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# Theoretical and Experimental Study of Coiled Finned tube Heat exchanger for Helium Liquefaction Plant

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**Abstract.** The performance of a liquefier is governed by the effectiveness of its heat exchangers. Coiled finned tube heat exchangers are used in medium capacity helium liquefiers based on Collins cycle. Experimental and theoretical analysis of the first heat exchanger in helium liquefaction cycle is presented in this work. The objective of this study is to design, optimize and test the heat exchanger to achieve the desired effectiveness and therefore the drop in the temperature of the gas on the hot side. A transient numerical model is employed to optimize the geometrical configuration and operating parameters of the coiled finned tube heat exchanger. Effect of variable physical properties, axial conduction and radiative heat transfer of the outer shell is also considered. The numerical model is validated with experimental results found in open literature. The numerical model is then used for deriving the optimum geometrical configuration of the coiled heat exchanger. Experimental observations from the tests carried out on a fabricated prototype are also reported in this work. Finally, results of the numerical model are also compared with the experimental results obtained in this study.

## 1. Introduction

Helium is a very rare and expensive gas. Its consumption is of the order of 100 million cubic meters per annum and it is increasing by 4 to 5 % every year [1]. Therefore, to conserve it, every research institute using helium on large scale should have a helium liquefier. Heat exchangers play a significant role in determining liquefier performance. Atrey [2] estimated a 12 % decrease in liquid helium yield when the effectiveness of the heat exchanger dropped from 0.97 to 0.95. In the liquefaction process of helium, no liquid yield is obtained if the heat exchanger effectiveness is less than 0.85 [3]. Collins had used coiled heat exchanger for the first time in a helium liquefaction plant [4]. Several authors [5-10] have presented numerical analysis and optimization of helically coiled heat exchangers for Joule-Thomson coolers with nitrogen and argon as working fluids. Experimental and CFD analysis of helically coiled heat exchangers with water as working fluid were reported by Jayakumar et al. [11]. Experimental studies on the pressure drop and heat transfer coefficients in coiled heat exchangers with helium have been reported by Gupta et al. [12] [13]. They also presented theoretical studies with flow area calculations to optimize the heat exchanger by fixing NTU (Number of Transfer Units) of the first heat exchanger in a Helium liquefaction cycle [14]. However, optimization of geometrical and operating parameters with variable thermo-physical properties along the length of heat exchanger, considering a transient approach, has not been reported so far. Recently, Haskins and El-Genk [15] presented a CFD analysis and correlation of pressure loss on shell side of coiled tubes heat exchanger in case of isothermal flows of liquid sodium, water and helium. The objective of this work is to explore different combinations of the geometrical and operating parameters which can lead to the same thermal performance of the first heat exchanger in a helium liquefaction cycle. For this purpose, in this work, a one-dimensional transient numerical model has been employed to simulate a coiled finned tube heat exchanger for medium capacity helium plant. The numerical model is validated with experimental data,



of the first heat exchanger in the helium liquefaction cycle, available in the literature. Thereafter, the numerical model is applied to optimize the mass flow rate and geometrical parameters to achieve the required thermal performance in terms of temperature drop and effectiveness of the heat exchanger. A coiled finned tube heat exchanger is fabricated based on the optimum geometrical configuration derived from the numerical analysis. Experiments are carried out to find the effectiveness of heat exchanger. Finally, the experimental results are compared with those obtained with the numerical model for specific operating conditions.

## 2. Coiled finned tube heat exchanger

The main parts of a coiled tube heat exchanger shown in Figure 1 are: a finned tube with hot high pressure gas; a mandrel over which finned tube is wound; and an external covering cylinder (shell) which forms external annular space for returning low pressure gas. The working fluid helium, at high pressure, flows inside the coiled finned tube and the low pressure cold gas flows over the fins in the opposite direction. The operating parameters (e.g., pressure and mass flow rate) and the physical parameters like mean coil diameter, tube diameter, shell diameter, fin height, fin density, diametrical clearance etc., have significant effect on heat exchanger performance.

## 3. Numerical model

In the present work, a transient one-dimensional numerical model is employed for numerical simulation of a recuperative heat exchanger employed in a helium liquefaction cycle. Variation of thermo-physical properties of the working fluid is also considered over the operating range of temperature (77-300 K) and pressure (10-15 bar). Axial conduction for the solid elements taken into account along with the radiative heat transfer from the outer surface of the shell. The details of the numerical method, association of CV (control volume) of finned tube and outer fluid, position of the nodes for CVs and with the global resolution algorithm are detailed in [10]. The flow related parameters i.e., flow area, hydraulic diameter, etc. are evaluated according to Gupta et al. [12]. Thermo-physical properties of the working fluid over the range of temperature and pressure are generated from the aspenOne software [17]. The numerical model developed under present work is validated against the experimental results available in literature [12] [14]. The same geometrical configuration as in [12] and [14] was used for both helium and nitrogen as working fluids. In this work, the steady state is achieved by marching in time from the initial ambient conditions of the heat exchanger at time  $t=0$ . Table 1 shows the comparison of steady state temperatures obtained with numerical simulation with the corresponding experimental values available in the literature. In this work,  $x$  is the Cartesian coordinate and  $L$  is the axial length of the heat exchanger while  $T$  and  $p$  are temperature and pressure respectively. Subscripts  $h$ ,  $c$ ,  $amb$ ,  $in$ , and  $out$  denote hot, cold, ambient, inlet and outlet conditions respectively. The relative differences in the hot side outlet temperatures, for both cases, are less than 3.6 % and that for the cold side are less than 2.8 %. It is observed that as the clearance reduces there is a reduction in  $T_{h,out}$  and increase in  $T_{c,out}$  values. This is due to less leakage flow with lower clearance values. Although, the variation is not significant, it shows that the numerical model is giving physically realistic results. A similar observation has been reported by Gupta [14] with their theoretical predictions. Therefore, the numerical model presented in this paper is able to capture the physics of the fluid flow and heat transfer in the recuperative heat exchanger.

