Temperature, Pressure and Velocity measurements on the Ranque-Hilsch Vortex Tube

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Abstract. Temperatures, pressures and velocities were measured in a Ranque-Hilsch vortex tube. Results show that the cooling power is larger than the heating power due to a heat loss to the surroundings. This heat loss became the more dominant thermodynamic process at large cold fractions (the ratio of cold mass flow over total mass flow). The velocities were obtained by means of Laser Doppler Anemometry. By this method, the three dimensional velocities of the gas and their standard deviations in the vortex tube are revealed by an non-intrusive measurement method. The turbulent fluctuations, characterized by the standard deviations, show that the turbulence is isotropic in the core region of the vortex tube.

1. Introduction

The vortex tube, shown in Fig. 1, is a device that is used to generate cold gas for cooling applications [1–3]. Pressurized gas enters the vortex tube, which has two exits. From the exits, the expanded gas leaves the device at a lower and, respectively, higher temperature than before it enters the vortex tube. This energy separation process was first discovered by Ranque [4] in 1933 and was improved by Hilsch [5] in 1947. These days, the vortex tube is often named after them: the Ranque-Hilsch vortex tube (RHVT).
Numerous theories that explain the energy separation process are given in literature. As example, we list a few approaches.

Hilsch [5] was the first who explained energy separation by means of internal friction which forces the flow to become a solid body rotation. In the core region of the RHVT, internal friction causes a flow of energy from the core to the peripheral region, establishing a solid body rotation. This flow of energy makes the gas in the core region to cool down, while heating up the gas present in the peripheral region. Schultz-Grunow [6] explained energy separation by means of stratification caused by a radial pressure distribution, due to a centrifugal force. Ahlborn also addressed the radial pressure distribution and concluded that the energy separation is caused by a heat pump present in the RHVT [7; 8].

This article presents temperatures, pressures and velocity profiles that we have measured in an RHVT. The velocities were obtained in the tube, near the vortex chamber (Fig. 1), by means of Laser Doppler Anemometry. Results from our experiments may be used to validate numerical simulations and to support RHVT theories.

2. Methods and Apparatus

The RHVT that was used in the experiments during the pressure and temperature measurements, had an inner diameter of $2R = 40$ mm and a length of $L = 2.50$ m. The vortex chamber (Fig. 1) had a diameter of 40 mm and contained 12 rectangular nozzles with dimensions $0.7 \times 7$ mm. During the velocity measurements, the length was $L = 1.60$ m, and the vortex chamber had a diameter of 80 mm, containing 8 rectangular nozzles with dimensions $1 \times 7$ mm. For both experiments, the vortex tube was insulated, where possible, to avoid heat losses. For both experiments, the diameter of the cold exit was 15 mm.

The temperature behaviour of the RHVT was determined by measuring the inlet and exit temperatures in an experimental setup as shown in Fig. 2. We have used nitrogen gas from a nitrogen supply ($N_2$) as working fluid. During an experiment, we kept the plenum pressure, $p_{pl}$, constant. The inlet mass flow of nitrogen, $\dot{m}_i$, was measured via mass flow sensor, $F_{in}$, and the cold mass flow was measured with mass flow sensor $F_c$. A typical parameter for the flow characteristics in the RHVT is the cold fraction: the ratio of cold exit mass flow and inlet mass flow, $\varepsilon = \dot{m}_c/\dot{m}$, which was set by means of two exit valves that were controlled by a computer. Subscripts $in$, $c$, and $h$ denote the inlet-, cold-, and hot conditions. Both flow meters had a standard deviation of less than 1%. The temperature probes (pt1000) were calibrated to have an accuracy of 0.01 K. The pressure probes had an maximum absolute error of 0.2 mbar. All temperatures, pressures and flows were recorded simultaneously.

