THE SECOND LAW ANALYSIS OF NATURAL GAS BEHAVIOUR WITHIN A VORTEX TUBE

by

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Original scientific paper
DOI: 10.2298/TSCI110505082F

Vortex tube is a simple device without a moving part which is capable of separating hot and cold gas streams from a higher pressure inlet gas stream. The mechanism of energy separation has been investigated by several scientists and second law approach has emerged as an important tool for optimizing the vortex tube performance. Here, a thermodynamic model has been used to investigate vortex tube energy separation. Further, a method has been proposed for optimizing the vortex tube based on the rate of entropy generation obtained from experiments. Also, an experimental study has been carried out to investigate the effects of the hot tube length and cold orifice diameter on entropy generation within a vortex tube with natural gas as working fluid. A comparison has been made between air and natural gas as working fluids. The results show that the longest tube generates lowest entropy for natural gas. For air, it is middle tube which generates lowest entropy. Integration of entropy generation for all available cold mass fractions unveiled that an optimized value for hot tube length and cold orifice diameter is exist.

Key words: *vortex tube, natural gas, the second law analysis, energy recovery, natural gas pressure reduction system*

Introduction

Energy conservation and its efficient use are currently an important issue. The current reduction in oil reserves combined with the increase in its price have led in the past years to an increasing interest and research in the field of waste energy recovery. The pressure reduction points in natural gas (NG) industries are currently wasting energy spots [1-3].

There are many places in natural gas industries where NG pressure needs to be reduced. This pressure reduction is currently accompanied with entropy generation or exergy destruction. For example, as a NG distribution pipeline nears a customer, the high-pressure gas needs to be reduced to working level. Throttling valves are currently utilized to reduce the gas pressure. This causes that a valuable exergy of the gas to be wasted. Some studies [4-7] are proposed to utilize expansion machine instead of throttling valves, but due to high installation cost and the reliability, it is not feasible to install expansion machine at low consumption points.

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As NG pressure reduction takes places at throttling valves, the gas temperature is also lessened. There is possibility of taking advantage of low pressure gas stream as heat sink (*e. g.* [8]) if the gas temperature is low enough. But for the case where the NG pressure is reduced from distribution pressure (about 5 bar) to working level (about 1 bar), the gas temperature is not low enough for refrigeration purposes.

The vortex tube (VT) is a simple device without a moving part which makes it a reliable system [9, 10]. The VT is also capable of separating a high pressure gas into hot and cold gas streams. The cold stream then could be utilized for refrigeration. These characteristics make a VT as potential candidate for replacing the throttling valves in NG industries. In case of replacement, it would be possible to produce refrigeration especially for high buyers such as hotels and restaurants.

The VT, also known as Ranque-Hilsch vortex tube (RHVT) was first discovered by Ranque [11]. The German physicist Hilsch [12] improved the design, provided comprehensive experimental and theoretical studies intend to improve the efficiency of the VT. He methodically inspected the effect of the inlet pressure and the geometrical parameters of the VT on its performance and presented a possible explanation of the energy separation process. There have been a lot of researchers since then which studying VT aiming to enhance its performance. These researches on VT could be divided into two categories as experimental and theoretical study.

Experimentally, the works have been carried out to study effects of two types of parameters on the VT performance as thermophysical parameters and geometrical parameters. Saidi and Valipour [13] have classified the parameters affecting VT performance as the thermophysical parameters such as inlet gas pressure, type of gas and cold gas mass ratio, moisture of inlet gas and the geometrical parameters, *i. e.*, diameter and length of main tube diameter of the outlet orifice and shape of the entrance nozzle were designated and studied. Dincer *et al.* [14] have studied the effects of position, diameter and angle of a mobile plug, located at the hot outlet side experimentally to get best performance. The most efficient (maximum temperature difference) combination of parameters is obtained for a plug diameter of 5 mm, tip angle of 30° or 60°, by keeping the plug at same position, and letting the air enter into the VT through 4 nozzles. Increasing the inlet pressure beyond 380 kPa did not bring any appreciable improvement in the performance. Orhan and Muzaffer [15] have carried out a series of experiments to investigate effects of the length of the pipe, the diameter of the inlet nozzle, and the angle of the control valve on the performance of the counter flow VT for different inlet pressures. Experiments showed that the higher the inlet pressure, the greater the temperature difference of the outlet streams. It is also shown that the cold fraction is an important parameter influencing the performance of the energy separation in the VT. Optimum values for the angle of the control valve, the length of the pipe, and the diameter of the inlet nozzle are obtained. Nimbalkar and Muller [16] presented the results of a series of experiments focusing on various geometries of the "cold end side" for different inlet pressures and cold fractions. The experimental results indicated that there is an optimum diameter of cold-end orifice for achieving maximum energy separation. It was observed that for cold fraction less or equal than 60%, the effect of cold end orifice diameter is negligible and above 60% cold fraction it becomes prominent. The results also show that the maximum value of performance factor was always reachable at a 60% cold fraction irrespective of the orifice diameter and the inlet pressure. Stephan *et al*. [17] measured the temperature profiles at different positions along a VT axis and concluded that the length of the vortex tube would have an important influence on the transport mechanism inside. Promvonge and Eiamsa-ard [18] experimentally studied the energy and temperature separation in the VT with the snail entrance. Eiamsa-Ard and Promvonge [19] furthermore presented a complete overview of the past investigations of the mean flow and temperature behaviours in a turbulent VT in order to understand the nature of the temperature separation or Ranque-Hilsch effect. They have proposed optimum values for the cold orifice to the VT inlet diameter (*d/D*) of 0.5, the angle of the conical control valve of 50 degree, the length of the VT to the VT inlet diameter (*L/D*) of 20 and the diameter of the inlet nozzle to the VT inlet diameters (δ/D) of 0.33 for air as working fluid.

