Effect of Physical Properties of a Gas on the Refrigeration Temperature Drop of Vortex Tubes Used in Oil and Gas Fields

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ABSTRACT: The composition of natural gas can vary considerably across different oil and gas fields. Such compositional variation is primarily reflected in the distinctive physical properties of natural gas. However, during practical application in an oil and gas field, a refrigeration temperature drop in a vortex tube is often observed because vortex tubes generally have low intrinsic refrigeration efficiencies. When vortex tubes are applied in oil and gas fields, the utilization of the oil pressure of a natural gas wellhead is often desirable to avoid excessive energy usage from external devices. In this study, a numerical model of a vortex tube was developed, executed, and validated through laboratory experiments. The refrigeration temperature drop values of 12 gases with distinctive physical properties at a total inlet pressure of 0.3 MPa, an inlet temperature of 300 K, and a cooling mass flow ratio of 0.5 were analyzed. The importance of different physical properties was ranked based on the gray correlation method. Additionally, the synergetic effects of the physical properties on the refrigeration temperature drop were analyzed via regression fitting. The results indicate a significant impact of the gas physical properties on the refrigeration temperature drop in the vortex tube. The maximum and minimum refrigeration temperature drop obtained for different gases can differ by up to 16 K. Furthermore, the refrigeration temperature drop in the vortex tube does not change monotonically with any physical property. Instead, it depends on the synergetic effect from the physical properties, which have different levels of influence on it.

1. INTRODUCTION

A vortex tube can be used to reduce the temperature of a gas stream with a small driving drop. It has a simple structure without any moving parts and can be manufactured or repaired easily. Owing to these advantages, vortex tubes have been used for a wide range of applications, including light hydrocarbon recovery from gases in oil fields, skid-mounted liquid natural gas precooling, and dehydration. In practical applications of such tubes in oil and gas fields, the refrigeration temperature drop is often a more important factor than the refrigeration efficiency. Therefore, it is necessary to analyze the factors influencing the refrigeration temperature drop when the available oil pressure at the oil and gas wellhead is known. The refrigeration temperature drop in a vortex tube is primarily affected by three types of factors. The first type involves structural parameters, including the number and types of nozzles,1−3 the length and diameter of the tube at the hot end,4,5 the diameter of the cold orifice, and the structure of the valve at the hot end. The second type includes operational parameters, such as inlet gas pressure, temperature, and the pressure ratio between the hot and cold ends.10 Finally, the third type includes the physical properties of the gas, such as specific heat capacity and thermal conductivity. The first two types have been explored extensively in previous studies. However, the influence of gas properties has only been partially analyzed by a few studies.

Han et al.11 experimentally used different hydrofluorocarbons to analyze the characteristics of energy separation in vortex tubes. They demonstrated that certain working gas
properties, including inlet pressure, specific heat ratio, and thermal conductivity, play a significant role in energy separation in vortex tubes. Wang et al. investigated the energy separation performance of different gases in a vortex tube using computational fluid dynamics; their approach was focused on the number of atoms contained in the gas molecules. Their results indicated that an enhanced temperature separation effect was observed for gas molecules that had low specific heat capacities and contained only one or two atoms. Thakare et al. reported that the use of nitrogen and carbon dioxide as the working gas yielded the best and the worst temperature separation performance, respectively. In addition, they determined that pure helium, neon, methane, and krypton could be used as the working fluid in Ranque–Hilsch vortex tubes (RHVT). Kirmaci et al. and Saidi et al. performed comparative studies on vortex tubes using air, nitrogen, and argon. Their results showed that air yielded the best performance among all tested gases. Zhang et al. experimentally analyzed the energy separation effect of different gases in a vortex tube. They found that the refrigeration temperature drop and the heating temperature difference in the vortex tube were proportional to the molar mass of the working fluid. Experimental or numerical calculation methods are mainly used to qualitatively establish the correlation of a certain physical property with the refrigeration temperature drop for a specific testing gas (air is mostly used) based on the experimental or numerically obtained data of the temperature drop. Due to a limited number of testing gas species and physical properties involved in such methods, the action principle of these physical properties cannot be explained reasonably. Therefore, the influence of the physical properties on the refrigeration temperature drop in a vortex tube cannot be comprehensively and objectively reflected. For the quantitative investigation of the impact mechanism of physical properties on the refrigeration temperature drop in a vortex tube, a comprehensive analysis of different types of physical properties of various testing gases should be conducted. Furthermore, the effect of these physical properties on the refrigeration temperature drop should be explained considering their action mode and synergetic effects.

Figure 1. Actual device and numerical model: (a) vortex tube; (b) axial section; (c) chamber and rectification; (d) vortex three-dimensional (3D) model; (e) internal geometric computing model; (f) chamber and rectification computing model.

Figure 2. Geometric model: (a) schematic representation and dimensions (in mm units) and (b) the axis system.
Throughout the lifecycle of oil and gas fields, the physical properties of the gas vary owing to changes in gas composition. Such variations significantly affect the refrigeration temperature drop in vortex tubes. For effective use of vortex tubes in oil and gas production, in this study, the influence of the physical properties, including specific heat capacity, thermal conductivity, dynamic viscosity, and molar mass, of a gas on the refrigeration temperature drop of a vortex tube was investigated through a combination of numerical simulations and laboratory experiments. The importance of different physical properties was ranked based on the gray correlation method. Furthermore, the relationship between the refrigeration temperature drop and the physical properties was analyzed via regression fitting. Finally, the proposed empirical correlation was validated, and the synergetic effect of the physical properties was explored.

