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CFD analysis of straight and flared vortex tube

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Abstract. Vortex tube (VT) is a simple low refrigeration producing device having no moving part. However, the flow inside it is very complex. Recent studies show that the performance of VT improves with the increase in the divergence angle of a flared VT. To explore the temperature separation phenomenon in the VT, a three dimensional computational fluid dynamics (CFD) analysis of VT has been carried out. For the present work, a VT having diameter of 12 mm, length of 120 mm, cold outlet diameter of 7 mm and hot outlet annulus of 0.4 mm with 6 straight rectangular nozzles having area of 0.5 sq. mm each is considered. The turbulence in the flow field of the VT is modeled by standard k-e turbulence model considering Redlich-Kwong real gas model. The effect of variation of divergence angle of hot tube in the VT is studied and compared with the experimental results available in the literature. The temperature separation between the hot outlet and cold outlet, in both straight and 2 degree flared tube is studied. Analysis results indicate that for a hot mass fraction above 0.5, the flared tube shows better cold production capacity compared to the straight tube. Effect of important parameters like temperature gradient, velocities (axial, radial and tangential), velocity gradients, effective thermal conductivity and viscosity of fluid etc., on heat transfer and shear work transfer in the VT have been investigated. To understand the temperature separation mechanism, heat transfer and work transfer along the axial direction have been evaluated in both straight and flared tubes. The isentropic efficiency and COP as a refrigerator as well as a heat pump of straight tube and flared tube have been computed.

Keywords: Vortex tube; Temperature separation; CFD model; Redlich-Kwong real gas model.

1. Introduction

Vortex tube (VT) as shown in figure 1 is a simple and compact device used as a heat pump as well as a refrigeration device. Now-a-days, VT is extensively used in industries for small scale cooling or heating application, such as cooling of cutting tools, cooling of food products, dehumidification, electrical cabinets cooling, etc. In a VT, a high pressure gaseous fluid is injected tangentially through a set of nozzles to produce vortex. The fluid separatesinto two low pressure streams with one at a higher temperature and another at a lower temperature than inlet temperature. This separation into two different temperature streams is known as energy separation. Though it has a simple geometry and complex flow, it is difficult to understand clearly the energy separation phenomenon in a VT. Many researchers tried to explain the energy separation phenomenon and proposed mechanism about energy separation in the VT from experimental, numerical and analytical studies. Ranque [1] discovered and reported the energy separation in vortex tube and proposed that expansion and compression are the reasons for energy separation in VT. Later Hilsch [2] studied geometrical parameters to improve the

performance of VT and he proposed that the effect of inner friction is also an important factor with expansion and compression for energy separation. Gulyaev [3] experimentally investigated the effect of flared tube VT and reported improvement of about 20-25% over the straight tube. Shannak [4] suggested a new model based on energy and mass balance and proposed frictional loss as an important factor for the energy separation in VT. Aljuwayhel et al. [5] carried out a CFD analysis of VT to investigate the energy separation mechanism and explained that viscous shear force on a rotating control surface produced torque causing a work transfer between hot and cold fluid. Behara et al. [6] investigated experimentally and numerically to analyze the energy separation mechanism and flow phenomenon. They evaluated energy transfer to explain energy separation mechanism and effect of secondary flow on energy separation. Dutta et al. [7] numerically investigated energy separation in vortex tube using standard k-ε model with NIST real gas eqation of state and observed that model predicted results are closer to the experimental results.