Theoretical studies have been carried out in parallel with experiments. Most theories are based on results obtained from the related experimental work, some are based on numerical simulations. Balmer [20] who investigated theoretically the temperature separation phenomenon in a VT, used the second law of thermodynamics to show temperature separation effect with a net increase in entropy is possible when incompressible liquids are used in the tube. This was confirmed by experiments with liquid water which showed that temperature separation occurred when an inlet pressure was sufficiently high. Takahama *et al*. [21] study resulted in several formulas for determining the performance and efficiency of VT under a variety of operating conditions, which induced the optimum ratios of VT dimensions corresponding to the highest efficiency. Lin *et al.* [22] made an experimental investigation to study the heat transfer behaviour of a water-cooled VT with air. Piralishvili and Polyaev [23] made experimental investigations into this effect in so-called double-circuit VT. The possibility of constructing a double-circuit VT refrigeration machine as efficient as a gas expansion system was demonstrated. In 2002, the VT system was used to enrich the concentration of methane by Kulkarni and Sardesai [24]. They tried to separate methane and nitrogen gases using VT. This particular separation or the resulting enrichment of methane concentration has applications in the mining industry. In 2004, Poshernev and Khodorkov [25] have suggested to utilize the VT a pre-cooling system for NG liquefaction. The conical VT was further investigated theoretically by Khodorkov *et al.* [26] for chemical applications. Silverman [27] questioned whether the VT is a violation of the second law of thermodynamics or not. Usage of exergy concepts in evaluating the performance of energy systems are increasing nowadays due to its clear indication of loss at various locations which is more informative than energy analysis. Exergy is the work potential of energy in a given environment. It has been also employed in studying VT performance theoretically. Rosen and Dincer [28] studied the effect of dead state variation on energy and exergy analysis of thermal system and showed that the variation does not affect the energy and exergy values significantly. In exergy analysis losses are measured in terms of exergy destruction, which provide direct measure of thermodynamic inefficiencies. Saidi [29] studied the effect of inlet pressure on temperature difference in the VT and discussed the advantage of an exergy analysis. He also listed equation for calculating rates of entropy generation and total irreversibility. Kirmaci [30] studied the exergy analysis on VT for two different gases (air, oxygen) by using different inlet pressures and different nozzle numbers.

Briefly, the aim of all these studies are; firstly, to find empirical expressions for geometrical/thermophysical parameters which can be used for improving the VT performance; secondly, to apply the VT for wide application purposes, like cooling and heating, cleaning, purifying and separation, *etc*.; finally, to investigate the internal process and to understand the mechanism of the energy separation.

Farezaneh-Gord and Kargaran [31] have carried out an experimental study to investigate NT temperature behaviours in a VT. The effects of the VT cold orifice diameter on the VT thermal separation are also studied. Farzaneh-Gord *et al.* [3] have studied the effects of hot tube length on the VT thermal separation. In this study a combination of experimental and theoretical study has been carried out to optimize the VT performance. The theoretical study is based on the second law analysis. Due to desired application of the VT, the NG has been adopted as working fluid. Based on the experimental study, NG temperature behaviours within the VT have been investigated. The combined effects of hot tube length and cold orifice diameter on entropy generation in the VT have been studied.

The proposed application of the VT

Natural gas is delivered to the customers through distribution pipe line approximately at 5 bar in Iran (probably in most countries) but all NG equipments consume it at working pressure of slightly higher than 1 bar. Currently, the throttling valves are employed to reduce the pressure through a constant enthalpy process. A schematic diagram of a current typical reduction system is shown in fig. 1. Inlet gas has a high pressure (P_1 = 5 bar) and temperature (T_1) which is typically the ambient temperature. The standard outlet pressure is approximately 1 bar and the output temperature (T_2) is approximately 2 K lower that inlet temperature. As the temperature reduction is low, it is not possible to take advantage of this pressure reduction. The VT is a potential candidate to replace current system as a reliable system. In case of replacement, it would be possible to produce refrigeration especially for high buyers.

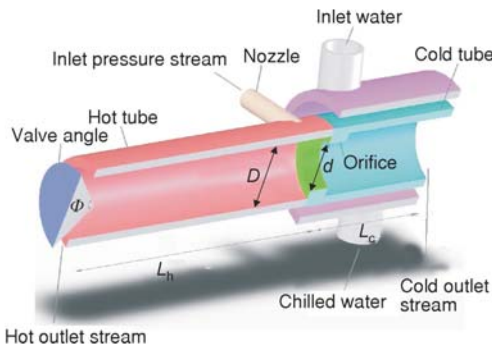

Figure 1. A schematic diagram of the proposed VT combined with a heat exchanger
(for color image see journal web site)

In this study, the throttling valve is proposed to be replaced by a combined VT and heat exchanger as shown in fig. 1. The high pressure NG introduced at inlet port and expanded through the VT. The gas is separated into two streams and flow through the hot and cold tube. The water is circulated around the cold tube. As the water is circulated, its temperature drops due to heat exchange with cold stream. Now the chilling water is available for refrigeration purposes. Finally the two low pressure NG streams could be combined and delivered to the equipments as current system.

Chemical composition of NG

NG composition (mixture) varies with location, climate, and other factors. The gas is refined before flowing into the pipe lines. Gas consumed in Shahrood (where the experiments were carried out) is totally processed in the Khangiran refinery. Table 1 details the chemical composition of the gas.

The vortex tube parameters

The parameters influence of which have been studied are:

– *The geometrical parameters*

Figure 1 shows a schematic diagram of a counter flow VT which has been constructed and used in this study. As shown in the figure, the geometrical parameters are inlet VT diameters (D), cold orifice diameter (d), inlet nozzle diameter (δ), conical controlling valve angle (Φ), cold tube length (L_c) and hot tube length (L_h).

