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Effect of Diameter and Diameter of Orifice on the Performance of Vortex Tube

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Abstract- In this paper, effect of energy separation has been investigated by using a Computational Fluid Dynamic technique. The equations have been solved in 2D and 3D compressible and high swirl turbulent model. The effects of thermo- physical and geometrical parameters also have been considered. Different diameters of vortex tube were modeled. The results show that the temperature separation will decrease by increasing the diameter. It was also found that increasing the cold mass fraction reduces the temperature separation and efficiency. An optimum value was found for the cold outlet orifice diameter. All the calculation for finding the optimum parameters are compared with the rate of axial and tangential work and heat transfer in different control volumes.

Keywords-*Vortex tube, Temperature separation, Energy separation, turbulance, CFD model.*

I. Introduction

Vortex tube or Ranque-Hilsch vortex tube is a simple mechanical device that by no movable part can separate a normal compressed air into a hot temperature and a cold temperature gas. This instrument can be used in many industrial applications like cooling and heating devices.

Aronson [1] used vortex tube near a drilling machine. He reported that by using such a device we can improve the tool life of the drilling and cutting machines for ten times. As stated above, this instrument has no movable part so it dose not need much maintenance and the production of it is not expensive. Crocker [2] used this device for liquidation of air. In this position, the air can be separated into oxygen and nitrogen.

This device is consisting of a simple tube, an inlet, and two outlets. The inlet is also consists of one or more nozzles. The pressure and the mass flow can be adjusted by using a valve at the end of tube. An orifice can also be used at the cold outlet. We can classify vortex tube by the flow behavior into a counter flow and the standard type. In the counter flow type, the cold and hot flow exits are at the opposite sides while in parallel or standard one they both exit at the same side.

After flow injection by a compressor, the air enters from nozzles with high angular velocity. Then the gas separates into a hot flow and a cold one. This effect is called energy separation or temperature separation. The hot temperature flow moves near the periphery and the cold one moves from center to the core that is near inlet in the standard vortex tube.

The separation effect first was discovered by a French physician Ranque (1933) [3]. He stated that the separation effect is due to the pressure of inner layers to the periphery layers. The inner layers become cold while the outer layers become hot by the adiabatic expansion and compression, respectively.

This theory was based on invisid, non-conducting fluid flow and rejected by Ranque himself. His new theory was based on migration of energy between different layers. Hilsch [4], a German physician, after Ranque, stated that the angular velocity gradients in the radial direction results in frictional coupling between different layers that can cause to the migration of energy by shear works from inner layers to the outer layers.

Kassner and Knoernschild (1948) [5] and Fulton (1951) [6], derived the laws of shear stress in circular flow. They revealed that the flow changes from a free vortex to a forced vortex due to shear stresses.

Scheper (1951) [7] stated a theory based on forced convection. Kurosaka (1982) [8], found a relation between the acoustic resonance frequencies and vortex motion. He stated that the separation effect is due to damping of acoustic streaming. Alborn (2000) [9], proposed a theory named ‘‘Secondary circulation flow’’. He believed that some amount of mass flow near the cold region and entrance in a close loop works like a refrigeration device. Some analysis also done based on CFD to explain the phenomena of energy separation.

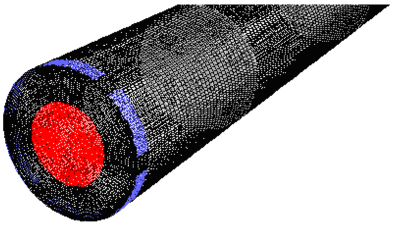
Promvonge (1997) [10] by ASM and $k-\varepsilon$ standard simulated vortex tube. Unger and Frohlingsdorf (1999) [11], numerically simulated vortex tube by a CFX code. Aljuwayhel et al. (2005) [12] used two different RANS models to predict the flow inside vortex tube. To understand the rate of work and heat transfer he separated the volume of tube into three regions. Behera et al. (2005)[13], experimentally and numerically studied the effect of nozzles (shape and number of nozzles) on the temperature separation in a counter-flow vortex tube. Skye et al. (2006) [14], simulated vortex tube in some different cold mass fractions. Saidi and Yazdi [15], experimentally studied the effect of nozzles and cold mass fraction on the flow and temperature separation.