The present study is carried out to understand the effect of divergence angle of hot tube of flared (Divergent) VT and the heat and work transfer in the vortex tube in both straight and flared tube. CFD software Ansys Fluent 14.0 is used for simulation. In this work Redlich-Kwong real gas equation of state is used to predict better results closer to experimental results [9]. The analytical results are compared with experimental results available in literature.
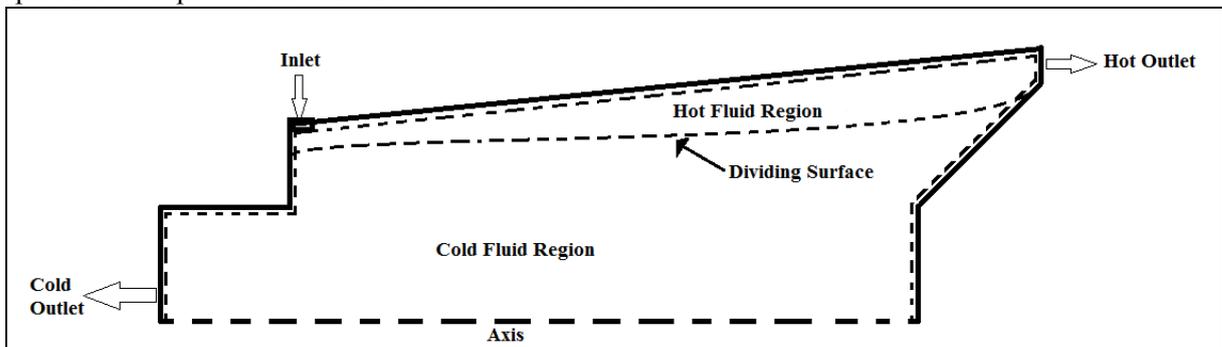

Figure 1. Counter flow flared vortex tube with control volume boundary.

2. CFD modeling of the vortex tube

A three-dimensional CFD model of 60° sector of a vortex tube was developed as shown in figure 2. The flow inside the VT is assumed to be at steady state condition. Air is used as a compressible and pure working fluid. The continuity, momentum, energy and Redlich-Kwong equation of state is solved by CFD Ansys Fluent 14.0 [9].

There are several models available in Fluent for modeling the turbulent flow of a fluid. Dutta et al. [8] suggested that the standard k-ε model predicts closer to experimental results and reasonably good for practical design purpose. Hence, in this work the standard k-ε model is used. The analysis is carried out considering Redlich-Kwong real gas equation of state with density based solver.
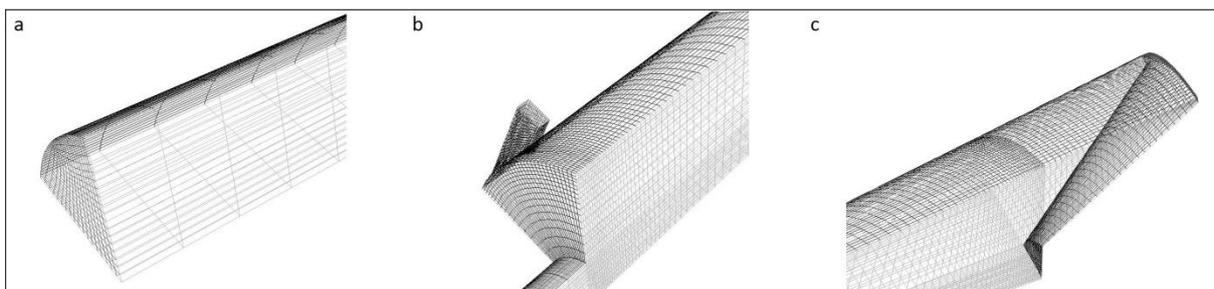

Figure 2. Mesh at (a) cold outlet, (b) inlet and (c) hot outlet of VT

For the CFD analysis of VT, the geometrical parameters of VT are following. Length of tube (L) is 120 mm, diameter of the tube (D) is 12 mm, cold end diameter (Dc) is 7 mm, inlet area of 6 rectangular nozzles is 0.5 sq. mm each and hot end annular gap is 0.4mm. An angle of divergence of hot tube (β) is 2° in case of the flared tube only.

The boundary conditions for study of VT are as follows:
- Stagnation boundary condition at the inlet is specified with total pressure of 542 kPa (a) and total temperature of 300 K.
- Pressure boundary condition at cold outlet is specified with static pressure of 136 kPa (a).
- Pressure boundary condition at hot outlet is specified with different pressure to vary the hot mass fraction.
- Zero gradient of temperature at both hot outlet and cold outlet.
- No slip and adiabatic condition at the wall.