Table 1. Chemical composition of natural gas from the Khangiran refinery

Component	Chemical formula	Percent mole fraction
Carbon dioxide	CO_2	0.055
Nitrogen	N_2	0.428
Methane	CH_4	98.640
Ethane	C_2H_6	0.593
Propane	C_3H_8	0.065
Isobutane	C_4H_{10}	0.015
Normal butane	C_4H_{10}	0.034
Isopentane	C_5H_{12}	0.026
All hydrocarbon compounds with more than 5 carbon in their chemical formula	C_6+	0.125
	Total	100%

(Source: Khangiran refinery website)

– *The flow parameters*

As mentioned by Eiamsa-ard and Promvonge [19], the most important flow (thermophysical) parameter is believed to be cold mass fraction which defined as:

$$\mu_c = \frac{\dot{m}_c}{\dot{m}_i} \tag{1}$$

where, $\dot{m}_c$ and $\dot{m}_i$ are the mass flow rates at the inlet of the VT and at the cold outlet, respectively.

– *The performance parameters*

There are various definitions to examine the performance of a VT. Considering the definition of the VT performance, (*e. g.* isentropic efficiency, coefficient of performance ..., see [19]). It could be noted that these definitions comprise following two parameters:

– the hot temperature difference which is temperature difference between hot (T_h) stream and cold (T_c) streams:

$$\Delta T_h = T_h - T_c \tag{2}$$

– the cold temperature difference which is temperature difference between inlet (T_i) and cold (T_c) streams:

$$\Delta T_c = T_i - T_c \tag{3}$$

It could be realized that these two parameters (hot and cold temperature difference) are affected by thermophysical/geometrical parameters and they have effects on the VT performance. So here, these two parameters have been called performance parameters.

The thermodynamic analysis of a VT

To examine the performance of a VT, it is necessary to invoke the first and second law of thermodynamics. The first law could be employed to define thermal efficiency or to evaluate temperature change due to the gas expansion.

Figure 2 illustrated the control volume considered in the thermodynamic analysis. For the analysis, only the gas properties at the inlet and outlets are of interest and the details of the in-

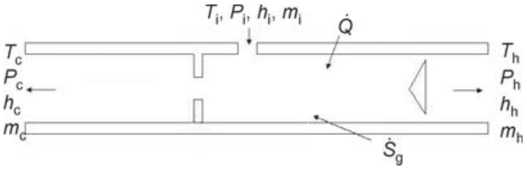

Figure 2. A schematic diagram of the thermodynamic system (the inlet is tangential to the tube axis)

ternal process do not need to be considered. In this system, there are three open boundaries namely inlet, cold and hot outlets.

In the steady state situation, the conservation of the mass could be expressed as below:

$$\dot{m}_\mathrm{i} = \dot{m}_\mathrm{c} + \dot{m}_\mathrm{h} \tag{4}$$

The first law analysis

Neglecting the kinetic and the potential energy and assuming steady-state condition, the first law of thermodynamics could be expressed as:

$$\dot{Q} + \dot{m}_\mathrm{i} h_\mathrm{i} = \dot{m}_\mathrm{c} h_\mathrm{c} + \dot{m}_\mathrm{h} h_\mathrm{h} + \dot{W} \tag{5}$$

For the VT, the work term is zero and the system could be assumed adiabatic, as a result, the eq. (5) could be simplified as below:

$$\dot{m}_\mathrm{i} h_\mathrm{i} = \dot{m}_\mathrm{c} h_\mathrm{c} + \dot{m}_\mathrm{h} h_\mathrm{h} \tag{6}$$

Applying the mass conservation and considering the definition of the cold mass fraction, the final form of the energy equation could be expressed as:

$$h_\mathrm{i} = \mu_\mathrm{c} h_\mathrm{c} + (1 - \mu_\mathrm{c}) h_\mathrm{h} \tag{7}$$

or as

$$\mu_\mathrm{c} = \frac{h_\mathrm{h} - h_\mathrm{i}}{h_\mathrm{h} - h_\mathrm{c}} \tag{8}$$

when the working fluid could be assumed as prefect gas, the specific enthalpy depends linearly on the temperature ($h = c_p T$), hence the energy equation could be more simplified to obtain an equation as:

$$\mu_\mathrm{c} = \frac{T_\mathrm{i} - T_\mathrm{h}}{T_\mathrm{c} - T_\mathrm{h}} \tag{9}$$

Equation (5) (or 4) could be employed in an experimental study to evaluate the cold mass fraction without measuring the flow rates.

If the aim is to produce maximum refrigeration, the term $\mu_\mathrm{c}(T_\mathrm{i} - T_\mathrm{c})$ should be maximized when the cold temperature is also fulfilled the requirement for low sink temperature. There have been many researches in which the VT optimization was based on this term. Lewins and Bejan [22] mentioned that the cooling effect can be used two ways. Firstly, it might simply be used to take heat from a body and raising the cold temperature accordingly or secondly, the cold stream might be used in a heat exchanger to cool the incoming stream itself, in a regenerative fashion, thus lowering the exit temperature still further. Based on these assumptions, they have developed an expression for optimizing the cooling effects. The expression was based on cold mass fraction (thermophysical parameter) and the performance parameters. Here the optimization is going to be based on geometrical parameters.