II. GEOMETRICAL DOMAIN AND ASSUMPTIONS

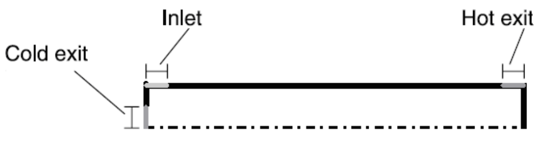
For our computation some assumptions are made:

- The gas is ideal and C_p is constant.
- The flow is supersonic.
- In 3D model the flow enters from 6 nozzles but in 2D model a circumferential inlet is considered.
- A block valve was used instead of discharge control at the hot exit.
- The 2D model is axisymetric and a periodic boundary was used for 3D model.

In Fig.1 $V_r = 0.25V_n$, $V_\theta = 0.97V_n$ .V_n is the total velocity vector , V_r and V_θ are the radial and tangential velocity components. The hot exit area is 95 mm^2 so the width l_h can be calculated as $A_h = \pi D l_h$ and for the inlet the equivalent width of slot is calculated from conservation of mass as was mentioned in [14]. The length and the radius of it are 106 and 5.7 mm, respectively. The high of each slot is 0.97 mm and the width is 1.41 mm. The cold and hot exits are 30.2 and 95mm^2. The cold mass fraction is the ratio of mass flow rate to the mass flow rate of the inlet stream ($\varepsilon = \dot{m}_c / \dot{m}_i$) and the temperature separation is defined as the difference in temperature between inlet flow and cold flow temperature ($\Delta T_{i,c}$ =T_{in}-T_c).



(a)



(b)

Fig 1. (a) 3D model, (b) 2D model

III. BOUNDRY CONDITION

- The cold outlet boundary assumed pressure outlet. The pressure of this boundary is constant for each cold mass fraction.
- The total inlet temperature is 297k.
- The cold exit and inlet static pressures were specified at experimental data of [14].
- The hot exit pressures change iteratively for getting the right cold mass fraction.
- No slip, adiabatic boundary was considered for tube walls.

IV. GOVERNING EQUATION

For motion in a compressible form, the equations are written in Favre's averaged mean motion. For 3D&2D compressible flow, the conservation of mass, momentum, energy and the state equation can be written as:

$$\frac{\partial \bar{\rho}}{\partial t} + \frac{\partial}{\partial x_i}(\bar{\rho}\,\tilde{u}_i) = 0 \quad (1)$$

$$\frac{\partial}{\partial t}(\bar{\rho}\,\tilde{u}_i) + \frac{\partial}{\partial x_j}(\bar{\rho}\,\tilde{u}_i\tilde{u}_j) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\bar{t}_{ji} - \overline{\rho u''_j\, u''_i}\right] \quad (2)$$

$$\frac{\partial}{\partial t}(\bar{\rho}E) + \frac{\partial}{\partial x_j}(\bar{\rho}\tilde{u}_j H) = -\frac{\partial}{\partial x_i}\left[-q_{L_i} - q_{T_j} + \overline{t_{ji}u''_i} - \overline{\rho u''_j \frac{1}{2} u''_i u''_i}\right] + \frac{\partial}{\partial x_j}\left[\tilde{u}_i(\bar{t}_{ij} + \tau_{ij})\right] \quad (3)$$

$$P = \bar{\rho} R \tilde{T} \quad (4)$$

V. TURBULENCE MODEL

The RANS shear stress transport (SST) $k-\omega$ model was used for our computations so we have:

$$\frac{D(\rho k)}{Dt} = \frac{\partial}{\partial x_i}\left[\Gamma_k \frac{\partial k}{\partial x_j}\right] + G_k - Y_k \quad (5)$$

$$\frac{D(\rho \omega)}{Dt} = \frac{\partial}{\partial xj}\left[\Gamma_\omega \frac{\partial \omega}{\partial x_j}\right] + G_\omega - Y_\omega + D_\omega \quad (6)$$

