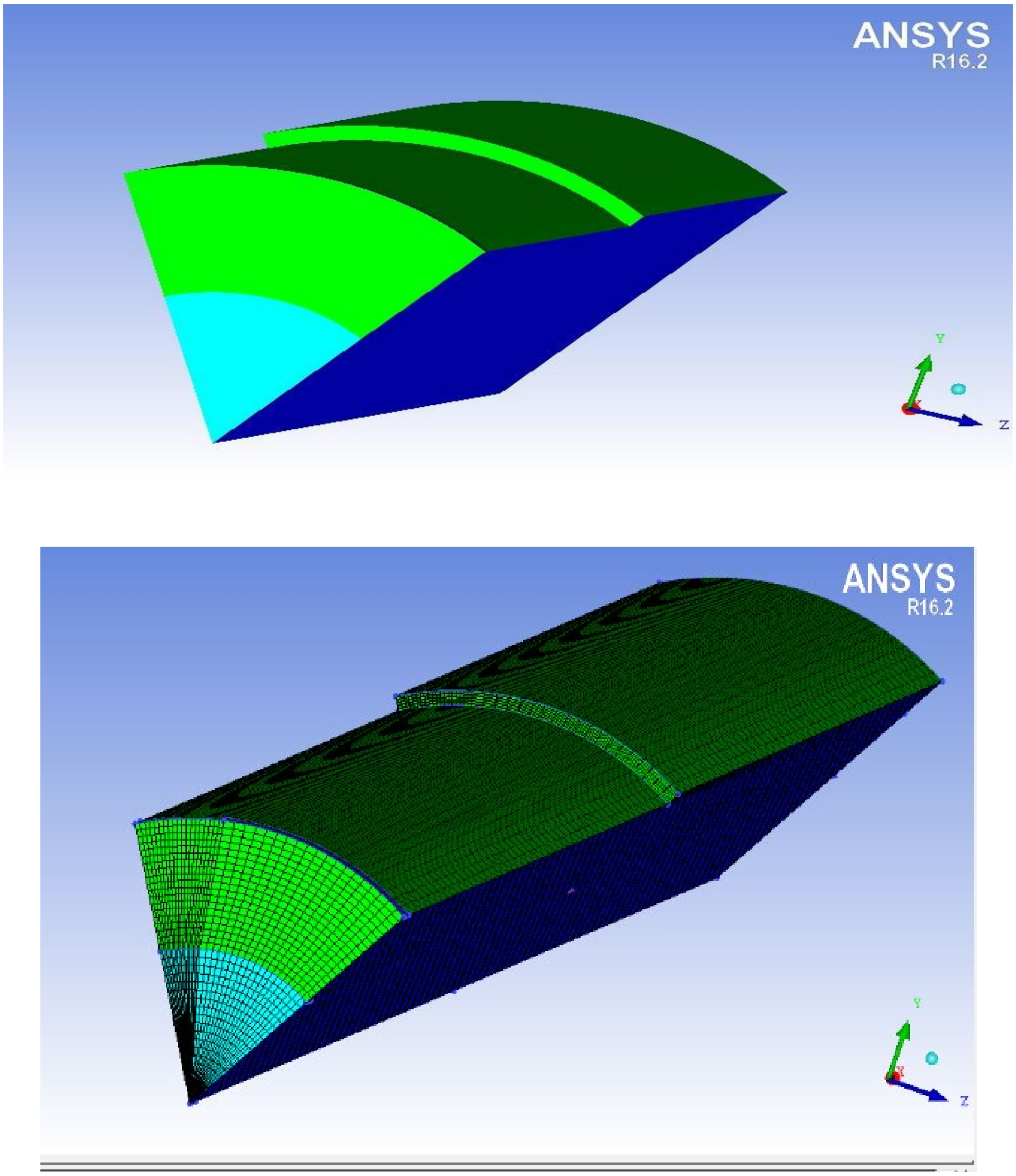


Figure 2. Mesh Geometry

6. Mesh Sensitivity

The studied geometry meshing is carried out by using ICEM software. To reach a good precision, the domain is meshed by adopting hexahedral elements. In the present study, six grid cases are tested namely 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= 66342** cells, therefore the mesh of **N= 120000** cells is adopted for the current computations.

Table. 1. Total temperature difference for different grids.

