The Ranque-Hilsch Effect: CFD Modeling

Vlad Bezprozvannykh[1*)], Hank Mottl[1*)]

DYCOR Technologies, 17944 – 106A Avenue, Edmonton, AB, Canada T5S1V3[1]

Email: vlad.bez@dycor.com, hank.mottl@dycor.com

ABSTRACT

The phenomenon of noticeable temperature distribution in confined steady rotating gas flows is referred as Ranque-Hilsch effect. The simple counter-flow Ranque-Hilsch tube consists of a long hollow cylinder with tangential nozzles at one end for injecting compressed gas. Rotating gas escapes the tube through two outlets – a central orifice diaphragm placed near the injection nozzle plane (cold stream) and a ring-shaped peripheral outlet placed at the opposite end of the tube (hot stream). The flow is essentially three-dimensional, turbulent, compressible, and spinning such that any theoretical simplifications are questionable, if at all possible.

Fluent suite of software was applied at Dycor Technologies, Canada to the task of numerical simulation of the Ranque-Hilsch effect that is part of Dycor's program of fundamental and applied research. The behavior of two types of fluids in the Ranque-Hilsch tube was investigated – air and water. Three-dimensional continuity, momentum, and energy equations were solved for incompressible and compressible flows for water and air cases correspondingly. The Reynolds stress turbulence model was originally applied to close the Reynolds-averaged Navier-Stokes equations.

The visualization of the velocity and temperature fields inside the vortex tube helped to understand the details of fluid flow. It was shown that CFD approach is applicable for simulating Ranque-Hilsch effect. It was found that various levels of complexity in turbulence modeling are suitable for vortex tube analysis. No vortex effect was observed for incompressible flow. The dependence of vortex tube cooling ability on initial gas pressure was investigated. Numerical simulation data are consistent with available experimental results.

1. RANQUE-HILSCH (VORTEX) EFFECT

Ranque [1] reported the phenomenon of noticeable temperature distribution in confined steady rotating gas flows in 1933. Hirsch [2] performed the detailed examination of the effect. He also suggested the working vortex tube design. The simple counter-flow vortex tube consists of a long hollow cylinder with tangential nozzles at one end for injecting compressed gas. Rotating gas escapes the tube through two outlets – a central orifice diaphragm placed near the injection nozzle plane (cold stream) and a ring-shaped peripheral outlet placed at the opposite end of the tube (hot stream).

The area of practical vortex effect applications is wide – ranging from the local cooling industrial devices to mixture separation equipment. As an energy re-distribution tool without any moving parts, the vortex tube is an attractive instrument for inventors and designers in spite of its low efficiency [3]. As to the theoretical description of processes governing the energy separation in the vortex tube, the number of suggested theories is comparable with the number of Ranque-Hilsch effect researchers. Even the basic physics of the phenomenon is still under question, as a vortex tube can be considered a practical embodiment of Maxwell's hypothetical demon, detaching the molecules with high kinetic energy from homogeneous gaseous medium [3]. Basically,

*) This work was supported under DYCOR, Canada Internal Development Project #00023. Authors gratefully acknowledge the support.

there are two schools of researchers explaining the Ranque-Hilsch effect. The first school, the majority, considers the interaction of compressibility, shear stresses and turbulent pulsations as the basis for energy separation [4]. The second school believes that the acoustic effects cause that separation [5].

2. THE MODELING APPROACH AND VORTEX TUBE PARAMETERS

Modern computational fluid dynamics provides researchers with an excellent means for analyzing complex fluid flows. CFD tools are especially helpful in cases like flow in a vortex tube when capabilities for modeling several simultaneous physical processes are desirable. The Ranque-Hilsch effect itself, on the other hand, represents the physical situation that can help to verify the ability of CFD software to simulate the real world processes. The flow is essentially three-dimensional, turbulent, compressible, and spinning such that any theoretical simplifications are questionable, if at all possible.

As to the authors' knowledge, Fluent Inc. at present time provides one of the most advanced universal fluid mechanics software tools able to handle complex multi-physics of fluid and gas flows. The Fluent suite of software was applied at Dycor Technologies, Canada to the task of numerical simulation of the Ranque-Hilsch effect that is part of Dycor's program of fundamental and applied research.

The modeled vortex tube's (Figure 1) relative geometrical parameters are considered to be optimal [6]: L/D = 5.5, where L is the tube length, and D is the diameter; two inlet tubular nozzles with their total relative area $S_n/(\pi D^2/4) = 0.14$; the cold air relative outlet diameter $d_c/D = 0.37$; the relative hot air outlet ring-shaped area $S_h/(\pi D^2/4) = 0.052$. Other researchers [7] believe that different sets of geometrical parameters are optimal.