![Figure 2: Schematic overview of the experimental setup to determine temperatures, flows, and pressures in the RHVT. $N_2$ is the nitrogen supply; $p_{pl}$ is the plenum pressure; $T$, $p$, and $F$ are the temperature, pressure, and mass flow respectively; $\varepsilon$ is the cold fraction. Subscripts $in$, $c$, and $h$ denote the inlet-, cold-, and hot conditions.](image)

The velocities inside the RHVT were determined by means of 3D Laser Doppler Anemometry (LDA). LDA is a non-intrusive measurement method that measures the velocity of seeding...
particles that are suspended in the flow. The experimental setup for LDA applied in the RHVT is shown in Fig. 3a. The optical layout is shown in Fig. 3b. We used a 2D transceiver probe and a 1D transceiver probe in combination with a receiver probe (RV). The receiver probe was used to measure the signal from the seeding particles in forward scattering. In this way, the signal to noise ratio was improved significantly. The measurement volume (the overlap of focal points of the probes) was moved through the vortex tube by moving the probes with a traverse system. To provide optical access, cylindrical windows (PMMA) replaced some part of the vortex tube.

![Experimental setup](image1)

![LDA layout](image2)

Figure 3: (a) Schematic overview of the experimental setup to determine the velocities in the RHVT. N\textsubscript{2} is the nitrogen supply; p\textsubscript{pl} is the plenum pressure; F\textsubscript{in} and F\textsubscript{c} are the inlet- and cold mass flow respectively; \(\varepsilon\) is the cold fraction. High pressure water (H\textsubscript{2}O) is injected in the nitrogen gas via high pressure nozzles to create seeding particles. The LDA measurements take place in a transparent section in the RHVT. (b) Layout of the configuration of the LDA probes (1D and 2D) and the receiver probe (RV) with respect to the vortex tube.

In this study, we have used water droplets as seeding particles. The droplets were created by spraying water through high pressure nozzles. These nozzles were mounted inside a pressure vessel, through which the nitrogen gas flows before it enters the RHVT. In the pressure vessel, the velocity of the nitrogen gas was low so as to give time for the droplets to (partly) evaporate while saturating the gas. This humidification was necessary to avoid the droplets to evaporate during transport and to increase the droplet lifetime after the gas/droplet mixture is injected in the RHVT [9]. The droplets were small enough (< 10 \(\mu\)m) to be considered as (apart from the centrifugal velocity) tracer particles.

3. Results

3.1. Thermodynamics

During the temperature and pressure measurements, we controlled \(\varepsilon\) to keep the mass flow as large as possible for a certain inlet pressure. The consequence was that the mass flow varied with \(\varepsilon\). The mass flow as functions of \(\varepsilon\) is shown in Fig. 4.

The temperatures are made dimensionless with the inlet temperature, \(T\textsubscript{in}\), according to

\[\frac{\Delta T}{T\textsubscript{in}} = \frac{T - T\textsubscript{in}}{T\textsubscript{in}}\]  \hspace{2cm} (1)

and are shown in Fig. 5a as functions of the cold fraction. The exit pressures are shown in Fig. 5b where we have made the cold and hot exit pressures dimensionless with the inlet pressure. The figures clearly show, that the cold exit temperature increases with the mass flow and decreases.
Figure 4: The mass flow as a function of $\varepsilon$.

Figure 5: (a) Dimensionless exit temperatures and (b) dimensionless exit pressures. The results are plotted as functions of $\varepsilon$.

with the cold exit pressure. The lowest temperature is found at the cold fraction where the pressures are the lowest. At this cold fraction, all the valves were completely opened and the maximum mass flow was possible (Fig. 4).