The geometry is created and meshed using Gambit 2.3.6. A grid independence study was carried out to remove the error due to the coarseness in meshing. Mesh near the boundaries is refined. It was found that after 160000 no. of cells there is no change in the total temperature difference between hot and cold end as shown in figure 3. So the 160000 numbers of control volume cells are used for mesh generation.
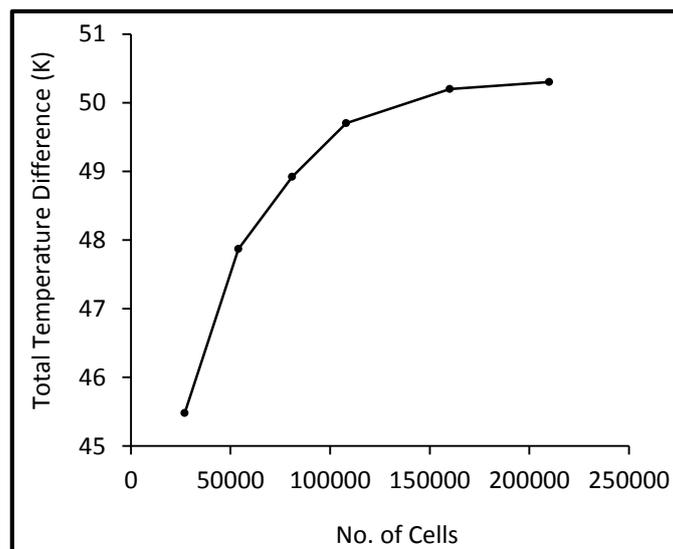

Figure 3. Grid independence study on Total temperature difference.

3. Results and discussions

A CFD simulation was carried out with the above model and predicted results are compared with the experimental results available in the literature to validate the model [14]. The cold and hot outlet temperatures based on CFD analysis of present work are compared with the experimental results. It is observed from the figure 4 (a) and 4 (b) that the maximum deviation is under 5% and thus there is good agreement between the model predicted results and experimental results. As shown in figure 4 (b), the cold outlet temperature increases with increase in the hot mass fraction above 0.5. This is also observed by Dutta et al.[8]. This happened because of backflow of ambient air at cold outlet due to larger cold outlet diameter (7 mm) and low cold mass flow rate. So the average temperature at cold outlet due to backflow is increased.
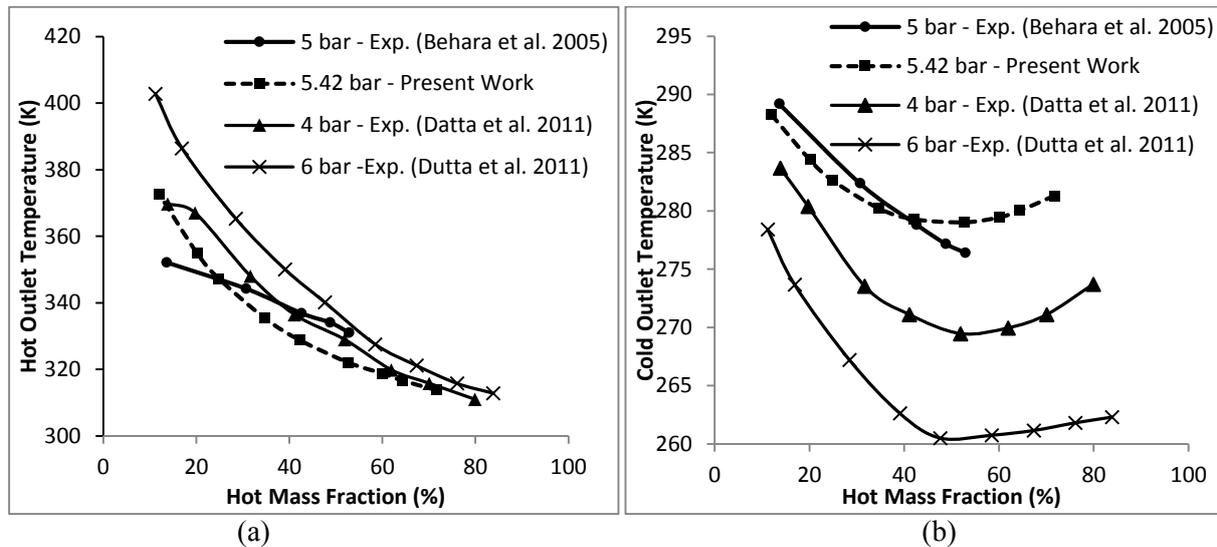
Figure 4. Comparison of experimental and CFD results.