The second law analysis

The first law analysis accounts only for heat balance and cooling or heating effect. The second law analysis based on irreversibility approach, considers also the losses due to separation effect as well as the pressure drop losses. As mentioned by Saidi and Yazdi [29], the separation losses account for a significant portion of VT irreversibility. Therefore, the VT optimal design couldn't be accessible following a first law analysis. Assuming steady state condition and neglecting heat transfer, the second law of thermodynamics for the system could be expressed as:

$$\dot{S}_{\mathrm{gen}} = \dot{m}_{\mathrm{h}} s_{\mathrm{h}} + \dot{m}_{\mathrm{c}} s_{\mathrm{c}} - \dot{m}_{\mathrm{i}} s_{\mathrm{i}} \tag{10}$$

Applying the mass conservation and considering the definition of the cold mass fraction, the above equation could be re-expressed as:

$$\dot{S}_{\mathrm{gen}} = \dot{m}_{\mathrm{i}} [(1-\mu_{\mathrm{c}}) s_{\mathrm{h}} + \mu_{\mathrm{c}} s_{\mathrm{c}} - s_{\mathrm{i}}] = \dot{m}_{\mathrm{i}} [(1-\mu_{\mathrm{c}})(s_{\mathrm{h}} - s_{\mathrm{i}}) + \mu_{\mathrm{c}}(s_{\mathrm{c}} - s_{\mathrm{i}})] \tag{11}$$

Equation (11) could be non-dimensionalized as:

$$s_g = \frac{\dot{S}_{gen}}{\dot{m}_i c_p} = \frac{1}{c_p [(1-\mu_{\mathrm{c}})(s_{\mathrm{h}} - s_{\mathrm{i}}) + \mu_{\mathrm{c}}(s_{\mathrm{c}} - s_{\mathrm{i}})]} \tag{12}$$

For the case of a ideal gas, by employing the following $T\mathrm{d}s$ relation:

$$s - s_{\mathrm{i}} = c_p \ln \frac{T}{T_{\mathrm{i}}} - \mathrm{R} \ln \frac{p}{p_{\mathrm{i}}} \tag{13}$$

Equation (12) could be simplified as:

$$s_{\mathrm{g}} = (1-\mu_{\mathrm{c}}) \ln \frac{T_{\mathrm{h}}}{T_{\mathrm{i}}} - (1-\mu_{\mathrm{c}}) \frac{\gamma - 1}{\gamma} \ln \frac{p_{\mathrm{h}}}{p_{\mathrm{i}}} + \mu_{\mathrm{c}} \ln \frac{T_{\mathrm{c}}}{T_{\mathrm{i}}} - \mu_{\mathrm{c}} \frac{\gamma - 1}{\gamma} \ln \frac{p_{\mathrm{c}}}{p_{\mathrm{i}}} \tag{14}$$

In most cases (as here) $p_{\mathrm{c}} \approx p_{\mathrm{h}}$, this makes the above equation to be more simplified as:

$$s_{\mathrm{g}} = \ln \frac{T_{\mathrm{h}}^{1-\mu_c} T_{\mathrm{c}}^{\mu_c}}{T_{\mathrm{i}}} - \ln \left(\frac{p_{\mathrm{c}}}{p_{\mathrm{i}}} \right)^{\frac{\gamma-1}{\gamma}} \tag{15}$$

If the process could be assumed reversible, then:

$$T_{\mathrm{cs}} = T_{\mathrm{h}}^{1-\mu_c} T_{\mathrm{c}}^{\mu_c} = T_{\mathrm{i}} \left(\frac{p_{\mathrm{c}}}{p_{\mathrm{i}}} \right)^{\frac{\gamma-1}{\gamma}} \tag{16}$$

For the case of an irreversible process:

$$T_{\mathrm{cs}} = T_{\mathrm{h}}^{1-\mu_c} T_{\mathrm{c}}^{\mu_c} = \mathrm{e}^{s_g} T_{\mathrm{i}} \left(\frac{p_{\mathrm{c}}}{p_{\mathrm{i}}} \right)^{\frac{\gamma-1}{\gamma}} \tag{17}$$

Equations (17) and (9) could be combined to obtain an equation for determining cold stream temperature in case of knowing dimensionless entropy generation as:

$$\left(\frac{T_{\mathrm{i}} - \mu_{\mathrm{c}} T_{\mathrm{c}}}{1 - \mu_{\mathrm{c}}} \right)^{1-\mu_c} T_{\mathrm{c}}^{\mu_c} = \mathrm{e}^{s_g} T_{\mathrm{i}} \ln \left(\frac{p_{\mathrm{c}}}{p_{\mathrm{i}}} \right)^{\frac{\gamma-1}{\gamma}} \tag{18}$$

The entropy generation rate is a measure to calculate irreversibility in the system. The minimization of the entropy generation could be used as a tool to optimise the VT. Practically, the goal is to optimise geometrical parameters over all range of cold mass fraction. For optimizing the geometrical parameters, integration over range of cold mass fraction may be carried out as:

$$Ns_{\mathrm{g}} = \int_0^1 s_{\mathrm{g}} \mathrm{d}\mu_{\mathrm{c}} \tag{19}$$

Lewins and Bejan [10] have also carried out the optimization of the VT by employing the second law. But again, their optimization was not carried out over geometrical parameters. Saidi and Yazdi [29] have employed exergy analysis and the related experimental work to optimize the VT over geometrical parameters but the optimization was carried out for fixed cold mass fraction.

As mentioned, the most theories are based on results obtained from the related experimental work. Here based on experimental values, the non-dimensional entropy generation created by the system is calculated using eq. (15) for all values of cold mass fractions. Then by employing eq. (19), a single value for total non-dimensional entropy generation is computed. The total non-dimensional entropy generation could be treated as a tool for optimizing the VT. The employed method could be used to optimize combined effects of geometrical parameters.

Experimental apparatus

Table 2 shows the detailed geometrical parameters. These values are selected based on proposed optimum value by Eiamsa-Ard and Promvonge [19] for air as working fluid. As it can be realized, the hot tube length and cold orifice diameter have been varied.