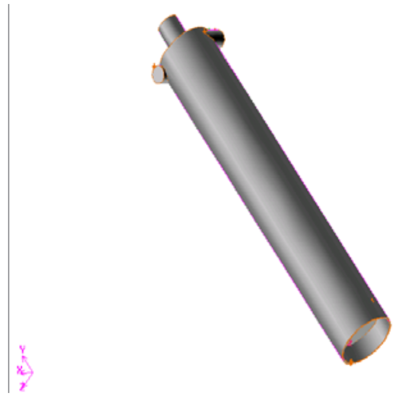


Figure 1. Simple vortex tube with two tangential inlet nozzles

The behavior of two types of fluids in the Ranque-Hilsch tube was investigated – air and water. Water was modeled as incompressible fluid with constant density. Air was considered as a compressible fluid with the ideal gas law for governing density-pressure-temperature relationship. Three-dimensional continuity, momentum, and energy equations were solved for incompressible and compressible flows for water and air cases correspondingly. The Reynolds stress turbulence model was applied to close the Reynolds-averaged Navier-Stokes equations. The concept of Reynolds' analogy of turbulent heat transport to turbulent momentum transport was applied with the Reynolds stress model for convective heat transfer modeling. It was expected that this model of turbulence would produce more accurate results compared to one- or two-equation models because it does not use the isotropic eddy-viscosity hypothesis and thus takes into account the effects of swirl and rapid changes in strain rate.

3. THE RESULTS OF NUMERICAL SIMULATIONS

The visualization of the velocity and temperature fields inside the vortex tube helped to understand the details of fluid flow. The following two figures illustrate the distribution of air velocities and temperatures in the interior of the Ranque-Hilsch tube for the case of 0.5 MPa inlet pressure.

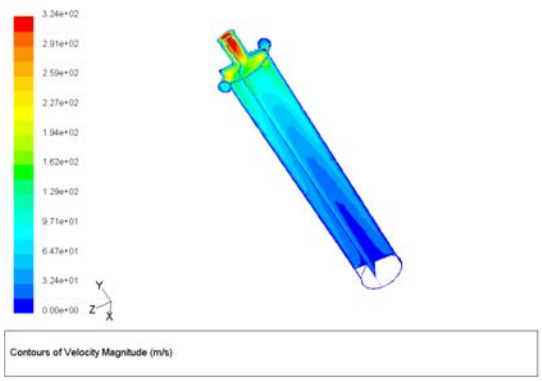


Figure 2. Air velocity contours inside the vortex tube

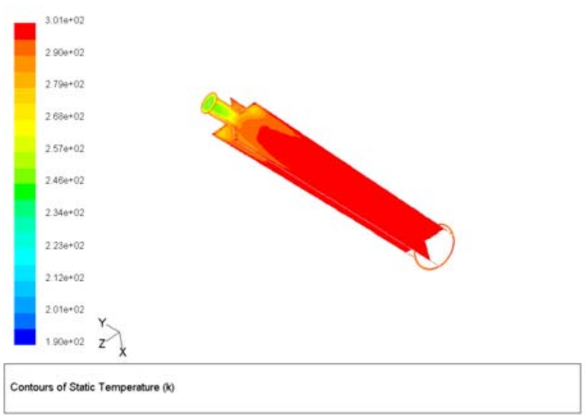


Figure 3. Temperature distribution inside the vortex tube

The contours of the air stagnation temperature are shown at Figure 4. The area of minimal energy is around the tube axis near the inlet nozzles. The area of maximal gas energy is near the hot outlet ring. The radial differential of the stagnation temperature peaks at the inlet nozzles section. This physical picture is confirmed by the measurements of the stagnation temperature in counter flow vortex tubes (see [7]).

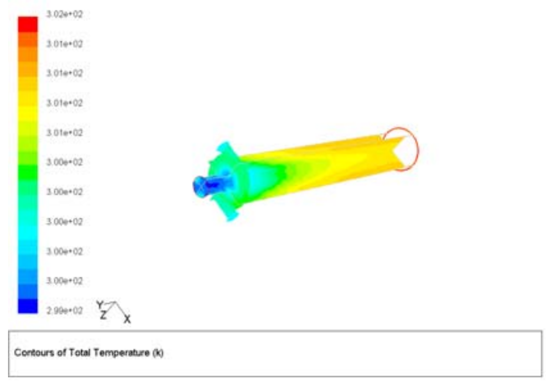


Figure 4. Distribution of the stagnation temperature

Various levels of accuracy in turbulence modeling for simulating Ranque-Hilsch effect were tested. It was found that k-ω two-equation turbulence model and Spallart-Allmaras one-equation model (known as giving good results for boundary layers subjected to adverse pressure gradients) provide the velocity and temperature distributions similar to those built with Reynolds stress model of turbulence. As to the attempts to get the vortex effect for incompressible fluid, they are failed. For water, no measurable temperature distribution in the tube inlet cross-section was found.

