NUMERICAL STUDY OF VORTEX TUBE PROPERTIES

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Abstract
The efforts of many researchers and designers, working with vortex tubes with compressed air as the working medium, have been focused on improvement of their efficiency by changing the parameters affecting vortex tube operation. The effective parameters are geometrical and thermo-physical. Numerical simulation can help to optimize the design of the vortex tube and to know the behaviour in various regimes.

1 Introduction

The Vortex Tube (also known as Ranque-Hilsch vortex tube) is a simple mechanical device operating as a refrigerating and heating machine without any moving parts. A flow that rotates about an axis (like a tornado) is called a vortex. In the late nineteenth century, James Clerk Maxwell theorized that it should be possible to produce hot and cold air from the same device if one could find a way to intelligently separate the hot and cold molecules. This theory was dubbed 'Maxwell's Demon', referring to a thermodynamic demon who would perform this work. The greatest step towards uncovering this demon came in 1929 when French physicist Georges Ranque [1] observed the strange behaviour of vortices within a tube when air exits are intelligently blocked. His ideas were not widely read and remained in obscurity until 1945 when German physicist Rudolph Hilsch [2] published a widely read paper describing the actions of the vortex tube. In his landmark paper, Hilsch proposed that the so-called Ranque effect was in fact an incarnation of Maxwell's Demon. Hilsch received wide acclaim for his paper and the device was re-christened the Ranque-Hilsch Vortex tube, as we know it today. The tube was hailed as a breakthrough device with applications throughout industry.

2 Influences in the Vortex Tube

2.1 Physical Description

The vortex tube creates a vortex from compressed air and separates it into two air streams, one hot and one cold. Compressed air enters a cylindrical generator, which is of proportionately larger diameter than the hot (long) tube where it causes the air to rotate. Then, the rotating air is forced to flow down the inner wall of the hot tube at speeds reaching a sonic value. At the end of the hot tube, a small portion of this air exits through a needle valve (control valve) as hot air exhaust.

Figure 1: Schematic diagram of a vortex tube

The remaining air is forced to flow back through the centre of the incoming air stream at a slower speed. The heat in the slower moving air is transferred to the faster moving incoming air, though here is a higher temperature. This cooled air flows through the centre of the generator and exits through the cold air exhaust.
port, as shown schematically in figure 1. The control valve located at the end of the hot tube is used to control the system. The vortex tube can be used as a refrigeration device or as a heating device.

Figure 2(a) shows the essential features of the counter flow vortex tube investigated by both its discoverers. Air is introduced into the tube through one, two or more tangential flow inlets. An alternative design for a temperature separating vortex tube, which we will term the uniflow variant, is shown in figure 2(b). The fundamental aspects of the device are the same as for the counter flow tube. Its distinguishing features are that the orifice and valve are combined at one end of the tube, while the other end of the tube, adjacent to the inlet nozzles, is sealed. Many investigators have suggested that the uniflow tubes perform less well than equivalently proportioned counterflow designs. Within the vortex tube, both vortices are rotating in the same direction, with the same angular momentum. [3,4].

Figure 2: Counterflow (a) and uniflow (b) vortex tubes

2.2 Effect of geometric and phys. parameters

The effect of various parameters, such as nozzle area, cold orifice area, hot end area and L/D ratio of the tube length to the tube diameter, on the performance of the vortex tube was investigated in [5]. There were experimentally tested variations of the cold air temperature \( T_c \) with respect to the change of the hot end area for \( L/D = 45,50,55 \). They showed the cold air temperature \( T_c \) at all the tested ratios L/D decreased if the hot end was opened.

It means the amount of cold air is reduced and its temperature gets lowered. Also there was observed that temperatures of cold air in tubes of \( L/D = 45 \) were the highest whereas in tubes \( L/D = 50 \) there was a minimum. Also, the same reference observed the variation of the cold air temperature with respect to changes in the hot end area. It showed the variation of \( T_c \) for \( L/D = 45 \), and there was observed that the temperature of cold air decreased if the hot end was opened. It means the amount of cold air is reduced and its temperature gets lowered.