Table 2. The geometrical parameters and theirs values

Parameter	D [mm]	d [mm]	δ [mm]	Φ [degree]	L_c [mm]	N	L_h [mm]
Value	25	8	8	50	50	1	250
		12.1					519
		17.7					769

Figure 3 shows a schematic diagram of the experimental apparatus and measuring devices. High pressure NT from distribution pipe line is directed tangentially into the VT. The high pressure gas expands in the vortex tube and separates into cold and hot streams. The cold gas leaves the central orifice near the entrance nozzle, while the hot gas discharges the periphery at the far end of the tube. The control valve is being used to control the flow rate of the hot stream. This would help to regulate cold mass friction. Two orifice flow meters which constructed according to ISO5167 are employed to measure the mass flow rate of the hot and cold streams. 3 PT100 temperature sensors are installed to measure inlet, hot and cold streams temperatures. The temperature sensors are installed within the tubes which carried out flow. Two pressure transmitters are utilized to quantify inlet pressure and outlet pressure of hot streams. The pressure sensors are installed same as temperature sensors. The description of a technique of experiment in more details are presented in Farezaneh-Gord and Kargaran [31] and Farzaneh-Gord *el al*. [3].

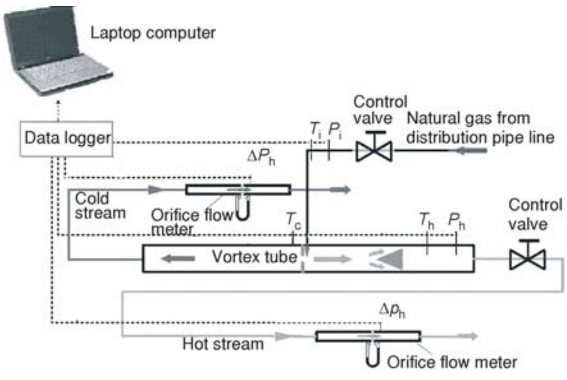

Figure 3. A schematic diagram of experimental set-up

Figure 4 shows the experimental test bed in operation. The experiments have been carried out in Shahrood University of Technology main campus. The VT was connected to NG distribution pipe line using a flexible pipe. Note from the figure, the hot length of the VT and hot stream flow meter were painted in red (right). On the other hand, the cold length of the VT and cold stream flow meters were painted in blue (left). The VT was made from steel with inlet diameters of 25 mm. During the tests, the cold orifice diameter and hot tube length of the VT was varied among 3 available one as detailed in tab. 2. It should be also pointed out that all tests have been carried out at inlet pressure of 5 bar which is absolute pressure within distribution NG pipe line.

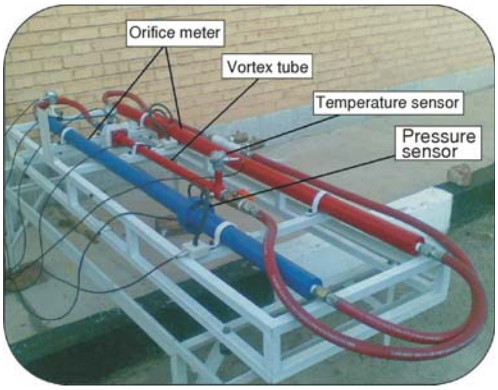

Figure 4. The experimental test bed in operation (for color image see journal web site)

The experimental test bed has been also moved into Koolab Toos company to investigate thermal separation of air. The company is able to provide high quantity of air at constant high pressure. In theses cases, the inlet pressure was also set at 5 bar. This situation provides the opportunity to compare results of NG and air for same VT.

Error analysis

The maximum possible errors in various measured parameters namely temperature and pressure were estimated by using the method proposed by Moffat [32]. Errors were estimated from the minimum values of output and the accuracy of the instrument. This method is based on careful specification of the uncertainties in the various experimental measurements. If an estimated quantity, Y, depends on independent variables like x_i then the error in the value of Y is given by:

$$\frac{\partial Y}{Y} = \sqrt{\sum_{1}^{n} \left(\frac{\partial x_i}{x_i} \right)^2} \tag{20}$$

where $\partial x_i/x_i$ are the errors in the independent variables, ∂x_i – the accuracy of the measuring instrument, and x_i – the minimum value of the output measured.

Error in temperature measurement

PT100 temperature sensors were used to measure the gas temperatures. Temperatures directly are logged in file with accuracy of 0.1 °C. The maximum possible error in the case of temperature measurement was calculated from the minimum values of the temperature measured and accuracy of the instrument. The error in the temperature measurement is:

$$\frac{\partial T}{T} = \sqrt{\left(\frac{\partial T_{T100}}{T_{\min}} \right)^2 + \left(\frac{\partial T_{\log}}{T_{\min}} \right)^2} = \sqrt{\left(\frac{0.5}{12} \right)^2 + \left(\frac{0.1}{12} \right)^2} = 0.04 = 4\% \tag{21}$$

Error in pressure measurement

Pressure transmitters were used to measure the gas pressure. Pressures directly are logged in file with accuracy of 0.01 bar. The error in the pressure measurement is:

$$\frac{\partial P}{P}=\sqrt{\left(\frac{\partial P_{\mathrm{tran}}}{P_{\min}}\right)^2+\left(\frac{\partial T_{\log}}{T_{\min}}\right)^2}=\sqrt{\left(\frac{0.01}{1.33}\right)^2+\left(\frac{0.01}{1.33}\right)^2}=0.01=1\% \tag{22}$$

Error in flow rate measurement

Flow measurement has been made using orifice flow meters. Uncertainty analysis was conducted according to the standard procedures reported in ISO5167. The analysis shows that the error in the flow rate measurement is 4.5%.

Results and discussion

Figure 5 shows effects of cold orifice diameter on temperature differences for NG as working fluid. It can be seen that the tube with L_h =769 mm creates highest temperature difference. It should be also pointed out there is a μ_c in each case which causes the temperature difference to be maximized. For the current configuration at $\mu_c \approx 0.65$ the highest temperature differences are encountered. This is in agreement with Nimbalkar and Muller [16] findings which showed that the maximum value of performance factor (defined as temperature difference there) was always reachable at a 60% cold fraction irrespective of the orifice diameter and the inlet pressure.