Where G is the generation of turbulence kinetic energy Y_k and Y_ω are the generation of K and ω, D_ω is the cross diffusion term.

$$\mu_t = \rho \frac{k}{\omega} \frac{1}{\max\left[1/\alpha^*, \Omega F_2 / \alpha_1 \omega\right]} \quad (7)$$

The Ω is mean rate of rotation and F_2 is blending function, α^* is given by

$$\alpha^* = \alpha^* \left(\alpha_0^* + \frac{\mathrm{Re}_t / R_k}{1 + \mathrm{Re}_t / R_k} \right) \quad (8)$$

Where $\mathrm{Re}_t = \frac{\rho k}{\mu \omega}$, $R_k = 6.0$, $\alpha_0^* = 0.024$.

VI. Results

To understand the phenomena of energy separation, the volume of the vortex tube was separated in three regions. 1- Hot region, 2- Cold region, 3- Re-circulating region. These three regions can be seen in Fig.2 Work transfer due to viscous shear in the tangential direction per unit length, heat, and work transfer due to axial distance can be calculated by equation (9-11) [12].

$$w_{s,\theta} = \iint_{\Omega} \tau_{r\theta} = 2\pi \mu_{eff} r_b^2 v_\theta \frac{d\omega}{dn} \quad (9)$$

$$Q = \iint_{\Omega} Q = 2\pi k_{eff} r_b \frac{dT}{dn} \quad (10)$$

$$w_{s,x} = \iint_{\Omega} \tau_{rx} = 2\pi \mu_{eff} r_b v_x \frac{dv_x}{dn} \quad (11)$$

Where μ_{eff} is the effective viscosity, v_θ is the swirl velocity and v_x, is axial velocity, k_{eff} and Ω are the effective heat transfer coefficient and area of the region.

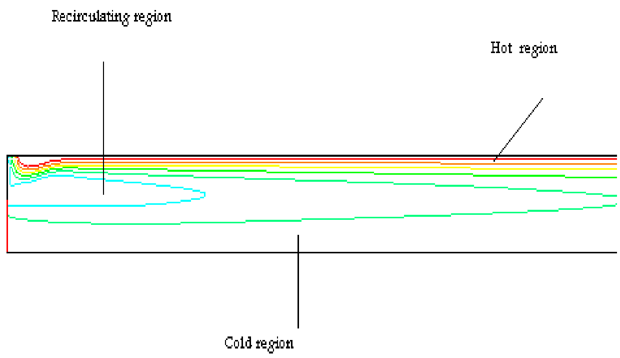


Fig.2 Three different regions for study work and heat transfer.

A. . Effect of Secondary Circulation Flow

As Alborn [13] mentioned, the existence of secondary circulation results in increasing efficiency but our CFD study with different cold mass fraction on the effect of secondary circulation flow in [16] shows that by increasing the cold mass fraction, the limit of secondary circulation fades and the energy separation increases. So the existence of secondary circulation flow will decrease the energy or temperature separation.

B. Effect of Diameter on temperature separation

To study the effect of diameter on energy separation three diameters of vortex tube were modeled (10, 16, 20mm). For each model the inlet and cold outlet pressure have the same pressure and the hot outlet pressure changes until we get to the cold mass fraction 0.37.

Table.1 shows how temperature separation changes by increasing the diameter of vortex tube. To investigate why temperature separation decreases by increasing the diameter, streamlines of three tubes was shown in Fig.3. It become obvious that by increasing the diameter, the area that was surrounded by secondary circulation flow becomes larger, so existence of such a flow mitigates the energy separation. Fig.4 shows the rate of work and heat transfer for two different tubes (10, 20 mm radius). The rate of tangential work transfer in 10 mm radius is much higher than the 20 mm one. The first reason that was mentioned before is the existence of secondary circulation flow. The second that is much important is that by increasing the diameter of the tube, the values of angular velocity and so the tangential work transfer descends. Maximum rate of tangential work transfer is about 15000 w in 10 mm diameter, while this value is about 6000 in a 20 mm diameter tube. The numerical efforts for simulation below diameters of 10 mm with the same inner pressure as other diameters simulated, was unsuccessful.