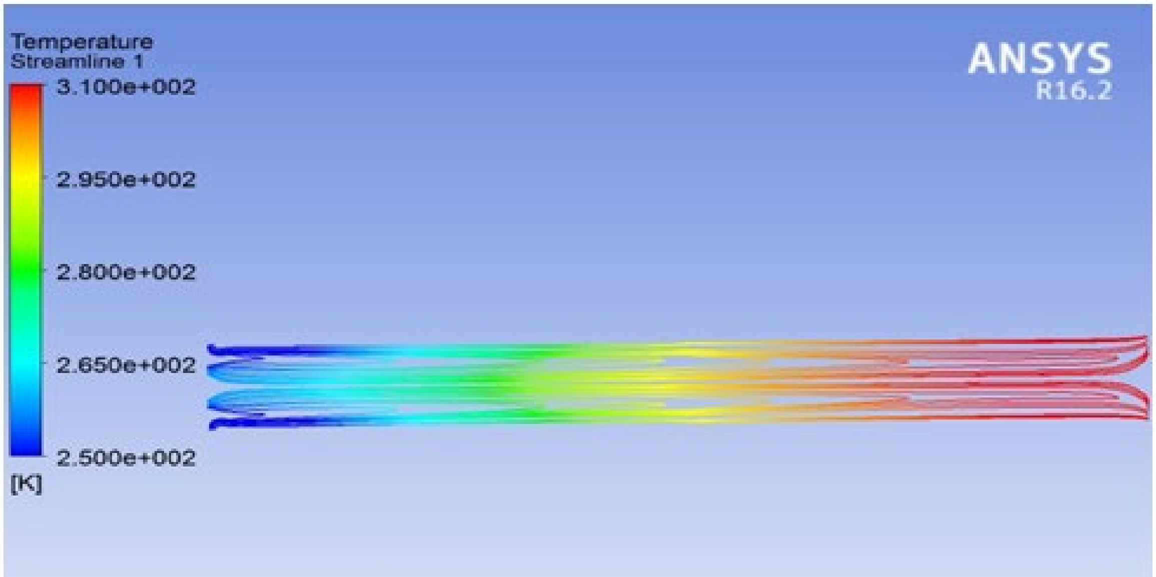
Cells	13268	26537	39805	66342	120000
ΔT	26.15	32.50	38	40	40

7. Boundary Conditions

In the current simulation, the cold and hot exits are kept constant t atmospheric pressure. Therefore to vary the cold mass fraction, it is necessary to increase the hot area pressures with constant value in each step of numerical simulation and keep the cold exit pressure constant. The mass flow rate at the tangential entrance is fixed at 8.35 g/s with total temperature is 294.2 K

8. Numerical results

All simulation in the current study have been carried out by using the commercial software CFX . The governing equations are discretized by using upwind schemes. The convergence is achieved when the residues reach 10^{-7} for the continuity and 10^{-6} for the remaining equations.