Figure 5 (a), 5 (b) and 5 (c) show the effect on cold outlet temperature, hot outlet temperature and temperature difference between hot and cold outlet, respectively, of both straight and flared tubes on variation of hot mass fractions. It is observed that total temperature difference and hot outlet temperature decrease monotonically with increase in hot mass fraction; whereas cold outlet temperature decreases till a hot mass fraction 0.5 and 0.6 in straight and flared tube, respectively and therefore it increase. Dutta et al. [8] reported that the increase in cold outlet temperature above a specified hot mass fraction is because of backflow in the vortex tubes. Maximum axial velocity is 127 m/s and 145 m/s in straight and flared tube, respectively for 0.65 hot mass fractions. Thus,the intensity of backflow is more in case of straight tube than flared tube at the cold outlet. From figure 5 (a) and 5 (c), it is observed that there is a crossover at a hot mass fraction of 0.50 and 0.65 for cold outlet temperature and total temperature difference, respectively. The analysis results indicate that the drop in temperature at cold outlet is the maximum in the flared tube. From these results it is clear that for a hot mass fraction higher than 0.5 flared tube plays abetter role as a refrigerator. Similar observation have also been made by Pouraria and Zangooee [10].
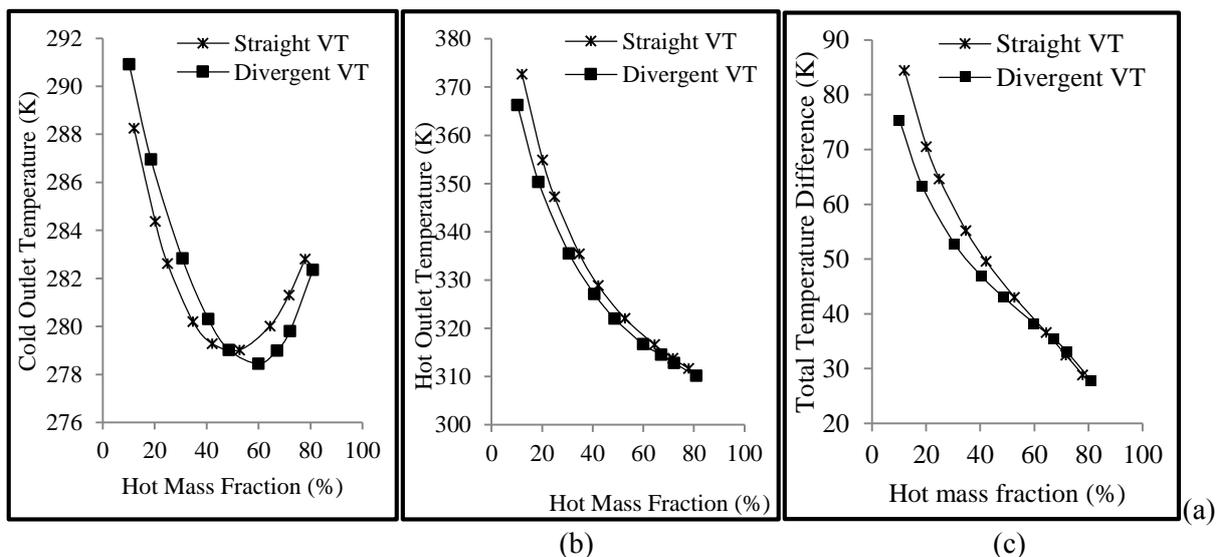

Figure 5. Comparison of (a) cold outlet temperature (b) hot outlet temperature and (c) total temperature difference respectively, with hot mass fraction for both straight and flared tubes.