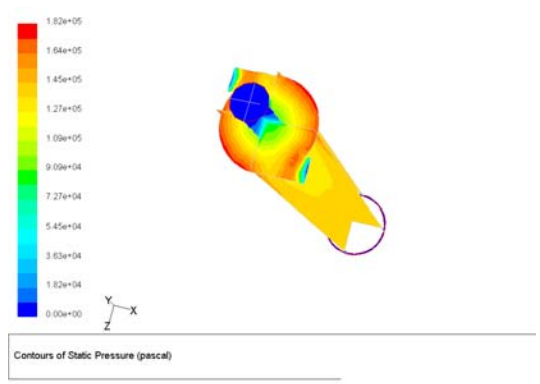


Figure 5. Static pressure distribution

According to the results of numerical simulations the following qualitative explanation of the Ranque-Hilsch effect seems to be reasonable. Expanding from inlet nozzle stream of compressed air transforms into highly intensive swirl flow with significant radial pressure gradient (data on pressure distribution are shown at Figure 5). The flow is turbulent, and turbulent eddies can travel in the tube cross-section to the center of tube as well as to the peripheral layers. Micro-volumes of fluid traveling from the central core of the vortex to its peripheral area with relatively higher pressure are compressed with corresponding heating. Fluid micro-volumes moving to the center of the tube are expanding with cooling. The higher values of initial gas pressure will intensify all the processes responsible for energy exchange and increase the cooling ability of the vortex tube. Figure 6 represents numerical data on the level of increasing vortex tube cooling ability with rising pressure at the inlet nozzle. Numerical analysis results are consistent with original experimental data of R.Hilsch for similar geometry vortex tube.

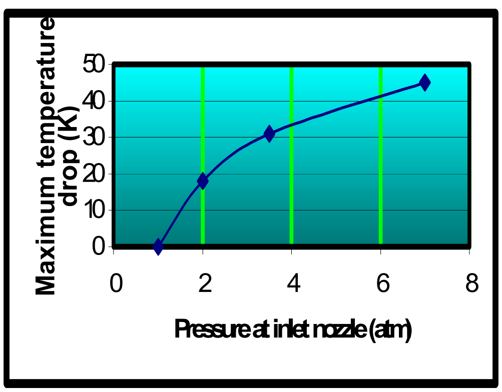


Figure 6. Dependence of vortex tube cooling ability on inlet nozzle pressure (line – numerical simulation results, points – original R.Hilsch's data

In conclusion, it is shown that CFD approach is applicable for simulating Ranque-Hilsch effect. The details of flow velocity and temperature distribution are presented. It is found that various levels of complexity in turbulence modeling are suitable for vortex tube analysis. No vortex effect is observed for incompressible flow. The dependence of vortex tube cooling ability on initial gas pressure is investigated. Numerical simulation data are consistent with available experimental results.

REFERENCES

1. Ranque G. Expériences sur la détente giratoire avec productions simultanées d'un énchappement d'air chaud et d'un énchappement d'air froid. J. de Physique el de la Radium 1933; 4. P.1125.

2. Hilsch R. The use of the expansion of gases in a centrifugal field as a cooling process. The Review of Scientific Instruments. 1947 February; 18(2). p.108.

3. Coccerill T. Thermodynamics and fluid mechanics of a Ranque-Hilsch vortex tube. Cambridge: University of Cambridge; 1998.

4. Deissler R, Permutter M. Analysis of the flow and energy separation in a turbulent vortex. International Journal of Heat and Mass Transfer. 1960; 1. p.173.

5. Kurosaka M. Acoustic streaming of swirling flow. Journal of Fluid Mechanics. 1982 November; 124. p.139.

6. Gupta K, Lilly D, Syrod N. Swirl flows. England: Abacus Press; 1984.

7. Gutsol A. The Ranque effect. Physics – Uspekhi. 1977; 40(6). p.639.

An Experimental Performance Evaluation of Vortex Tube

P K Singh, *Non-member*
R G Tathgir, *Non-member*
D Gangacharyulu, *Non-member*
G S Grewal, *Non-member*

The vortex tube is a simple device, having no moving parts, which produces hot and cold air streams simultaneously at its two ends from a source of compressed air. Literature review reveals that there is no theory so perfect, which gives the satisfactory explanation of the vortex tube phenomenon as explained by various researchers. Therefore, it was thought to carryout experimental investigations to understand the heat transfer characteristics in a vortex tube with respect to various parameters like mass flow rates of cold and hot air, nozzle area of inlet compressed air, cold orifice area, hot end area of the tube, and L/D_T ratio. The experimental investigations were carried out based upon two designs, ie, maximum temperature drop tube design and maximum cooling effect tube design. It is observed that the effect of nozzle design is more important than the cold orifice design in getting higher temperature drops. Cold fraction as well as adiabatic efficiency are more influenced by the size of the cold orifice rather than the size of the nozzle. Higher temperature drops are obtained in vortex tube made of maximum temperature drop tube design, whereas, more cold fraction and higher adiabatic efficiency are obtained with maximum cooling effect tube design. Length of the tube has no effect on the performance of the vortex tube in the range of 45 to 55 (L/D_T ratio).