Cold

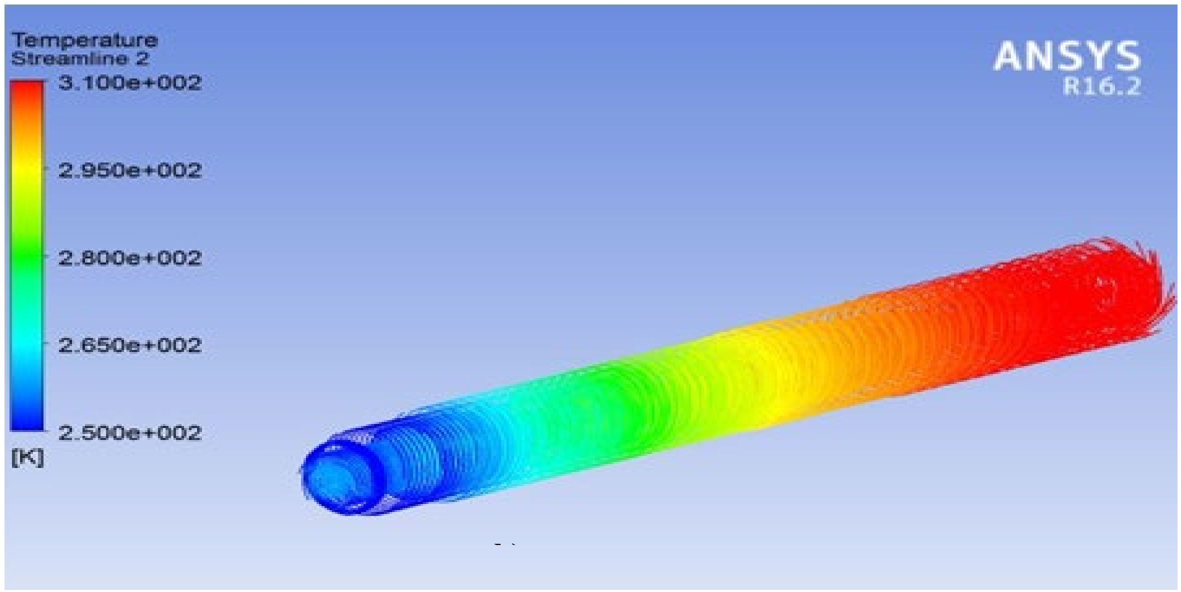


Fig.3. Flow behaviour inside Vortex Tube

Figures 3 show the streamlines inside the vortex tube in the central plane. It is clear that the existence of secondary flow in neighbouring of the cold side. The figure 3 (b) gives an exemple of the spiral flow inside the vortex tube a anoticeable increase in the cold flow rate is expected in comparison to the case of classical vortex

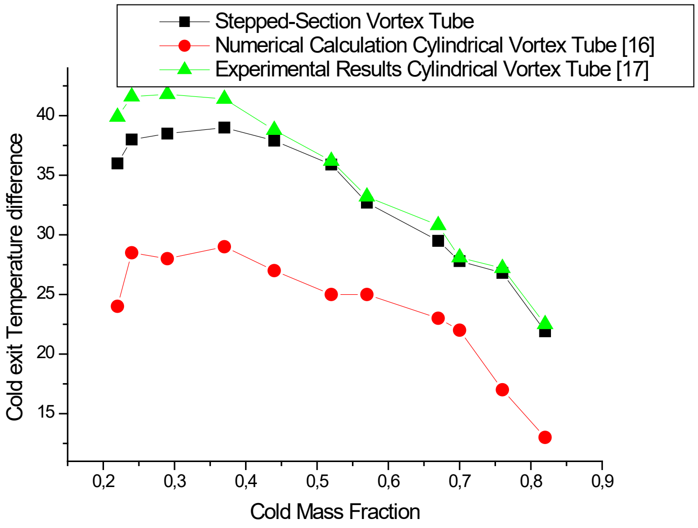


Fig.4. Cold exit temperature difference

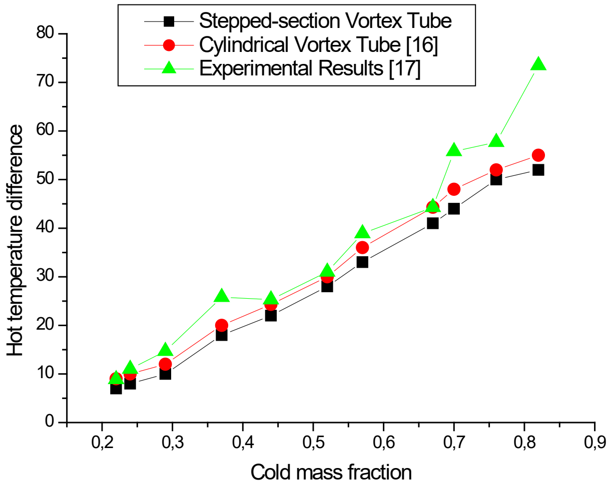


Fig.5. Hot exit temperature difference

Figures 4 and 5 show the temperature separation. These results were compared respectively with the experimental and computational results of [20]. As seen in figures 4 the cold exit predicted temperature separation agrees with the experimental results, excepting for low value of cold mass fraction. It is clearly seen that Stepped-Section shape has an influence on vortex tube performance