The cooling and heating power, $Q_c$ and $Q_h$ respectively, are defined as

$$Q_c = \varepsilon \dot{m} c_p (T_{in} - T_c)$$
$$Q_h = (1 - \varepsilon) \dot{m} c_p (T_h - T_{in})$$

where $c_p$ is the heat capacity at constant pressure. The maximum work that can be delivered by pressurized gas is obtained by isentropic expansion of the gas. The isentropic work, $W_{is}$ is defined as

$$W_{is} = \dot{m} c_p T_{in} \left( 1 - \frac{p_{ex}}{p_{in}} \right)^{\frac{\gamma - 1}{\gamma}}$$

where $\gamma = c_p/c_v$ is the isentropic exponent and is the ratio between the specific heat at constant pressure and constant volume, $c_v$. The gas expands from inlet pressure towards the exit pressure,
Because the hot and cold exit pressures are different, $p_{ex}$ can be the cold or the hot exit pressure. Now we define two isentropic efficiencies of the vortex tube, for the RHVT as cooling, $\eta_c$, or heating, $\eta_h$, device:

$$\eta_c = \frac{Q_c}{W_{is}} = \frac{\varepsilon (T_{in} - T_c)}{T_{in} \left(1 - \frac{p_c}{p_{in}} \frac{1}{1 - \varepsilon}\right)}$$

$$\eta_h = \frac{Q_h}{W_{is}} = \frac{(1 - \varepsilon) (T_h - T_{in})}{T_{in} \left(1 - \frac{p_h}{p_{in}} \frac{1}{1 - \varepsilon}\right)}$$

Both cooling and heating powers are shown in Fig. 6a, and the isentropic efficiencies are shown in Fig. 6b. Although the cooling power is larger than the heating power, the isentropic efficiency is larger for the RHVT as heater for $\varepsilon < 0.8$. The explanation is that the hot exit pressure is higher than the cold exit pressure (Fig. 5b), consequently, $W_{is}$ is lower, resulting in a higher efficiency.

While gas expands through the RHVT, no work is extracted from the device and there is no internal heat source or sink. Although we have insulated the RHVT to avoid heat losses to the surroundings, the losses are larger than zero. Therefore, the heating and cooling powers are unequal (especially at larger cold fractions).

According to the first law of thermodynamics, the heat loss to the surroundings, $Q_{loss}$, is

$$Q_{loss} = m c_p (T_{in} - \varepsilon T_c - (1 - \varepsilon) T_h) = Q_c - Q_h$$

where we have assumed that $c_p$ is constant. $Q_{loss}$ is shown in Fig. 7a, which shows that the heat loss increases with increasing $T_h$. This is to be expected because the hot gas is located near the walls of the RHVT, heating up the wall material. Consequently, the temperature of the tube and the insulation becomes higher than that of the ambient air, resulting in convective heat losses. The larger this temperature difference, the larger the heat loss. Because of this, we observe an increase in $Q_{loss}$ with increasing $\varepsilon$.

Especially the hot exit temperature is influenced by $Q_{loss}$, because the highest heat loss occurs at large cold fractions, where the tube temperature is the highest. The ‘contribution’ of the heat loss in the hot exit temperature, compared to the energy transfer process in the RHVT, is found by dividing the heat loss by the heating power, $Q_{loss}/Q_h$, shown in Fig. 7b. For most
cold fractions, the relative heat loss is less than 1. Here, the dominant thermodynamic process is the energy separation process, rather than the heat loss. At higher cold fractions, however, the relative heat loss significantly increases and becomes more important. At some ε, the heat loss and the heating power are equal (\(Q_{\text{loss}}/Q_h = 1\)) and the hot exit temperature reaches its maximum. For \(Q_{\text{loss}}/Q_h > 1\), the hot exit temperature should decrease, what is confirmed by Fig. 5a. The same was done by using the cooling power instead of the heating power, to see which process was dominant. In that case, the relative heat loss remains less than 1 (not shown here). This shows that the heat loss has only minor influence on the cold exit temperature, confirmed by Eiamsa-ard et al [10]. They have cooled the RHVT with water and found only a minor (\(\sim 0.5\%\) of absolute temperature change) reduction of the cold exit temperature.