Keywords: Vortex tube; Heat transfer characteristics; Nozzle diameter; Cold orifice diameter; Cold and hot air temperatures

NOTATION

A_C, A_H : areas of the cold orifice and at the hot end, respectively, mm^2

A_N, A_T : areas of the nozzle and the tube, respectively, mm^2

D_C, D_N, D_T : diameters of the cold orifice, the nozzle and the tube, respectively, mm

L : length of the tube, mm

m_C, m_H : mass flow rates of cold and hot air, respectively, kg/s

m_i : mass flow rate of air at inlet nozzle, kg/s

T_a : ambient temperature, °C

T_C, T_H : temperature of air at cold and hot ends, respectively, °C

T_o : temperature of air inside the compressor tank, °C

v_C, v_H : velocity of the air at cold orifice and at hot end, respectively, m/s

ΔT_C : temperature drop in vortex tube, °C

$\Delta T'_C$: temperature drop due to adiabatic expansion, °C

ρ_C, ρ_H : density of air at cold orifice and at hot end, respectively, kg/m^3

α_1 : A_N/A_T

α_2 : A_C/A_T

α_3 : L/D_T

α_4 : hot end area/tube area, A_H/A_T

β_1 : m_C/m_i

β_2 : $\beta_1 \times \Delta T_C / \Delta T'_C$

γ : specific heat ratio

P K Singh is with Department of Mechanical Engineering, Government Polytechnic GTB Garh, Moga, Punjab; R G Tathgir and D Gangacharyulu are with Mechanical and Chemical Engineering Departments, respectively, Thapar Institute of Engineering and Technology, Patiala 147 001; while G S Grewal was with Thapar Centre for Industrial Research and Development, Patiala 147 001.

This paper was received on August 19, 2002. Written discussion on the paper will be entertained till April 30, 2004.

INTRODUCTION

The vortex tube also known as Hilsch or Ranque tube[1,2,3,9,11,13] is a simple device having no moving parts, which produces hot and cold air streams simultaneously at its two ends from a source of compressed air. It consists of a long tube having a tangential nozzle near one end and a conical valve at the other end, as shown schematically in Figure 1.

A diaphragm called cold orifice, with a suitable sized hole in its centre is placed immediately to the left of the tangential

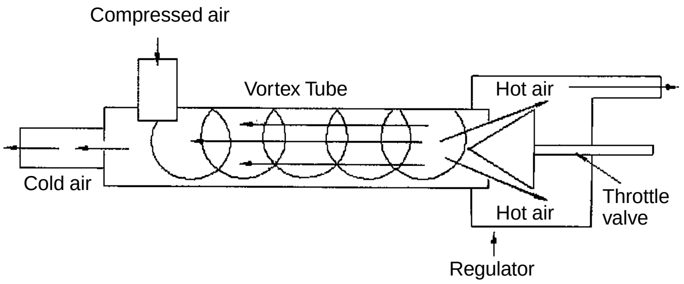


Figure 1 Schematic diagram of a vortex tube

inlet nozzle. The compressed air is then introduced into the tube through this nozzle. The tangential flow imparts a whirling or vortex motion to the inlet air, which subsequently spirals down the tube to the right of the inlet nozzle. Conical valve at right end of the tube confines the exiting air to regions near the outer wall and restricts it to the central portion of the tube from making a direct exit. The central part of the air flows in reverse direction and makes exit from the left end of the tube with sizeable temperature drop, thus creating a cold stream. The outer part of the air near the wall of the tube escapes through the right end of the tube and is found to have temperature higher than that of inlet air.

Literature review[1-15] reveals that there is no theory so perfect, which gives the satisfactory explanation of the vortex tube phenomenon as explained by various researchers. Therefore, it was thought to carryout experimental investigations for understanding the heat transfer characteristics in a vortex tube. An attempt has been made to study the effect of various parameters like mass flow rates of cold and hot air, nozzle area of inlet compressed air, cold orifice area and hot end area of the tube. The study also investigates the effect of L/D_T ratio on the performance of the vortex tube.

DESIGN AND CONSTRUCTIONAL FEATURES

In general, there are two design features associated with a vortex tube, namely, (a) maximum temperature drop vortex tube design for producing small quantity of air with very low temperatures; and (b) maximum cooling effect vortex tube design for producing large quantity of air with moderate temperatures. These two design considerations have been used in the present study. The parameters investigated in the study, to understand their inter-relationships and their effect on the performance of the vortex tube, are:

- nozzle diameter;
- cold orifice diameter;
- mass flow rates of cold and hot air;
- length of the tube; and
- area at the hot end.

