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Development of Two Opposing Nozzle Vortex Tube Using Air as Working Fluid

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Abstract: *One of the practical applications of thermodynamics is refrigeration where heat is transferred from low temperature region to high temperature region through the working fluid known as refrigerant. Vapour compression and vapour absorption refrigeration systems are two commonly employed conventional systems in almost all the major applications of refrigeration and air-conditioning. The chronological literature review describes the past studies on the various issues of vortex tubes including theory, operation and design. The performance of vortex tube depends on various geometric parameters, operating parameters and gaseous properties. The magnitude of the energy separation increases as the length of the vortex tube increases to a critical length. However, a further increase of the vortex tube length beyond the critical length does not improve the energy separation. A very small diameter vortex tube leads to low diffusion of kinetic energy which also means low temperature separation. A very large tube diameter would result in lower overall tangential velocities both in the core and in the periphery region that would produce low diffusion of mean kinetic energy and also low temperature. There must be an optimum value of cold orifice diameter so that we get desired performance of vortex tube.*

Keywords: *Ranque–Hilsch vortex tube, refrigeration, counter-flow type vortex tube, energy separation.*

1. Introduction

The vortex tube (also called the Ranque–Hilsch vortex tube) is a mechanical device operating as a refrigerating machine without any moving parts, by separating a compressed gas stream into two low pressure stream, the temperature of which are respectively higher and lower than inlet stream. Such a separation of the flow into regions of low and high total temperature is referred to as the temperature (or energy) separation effect. Much earlier, the great nineteenth century physicist James Clerk Maxwell, postulated that since heat involves the movement of molecules, we might someday be able to get hot and cold air from the same device with the help of a "friendly little demon" who will sort out and separate the hot and cold molecules of air [1]. The vortex tube was invented by accident in 1928 by George Ranque, a French metallurgist as well as physicist, who were experimenting with a vortex type pump; he had developed; when he noticed warm air exhausting from one end and cold air from other. Ranque started a small firm to exploit the commercial potential of this strange device that produced hot and cold air without moving any parts. Ranque was granted a French patent for this device in 1932 and a United States patent in 1934[2]. The initial reaction of the scientific and engineering communities to his invention was disbelief and apathy. Since the vortex tube was thermodynamically highly inefficient, it was abandoned for several years. Vortex tube slipped into obscurity until 1945; when a German physicist and engineer, Hilsch reported on account of his own comprehensive experimental and theoretical studies aimed at improving the efficiency of the vortex tube. He systematically examined the effect of the inlet pressure and the geometrical parameters of the vortex tube on its performance and presented a possible explanation of the energy separation process [3]. Since vortex flow phenomenon taking place in a vortex tube is compressible and complex, the simulation and solution of turbulent vortex flows is a difficult and challenging task. Thus, the vortex tube has been variously known as the “Ranque vortex tube”, the “Hilsch Tube”, the “Ranque-Hilsch Tube” and “Maxwell’s Demon”. By any name, it has in recent past gained acceptance as a simple, reliable and low cost answer to a wide variety of industrial spot cooling problems.

2. PREVIOUS WORK

Lewins J. and Bejan A. [7] suggested that the angular velocity gradients in the radial direction give rise to frictional coupling between different layers of the rotating flow resulting in a migration of energy via shear work from the inner layers to the outer layers.

Saidi M. S. and Yazdi N. [8] had used a thermodynamic model to investigate vortex tube energy separation. An equation has been derived for the rate of entropy generation. This equation is used to model the irreversibility term.

Piralishvili S. A. and Fuzeeva A. A. [9] had derived regression equation for calculating the relative cooling of a gas by its thermodynamic parameters. However, this equation was obtained with no account for the geometry of a vortex tube and the differential pressure in it because these quantities were held constant in the experiment. To derive a more general regression equation with account for all determining parameters, it is necessary to perform additional experimental investigations.

Pinar A. M. et al. [10] investigated the application of Taguchi method in assessing maximum temperature gradient for the Ranque–Hilsch counter flow vortex tube performance. The experiments were planned based on Taguchi‟s L27 orthogonal array with each trial performed under different conditions of inlet pressure, nozzle number and fluid type. In this study, a counter flow type Ranque–Hilsch vortex tube with (L=150 mm, D=10 mm) L/D ratio equal to 15 was used. Three different orifices with different nozzle numbers (2, 4, and 6) have been manufactured. ANOVA is a method most widely used for determining significant parameters on response and measuring their effects.

Singh P. K. et al. [11] proposed two design features associated with a vortex tube, namely, (a) maximum temperature drop vortex tube design (b) maximum cooling effect vortex tube design. While investigating, 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. 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. Length of the tube has no effect on the performance of the tube when it is increased beyond 45 up to 55 DT.