Figure 6 shows effects of inlet pressure on temperature difference for L_h = 519 mm for air as working fluid. As shown, increasing the inlet pressure has increased the temperature gradient between the hot and the cold outlets. It is worth noting that the temperature difference for various inlet pressures has a similar trend and it can be seen temperature difference at the cold mass fraction between 0.65 and 0.75, having a maximum value. This finding is slightly differed from finding of Nimbalkar and Muller [16] who reported that maximum temperature difference was always reachable at a 60% cold fraction irrespective of the inlet pressure.

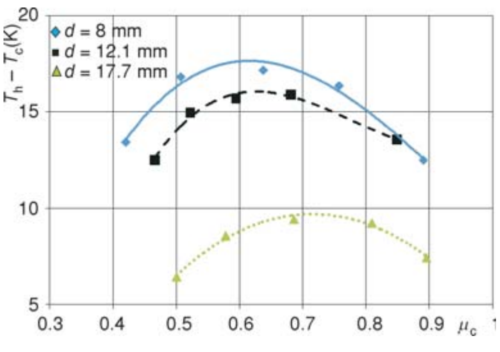

Figure 5. Effects of cold orifice diameter on temperature difference for NG

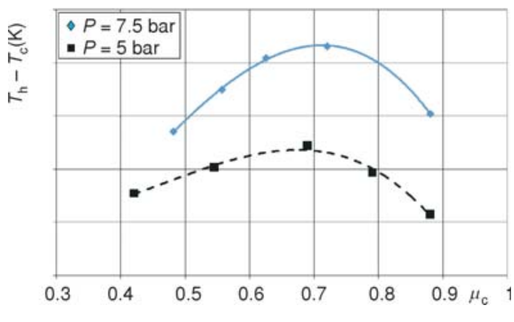

Figure 6. Effects of inlet pressure on temperature difference for air

Figure 7 shows effects of hot tube length on amount of entropy generation for d =17.7 mm where NG is working fluid and inlet pressure is 5 bar. Saidi [29] showed that entropy generation decreases with increase in tube length due to the increase in temperature difference. Comparing among 3 hot tube lengths, it could be realized the longest tube generates lowest entropy in most of μ_c value which in agreement Saidi [29] finding. Note from figure, minimum entropy generation is always reachable at a 70% cold fraction irrespective of tube lengths.

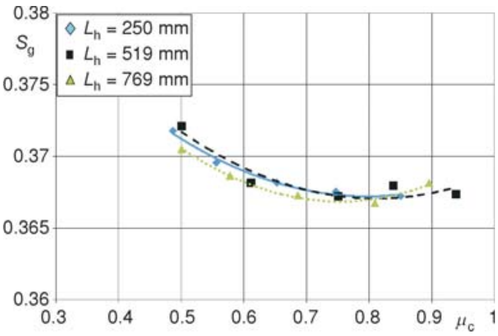

Figure 7. Effects of hot tube length on amount of entropy generation for NG

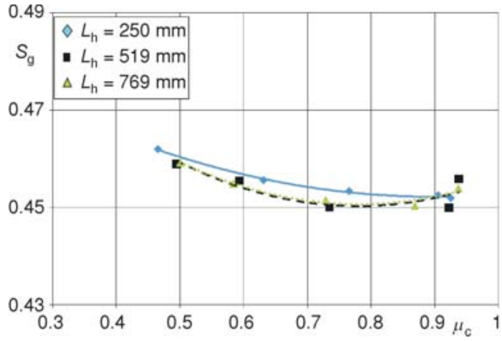

Figure 8. Effects of hot tube length on amount of entropy generation for air

Figure 8 shows effects of hot tube length on amount of entropy generation for d = 17.7 mm and air as working fluid. In this case, with increasing hot tube length, temperature differences increase and entropy generation decreases. It should be mentioned that entropy generation could be lowered by improving the VT performance by methods suggested by Wu *et al*. [33] and [15]. As shown, the tubes with L_h = 250 mm generates most entropy than L_h = 512 and 769 mm.

Figure 9 and 10 show effects of cold orifice diameter on amount of entropy generation for L_h = 769 mm and NG and air as working fluid, respectively. Note from the figures, minimum entropy generation is always reachable $\mu_c \approx 6.5$ irrespective of orifice diameter for both air and NG. For NG, the orifice with d = 12.1 mm generates lowest entropy while for air the lowest amount of entropy generation is encountered at d = 12.1 mm.

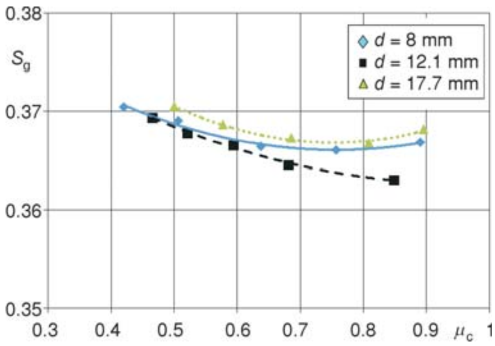

Figure 9. Effects of cold orifice diameter on amount of entropy generation for NG

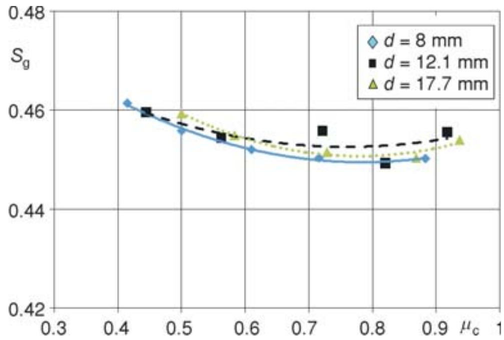

Figure 10. Effects of cold orifice diameter on amount of entropy generation for air

Figure 11 shows comparison between air and NG as working fluid on amount of entropy generation for L_h = 769 mm and pressure 5 bar. As shown, that amount of entropy generation for air is higher than NG for all value of μ_c that is because entropy generation varies according to the temperature gradient between the hot and cold outlets and as it can be seen in figs. 5 and 6, the temperature difference for air is more than NG .