The rate of energy transfer between different fluid layers in both straight and flared vortex tube is computed by considering control surfaces along axial direction at radial locations 0.5 mm to 5.8 mm in steps of 0.5 mm for a hot mass fraction of 0.65. The energy transfer per unit length in straight vortex tube at radial location 5.5 mm along axial length is shown in figure 6. The tangential shear work due to viscous shear has an important contribution in energy separation. Heat transfer is negative in vortex tube along axial direction. As observed by Dhillon [12] and Behara [13], the static temperature in the VT decreases radially i.e. inner core flow layers are at higher static temperature than the peripheral layers except for the layers near the wall. This static temperature gradient leads to the transfer of heat from higher static temperature region, i.e. core flow cold fluid layer, to the lower static temperature region, i.e. peripheral layers of hot fluid stream. Figure 7 shows the energy transfer in straight and flared vortex tubes. The following expressions are used to calculate heat and work transfer rate per unit length [12].

3.1. *Sensible heat transfer rate per unit length*

$$\frac{\delta q}{\delta z} = 6 \times \int_{0}^{\pi/6} -K_{eff}.\frac{\partial T_s}{\partial r}.r.d\theta$$

3.2. *Tangential shear work transfer rate per unit length*

$$\frac{\delta w_\theta}{\delta z} = 6 \times \int_{0}^{\pi/6} -\mu_{eff}.\left(\frac{\partial w}{\partial r} - \frac{w}{r}\right).w.r.d\theta$$

3.3. *Axial shear work transfer rate per unit length*

$$\frac{\delta w_z}{\delta z} = 6 \times \int_{0}^{\pi/6} -\mu_{eff}.\frac{\partial u}{\partial r}.u.r.d\theta$$

The maximum net energy transfer in straight and flared vortex tubesare 103 watts and 134 watts, respectively. From these results, it is observed that the energy separation in flared tube is more than that in the straight tube.
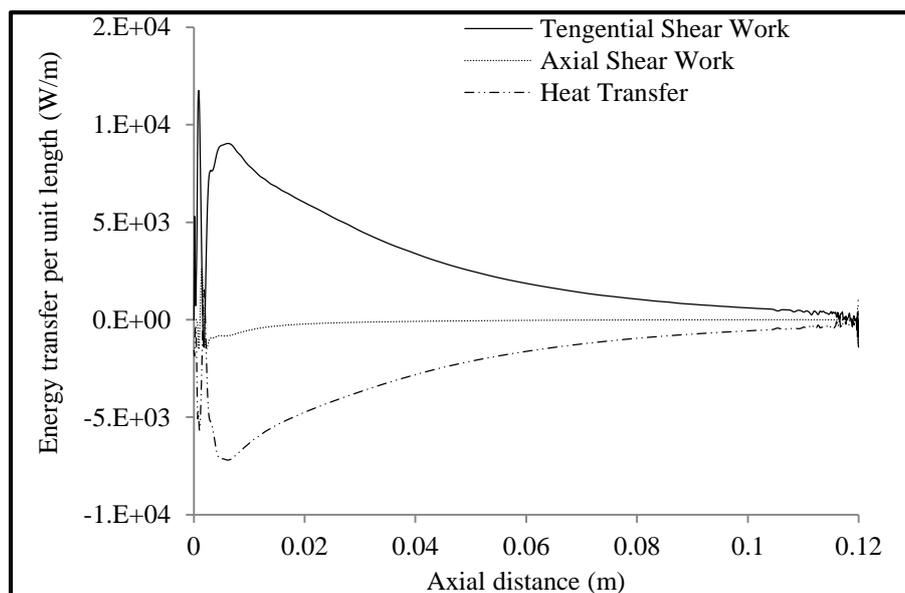