Deissler R. G. [12] presented mathematical analysis based on the turbulent N-S equation. Based on his the analysis, he stated that that heat transfer between flow layers by temperature gradients and by pressure gradients due to turbulent mixing as well as turbulent shear work done on elements; are the main reasons for the energy separation.

Eiamsa-ard S. et al. [13] predicted compressible vortex-tube flow using numerical simulations with both the standard k-ε model and the acoustic streaming model (ASM). Predictions with the ASM are in closer agreement with measurements than those with the k-ε model for the quantities compared. Only in terms of overall local flow properties and temperature, the ASM has better conformity to the experimental data due to its ability to introduce non-isotropic turbulence effects

Baghdad M. et al. [17] carried out a numerical study to investigate the energy separation mechanism and flow phenomena within a vortex tube using four different turbulence models, namely, standard k-ε, k-ω, SST k-ω models and RSM model. The predicted mean temperature difference and compared this with the available experimental data. Results showed that all the models were able to reproduce the general dynamics and separation of energy within the vortex tube. However, the prediction of the temperature difference between both exits is largely over predicted by the two-equation models were tested herein. This also draws attention to the fact that these models should be used with extreme caution for device design purposes.

Pouraria H. et al. [18] presented numerical studies where result showed that the magnitude of the swirl velocity is higher than axial and radial velocities. The distribution of angular velocity in radial direction indicates the transfer of shear work from the cold inner region to the hot peripheral region, except at locations very close to the inlet. The existence of heat transfer between the cold inner region and the hot peripheral region was confirmed. The numerical simulation indicates that the performance of the vortex tube refrigerator can be improved by using a divergent hot tube. Present results indicate that an increase in divergent tube angle results in an increase in cooling performance of vortex tube. However, there was critical angle, so that further increase will lead to the reduction in cooling performance of the device.

3. GOVERNING EQUATIONS & PARAMETERS

Takahama has proposed the following correlations for optimized RHVT for larger temperature difference, given as;

Cold Drop Temp. $\Delta T_c = T_i - T_c$.

Hot Rise Temp. $\Delta T_h = T_h - T_i$

Temp Drop At Two Ends

$$\Delta T = T_h - T_c.$$

Cold Mass Fraction

$$\mu = \frac{mc}{mi}$$

Tube length-The length of the vortex tube affects performance significantly. Optimum L/D is a function of geometrical and operating parameters. The magnitude of the energy separation increases as the length of the vortex tube increases to a critical length. However, a further increase of the vortex tube length beyond the critical length does not improve the energy separation.

Tube diameter- In general smaller diameter vortex tubes provide more temperature separation than larger diameter ones. A very small diameter vortex tube leads to low diffusion of kinetic energy which also means low temperature separation. Avery large tube diameter would result in lower overall tangential velocities both in the core and in the periphery region that would produce low diffusion of mean kinetic energy and also low temperature.

Number of nozzles-For maximum temperature drop the inlet nozzles should be designed so that the flow will be tangentially entering into vortex tube. The increase of the number of inlet nozzles leads to higher temperature separation. The inlet nozzle location should be as close as possible to the orifice to yield high tangential velocities near the orifice.

Cold orifice-Using a small cold orifice (D_C/D= 0.2, 0.3, and 0.4) yields higher backpressure while a large cold orifice (D_C/D= 0.6, 0.7, 0.8, and 0.9) allows high tangential velocities into the cold tube, resulting in lower thermal/energy separation in the tube. Dimensionless cold orifice diameter should be in the range of 0.4 to 0.6 for optimum results.

Hot flow control valve- The hot-end plug is not a critical component in VT. Optimum value for the angle of the cone-shaped control valve (α) is approximately 45°.

Tube geometry- Tapered vortex tube contributes separation process in vortex tubes used for gas separation. In divergent vortex tubes, there exists an optimal conical angle and this angle is very small (3°). Rounding off the tube entrance improves the performance of the RHVT.

4. EXPERIMENTATION

Experimentations were performed at various operating conditions. Initially compressor was put on to get the compressed air at desired pressure continuously from the receiver. The FRL unit is used to control the inlet pressure. After setting the supply air pressure, measured the reading at supply pressure for both Rotameter and multiply by multiplication factor for inlet Rotameter reading from calibration chart, cold end Rotameter already calibrate at atmospheric condition, no need to multiply by multiplication factor, from this we obtain exact cold mass fraction. Desired cold mass fraction is obtained with the help of hot end valve. Two minutes were allowed to stabilize the flow and temperature to reach on steady state. The inlet temperature (T_i) is noted before pneumatic connector, from this compressed air supplied double inlet nozzle of vortex tube. After setting of cold mass fraction from fully closed to fully open, the temperature at cold end (T_c) and hot end (T_h) are noted. Based on the recorded data the performance of system is calculated in terms of coefficient of performance (COP) and isentropic efficiency of system for air and geometric parameters

An experimental set-up is developed to carry out the experiments of two nozzle vortex tubes using air as the working fluid. Three different configuration vortex tubes have been developed and tested. Each vortex tube is tested at various operating condition with air as working substance. A series of experiments are performed to evaluate the performance of the system and to optimize the geometrical parameters. Experiments are performed under two parts, in first part experiment carried out using different diameters of cold orifice i.e.3, 4 and 5 mm to optimize cold orifice and in second part, experiments are carried out on optimized cold orifice by varying different geometric parameters such as diameter and single or double inlet nozzles for optimizing L/D ratio.