In the present investigation, a nozzle area to tube area ratio of 0.11 ± 0.01 for maximum temperature drop and a ratio of 0.084 ± 0.001 for achieving maximum efficiency has been considered, as suggested by Soni and Thomson[9]. Further, they suggested that the ratio of cold orifice area to tube area should be 0.080 ± 0.001 for achieving maximum temperature drop and it will be 0.145 ± 0.035 for attaining the maximum efficiency. Also, the same researchers suggested that the length of the vortex tube should be greater than 45 times the tube diameter but no upper limit was specified.

Keeping these suggestions in view, a vortex tube of size 15.30 mm diameter of PVC was selected. The casing of the vortex tube was a 64 mm long CI cylinder. Two sets of nozzles were made and each set consisted of two nozzles. The first set of nozzles had a 3.1 mm diameter hole and was 40 mm long, whereas the second set nozzles had a 3.5 mm diameter hole and was 38 mm long. Similarly, two cold orifices were used having hole diameters of 4.5 mm and 5.8 mm, respectively. A conical valve made of mild steel was provided on the right hand side of the tube to regulate the flow. Also, a muffler was provided to reduce the noise levels at the hot end.

EXPERIMENTAL SETUP

To study the effect of various parameters such as mass flow rates of cold and hot air, nozzle area, cold orifice area, hot end area and L/D_T ratio on the performance of the vortex tube, an experimental set up was prepared as per design described earlier. The vortex tube components were made in a manner such that the geometry of the tube could be changed from maximum temperature drop tube design to maximum cooling effect tube design. The line diagram of the experimental setup is shown in Figure 2. The temperature of air at cold and hot ends was measured with digital thermometers with an accuracy of $\pm 0.1\,^{\circ}C$. The air velocities were measured made by an anemometer. The pressures were measured by the Bourden tube pressure gauges. All the instruments were calibrated before the measurements were actually observed/ recorded.

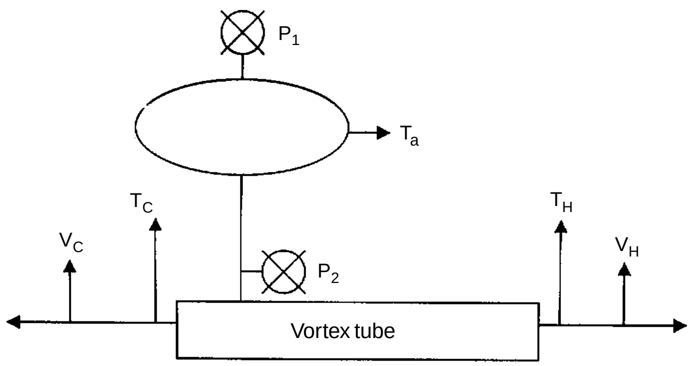
Throughout the experiment, the inlet compressed air pressure was maintained at 4.1 bar. The operating parameters noted during the experiment for each vortex tube design were:

- air pressure in compressor receiver, bar;
- air pressure near inlet to vortex tube, bar;
- compressed air temperature, $^{\circ}C$;
- cold air temperature, $^{\circ}C$;
- hot air temperature, $^{\circ}C$;
- cold air velocity, m/s;
- hot air velocity, m/s;
- hot end area, mm^2; and
- cold orifice area, mm^2.

TEST PROCEDURE

The various parameters, which affect the performance of the vortex tube, were divided into following input and output variables:

Input variables: Ratio of nozzle area to tube area (α_1), ratio of cold orifice area to tube area (α_2), L/D_T ratio of the tube (α_3) and ratio of hot end area to the tube area (α_4).



P_1 : air pressure in compressor receiver, kg/cm^2; P_2 : air pressure near inlet to vortex tube, kg/cm^2; T_a : compressed air temperature, °C; T_C : cold air temperature, °C; T_H : hot air temperature, °C; v_C : cold air velocity, m/s; v_H : hot air velocity, m/s

Figure 2 Line diagram of the experimental setup

Output variables: Temperature of cold air (T_C), cold fraction (β_1), and adiabatic efficiency (β_2).

The compressor was initially run for about 30 min to get a stable compressor air tank pressure of 5.1 bar and a line pressure of 4.1 bar. Firstly, two sets of nozzle diameters were chosen. First value represented the nozzle size of the 'maximum temperature drop tube design' whereas the second value represented the nozzle of the 'maximum cooling effect tube design'. Then for each set of nozzles, two values of cold orifices were chosen as given by the two design criteria described earlier. Now for each cold orifice area, three lengths of the tubes were selected. Further, for each length of tube four hot end openings were chosen. For each reading, the value of one input variable was changed, keeping the other input variables constant. Two readings of each output variable were taken for each run of the experiment. The values of density of air at both ends were calculated assuming the flow to be adiabatic.