C. Effect of orifice diameter to tube diameter ratio

To investigate the effect of orifice diameter to tube diameter ratio, four models with dc/D (0.15, 0.45, 0.55, 0.71) and with 106 mm length have been studied. It can be seen in Fig.5 that on dc/D=0.45, the temperature separation is maximum. Before and after this value the temperature separation decreases. Fig.6 shows streamlines of four different models. For dc/D >0.45, the air rapidly goes out after its injection and high pressure on the cold end cause a reverse flow as its shown in Fig.7.

TABLE I. Effect of diameter on temperature separation.

D(mm)	**Temperature separation**
10	47.5
16	41
20	30

For dc/D<.5, the area that is surrounded by secondary circulation flow becomes bigger so it has negative effect on temperature separation.

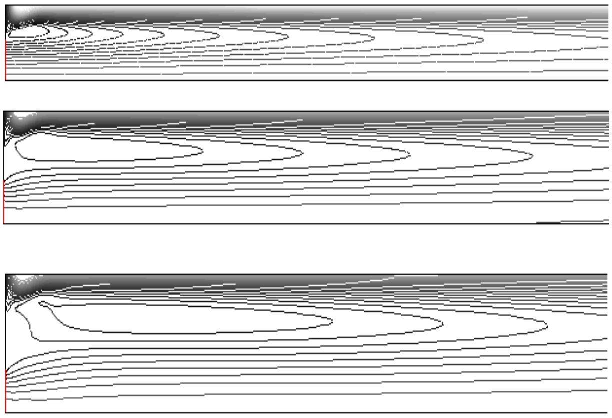
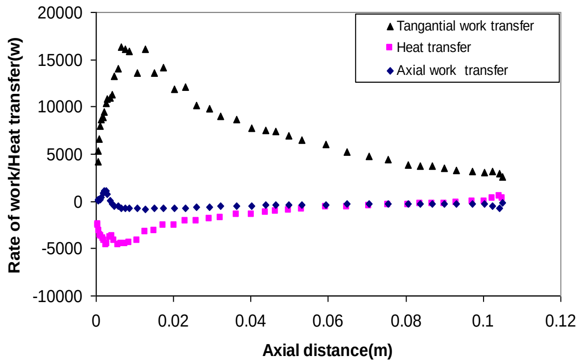
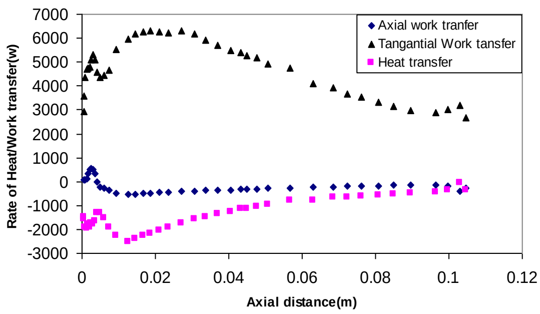


Fig 3. Streamlines of three different model of vortex tube.



(a)



(b)

Fig .4 Rate of work and heat transfer for two different radiuses (a) 5 mm (b) 10 mm.