Figure 12 shows the combination effect of cold orifice diameter and hot tube length on entropy generation. From author knowledge, there has been no previous research which

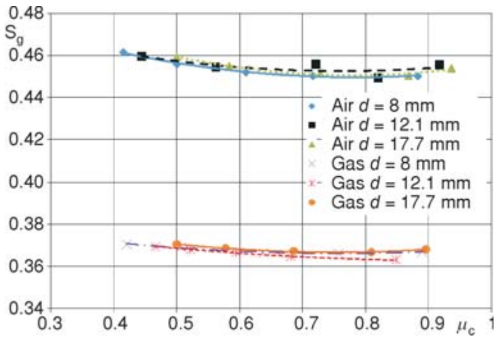

Figure 11. Comparing amount of entropy generation for NG and air when L_h = 769 mm

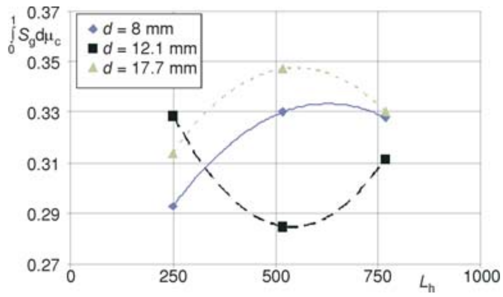

Figure 12. Effects of cold orifice diameter and hot tube length on amount of entropy generation for NG

combined effects of various parameters as presented here. Presenting results as illustrated in fig. 12 have a few advantages. The main advantage is capability of optimizing the geometrical parameters all in once. Note from this figure, it could be easily concluded that the case with d = 12.1 mm and L_h = 512 mm generates the minimum amount of entropy. This indicates the optimize point for combination cold orifice diameter and hot tube length. Eiamsa-Ard and Promvonge [19] proposed optimum values for the cold orifice to the VT inlet diameter (d/D) of 0.5 and the length of the VT to the VT inlet diameter (L_h/D) of 20 for air as working fluid. As D = 25 mm in this study, the results from fig. 10 is well supported by findings of Eiamsa-Ard and Promvonge [19] for air as working fluid.

Conclusions

As a new application of the VT has been proposed in this work, an experimental study has been carried out to investigate NG temperature behaviours within a VT and find the optimize values for cold orifice diameter and hot tube length. Further, based on the second law analysis, the amount of entropy generation has been calculated. The entropy generation has been integrated over all possible cold mass fractions (flow parameter) to study effect of geometrical parameters only. A comparison has been made for thermal behaviour of air and NG within the VT.

The results indicated that as hot tube length or inlet pressure increases the temperature difference increases too. The second law analysis shows that the longest tube generates lowest entropy for NG. For air, it is middle tube which generates lowest entropy. Integration of entropy generation for all available cold mass fractions unveiled that the optimized value are middle hot tube length and cold orifice diameter. This is consistent with previous researches which proposed optimum values for the cold orifice to the VT inlet diameter (d/D) of 0.5 and the length of the vortex tube to the VT inlet diameter (L_h/D) of 20 for air as working fluid.

Finally, proposed application of the VT (which is replacement of throttling valve in natural gas distributions pipelines) is capable of generating refrigeration from current pressure reduction system.

Acknowledgment

This study has been supported by Shahrood University of Technology. Special thanks also go to Koolab Toos company.

Nomenclature

D	– vortex tube inlet diameter, [mm]
d	– cold orifice diameter, [mm]
L	– vortex tube length, [mm]
L_c	– cold tube length, [mm]
L_h	– hot tube length, [mm]
$\dot{m}$	– mass flow rate, [gs^{-1}]
P	– pressure, [bar]
S	– entropy, [kJK^{-1}]
T	– temperature, [°C, K]
ΔT	– temperature difference, [K]
ΔT_{h}	– hot temperature difference, [K]

Greek symbols

δ	– inlet nozzle diameter, [mm]
Φ	– conical controlling valve angle, [degree]
μ_{c}	– cold mass fraction (= $\dot{m}_{\mathrm{c}}/\dot{m}_{\mathrm{i}}$)

Subscript

c	– cold stream
h	– hot stream
i	– inlet stream
gen	– generation
o	– dead state
1	– inlet gas condition of the pressure drop system
2	– outlet gas condition of the pressure drop system