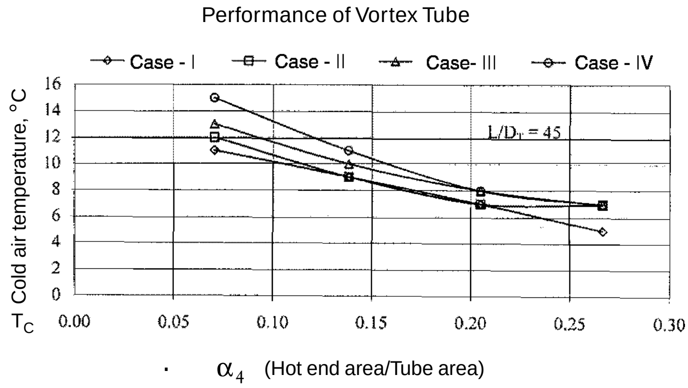
RESULTS AND DISCUSSION

Since, the study was conducted based upon two tube-designs, the effect on output variables needed to be concluded. Three output variables, namely, temperature of cold air T_C, cold fraction (β_1) and adiabatic efficiency (β_2) were evaluated and plotted for different tube designs against the hot end area. Also, the effects of parameters such as nozzle area and cold orifice area on the output variables were studied by mixing the two tube designs.

Figure 3 shows the variation of temperature of cold air (T_C), with respect to change in the hot end area for four design conditions. It shows the variation of (T_C) for a L/D_T ratio of 45 and it is observed that the temperature of cold air in all the cases decreases as the hot end is opened, which means as the amount of cold air is reduced its temperature gets lowered. It is also observed that temperatures of cold air in Case 1 were highest whereas in Case 2 it was minimum. It clearly indicates that for obtaining low temperatures, nozzle as well as cold orifice should be of 'maximum temperature drop tube design'.

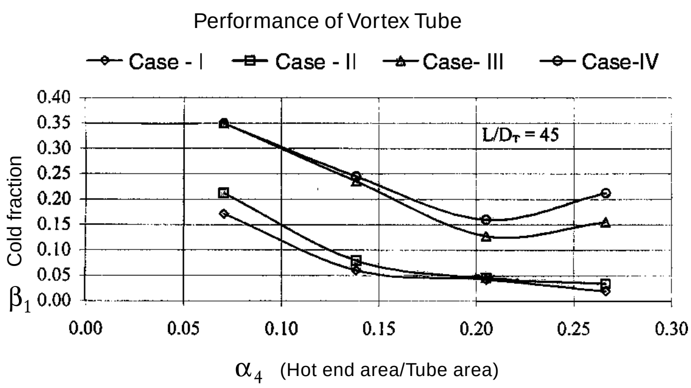
The temperatures of cold air observed in Case 3 (where nozzle is of 'maximum temperature drop tube design') were higher as compared to that of Case 4 (where nozzle is of 'maximum cooling effect tube design') for all the values of hot end area. It indicates that the effect of nozzle area is more prominent in getting higher temperature drops. Similar, effects were observed for vortex tube having L/D_T of 50 and 55.

Figure 4 shows the variation of cold fraction (β_1) with respect to change in hot end area for four design conditions as explained earlier. The figure has been plotted for L/D_T = 45. It is observed from the graph that the cold fraction is highest in the Case 2 (where nozzle and cold orifice were of 'maximum cooling effect tube design'). Also, the cold fractions observed in Case 3 were higher as compared to that of the Case 4. It



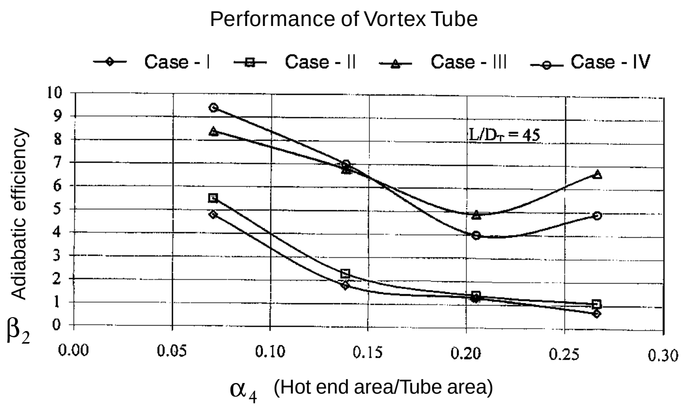
Case 1 : nozzle of 'maximum temperature drop tube design' ; cold orifice of 'maximum temperature drop tube design'; Case 2 : nozzle of 'maximum cooling effect tube design' ; cold orifice of 'maximum cooling effect tube design'; Case 3 : nozzle of 'maximum temperature drop tube design' ; cold orifice of 'maximum cooling effect tube design'; and Case 4 : nozzle of 'maximum cooling effect tube design' ; cold orifice of 'maximum temperature drop tube design'

Figure 3 Variation of temperature of cold air with respect to change in hot end area



Case 1 : nozzle of 'maximum temperature drop tube design' ; cold orifice of 'maximum temperature drop tube design'; Case 2 : nozzle of 'maximum cooling effect tube design' ; cold orifice of 'maximum cooling effect tube design'; Case 3 : nozzle of 'maximum temperature drop tube design' ; cold orifice of 'maximum cooling effect tube design'; and Case 4 : nozzle of 'maximum cooling effect tube design' ; cold orifice of 'maximum temperature drop tube design'