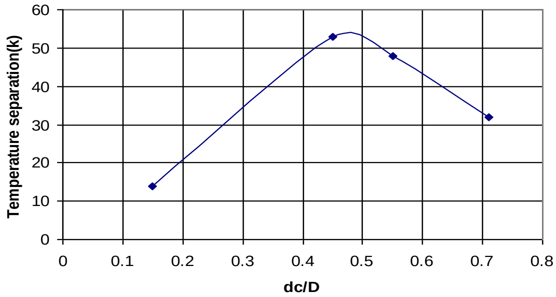


Fig .5 Effect of cold orifice diameter to tube diameter ratio on temperature separation

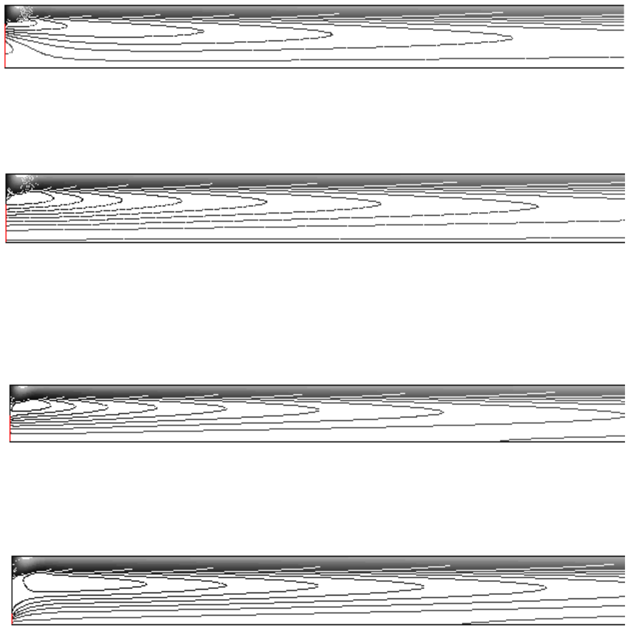


Fig .6 streamlines of four models with dc/D 0.71, 0.55, 0.45, and 0.15

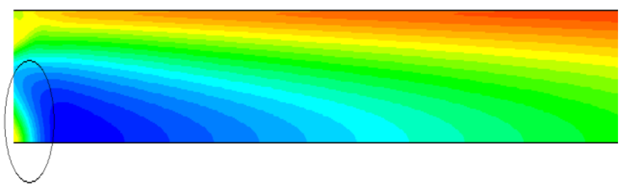


Fig .7 Reverse flow and its effect on cold exit temperature

VII. CONCLUSION

To study the effect of diameter on energy separation three different vortex tubes were modeled. By increasing the diameter of vortex tube, values of angular velocity and so tangential work transfer decreases. By increasing the diameter the area that is surrounded by secondary circulation flow becomes larger and as was described earlier the energy separation decreases by existence of such this flow. An optimum value for the best performance of tube was found on the orifice diameter of dc/Dc=0.45. On the other orifice diameters the temperature separation descends because of reverse flow or decreasing in the value of angular velocity.

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Index

D	- Diameter of vortex tube [mm]
K	- Turbulence kinetic energy($m^2\,s^{-2}$)
L	- Length [mm]
l_h	- Width of hot exit [mm]
P	- Pressure [Pa]
r	- Radial distance from axis [mm]
C_p	-Specific heat capacity[$j\ kg^{-1}\ k^{-1}$]
k_{eff}	-Effective thermal conductivity [$W\ m^{-1}K^{-1}$]
$\dot{m}$	- Mass flow rate [$kg\ s^{-1}$]
T	- Temperature [K]
V	- Velocity [$m\ s^{-1}$]
R	- Ideal gas constant [$j\ kg^{-1}\ K^{-1}$]
R,z,Θ	- Components of cylindrical coordinate system [m]
V_Θ	- Tangential velocity [$m\ s^{-1}$]
V_z	- axial velocity [$m\ s^{-1}$]
V_r	- Radial velocity [$m\ s^{-1}$]

Greek symbols

ε	- Turbulent dissipation rate [$m^2\ s^{-1}$]
μ_t	- Turbulent viscosity [$kg\ m^{-1}\ s^{-1}$]
τ	- Shear stress [$N\ m^{-2}$]
ρ	- Density [$kg\ m^{-3}$]
μ	- Dynamic viscosity [$kg\ m^{-1}\ s^{-1}$]
ξ	- Mass flow fraction
σ	- Stress [$N\ m^{-2}$]

Subscripts

i	- Inlet gas
c	- Cold gas
h	- Hot gas