Experiments are performed under following conditions:

- Inlet pressures range : 02 bar – 06 bar
- Cold mass fraction : 0 – 1
- L/D Ratio by varying diameter : 12.5, 13.5, 17.5
- Number of inlet nozzle : 2
- Working substance : Air

Following are the observations we got while performing an experiment.

Table I. *(Reading at Various Pressures)*

Parameters Varied		Observations			Calculations		
Pressure	L/D Ratio	Inlet Temp	Temp. Of Hot Air	Temp. Of Cold Air	Cold Drop Temp	Hot Rise Temp	Temp Drop At Two Ends
bar	mm	Ti	Th	Tc	ΔT_c	ΔT_h	ΔT
2	12.5	25	25.4	24.4	0.6	0.4	-0.2
	13.5	25	25.4	24.3	0.7	0.4	-0.3
	17.5	25	25.7	24.2	0.8	0.7	-0.1
3	12.5	25	27.8	23.9	1.1	2.8	1.7
	13.5	25	26	24	1	1	0
	17.5	25	27.1	24	1	2.1	1.1
4	12.5	25	27.3	23.9	1.1	2.3	1.2

	13.5	25	25.5	23.7	1.3	0.5	-0.8
	17.5	25	27	23.3	1.7	2	0.3
5	12.5	25	25.6	24.1	0.9	0.6	-0.3
	13.5	25	26.3	24.3	0.7	1.3	0.6
	17.5	25	28.2	23.8	1.2	3.2	2
6	12.5	25	26.7	24.1	0.9	1.7	0.8
	13.5	25	25	23.6	1.4	0	-1.4
	17.5	25	25.6	21.8	3.2	0.6	-2.6
7	12.5	25	27.5	23.8	1.2	2.5	1.3
	13.5	25	26	24.1	0.9	1	0.1
	17.5	25	25.6	22.7	2.3	0.6	-1.7
8	12.5	25	29	23.7	1.3	4	2.7
	13.5	25	26.8	24.3	0.7	1.8	1.1
	17.5	25	27.1	23.1	1.9	2.1	0.2
9	12.5	25	30	22.9	2.1	5	2.9
	13.5	25	28	23.9	1.1	3	1.9
	17.5	25	28	21.5	3.5	3	-0.5
10	12.5	25	30.1	22.5	2.5	5.1	2.6
	13.5	25	29.5	24	1	4.5	3.5
	17.5	25	30.2	22.3	2.7	5.2	2.5

Table II. *(Obeservations at various pressures)*

Parameters Varied		**Observations**					
Pressure	**L/D Ratio**	**Cold Mass Fraction**			**Atm.Pressure**	**Inlet Pressure**	**Compressor Energy Input**
bar	**mm**	**mc**	**mi**	**µ**	**Pa**	**Pi**	**Watt**
2	12.5	110	160	0.6875	1.01325	2	745.7
	13.5	150	180	0.83333	1.01325	2	745.7
	17.5	160	190	0.84211	1.01325	2	745.7
3	12.5	50	170	0.29412	1.01325	3	745.7
	13.5	50	120	0.41667	1.01325	3	745.7
	17.5	90	150	0.6	1.01325	3	745.7
4	12.5	100	190	0.52632	1.01325	4	745.7
	13.5	140	170	0.82353	1.01325	4	745.7
	17.5	160	210	0.7619	1.01325	4	745.7
5	12.5	170	210	0.80952	1.01325	5	745.7
	13.5	80	150	0.53333	1.01325	5	745.7
	17.5	100	190	0.52632	1.01325	5	745.7
6	12.5	45	110	0.40909	1.01325	6	745.7
	13.5	135	170	0.79412	1.01325	6	745.7
	17.5	260	310	0.83871	1.01325	6	745.7
7	12.5	150	280	0.53571	1.01325	7	745.7
	13.5	50	105	0.47619	1.01325	7	745.7
	17.5	120	220	0.54545	1.01325	7	745.7
8	12.5	50	250	0.2	1.01325	8	745.7
	13.5	55	140	0.39286	1.01325	8	745.7
	17.5	240	270	0.88889	1.01325	8	745.7