2. METHODOLOGY

2.1. Geometric Model of the Vortex Tube. Images of a real vortex tube and its geometric model (to be used for the numerical simulation) are presented in Figure 1. The vortex tube used in the experiment was cut along the central axis to observe its internal structure and to ensure that the geometric model was sufficiently accurate. The internal structural dimensions were measured and used to build a full-scale numerical model.

The geometric model was built in a three-dimensional Cartesian coordinate system, as shown in Figure 2, with the center of the circular cross-section near the cold end of the vortex chamber as the origin. The axis of rotation of the vortex tube model was considered as the z-axis; the hot outlet end was considered as the positive direction.

The internal structural dimensions of the vortex tube are summarized in Table 1.

2.2. Governing Equations of Turbulent Flow and Boundary Conditions. Numerical calculation of flow and heat transfer in the vortex tube is based on FLUENT18.0. In this work, the mass, momentum, and energy conservation for the compressible turbulent flows in the RHVT were calculated as follows:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} (\rho u_i) = 0 \quad (1)$$

$$\frac{\partial (\rho u_i)}{\partial t} + \frac{\partial}{\partial x_j} (\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \mu \frac{\partial u_i}{\partial x_j} + \frac{\partial u_i}{\partial x_j} \right) \quad (2)$$

$$\frac{\partial (\rho H)}{\partial t} + \frac{\partial}{\partial x_i} (\rho u_i H) = \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i} \left( \frac{k}{c_p} \frac{\partial H}{\partial x_i} - \rho u_i \frac{\partial H}{\partial x_i} \right)$$

where $\rho$ is the density (kg·m⁻³), $p$ is the static pressure (Pa), $H$ is the total enthalpy (J), $k$ is the thermal conductivity (W·m⁻¹·K⁻¹), and $c_p$ is the specific heat (kJ·kg⁻¹·K⁻¹).

As demonstrated in previous studies, the flow pattern of the gas inside a vortex tube can be described reasonably by the standard $k$ $-\varepsilon$ turbulent model and is expressed as follows:

$$-\rho u_i u_i = \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} (\rho k + \mu) \frac{\partial u_i}{\partial x_i} \delta_{ij}$$

$$\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho u_i k)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \mu + \frac{\mu_t}{\alpha_t} \frac{\partial k}{\partial x_i} \right) + G_k + G_b - \rho \varepsilon - Y_M + S_k$$

Table 1. Structural Parameters of the Vortex Tube

| Chamber Diameter (mm) | Vortex Chamber Height (mm) | Hot End Diameter (mm) | Hot End Length (mm) | Cold Orifice Diameter (mm) | Cold Orifice Length (mm) | Transition Diameter (mm) | Cold End Diameter (mm) |Cold End Length (mm) |
|-----------------------|---------------------------|-----------------------|---------------------|---------------------------|-------------------------|--------------------------|-----------------------|---------------------|
| 7                     | 0.8                       | 7.3                   | 45                  | 2.9                       | 4                       | 14                       | 23                    |

Table 2. Physical Properties of the Gases Analyzed in this Study

| Gas                  | Specific Heat Capacity (kJ·kg⁻¹·K⁻¹) | Thermal Conductivity (W·m⁻¹·K⁻¹) | Dynamic Viscosity (kg·m⁻¹·s⁻¹) × 10⁻³ | Molar Mass (g·mol⁻¹) |
|----------------------|--------------------------------------|----------------------------------|----------------------------------------|----------------------|
| Hydrogen (H₂)        | 14.2874                              | 0.17892                          | 0.89445                                | 2.016                |
| Methane (CH₄)        | 2.2518                               | 0.03446                          | 1.12496                                | 16.043               |
| Air                  | 1.008                                | 0.02611                          | 1.85429                                | 28.95                |
| Ethane (C₂H₆)        | 1.8045                               | 0.02176                          | 0.94146                                | 30.07                |
| Oxygen (O₂)          | 0.9236                               | 0.0266                           | 2.0725                                 | 31.999               |
| Hydrogen Sulfide (H₂S)| 1.0527                               | 0.01459                          | 1.26724                                | 34.082               |
| Fluorine (F₂)        | 0.8293                               | 0.02607                          | 2.32629                                | 37.997               |
| Carbon Dioxide (CO₂) | 0.8772                               | 0.02061                          | 1.50584                                | 40.01                |
| Sulfur Dioxide (SO₂) | 0.6869                               | 0.00962                          | 1.30065                                | 64.065               |
| Chlorine (Cl₂)       | 0.517                                | 0.00896                          | 1.35099                                | 70.905               |
| Difluoromethane (CH₂F₂)| 0.8783                               | 0.01107                          | 1.41767                                | 52.024               |
| Tetrafluoroethane (CH₂F₂)| 0.9068                               | 0.01327                          | 1.24621                                | 102.031              |
where $\rho$, $t$, $\mu$, and $\mu_t$ are the density (kg·m$^{-3}$), time (s), dynamic viscosity (Pa·s), and turbulent viscosity (Pa·s), respectively; $u_i$ and $u_j$ are the time-averaged velocities (m·s$^{-1}$); $\delta_{ij}$ is the Kronecker delta; $G_{k}$ and $G_{b}$ represent the turbulence kinetic energy produced by the gradient of the average velocity and the buoyancy force, respectively; $\sigma_{k}$ and $\sigma_{\varepsilon}$ are the Prandtl number associated with the turbulent kinetic energy $k$ and the dissipation rate $\varepsilon$, respectively; $C_{1k}$, $C_{2k}$, and $C_{\varepsilon}$ are empirical coefficients; $Y_M$ represents the contribution from pulsating expansion; and $S_k$ and $S_\varepsilon$ are the source terms.