Figure 4 Variation of cold fraction with respect to change in hot end area



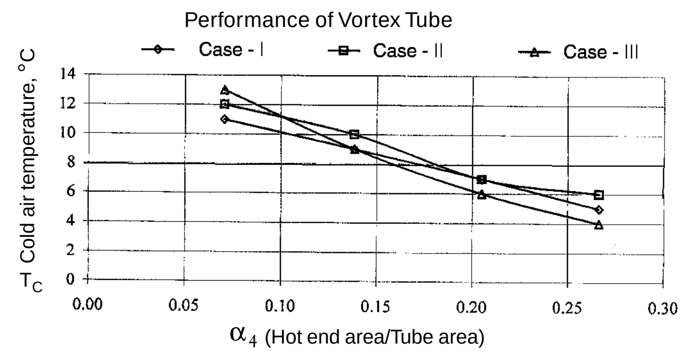
Case 1 : nozzle of 'maximum temperature drop tube design' ; cold orifice of 'maximum temperature drop tube design'; Case 2 : nozzle of 'maximum cooling effect tube design' ; cold orifice of 'maximum cooling effect tube design'; Case 3 : nozzle of 'maximum temperature drop tube design' ; cold orifice of 'maximum cooling effect tube design'; and Case 4 : nozzle of 'maximum cooling effect tube design' ; cold orifice of 'maximum temperature drop tube design'

Figure 5 Variation of adiabatic efficiency with respect to change in hot end area

indicates that the cold fraction is more influenced by the size of the cold orifice rather than the size of the nozzle. Similar, effects have been observed in case of vortex tube having L/D_T of 50 and 55.

Figure 5 shows the variation of adiabatic efficiency (β_2) with respect to change in hot end area for four design conditions. The figure shows similar pattern as observed in the case of cold fraction. It is observed from the figure that adiabatic efficiency is highest in the Case 2 (where nozzle and cold orifice were of 'maximum cooling effect tube design'). Also, the adiabatic efficiencies observed in Case 3 were higher as compared to those observed in Case 4. It indicates that adiabatic efficiency is also influenced by the size of the cold orifice rather than the size of the nozzle. Similar, effects have been observed for vortex tube having L/D_T of 50 and 55.

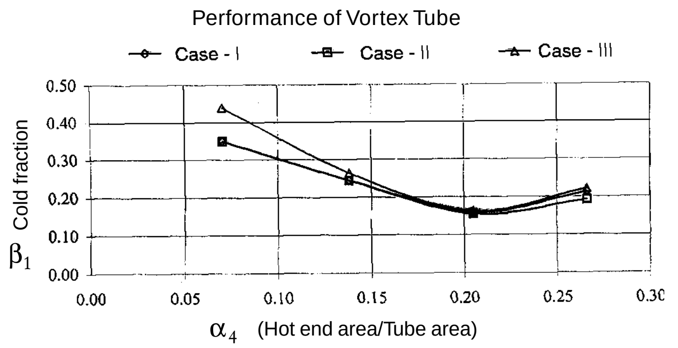
Figure 6 shows the variation of temperature of cold air (T_C) with respect to change in the hot end area for the three cases.



nozzle of 'maximum temperature drop tube design' ; cold orifice of 'maximum temperature drop tube design'

Case 1 : L/D_T = 45 ; Case 2 : L/D_T = 50; and Case 3 : L/D_T = 55

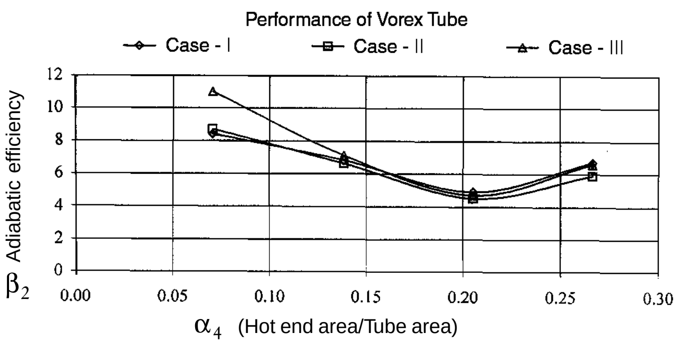
Figure 6 Variation of cold air temperature with respect to change in hot end area



nozzle of 'maximum cooling effect tube design' ; cold orifice of 'maximum cooling effect drop tube design'

Case 1 : L/D_T = 45; Case 2 : L/D_T = 50; and Case 3 : L/D_T = 55

Figure 7 Variation of cold fraction with respect to change in hot end area



nozzle of 'maximum cooling effect tube design' ; cold orifice of 'maximum cooling effect drop tube design'

Case 1 : L/D_T = 45; Case 2 : L/D_T = 50; and Case 3 : L/D_T = 55

Figure 8 Variation of adiabatic efficiency with respect to change in hot end area

Case 1 is for $L = 45\ D_T$, Case 2 is for $L = 50\ D_T$, and Case 3 is for $L = 55\ D_T$. It is observed that all the cases show very similar trends, which indicate that the length of the tube has no effect on the performance of the tube when the length is increased beyond 45 D_T up to 55 D_T. Similar, results have been observed for cold fraction (β_1) (Figure 7) and for adiabatic efficiency (β_2) (Figure 8).