9	12.5	110	250	0.44	1.01325	9	745.7
	13.5	60	160	0.375	1.01325	9	745.7
	17.5	250	120	2.08333	1.01325	9	745.7
10	12.5	50	180	0.27778	1.01325	10	745.7
	13.5	55	140	0.39286	1.01325	10	745.7
	17.5	200	100	2	1.01325	10	745.7

Table IIII. *(Calculations at various pressures)*

Parameters Varied		**Calculations**				
Pressure	**L/D Ratio**	**Static Temp. Drop due to expansion**	**Relative Temperature drop**	**Isentropic efficiency**	**Compressor efficiency**	**C.O.P**
bar	**mm**	**$\Delta T'_c$**	**ΔT_{rel}**	**η_{is}**	**η_{ac}**	
2	12.5	4.47425	0.1341	0.09219	0.75	0.05677073
	13.5	4.47425	0.15645	0.13038	0.75	0.08028185
	17.5	4.47425	0.1788	0.15057	0.75	0.09271648
3	12.5	6.75127	0.16293	0.04792	0.75	0.02623507
	13.5	6.75127	0.14812	0.06172	0.75	0.03378759
	17.5	6.75127	0.14812	0.08887	0.75	0.04865413
4	12.5	8.21194	0.13395	0.0705	0.75	0.03550707
	13.5	8.21194	0.15831	0.13037	0.75	0.0656596
	17.5	8.21194	0.20702	0.15773	0.75	0.07943746
5	12.5	9.26392	0.09715	0.07865	0.75	0.03712745
	13.5	9.26392	0.07556	0.0403	0.75	0.01902478
	17.5	9.26392	0.12953	0.06818	0.75	0.03218478
6	12.5	10.0743	0.08934	0.03655	0.75	0.01636446
	13.5	10.0743	0.13897	0.11036	0.75	0.04941424
	17.5	10.0743	0.31764	0.26641	0.75	0.11928913
7	12.5	10.7269	0.11187	0.05993	0.75	0.02566152
	13.5	10.7269	0.0839	0.03995	0.75	0.01710768
	17.5	10.7269	0.21441	0.11695	0.75	0.05007884
8	12.5	11.269	0.11536	0.02307	0.75	0.00950409
	13.5	11.269	0.06212	0.0244	0.75	0.0100524
	17.5	11.269	0.1686	0.14987	0.75	0.06173595
9	12.5	11.7301	0.17903	0.07877	0.75	0.03135876
	13.5	11.7301	0.09378	0.03517	0.75	0.01399945
	17.5	11.7301	0.29838	0.62162	0.75	0.24746494
10	12.5	12.1294	0.20611	0.05725	0.75	0.02210631
	13.5	12.1294	0.08244	0.03239	0.75	0.01250586
	17.5	12.1294	0.2226	0.4452	0.75	0.17189866

5. CONCLUSION

Effect of Geometrical Parameters

I) Effect of L/D Ratio

L/D ratio is varied with change in diameter by keeping length constant. The L/D ratios selected as

12.5, 13.5 and 17.5 for the lengths of 125,175 & 245 mm and diameters as 10mm, 13mm and 14 mm respectively. Results that for each L/D ratio, as pressure increases the cold end temperature drop also increases.. For L/D ratio of 17.5, energy diffusion from inner cold vortex to outer hot vortex increases, simultaneously the angular momentum also increases and hence due to this we get maximum temperature drop at cold end. But the cold end temperature drop for 13.5 is lower than that for 12.5, because as we decrease the diameter by keeping length constant, the intermixing of two layers starts taking place and in turn we get reduced cold end temperature drop. And in the case of L/D ratio of 17.5, though the diameter is smaller than remaining two ratios, we get better cold end temperature drop, because as diameter is decreased, the rate of increase of angular momentum as well as diffusion of energy becomes more than the rate of increase of intermixing of two layers and hence cold end temperature drop increases.

II) Effect of cold orifice diameter

Three different configuration vortex tubes have been developed and tested. Each vortex tube is tested at various operating condition with air as working substance. A series of experiments are performed to evaluate the performance of the system and to optimize the geometrical parameters. Experiments are performed under two parts, in first part experiment carried out using different diameters of cold orifice i.e. 4,5 and 6 mm to optimize cold orifice and in second part, experiments are carried out on optimized cold orifice by varying different geometric parameters such as diameter and double inlet nozzles for optimizing L/D ratio. Cold end orifices diameters 4 mm, 5 mm and 6 mm are tested but for 6 mm cold end orifice maximum cold end temperature drop of 21.5°C. Experimental investigation shows that the double inlet nozzle gives the maximum cold end temperature drop. L/D ratio affects performance of vortex tube. The optimum value of L/D ratio is found to be 17.5 for 6 mm orifice of vortex tube .

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