The boundary conditions used in this study include a pressure inlet and outlet. The turbulent conditions at the inlet and outlet boundaries of the vortex tube were set based on the hydraulic diameter and the turbulence intensity. The numerical convergence criteria used in this study was $10^{-6}$.

The wall treatment function was the standard wall function. The wall of the vortex tube was assumed to be thermally insulated, nonslipping, and smooth. The exchange of heat and work between the vortex tube and its surrounding environment was neglected in this study.

### 2.3. Determination of the Physical Properties of the Gas

The physical properties of the gases analyzed in this study include the specific heat capacity (at constant pressure), thermal conductivity, dynamic viscosity, and molar mass. To analyze the effects of the gas properties on the refrigeration temperature drop in the vortex tube, 12 gases with different physical properties were considered. The initial physical properties of each gas are listed in Table 2.

For the numerical simulations, the pressure/volume/temperature properties of each gas were further calculated using the equation of state for ideal gases. The initial specific heat capacity, dynamic viscosity, and thermal conductivity of each gas shown in Table 2 were calculated using the Peng–Robinson equation of state for real gases, the Wilke method, and the Mason–Saxena method, respectively. The molar mass of each gas was obtained by checking the corresponding CAS number.

### 2.4. Meshing and Mesh Independence Study

In general, the interior of a vortex tube comprises multiple interconnected structures as well as a petal-shaped rectification at the hot end. These complex geometric features make it difficult to generate a structured mesh for the entire model. Furthermore, mesh quality is often quite poor. To address these issues, an unstructured mesh was generated for different regions of the vortex tube based on the characteristics of the internal flow field. A higher mesh density was used in the vortex chamber, the vortex reducer, and sections near the surface of the hot and cold tubes, as shown in Figure 3. To ensure that an appropriate wall function was employed in the simulation, $y^+$ was controlled within the range of 30–300 near the surface of the cold and hot tubes.

The simulation results can be substantially affected by the number of nodes in the entire mesh. To ensure that the simulation results were sufficiently accurate, a mesh independence study was performed prior to the actual
simulations. Specifically, the refrigeration temperature drop and the cold outlet temperature were calculated using different mesh sizes at an inlet total air pressure of 0.3 MPa (g), a temperature of 300 K, and a cooling mass flow ratio of 0.5, which indicates the ratio of cold outlet mass flow rate to inlet mass flow rate. The corresponding results are shown in Figure 4. The refrigeration temperature drop and the cold outlet temperature become relatively stable when the total number of grid nodes reaches 608940. Therefore, the mesh independence criterion is satisfied under this condition.

2.5. Reliability Validation of the Numerical Model. Selecting appropriate governing equations and setting reasonable boundary conditions have a considerable impact on the simulation results. To validate the reliability of the numerical model developed in this study, the simulation results were compared with the experimental data collected from laboratory tests as well as the results obtained in previous studies.

2.5.1. Experimental Setup and Procedures. 2.5.1.1. Test Gas Selection. Due to experimental limitations and safety concerns, air was selected as the working fluid in our experiments. The cold outlet temperature and the refrigeration temperature drop were measured through laboratory tests conducted under a certain inlet pressure, temperature, and cooling mass flow ratio. These experimental results were then used to validate the numerical simulation.

2.5.1.2. Experimental System. The experimental system used in this study consisted of a gas/liquid supply system, a heating system, an experimental section, and a measurement system, as shown in Figures 5 and 6. The gas/liquid supply system contained an air compressor with a rated displacement of 0.2 m³·min⁻¹ and a metering pump with a rated flow rate of 0.003 m³·h⁻¹. The heating system was composed of a small heater with a rated power of 1.8 kW. The experimental section was the vortex tube. The inlet and outlet of the vortex tube were wrapped with insulation layers. The measurement system consisted of flow rate, pressure, and temperature measurement units.

2.5.1.3. Experimental Instruments and Accuracy. The technical specifications of the measurement devices are given in Table 3.

Table 3. Technical Properties of the Measurement Devices

| Device Type       | Property | Sensibility   |
|-------------------|----------|---------------|
| Precise Pressure  | YB-150   | 0–1.0 MPa     | ±0.25% MPa    |
| Gauges            |          |               |               |
| Bimetallic        | DTM/WST-491 | 223.15–473.15 K | ±0.5% K       |
| Thermometers      |          |               |               |
| Vortex Flow Meter | LUGB     | 4–20 m³·h⁻¹   | ±1.5% m³·h⁻¹  |

2.5.1.4. Experimental Procedures and Parameters. A two-phase experimental system containing both the liquid and gas was used in this study. For performing a single-phase air experiment, the metering pump was shut down, and no liquid was injected into the experimental system. Specifically, the air compressor was switched on to pump compressed air into the storage tank. The pressure of the compressed air inside the storage tank was maintained at 0.3 MPa by
adjusting the regulation valve. Subsequently, the heater was turned on and the heating power was adjusted to achieve the desired inlet gas temperature. The flow rate, pressure, and temperature of the inlet gas were measured using a vortex flow meter, a high precision pressure gauge, and a bimetallic thermometer, respectively, which were installed on the inlet pipeline. An identical set of measurement instruments was also installed on the cold/hot outlet pipeline. The cold outlet of the vortex tube was exposed to the ambient atmosphere. The backpressure at the hot end was adjusted to maintain a cooling mass flow ratio of 0.5. By changing the inlet gas temperature at this cooling mass flow ratio, different outlet temperatures were measured at the cold end of the vortex tube. The refrigeration temperature drop of the vortex tube and cooling mass flow ratio is defined as

\[ \Delta T_i = T_i - T_c \]  

(7)

\[ \eta = \frac{m_i}{m_i} \]  

(8)

where \( T_i \) and \( T_c \) are the inlet temperature (K) and the cold outlet temperature (K) and \( m_i \) and \( m_c \) are the inlet mass flow rate (kg s\(^{-1}\)) and the cold outlet mass flow rate (kg s\(^{-1}\)), respectively.