References

[1] Farzaneh-Gord, M., *et al.*, Energy Destruction in Iran's Natural Gas Pipe Line Network, *Energy Exploration and Exploitation, 25* (2007), 6, pp. 393-406
[2] Farzaneh-Gord, M., Magrebi, M. J., Exergy of Natural Gas Flow in Iran's Natural Gas Fields, *International Journal of Exergy, 6* (2009), 1, pp. 131-142
[3] Farzaneh-Gord, M., *et al.*, Feasibility of Accompanying Uncontrolled Linear Heater with Solar System in Natural Gas Pressure Drop Stations, *Energy, 41* (2012), 1, pp. 420-428
[4] Farzaneh-Gord, M., Dashtbayaz, M. D., Recoverable Energy in Natural Gas Pressure Drop Stations: A Case Study of the Khangiran Gas Refinery, *Energy Exploration & Exploitation, 26* (2008), 2, pp. 71-82
[5] Farzaneh-Gord, M., Deymi-Dashtbayaz, M., A New Approach to Enhancing Performance of a Gas Turbine (case study: Khangiran Refinery), *Applied Energy, 86* (2009), 12, pp. 2750-2759
[6] Farzaneh-Gord, M., Deymi-Dashtbayaz, M., Effect of Various Inlet Air Cooling Methods on Gas Turbine Performance Energy, *Energy, 36* (2011), 2, pp. 1196-1205
[7] Farzaneh-Gord, M., *et al.*, Feasibility of Employing Solar Energy in Natural Gas Pressure Drop Stations, *Journal of the Energy Institute, 84* (2011), 3, pp. 165-173
[8] Farzaneh-Gord, M., *et al*., A New Inlet Air Cooling Method for Improving Gas Turbine Efficiency (case study: Khangiran Refinery), *Journal of the Energy Institute, 82* (2009), 3, pp. 150-158
[9] Cockerill, T., The Ranque-Hilsch Vortex Tube, Ph. D. thesis, Cambridge University, Engineering Department, Sunderland, UK, 1995
[10] Lewins, J., Bejan, A., Vortex Tube Optimization Theory, *Energy, 24* (1999), 11, pp. 931-943
[11] Ranque, G. J., Experiments on Expansion in a Vortex with Simultaneous Exhaust of Hot Air and Cold Air, *J Phys Radium* , 4:112-4 S-115 (1933); also translated as General Electric Co., Schenectady Works Library 1947; T. F. 3294, (1933)
[12] Hilsch, R., The use of Expansion of Gases in a Centrifugal Field as a Cooling Process, *Rev Sci Instrum., 18* (1947), 2, pp. 108-113
[13] Saidi, M. H., Valipour, M. S., Experimental Modelling of Vortex Tube Refrigerator, *Applied Thermal Engineering, 23* (2003), 15, pp. 1971-1980
[14] Dincer, K., *et al.*, Experimental Investigation of the Performance of a Ranque-Hilsch Vortex Tube with Regard to a Plug Located at the Hot Outlet, *International Journal of Refrigeration, 32* (2009), 1, pp. 87-94
[15] Orhan, A., Muzaffer, B., An Experimental Study on the Design Parameters of a Counter Flow Vortex Tube, *Energy, 31* (2006), 14, pp. 2763-2772
[16] Nimbalkar, S. U., Muller, M. R., An Experimental Investigation of the Optimum Geometry for the Cold end Orifice of a Vortex Tube, *Applied Thermal Engineering, 29* (2009), 2-3, pp. 509-514
[17] Stephan, K., *et al*., An Investigation of Energy Separation in a Vortex Tube, *Int. J. Heat Mass Transfer, 26* (1983), 3, pp. 341-348
[18] Promvonge, P., Eiamsa-Ard, S., Experimental Investigation of Temperature Separation in a Vortex Tube Refrigerator with Snail Entrance, *ASEAN J. Sci. Technol. Dev., 21* (2004), 4, pp. 297-308
[19] Eiamsa-Ard, S., Promvonge, P., Review of Ranque-Hilsch Effects in Vortex Tubes, *Renewable and Sustainable Energy Reviews, 12* (2008), 7, pp. 1822-1842

[20] Balmer, R.T., Pressure Driven Ranque-Hilsch Temperature Separation in Liquids, *J Fluids Eng., 110* (1988), 2, pp. 161-164
[21] Takahama, H., *et al.*, Performance Characteristics of Energy Separation in a Steam-Operated Vortex Tube, *Int J. Eng. Sci., 17* (1979), 6, pp. 735-744
[22] Lin, S., *et al*., A Heat Transfer Relation for Swirl Flow in a Vortex Tube, *Can J Chem Eng, 68* (1990), 6, pp. 44-7
[23] Piralishvili, S. A., Polyaev, V. M., Flow and Thermodynamic Characteristics of Energy Separation in a Double-Circuit Vortex Tube – An Experimental Investigation, *Exp Thermal Fluid Sci, 12* (1996), 4, pp. 399-410
[24] Kulkarni, M. R., Sardesai, C. R., Enrichment of Methane Concentration via Separation of Gases Using Vortex Tubes, *J. Energy Engrg, 128* (2002), 1, pp. 1-12
[25] Poshernev, N. V., Khodorkov, I. L., Natural-Gas Tests on a Conical Vortex Tube (CVT) with External Cooling, *Chemical and Petroleum Engineering, 40* (2004), 3-4, pp. 212-217
[26] Khodorkov, I. L., *et al*., The Vortex Tube a Universal Device for Heating, Cooling, Cleaning, and Drying Gases and Separating Gas Mixtures, *Chemical and Petroleum Engineering, 39* (2003), 7-8, pp. 409-415
[27] Silverman, P. M., The Vortex Tube: A Violation of the Second Law, *Eur. J. Phys., 3* (1982), 2, pp. 88-92
[28] Rosen, M. A., Dincer, I., Effect of Varying Dead-State Properties on Energy and Exergy Analyses of Thermal Systems, *International Journal of Thermal Sciences, 43* (2004), 2, pp. 121-133
[29] Saidi, M. H., Yazdi, A., Exergy Model of a Vortex Tube System with Experimental Results, *International Journal of Energy, 24* (1999), 7, pp. 625- 632
[30] Kirmaci, V., Uluer, O., An Experimental Investigation of the Cold Mass Fraction, Nozzle Number and Inlet Pressure Effects on Performance of Counter Flow Vortex Tube, *Journal of Heat Transfer, 131* (2009), 8, pp. 081701- 081709
[31] Farzaneh-Gord, M., Kargaran, M., Recovering Energy at Entry of Natural Gas into Customer Premises by Employing a Counter-Flow Vortex Tube, *Oil & Gas Science and Technology – Revue de l'IFP, 65* (2010), 6, pp. 903-912
[32] Moffat, R. J., Using Uncertainty Analysis in the Planning of an Experiment, *J. Fluids Eng., 107* (1985), 2, pp. 173-178
[33] Wu, Y. T., *et al.*, Modification and Experimental Research on Vortex Tube, *Int. J. Refrigeration* (2007), 6, pp. 1042-1049

Paper submitted: May 5, 2011
Paper revised: April 26, 2012
Paper accepted: May 12, 2012