3.2. Velocimetry
During the experiment where we measured the velocities and its statistics, \(\dot{m}, \varepsilon\), and \(p_{\text{pl}}\) were kept constant. Every measurement series was taken at a constant axial position (\(z\), as defined in Fig. 1) for several radial positions, \(r\). Each measurement contained velocity data of 10,000 to 100,000 individual seeding particles. With this amount of individual measurements, the statistical error was determined to be less than 1%. The axial and radial positions are made dimensionless with the RHVT radius, \(R\), and its length, \(L\), respectively. The results shown in this section were obtained with a mass flow of \(\dot{m} = 34.7\) gr/s and a cold fraction of \(\varepsilon = 0.35\).

The mean (mean quantities are denoted with chevrons, \(\langle \cdot \rangle\)) of the velocity components are shown in Fig. 8 for two axial positions (\(z/L = 0.04\) and \(z/L = 0.08\)). The increasing (and positive) radial velocity, \(\langle U_r \rangle\) (Fig. 8a), shows that the seeding particles move away from the tube axis. Although the seeding particles (droplets) are very small, the centrifugal force is large enough to have influence on the radial droplet velocity. Quantitative results for the radial velocity of the nitrogen gas can only be given if the droplet size is known and the flow is axisymmetric. In confined vortex flow, the flow is not necessarily axisymmetric (see e.g. IJzermans [11]). Because the measurement method gives velocity information in a point, rather than a plane, we can not guarantee this symmetry condition. Therefore, we are unable to give quantitative results for the radial gas velocity.

The mean swirl velocity is shown in Fig. 8b. The graph shows that the maximum swirl velocity decreases with \(z\). This is caused by viscous friction in the peripheral region. Viscous
Figure 8: Mean velocities as functions of the dimensionless radial coordinate for two axial distances from the inlet. (a) Radial velocity, $\langle U_r \rangle$; (b) Swirl velocity, $\langle U_\theta \rangle$; (c) Axial velocity, $\langle U_z \rangle$.

effects (shear stresses) also cause a spin up of the vortex core and tend to make the gas rotate as a solid body [5].

The third velocity component, the axial velocity, is negative in the core region because the gas located near the axis flows through the cold exit (Fig. 8c). In the peripheral region, the nitrogen flows towards the hot exit. This is an indication for the existence of secondary circulation [12–14].

The standard deviations, $\sigma$, of the three velocity components are shown in Fig. 9. The standard deviations are similar in all directions, in the region $\varepsilon < 0.6$ and increase with $r$. Here, the turbulent fluctuations have similar amplitudes, showing that the turbulence is isotropic, already observed by Gao [14]. The results show that turbulent fluctuations increase, not only in radial direction, but also in axial direction. In a few centimeters of the tube, $\sigma$ increases with about 30% in the core region.

Figure 9: Standard deviation of the velocity. (a) Radial velocity, $\sigma_r$; (b) Swirl velocity, $\sigma_\theta$; (c) Axial velocity, $\sigma_z$.

We have performed a large number of experiments with different flow settings. The results
are used to validate numerical simulations and a new developed analytical thermal model.

4. Conclusions

Temperatures and pressures were obtained from a RHVT. The results show that the cold exit temperature is minimum where the exit pressures are low and the mass flow is high. The hot exit temperature increases with $\varepsilon$, and due to the high temperature of the gas near the walls of the RHVT, there is a heat loss to the surroundings. Due to the heat loss, the cooling power is larger than the heating power. The isentropic efficiency, however, is larger for the RHVT as heating device due to the higher hot exit pressure. For large $\varepsilon$, the relative heat loss increases significantly and becomes the more dominant thermodynamic process. The cold exit temperature is only slightly influenced by the heat loss.

All three velocity components were measured in the RHVT by means of Laser Doppler Anemometry. Results show that the radial velocity of seeding particles is positive. However, it was impossible to give quantitative results for the radial gas velocity. Due to viscous effects, the maximum swirl velocity decreases with the distance from the inlet and the swirl velocity in the core region increases. The axial flow is negative in the core region because of flow through the cold exit. Near the wall, the axial velocity is positive. The standard deviations of the three velocity components are similar in the core region, implying that the turbulence is isotropic.

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