**2.5.1.5. Uncertainty Analysis of the Experiments.** The measurements are evaluated by uncertainty analysis. The average of the measurement (\( \bar{X} \)) is

\[ \bar{X} = \frac{\sum X_m}{n} \]  

(9)

where \( n \) is the number of the measurements and \( X_m \) is the measurement. Standard deviation (SD) is

\[ SD = \sqrt{\frac{\sum_{m=1}^{n} (X_m - \bar{X})^2}{n - 1}} \]  

(10)

uncertainty (\( U \)) was given by eq 11

\[ U = \frac{SD}{\sqrt{n}} \]  

(11)

The inlet pressure and cold outlet pressure were accepted as 0.3 MPa and ambient atmosphere, respectively. Therefore, the uncertainties of the pressure meters after the inlet and cold outlet were not calculated. The uncertainty of hot outlet pressure was also not calculated because its main function is to adjust the flow.

The uncertainties of the measurements are given in Table 4.\(^{1,2,22}\) According to the findings of uncertainty analysis, it was determined that the obtained results of the measurements were acceptable for use in the RHVT.

**2.5.2. Experimental Data Analysis and Model Validation.**

Table 5 presents the variation in the cold outlet temperature and refrigeration temperature drop with respect to varying inlet gas temperatures; these values were measured under an inlet gas pressure of 0.3 MPa, with the cold end exposed to the ambient atmosphere, and a cooling mass flow ratio of 0.5.

As shown in Table 5, an increase in the inlet gas temperature led to an increase in both the cold outlet temperature and the refrigeration temperature drop. However, the refrigeration temperature drop increases slightly with increasing inlet temperature. This suggests that increasing the inlet gas temperature alone has a minor effect on the refrigeration temperature drop.

To validate the reliability of the simulation results, they were compared with the experimental data obtained at a cooling mass flow ratio of 0.5, a total inlet pressure of 0.3 MPa, and an inlet temperature varying between 290 and 310 K, as shown in Figure 7a. This comparison revealed that the simulation and experimental results exhibited the same increasing trend with an increase in the total inlet temperature. This trend is attributed to the higher base separation temperature between the hot and cold streams caused by the increasing inlet temperature. The maximum and average relative errors between the simulation and experimental results are 0.7 and 0.5%, respectively. These deviations are likely induced by measurement errors. Nevertheless, these errors are within a reasonable range, which indicates that the proposed model is sufficiently reliable. As shown in Figure 7b, the refrigeration temperature drop increases as the inlet temperature increases. In particular, when the inlet temperature increases by 20 K, the refrigeration temperature drop increases by 1.4 and 2.8 K in the simulation and experimental results, respectively, which is consistent with the findings of Gong et al. and Wang et al.\(^{23,24}\)

On the other hand, the obtained values from the numerical modeling can be assessed in point of statistical fact. For this purpose, the performance of some bimetallic thermometers in a numerical model is determined using statistical methods like the root-mean-square error (RMSE), the coefficient of variation (cov), the absolute percentage change (R\(^2\)), and the mean percentage error (MPE), as given in the following equalities

\[ \text{RMSE} = \sqrt{\frac{\sum_{n=1}^{n} (y_{\text{output}} - y_{\text{output}})^2}{n}} \]  

(12)

Table 4. Uncertainties of the Measurements

| Measurement Devices | Uncertainty (U) |
|---------------------|-----------------|
| Bimetallic thermometers (\( T_i \)) | 0.23 K |
| Bimetallic thermometers (\( T_c \)) | 0.33 K |
| Bimetallic thermometers (\( T_h \)) | 0.28 K |
| Vortex flow meter (\( Q_i \)) | 0.039 m\(^3\)h\(^{-1}\) |
| Vortex flow meter (\( Q_c \)) | 0.021 m\(^3\)h\(^{-1}\) |
| Vortex flow meter (\( Q_h \)) | 0.027 m\(^3\)h\(^{-1}\) |

Table 5. Variation of the Experimental Data

| Inlet Temperature (K) | Cold Outlet Temperature (K) | Refrigeration Temperature Drop (K) |
|-----------------------|-----------------------------|-----------------------------------|
| 290                   | 266.95                      | 23.05                             |
| 292                   | 268.75                      | 23.25                             |
| 294                   | 270.45                      | 23.55                             |
| 296                   | 272.05                      | 23.95                             |
| 298                   | 273.75                      | 24.25                             |
| 300                   | 275.45                      | 24.55                             |
| 302                   | 277.25                      | 24.75                             |
| 304                   | 278.85                      | 25.15                             |
| 306                   | 280.65                      | 25.35                             |
| 308                   | 282.35                      | 25.65                             |
| 310                   | 284.15                      | 25.85                             |

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results (bimetallic thermometers in RHVT showed the acceptable
viscosity, and molar mass) of the gas were normalized using
capacity at constant pressure, thermal conductivity, dynamic
properties, the original physical properties (i.e., specific
heat capacity), and the average cold outlet experimental temperature (K), and
is the number of values.
According to Table 6, the statistical results (Table 6) of bimetallic thermometers in RHVT showed the acceptable
results (R² = 0.988, RMSE = 0.005). This indicates that the simulation results were statistically reliable.