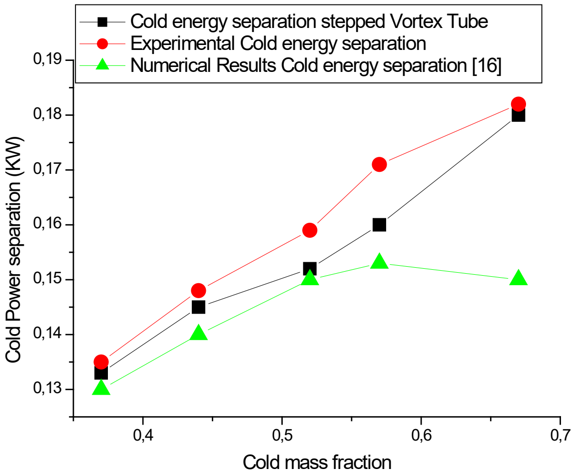


Fig. 6. Cold energy separation

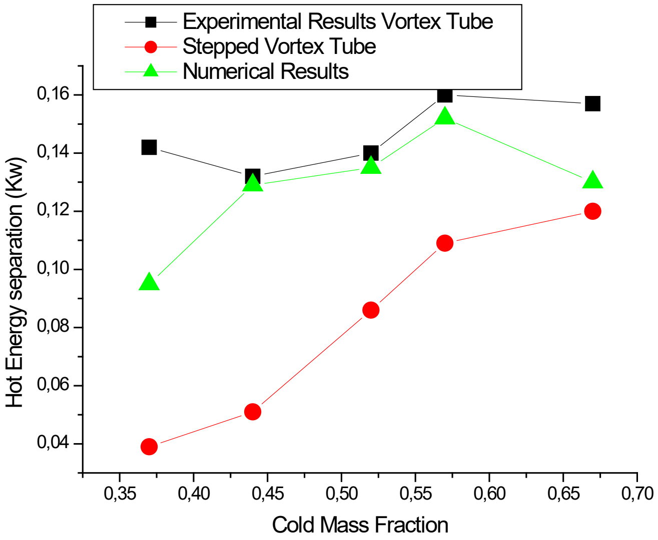


Fig. 7. Hot energy separation

The main results are shown in figures 6 and 7 the cold energy separation shows an increase in the stepped geometry comparing to the cylindrical shape. It is clearly shown that the energy separation improved due to the gain in cold temperature. On the other hand the decrease of the hot temperature separation in stepped vortex tube shape provokes a lowering in the hot power energy separation comparing to the others vortex tube geometries.

9. Coefficient of performance

The Coefficient of performance COP is a physical parameter which defines the cooling capacity in vortex tubes. If the compressed gas is air, the max COP reaches 2.5. In the case of another refrigerant, used as working fluid such as R-114, the COP max = 11.11, R-218 COP max = 15 and the n-Hptane COP max = 20 [22].

The coefficient of performance, COP, is defined by the following relationship

$$COP = \left[\left(1 - \frac{T_{Cold}}{T_{Inlet}} \right) \Big/ \ln\left(\frac{T_{Inlet}}{T_{Cold}} \right) \right]$$

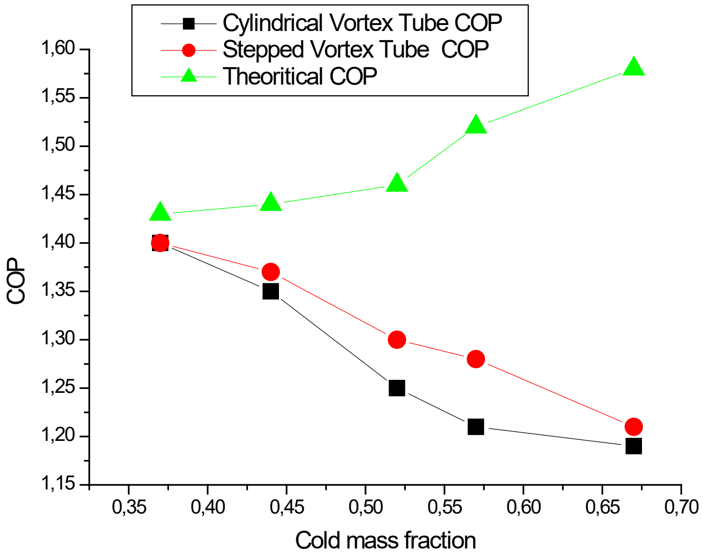


Fig..8. Coefficient of performance vs. cold mass fraction

Figure 8. represent the Coefficient of performance predicted by the experimental investigation. As seen in figure it showed a significant improvement in cooling capacity the stepped configuration comparing with cylindrical shape. It is noticed a significant gap between the maximal Theoretical COP and the other curves which due to geometrical discontinuities in steeped vortex tube sections which provokes a considerable losses.

10. Conclusions

The present purely experimental investigation of a stepped vortex tube has revealed the following results compared with those of classical vortex tube:

- An improvement in cold temperature separation from
- A significant drop in the hot temperature separation

The present results should be further enhanced by a series of parametric studies on this new configuration of vortex tube such as:

- increasing inlet nozzles
- Variation of the diameter of the two exits cold and hot
- Numbers and diameters of stepped sections
- Shape and conical angle of the valve control

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