Figure 6. Energy transfer along the axial length at a radial location of 0.005 m.
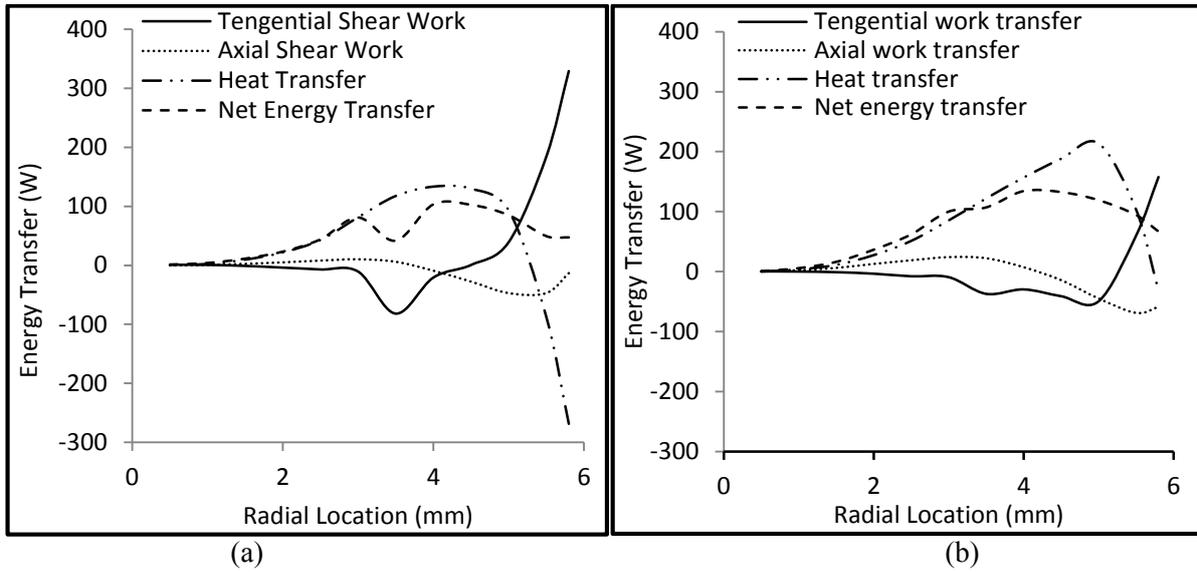
Figure 7. Energy transfer at different radial location for (a) straight VT (b) flaredVT.
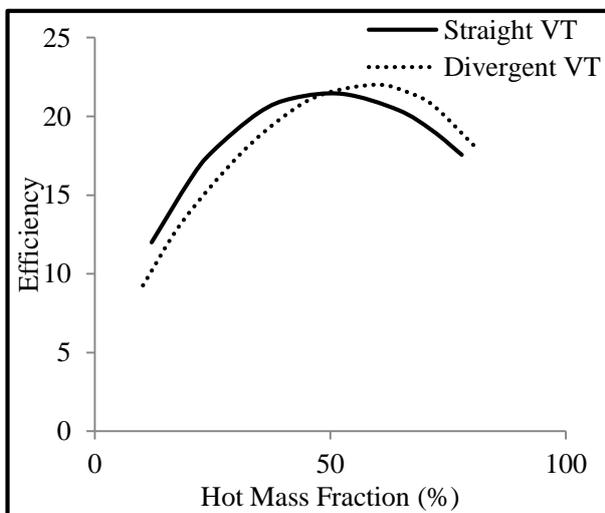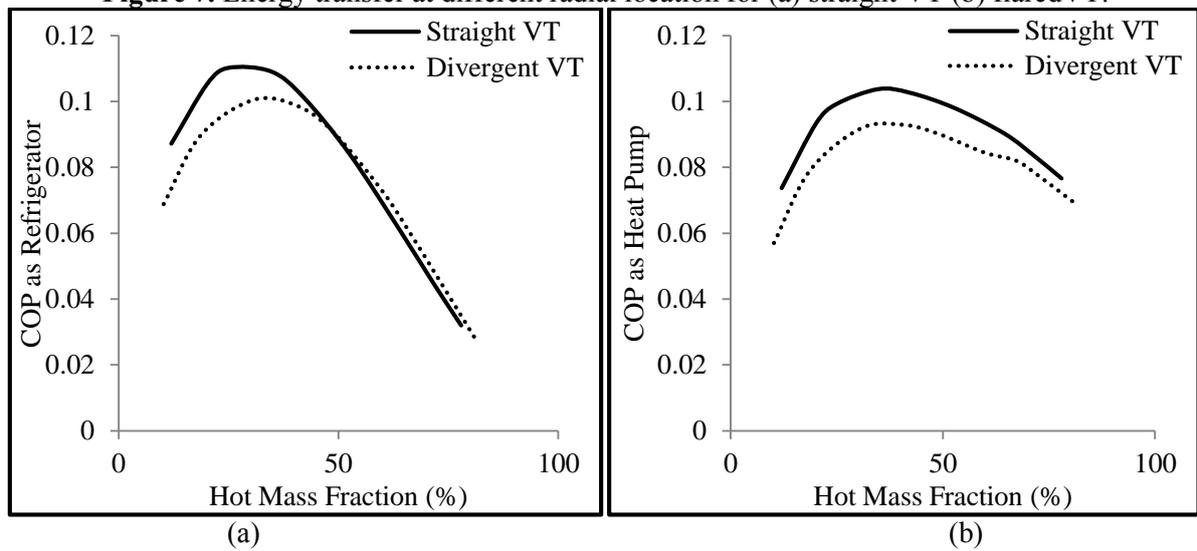
Figure 8. Comparison of (a) COP as refrigerator, (b) COP as Heat Pump and (c) efficiency of straight VT and flared VT.