3. RESULTS AND DISCUSSION
To eliminate dimensional effects between different physical properties, the original physical properties (i.e., specific heat capacity at constant pressure, thermal conductivity, dynamic viscosity, and molar mass) of the gas were normalized using
standard min–max method, as shown in eq 16

\[ x'_i = \frac{x_i - \min(x_i)}{\max(x_i) - \min(x_i)} \] (16)

where max and min represent the maximum and minimum values of the sample data, respectively.

3.1. Characteristics of the Influence of Physical Properties. Under the same structural parameters and operating conditions for the vortex tube, the refrigeration temperature drop is mainly affected by the physical properties of the gas.

The inlet total pressure and temperature of the vortex tube were kept at 0.3 MPa and 300 K, respectively. The static pressure at the hot outlet was adjusted to maintain a cooling mass flow ratio of 0.5, as shown in Table 7.

Figure 8 shows the distribution of the refrigeration temperature drop with respect to different physical properties.

As shown in Figure 8, no physical property investigated in this study has a particularly significant influence on the refrigeration temperature drop, so that it can cover the influence of other physical properties and make the refrigeration temperature drop monotonously with it; instead, the data are rather scattered and distributed in a disordered manner. Therefore, the refrigeration temperature drop is considered to be affected by all physical properties. The influence of the specific heat capacity is primarily reflected in the temperature drop rate. When the other properties remain constant, a smaller specific heat capacity represents a lower amount of energy required to decrease the gas temperature per unit degree. Thus, the temperature of the swirling gas decreases faster during energy conversion at the cold end. The influence of thermal conductivity on the refrigeration temperature drop is mainly reflected in the internal energy exchange rate between the inner and outer flows of the swirling gas. When the other properties remain constant, a higher thermal conductivity represents a faster exchange of the internal energy and, thus, a more rapid temperature drop of the gas in the inner swirling flow. The influence of dynamic viscosity on the refrigeration temperature drop is mainly reflected in the conversion from kinetic to internal energy. It is assumed that the flow field in the vortex tube remains stable, the vortex tube is stationary on a short time scale, and the working fluid is an ideal gas. The gas in the vortex tube undergoes an adiabatic and isenthalpic process

Table 6. Statistical Results \(T_c\) of Bimetallic Thermometers in RHVT

| parameter | cov | MPE | RMSE | \(R^2\) |
|-----------|-----|-----|------|-------|
| results   | 31.776 | 7.480 | 0.005 | 0.988 |

Table 7. Static Pressure at the Hot Outlet

| gas       | H₂  | CH₄ | air  | C₂H₆ | O₂  | H₂S | F₂  | CO₂ | SO₂ | Cl₂ | CH₃F₂ | CH₂FCF₃ |
|-----------|-----|-----|------|------|-----|-----|-----|-----|-----|-----|-------|--------|
| hot outlet pressure (MPa) | 0.072 | 0.076 | 0.074 | 0.081 | 0.074 | 0.077 | 0.075 | 0.078 | 0.079 | 0.079 | 0.062 |

Figure 7. Reliability verification: (a) the cold outlet temperature with respect to the inlet temperature and (b) the refrigeration temperature drop with respect to inlet temperature.
due to the irreversible viscous dissipation. Such a process can be described by eq 17. Considering that the gas flow in the vortex tube is adiabatic, has a high velocity, and is compressible, the Brinkman number is often greater than unity (i.e., $B_r \geq 1$). Therefore, viscous heating plays a dominating role in the energy transport inside a vortex tube

$$h_{v} + \frac{v_{2}^{2}}{2} = h_{1} + \frac{v_{1}^{2}}{2}$$

$$h = u + pV$$

where $h$ is the static enthalpy; $v$ is the gas flow velocity; and the subscripts $e$ and $f$ represent two arbitrary locations in the vortex tube, $u$ is the gas internal energy; $p$ is the gas pressure; and $V$ is the gas volume. As shown in eq 17, the energy conversion process in the vortex tube is simply an exchange between kinetic and internal energy. Figure 9 shows the meridian and equatorial plane distribution of the tangential velocity and static temperature of the air, obtained at a total inlet pressure of 0.3 MPa, a temperature of 300 K, and a cooling mass flow ratio of 0.5. It is evident that the tangential velocity and kinetic energy of the gas decrease due to the viscous shear effect as the gas flows in the axial direction. Meanwhile, the static temperature and internal energy increase in the direction toward the hot outlet. These features represent a conversion between kinetic and internal energy during the gas flow in the vortex tube.

The heat generated by viscous dissipation not only increases the internal energy of the gas but also provides a basis for the axial temperature separation between the inner and outer swirling flows. When the other physical properties remain constant, a higher dynamic viscosity represents a greater amount of energy converted into internal energy. In this case, the adiabatic expansion of the inner swirling gas also enables additional internal energy to be transported to the outer swirling gas, which results in a greater temperature drop. The influence of the molar mass on the refrigeration temperature drop is mainly reflected in the velocity field in

Figure 8. Variation in the refrigeration temperature drop with respect to physical properties: (a) specific heat capacity; (b) thermal conductivity; (c) dynamic viscosity; and (d) molecular weight.