The effects of length and diameter on the principal vortex tube are considered in [6]. They showed variation of efficiency versus different L/D of vortex tube. The efficiency is defined as \( \eta = \Delta T_c / \Delta T_{cs} \), where \( \Delta T_c \) is difference between the inlet temperature and the cold one, and \( \Delta T_{cs} \) is difference between the inlet temperature and isentropic temperature at the cold end. They found that for \( L/D \leq 20 \) the energy separation decreased that led to the decrease of the cold air temperature difference \( \Delta T_c \), and efficiency decreased as well. For \( L/D \geq 55.5 \), the variation of efficiency with L/D is not considerable. Consequently, the optimum value of L/D is within the following ranges: \( 20 \leq L/D \leq 55.5 \).

The effect of the cold air orifice diameter \( d_c \) on the cold air temperature difference \( \Delta T_c \) and efficiency were presented also in [6]. It showed the cold air temperature difference and the efficiency versus the dimensionless cold air orifice diameter \( d_c^* = d_c / D \). They found that for \( d_c^* < 0.5 \) the increasing \( d_c^* \) causes increasing of the cold air temperature difference and the efficiency as well, and for \( d_c^* > 0.5 \) the increasing \( d_c^* \) tends to decreasing of the cold air temperature difference and of the efficiency. An experimental investigation of the energy separation process in the vortex tube with air as a working medium was conducted in [7]. Experimentally tested, the temperature differences between the hot exit air and the inlet air, \( \Delta T_h = T_h - T_o \), and between the inlet air and the cold exit air, \( \Delta T_c = T_o - T_c \), as functions of the cold air mass ratio is defined as follows: \( \mu_c = m_c / m_o \), where \( m_c \) is a mass flow rate of the cold air and \( m_o \) is a total air mass flow rate,
with the pressure of the inlet air as a parameter. They found that without the outflow of the cold air, $\mu_c = 0$, the temperature measured at the centre of the central orifice used for exit of the cold air is lower than the temperature of the inlet air, similarly without the outflow of the hot air, $\mu_c = 1$, the temperature of the vortex tube wall at the hot air exit end is higher than that of the inlet air.

Experimental results of the energy separation in the vortex tube under different operating conditions presented in [8] showed the changes of the temperature of cold and hot streams as a function of the inlet pressure. In this study there was found that the temperature of the hot stream increased with the inlet pressure increasing, and that the temperature of the cold stream decreased with the inlet pressure. Further, the same research deduces that the cold stream temperature generally increases slightly with $\mu_c$ and attains its extreme value in the range of about $\mu_c = 30 \sim 43 \%$. And illustrates the temperature difference between the hot stream and the cold one approaches its maximum value in the range of about $\mu_c = 70 \sim 80 \%$. In [6] there are also presented variations of the cold air temperature difference $\Delta T_c$ versus cold mass ratios $\mu_c$ for the different inlet pressures. It shows that increasing the inlet pressure causes the cold air temperature difference to increase.

### 2.3 Results and Discussions

- **The effect of inlet pressure**

  References find that increasing the inlet pressure causes the cold air temperature difference to increase. The compressed air begins its vortex flow as soon as it is introduced into the vortex tube. Because of the centrifugal characteristics of the forced vortex flow, the tangential velocity of the air near the tube wall would be larger than that in the central region. This would naturally cause the temperature near the tube wall to be higher than that in the central region. Also, the higher frictional force among fluid particles as well as among the fluid particles and the tube wall near the wall region is responsible for part of this phenomenon.

The efficiency increases by increasing the inlet pressure but above a certain value it decreases. By increasing the inlet pressure, the flow velocity in the outlet of the entrance nozzle increases up to the point at which it becomes choked.