The performance characteristics such as COP as refrigerator and heat pump and efficiency of both straight and flared VTs (shown in figure 8) are calculated using following expression[12] :

3.4. Isentropic efficiency. It is the ratio of actual cooling gained in VT to the possible cooling with isentropic expension.

$$\eta_{is} = \frac{T_i - T_c}{T_i(1 - (P_a/P_i)^{(\frac{\gamma-1}{\gamma})})}$$

3.5. Coefficient of performance as refrigerator. It is defined as amount of energy available for cooling to the work required to pressurise gas.

$$COP_{ref} = \frac{\gamma}{\gamma - 1} \frac{(1 - \dot{m_h})(T_i - T_c)}{T_i \ln \frac{P_i}{P_c}}$$

3.6. Coefficient of performance as heat pump. It is defined as amount of energy available for heating to the work required to pressurise gas.

$$COP_{hp} = \frac{\gamma}{\gamma - 1} \frac{\dot{m_h}(T_h - T_i)}{T_i \ln \frac{P_i}{P_c}}$$

The COP as refrigerator and heat pump vary from 0.028 to 0.110 and 0.057 to 0.104, respectively, in both type of vortex tubes. As shown in figure 8 (a) and 8 (c), energy separation is better in the straight VT than that in the flared VT upto a hot mass fraction 0.5. And, above 0.5 hot mass fraction there is a crossover. As a result, the COP as refrigerator and the efficiency of flared VT is higher than those of the straight VT above a hot mass fraction of 0.5. However, with respect to COP as refrigerator, the flared VT performs marginally better than the straight VT. And with respect to COP as heat pump, the straight VT performs better even above 0.5 hot mass fraction, while the difference between the COP curves of straight and flared VT gradually decreases beyond 0.5 hot mass fraction. Maximum COPs (refrigerator, heat pump) are (0.110, 0.104) and (0.100, 0.093) in straight and flared vortex tubes, respectively. The efficiency of vortex tubesvary from 9.27 to 22.02 % and maximum efficiency is found to be 22.02 % in flared tube.

4.   Conclusions
In this study, the temperature separation phenomenon is modeled using CFD for both straight and flared tubes. Major findings in this investigation are following:
- Flared tube performs better than a straight tube for a hot mass fraction above 0.5.
- Energy transfer along the axial direction in flaredVT is more than straight VT.
- Maximum COP as refrigerator and heat pump is observed to be 0.11 and 0.10, respectively in a straight VT and maximum efficiency is observed to be 22.02% in a flaredVT.

Nomenclature:

COP	Coefficient of performance	T	Temperature (K)
k_{eff}	Effective thermal conductivity	u	Tangential velocity (ms^{-1})
$\dot{m}$	Mass fraction	v	Radial velocity (ms^{-1})
P	Pressure (Pa)	W	Rate of work Transfer (W)
q	Rate of sensible heat transfer (W)	w	Axial velocity (ms^{-1})
r	Radial coordinate (m)	x	Cartesian coordinate

z	Axial coordinate (m)	c	Cold outlet
Greek Symbols		eff	Effective
γ	Specific heats ratio (c_p/c_v)	h	Hot outlet
η_{is}	Isentropic efficiency	i	Inlet
μ	viscosity ($\mathrm{kgm^{-1}s^{-1}}$)	ref	Refrigeration
		z	Axial
Subscript		θ	Tangential

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