Figure 9. Axial distribution of tangential velocity and static temperature: (a) the distribution of tangential velocity and (b) the distribution of static temperature.
sections in the vortex tube for different gases. The results indicate significantly different velocity fields in the vortex tube. In particular, for a small molar mass, the tangential, axial, and radial gas flow velocities are high. When using carbon tetrafluoride, which has the highest molar mass (102.031 g · mol⁻¹), as the working fluid, the tangential, axial, and radial absolute velocities are 1128, 810, and 97 m · s⁻¹, respectively. These results indicate significant velocity differences in the vortex tube for different gases. Higher tangential, axial, and radial velocities can lead to a strong temperature separation between the inner and outer swirling flows, therebyyielding a large refrigeration temperature drop.

To further investigate the effect of physical properties on the refrigeration temperature drop of the different gases investigated in this study, the impact of each property was quantified using the Gray relational analysis and ranked accordingly based on the level of influence. The subsequence of the original gas property sequence is $X_w^{(0)}(t) = \{X_1^{(0)}(t), X_2^{(0)}(t), ..., X_m^{(0)}(t)\}$, $w = 1, 2, ..., m$, $t = 1, 2, ..., N$. Here, $N$ is the length of each sequence, which represents a total of 12 different gases; $m$ is the number of elements in the sequence (the number of physical properties), which represents the specific heat capacity, thermal conductivity, dynamic viscosity, and molar mass of each gas. The parent sequence $\{X_0^{(0)}(t)\}$, $t = 1, 2, ..., N$ represents the refrigeration temperature drop of the 12 different gases. The sequence matrix of the original data is indicated by the matrix on the right above. The average transformation is performed using the following equations.

$$\tilde{X}_w(t) = \frac{X_w^{(0)}(t)}{X_w}, \quad \tilde{X}_w = \frac{1}{N} \sum_{t=1}^{N} X_w^{(0)}(t)$$  \hspace{1cm} (19)

Following an average transformation of the original data sequence matrix, the converted data of the subsequence $\{X_w(t)\}$ and parent sequence $\{X_0(t)\}$ are obtained. The converted data sequence matrix is indicated by the matrix on the right above. The average transformation is performed using the following equations.

$$X_0(t) = \frac{X_0^{(0)}(t)}{X_0}, \quad X_0 = \frac{1}{N} \sum_{t=1}^{N} X_0^{(0)}(t)$$  \hspace{1cm} (20)

The converted data sequence matrix is further used to calculate the correlation coefficients. The correlation coefficient $L_w(t)$ between the parent sequence $\{X_0(t)\}$ and the subsequence $\{X_w(t)\}$ can be calculated as
Table 8. Physical Properties of the Five Additional Gases Investigated in This Study

| gas             | specific heat capacity (kJ·kg⁻¹·K⁻¹) | thermal conductivity (W·m⁻¹·K⁻¹) | dynamic viscosity (kg·m⁻¹·s⁻¹) × 10⁻⁴ | molar mass (g·mol⁻¹) |
|-----------------|--------------------------------------|----------------------------------|----------------------------------------|----------------------|
| nitrogen (N₂)   | 1.0461                               | 0.02562                          | 1.78005                                | 28.014               |
| carbon monoxide (CO) | 1.0478                       | 0.0251                           | 1.77631                                | 28.01                |
| nitric oxide (NO) | 1.0108                       | 0.02576                          | 1.92213                                | 30.006               |
| ammonia (NH₃)   | 2.2067                               | 0.02307                          | 1.01946                                | 17.031               |
| oxygen (O₂)     | 0.9236                               | 0.0266                           | 2.0725                                 | 31.999               |

\[ L_w(t) = \frac{\Delta_{\text{min}} + s \Delta_{\text{max}}}{\Delta_w(t) + s \Delta_{\text{max}}} \]  

where \( \Delta_w(t) = |X_w(t) - X_m(t)| \) (1 ≤ w ≤ m) is the absolute difference between the two sequences; \( \Delta_{\text{max}} = 7.074 \) and \( \Delta_{\text{min}} = 0.008 \) are the maximum and minimum absolute differences between all sequences, respectively; and \( s \) is an identification coefficient (typically set as 0.1). \( \Delta_w(t) \) and the matrix of calculated correlation coefficients \( L_w(t) \) are given below:

\[ \Delta_w(t) = \begin{bmatrix} 7.074 & 0.008 & 0.971 & 0.062 & 1.017 & 0.698 & 0.995 & 0.803 & 0.821 & 1.028 & 0.639 & 0.156 \\ 6.008 & 0.032 & 0.537 & 0.168 & 0.511 & 0.784 & 0.453 & 0.692 & 0.892 & 1.028 & 0.769 & 0.208 \\ 1.265 & 0.637 & 0.481 & 0.627 & 0.885 & 0.343 & 1.429 & 0.097 & 0.178 & 0.194 & 0.098 & 0.254 \\ 1.265 & 0.716 & 0.613 & 0.028 & 0.532 & 0.237 & 0.305 & 0.009 & 0.598 & 0.660 & 0.369 & 2.060 \\ 0.092 & 1.000 & 0.426 & 0.930 & 0.415 & 0.509 & 0.420 & 0.474 & 0.468 & 0.412 & 0.531 & 0.829 \\ 0.107 & 0.967 & 0.575 & 0.817 & 0.587 & 0.480 & 0.617 & 0.511 & 0.447 & 0.412 & 0.485 & 0.782 \\ 0.363 & 0.532 & 0.602 & 0.536 & 0.449 & 0.681 & 0.335 & 0.889 & 0.808 & 0.794 & 0.888 & 0.744 \\ 0.363 & 0.503 & 0.542 & 0.973 & 0.577 & 0.757 & 0.706 & 0.998 & 0.548 & 0.523 & 0.665 & 0.258 \\ 76.7900 \end{bmatrix} 

The correlation between the parent sequence \( \{X_n(t)\} \) and the subsequence \( \{X_w(t)\} \) can be expressed by the average correlation coefficient between the two sequences as follows:

\[ r_{ow} = \frac{1}{N} \sum_{t=1}^{N} L_w(t) \]  

As indicated by the calculation results for the correlation vector, the influence levels of the physical properties of four gases on the refrigeration temperature drop are in the following order: dynamic viscosity > molar mass > thermal conductivity > specific heat capacity.