- **The effect of various length and diameter**

  All references find that an efficient tube of either design must be many times longer than diameter. The [4] recommends a minimum length of 30 diameters, and [9] advises the length of the vortex tube should be greater than 45 diameters, but no upper limit is specified. And [6] gets optimum performance for $20 < L/D < 55$. This is comparable to [5] comment that the length of the tube has no effect on the performance of the vortex tube when the length is greater than 55D.

- **The effect cold air mass ratio**

  The experimental temperature of the hot and cold exit as a function of the cold air mass ratio $\mu_c$, were presented in [7, 8]. They concluded that the temperature of the exit air increased with increasing cold air mass ratio. With further increases of the cold air mass ratio, higher than 80 %, the hot exit air temperature increased to a maximum value and then decreased to its minimum value. The range of the cold air mass ratio, at which the hot exit air becomes a maximum, is from 70 to 80 %. At cold air mass ratio from 0.8 up to 1 a sharp temperature drop is measured for the hot air exit temperature. The typical temperature variation of the hot air can be explained as follows:

  In fluid flow there are basically two causes of the pressure variation in addition to the weight effect. These are acceleration and viscous resistance. In the range of the cold air mass ratio from 1 up to 0.8, hence in the relative increase of the hot mass flow rate, the acceleration in the inner pipe is not enough large to overcome the viscous resistance. The friction between the air molecules, namely the friction between the air and the pipe wall, is very high so that the temperature of the hot exit air increases and reaches its highest value. At the cold air mass ratio lower than 80 % more
air flows into the hot outlet pipe. The momentum is then enough large to overcome the viscous resistance. A decrease in the cold air mass ratio from 0.8 up to 0 causes a decrease in the wall friction effect; a temperature drop is a consequence.

3 Numerical simulations

3.1 History of the numerical simulations

Until recently, sufficient computational power has not been readily available to model the high-speed, turbulent, and compressible fluid dynamic and heat transfer problem that exists within a vortex tube. Investigators have relied on analytical solutions addressed mostly to simplified versions of the problem [10-13]. CFD modelling of the vortex tube has not been widely used, and those investigators that have employed this tool have either made broad assumptions that limit the quantitative utility of the model or have presented only a very limited set of results. Cockerel [14] presents a numerical analysis in which the fluid dynamic equations are derived from the energy equation. The flow model was determined using the vorticity-stream function formulation of the radial and axial Navier-Stokes equation and a standard k- turbulence model. Calculation of the temperature field was added separately. Although his solution technique allowed variation of density, most results were presented for the case of an incompressible fluid due to computational limitations. The modelling provides some insight into the vortex tube's behaviour but does not agree with the measured energy separation effect very well. Some publications show that it is possible to solve the vortex tube by a CFD program using a relatively simple axisymmetric model with two transport turbulence equations. Frohlingsdorf et al. [15] modelled the flow within a vortex tube using a CFD solver that included compressible and turbulent effects in an attempt to duplicate the experimental results of Bruun [16]. The numerical results qualitatively predicted the experimental data when a standard k- model was used. When the standard k- model was replaced by a correlation attributed to Keyes [17], that accounted for anisotropy of the turbulence the large radial pressure gradients, a better quantitative agreement was obtained. The predicted tangential velocity profiles are in accordance with measurements at several axial tube locations. However, a good agreement of the CFD predictions with the overall measurement of the vortex tube's energy separation performance was obtained only by increasing the turbulent Prandtl number up to 9.0.

3.2 Simplified geometry scheme for calculation

We investigated a number of geometrical arrangements to obtain a flow comparable to that found in the real vortex tube. The most successful design for the counterflow vortex tube is shown in figure 3.

![Figure 3. The geometry employed for counterflow calculations](image)

3.3 Description of vortex tube computations

The goal of our research has been to find a credible and simple CFD model to solve the vortex tube numerically. The simulations should be supported by the planned own experimental research in near future. The CFD system FLUENT in version 6.1 was used for numerical simulations of the mentioned cases. The vortex tube described in articles [15] and [18] was chosen for tuning up our numerical model to published results. Geometry of the vortex tube simulated by an axisymmetric CFD model and basic boundary conditions are depicted in figure 4. The parameters of the model were entered so as to correspond to the publication. The area of the
inlet was set to correspond to the mass flow rates in [15] and performed computations.
A question is, how to set the boundary conditions for the turbulent model equations. It is impossible to find these conditions in references. The inlet turbulent intensity in described model was set to 2%. Performed computations show that the value doesn’t exert influence upon the results, but the cases with the lower value of the inlet turbulence intensity are more stable.