All the graphs of Case 2 and Case 3 show the start of reversed trend when the hot end opening is increased beyond 20% of the tube cross sectional area.

CONCLUSIONS

1. The effect of nozzle design is more important than the cold orifice design in getting higher temperature drops.

2. Cold fraction as well as adiabatic efficiency is more influenced by the size of the cold orifice rather than the size of the nozzle.

3. Higher temperature drops are obtained in vortex tube made of maximum temperature drop tube design, whereas, more cold fraction and higher adiabatic efficiency are obtained with maximum cooling effect tube design.

4. Length of the tube has no effect on the performance of the tube when it is increased beyond 45 D_T up to 55 D_T.

5. Two design conditions (Case 1 and Case 2) show a start of reverse trend when the hot end area is increased beyond 20% of the tube area.

ACKNOWLEDGEMENTS

The authors express their gratitude to Dr S C Saxena, Director, Thapar Institute of Engineering and Technology and Dr M P Kapoor, Director, Thapar Centre for Industrial R&D, Patiala for providing necessary facilities and encouragement. The authors also express their sincere thanks to Mr Sohan Lal, Laboratory Superintendent and Ms Anu Parashar, Office-cum-Technical Assistant for their continuous help in experimentation and preparation of this manuscript, respectively.

REFERENCES

1. G J Ranque. 'Experiments on Expansion in a Vortex with Simultaneous Exhaust of Hot and Cold Air.' *Le Journal De Physique, et le Radium (Paris),* vol 4, June 1933, pp 1125-1130.

2. R Hilsch. 'The use of the Expansion of Aires in a Centrifugal Field as a Cooling Process.' *Review of Scientific Instruments,* vol 13, February 1947, pp 108-113.

3. C D Fulton. 'Ranque Tube.' *Journal of the ASRE, Refrigeration Engineering,* vol 58, May 1950, pp 473-479.

4. G W Scheper (Jr). 'The Vortex Tube Internal Flow Data and a Heat Transfer Theory.' *Journal of the ASRE, Refrigeration Engineering,* vol 59, October 1951, pp 985-989.

5. J P Hartnett and E R G Eckert. 'Experimental Study of the Velocity and Temperature Distribution in a High Velocity Vortex Tube Flow.' *Transactions of ASME,* vol 79, May 1957, pp 751-758.

6. B B Parulekar. 'The Short Vortex Tube.' *Journal of Refrigeration,* vol 4, 1961, pp 74-80.

7. R B Aronson. 'The Vortex Tube: Cooling with Compressed Air.' *Journal of Machine Design,* December 1976, pp 140-143.

8. J E Lay. 'An Experimental and Analytical Study of Vortex Flow and Temperature Separation by Superposition of Spiral and Axial Flow, Part 1 and Part 2.' *ASME Journal of Heat Transfer,* vol 81, no 3, August 1959, pp 316-317.

9. Y Soni and W J Thomson. 'Optimal Design of Ranque — Hilsch Vortex Tube.' *ASME Journal of Heat Transfer,* vol 94, no 2, May 1975, pp 316-317.

10. K Landecker. 'A Two Stage Refrigeration and Power Producing Arrangement Consisting of a Vortex Cooling Tube and a Thermoelectric Stage.' *Energy Conversion,* vol 77, no 3, 1977, pp 119-122.

11. R L Collins and R B Lovelace. 'Experimental Study of Two Phase Propane Expanded through the Ranque — Hilsch Tube.' *ASME Journal of Heat Transfer,* vol 101, no 2, May 1979, pp 300-305.

12. H Takahama and H Yokosawa. 'Energy Separation in Vortex with a Divergent Chamber.' *Transactions of ASME,* vol 103, May 1981, pp 196-203.

13. M Kurosaka. 'Acoustic Streaming in Swirling Flow and the Ranque — Hilsch Effect.' *Journal of Fluid Mechanics,* 1982, p 139.

14. Y D Raiskii and L E Tankel. 'Influence of Vortex Tube Configuration and Length on the Process of Energetic Air Separation.' *Journal of Engineering Physics,* vol 27, no 6, December 1974, pp 1578-1581.

15. V I Metenin. 'Investigations of Vortex Tube Type Compressed Air Separators.' *Soviet Physics — Technical Physics,* vol 5, 1961, pp 1025-1032.