### 3.2. Regression Model for the Physical Properties.

#### 3.2.1. Development and Validation of the Empirical Equation.

A multi-factor regression analysis was performed on the simulation results using SPSS. After eliminating nonsignificant terms at a significance level of \( \alpha = 0.05 \), the following empirical equation was obtained:

\[ Y = 22.83 + 7.98X_3 + 95.15X_1 \times X_5 - 466.31X_1 \times X_4 + 94.94X_2 \times X_4 - 31.09X_3 \times X_4 \]  

where \( Y \) is the refrigeration temperature drop (K) in the vortex tube at a cooling mass flow ratio of 0.5; \( X_3 \) is the specific heat capacity at constant pressure (kJ·kg⁻¹·K⁻¹); \( X_1 \) is the thermal conductivity (W·m⁻¹·K⁻¹); \( X_5 \) is the dynamic viscosity ((kg·m⁻¹·s⁻¹) × 10⁻⁴); and \( X_4 \) is the molar mass (g·mol⁻¹). The correlation coefficient of the regression equation is \( R = 0.99574 \). After adjustments, the correlation coefficient is \( R_5 = 0.992176 \), \( P = 0.0001 < 0.05 \), \( \text{DF}(5,6) \), and \( F = 26.7900 \). After checking the \( F \) distribution at \( \alpha = 0.05 \), \( F \) is determined to be \( 4.39 \), which is less than \( 139.9365 \). Thus, the empirical regression model provides a statistically significant prediction.

Figure 11 presents a comparison between the simulated and the corresponding calculated values using the gas properties and the empirical correlation. As shown in the figure, the calculated and simulated values exhibit the same trend and are in good agreement. For hydrogen sulfide, the maximum relative error is 4.43%, and the average relative error is 1.87%.

The empirical equation for the refrigeration temperature drop is further validated using five other working gases. The physical properties of these gases are listed in Table 8.

The physical properties listed in Table 8 were obtained using the same methods as those for the properties in Table 2.

As shown in Figure 12, the empirical equation can be used to predict the refrigeration temperature drop of different gases with high accuracy. For ammonia, the maximum relative error is 7.59%, while the average relative error is 2.58%. This small deviation further confirms the reliability of the proposed empirical equation.

#### 3.2.2. Analysis of the Synergetic Effect of the Physical Properties.

To further explore the synergetic effect, which indicated that the effect of the physical properties on the refrigeration temperature drop was not independent of the physical properties on the refrigeration temperature drop in the vortex tube, a two-dimensional isothermal contour map was plotted based on the results obtained using the empirical equation. Figure 13 shows the contour maps of the refrigeration temperature drop obtained under any two of the following four conditions: (1) a specific heat capacity of 2.1686 kJ·kg⁻¹·K⁻¹, (2) a thermal conductivity of 0.0323 W·m⁻¹·K⁻¹, (3) a dynamic viscosity of 1.44188 kg·m⁻¹·s⁻¹.
(x10^-5), and (4) a molar mass of 42.849 g·mol^-1. These values were considered as the average values of the physical properties listed in Table 2 and were normalized.

As shown in Figure 13a, the dynamic viscosity and molar mass have opposite effects on the refrigeration temperature drop. The maximum refrigeration temperature drop is obtained by increasing the dynamic viscosity and decreasing the molar mass. Changing only one of these two properties does not lead to a large refrigeration temperature drop. As shown in Figure 13b, the refrigeration temperature drop is inversely proportional to the specific heat capacity but proportional to the thermal conductivity. Thus, it can be maximized by decreasing the specific heat capacity and increasing the thermal conductivity. Figure 13c shows that a large refrigeration temperature drop can be obtained easily using a low specific heat capacity. Increasing the dynamic viscosity also increases the refrigeration temperature drop to a certain extent. However, this enhancement becomes weaker with a decrease in the specific heat capacity.

The results in Figure 13d suggest the existence of a certain synergetic effect between the specific heat capacity and molar mass. However, when the specific heat capacity is sufficiently low, it has a dominant impact on the refrigeration temperature drop. Figure 13e indicates a complete synergetic effect of the thermal conductivity and dynamic viscosity. Changing any of these two parameters can have a significant impact on the refrigeration temperature drop. Figure 13f indicates that the refrigeration temperature drop is dominated by the molar mass when it is considerably low. Furthermore, the minimum refrigeration temperature drop is obtained by decreasing the thermal conductivity while increasing the molar mass.

**4. CONCLUSIONS AND FUTURE WORK**

The influence of gas physical properties on the refrigeration temperature drop in a vortex tube was comprehensively investigated using a combination of numerical simulations and laboratory experiments. The following conclusions were drawn based on the results obtained in this study:

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![Figure 11. Empirical formula verification.](image1)

![Figure 12. Empirical formula prediction.](image2)

![Figure 13. Synergetic effect of physical properties: (a) dynamic viscosity and molar mass; (b) specific heat capacity and thermal conductivity; (c) specific heat capacity and dynamic viscosity; (d) specific heat capacity and molar mass; (e) thermal conductivity and dynamic viscosity; and (f) thermal conductivity and molar mass.](image3)
(1) A numerical model of the vortex tube was built. The accuracy of this numerical model was validated by comparing the simulation results with the results from laboratory experiments and previous studies. A comparative analysis indicated a maximum relative deviation of 0.7% and an average relative deviation of 0.5%. These small errors suggest that the proposed model can be used to calculate the refrigeration temperature drop under different gas properties with good accuracy.