Computational mesh has 10 400 cells. The analysis studying needed fineness of the grid showed that meshes with 41 600 and 166 400 cells render principally the same results as the coarse grid.
A coupled explicit solver was used for the computations. The initialising of the computation must be set suitably to obtain converging solution. The computations converged relatively slowly, the number of iterations leading to a converged solution was more than 10 000, final stabilization of computation was obtained after about 20 000 iterations.

The used turbulent model plays in numerical simulations of any vortex tube very important role because a tube doesn’t work without the turbulent mass and energy transfer, and a laminar CFD model of a vortex tube really doesn’t work as a vortex tube. Various turbulent models were tested (Spalart–Almaras, k-ε, RNG k-ε, Realizable k-ε and RSM) and results were compared with ones in article [15]. All solved cases with unmodified turbulent model showed working vortex tube with temperature rise and drop. The worst agreement in velocity and temperature fields with published results gave the RSM model, but differences between models were relatively small, while differences It seems, that the turbulent flow in the vortex tube has between simulations (generally) and experimental results were higher. This outcome corresponds well with results published in [15].
The standard k-ε model was selected for practical testing of some model parameters and their modifications to find useful form of model for practice. To obtain closer results of velocity profiles, the relation between turbulent and laminar viscosity was reduced to 500. The influence on the results is shown in the fig. 5, where it is possible to compare selected profiles of swirl velocity. We can see, that the simple reduction of the turbulent viscosity has very similar effect on the velocity profile as the turbulent model of Keyes.

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<th>(\Delta T_{t,c}) (K)</th>
<th>(\Delta T_{t,h}) (K)</th>
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<td>6.0</td>
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Table 1. Calculated and measured cold/hot gas total temperature differences related to the inlet total temperature for several Prandtl numbers

a specific character and requires a modification of the standard model. The authors recommended in publication [15] to increase
the value of turbulent Prandtl number up to value Pr_t = 9, to obtain better correspondence of measured and computed total temperatures. The table 1 shows obtained total temperatures in the outlets. The performed simulations give the best results for Pr_t = 3, but the computed temperature differences don’t agree so well, as in the work of Froehlingsdorf et al. For Pr_t = 9, the hot gas total temperature difference is too high. The cold mass flow fraction was as well as in the computation of Froehlingsdorf et al. (experiment). Figures 6 and 7 show obtained velocity and temperature fields, and typical results for vortex tube as well.

Figure 5. Comparison of the computed tangential velocity profiles with experiment in positions of the dimensionless tube length z/l = 0.23 and z/l = 0.98

Figure 6. Velocity vectors in the vortex tube (Pr_t = 3.0)

Figure 7. Distribution of the total temperature (K) in the vortex tube (Pr_t = 3.0)
4 Conclusions
The temperature difference between the outlet streams is dependent on a number of parameters, especially on the fraction of the inlet air leaving the tube through the cold exit and the pressure ratio between the inlet and cold outlet of the vortex tube. Other important factors influencing the performance include the length and diameter of the tube.

Performed work shows that the vortex tube flow processes is possible to simulate by means of accessible computation instruments, but it is necessary to modify the generally defined turbulent models. Taking into account special flow and energy transport in the vortex tube, the computing simulation requires an alternative definition of model constants to obtain a better agreement with the available experimental results. Published works and performed calculations show, that nearly all practically used turbulent models give too high turbulent viscosity and work with too low turbulent Prandtl number. To obtain much better results, there should be necessary to find complex redefinition of model constants in the turbulent model for vortex tubes or use a special formulation as in [15].

References