(2) At a total inlet pressure of 0.3 MPa, a temperature of 300 K, and a cooling mass flow ratio of 0.5, the maximum and minimum refrigeration temperature drops, which were obtained for air and carbon tetrafluoride, respectively, differ by up to 16 K. This indicates that the physical properties of the gas have a significant impact on the refrigeration temperature drop in a vortex tube.

(3) The physical properties exhibited a complicated influence on the refrigeration temperature drop of the vortex tube, and it is hard to tell that which one has a dominating effect. Instead, it is affected by all physical properties. The refrigeration temperature drop is negatively correlated with the specific heat capacity and molar mass, whereas it is positively correlated with thermal conductivity and dynamic viscosity. The influence level of the four physical properties on the refrigeration temperature drop, which was investigated under the working condition of study, was ranked based on the Gray relational analysis, and the following order was determined: dynamic viscosity > molar mass > thermal conductivity > specific heat capacity.

(4) The refrigeration temperature drop of 12 gases with distinctive physical properties in the vortex tube were calculated at a total inlet pressure of 0.3 MPa, a temperature of 300 K, and a cooling mass flow ratio of 0.5. A regression analysis was performed on the calculation results, and an empirical equation was developed and validated. Based on this empirical equation, two-dimensional isothermal contour maps were plotted to investigate the synergetic effects of the different gas properties. The results suggest that the specific heat capacity, thermal conductivity, dynamic viscosity, and molar mass have a synergetic effect on the refrigeration temperature drop in a vortex tube.

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**Notes**

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**NOMENCLATURE**

- $B_r$: Brinkman number
- $cov$: the coefficient of variation
- $c_p$: specific heat capacity at constant pressure (kJ·kg$^{-1}$·K$^{-1}$)
- $D$: degree of freedom
- $F$: F-distribution
- $G_b$: the turbulence kinetic energy produced by the buoyancy force
- $G_k$: the turbulence kinetic energy produced by the gradient of the average velocity
- $h$: static enthalpy (J)
- $m$: the number of elements in the sequence
- $m_{c_{out}}$: cold outlet mass flow rate (kg·s$^{-1}$)
- $m_{i}$: inlet mass flow rate (kg·s$^{-1}$)
- MPE: the mean percentage error
- $N$: the length of each sequence
- $P$: the probability
- $P_{c_{out}}$: cold outlet pressure (MPa)
- $Q_{c_{out}}$: cold outlet volume flow rate (m$^3$·h$^{-1}$)
- $Q_{b}$: cold outlet volume flow rate (m$^3$·h$^{-1}$)
- $Q_{i}$: inlet volume flow rate (m$^3$·h$^{-1}$)
- $R^2$: the percentage of absolute change
- $R$: the correlation coefficient
- $R_{adj}$: the correlation coefficient adjusted
- RMSE: root-mean square error
- $s$: the identification coefficient
- $S_{j}$: the source terms
- $S_{k}$: the source terms
- SD: standard deviation
- $T$: time (s)
- $T_{c_{out}}$: cold outlet temperature (K)
- $T_{h}$: hot outlet temperature (K)
- $T_{i}$: inlet temperature (K)
- $U$: uncertainty
- $X$: the average of the measurement
- $X_{m}$: the measurement
- $X_{c}$: the specific heat capacity at constant pressure (kJ·kg$^{-1}$·K$^{-1}$)
- $X_{T}$: the thermal conductivity (W·m$^{-1}$·K$^{-1}$)
- $X_{D}$: the dynamic viscosity (kg·m$^{-1}$·s$^{-1}$) × 10$^{-3}$
- $X_{M}$: the molar mass (g·mol$^{-1}$)
- $Y_{M}$: the contribution from pulsating expansion
- $Y_{output}$: the cold outlet calculated temperature (K)
- $Y_{output}$: the cold outlet calculated temperature (K)
the cold outlet experimental temperature (K)
y_{\text{actual}}
the average cold outlet experimental temperature (K)
Y
the refrigeration temperature drop (K)

\[ \rho \] density (kg\cdot m^{-3})
\[ \mu \] dynamic viscosity (Pa\cdot s)
\[ \mu_t \] turbulent viscosity (Pa\cdot s)
\[ u_i \] time-averaged velocity (m\cdot s^{-1})
\[ u_j \] time-averaged velocity (m\cdot s^{-1})
\[ \delta_i \] Kronecker delta symbol
\[ k \] the turbulent kinetic energy (m^2\cdot s^{-2})
\[ \varepsilon \] the dissipation rate (m^2\cdot s^{-3})
\[ \sigma_i \] the Prandtl number associated with the turbulent kinetic energy
\[ \sigma_e \] the Prandtl number associated with the dissipation rate
\[ \eta \] cooling mass flow ratio
\[ \nu \] velocity (m\cdot s^{-1})
\[ \alpha \] significant level

\section*{ABBREVIATIONS USED}

CFD computational fluid dynamics
RHVT Ranque—Hilsch vortex tubes

\section*{SUBSCRIPT}
c cold outlet
e arbitrary locations in the vortex tube
f arbitrary locations in the vortex tube
h hot outlet
i inlet

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