## 4. Parametric analysis

In the present study, numerical simulations are carried out by varying inlet pressure on hot side from 11 to 15 bar. The cold side pressure is fixed to 1.5 bar. The inlet temperature on the hot side ( $T_{h,in}$ ) is 300 K while the same on the cold side ( $T_{c,in}$ ) is taken to be 85 K. The mass flow rate ( $\dot{m}$ ) is varied from 1g/s to 5 g/s. The effect of geometrical parameters like fins density, axial length, and coil diameter is also studied. The finned tube inner and outer diameters are 10.8 mm and 11.5 mm respectively. Fin density ( $n$ ) is varied from 300-1000 fins/metre and axial length ( $L$ ) of the heat exchanger is varied between 600-2000 mm.

Table 1. Comparison of end temperatures on hot and cold sides

	c (mm)	$T_{h,in}$ (K)	$T_{c,in}$ (K)	$T_{c,out}$ (K)	$T_{h,out}$ (K)
Working fluid: Nitrogen ( $p_{h,in}=15$ bar, $m_f=9.5$ g/s)					
Experimental	1.2	296	151	284	165
Numerical	1.2	296	151	292.07	163.92
Numerical	0.5	296	151	292.85	163.12
Numerical	0.3	296	151	293.06	162.91
Working fluid: Helium ( $p_{h,in}=11$ bar, $m_f=1.6$ g/s)					
Experimental	1.2	293	82	282	88
Numerical	1.2	293	82	288.83	91.18

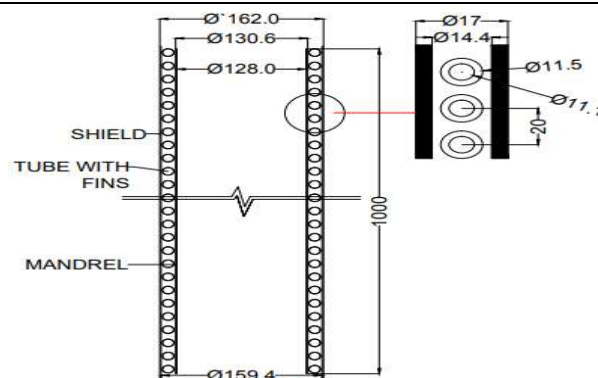


Figure 1: Coil finned tube heat exchanger

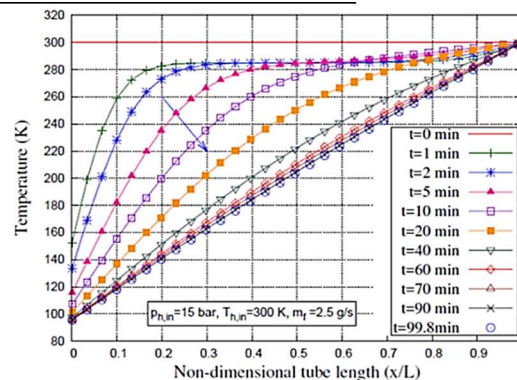


Figure 2: Temperature distribution of the hot steam

#### 4.1 Transient evaluation of the heat exchanger

The transient temperature evolution of the hot fluid in the finned tube is shown in Figure 2. For this case the fin density is 700 fins per meter and mass flow rate is 2.5 g/s. The pressures on the hot ( $p_{h,in}$ ) and cold ( $p_{c,in}$ ) sides are 15 bar and 1.5 bar respectively. As depicted in the Figure 2, initially at  $x=0$ , temperature  $T$  is equal to 300 K. With time, as  $x$  increases, temperature decreases quickly initially and later on rate of decrease in temperature reduces significantly. After the 1st minute the outlet temperature on the hot side falls to around 152 K. A steady state value of around 95 K is obtained after 99.8 min. Figure 2 shows that the temperature profiles after 90 min and those at steady state (99.8 min) coincide with each other. This shows that the convergence criteria for declaring steady state is satisfactory.

#### 4.2 Effect of geometrical parameters

The effect of mass flow rate and fin density on effectiveness and outlet temperature of hot fluid is plotted in Figure 3 and Figure 4 respectively. It is observed from Figure 3 that for the same mass flow rate effectiveness increases with fin density but range of increase in effectiveness decreases. To achieve an effectiveness higher than 0.95, mass flow rate in the range of 2-3 g/s is necessary. Similarly, a minimum fin density of around 700/m is also required. Figure 4 shows that lowest temperature drop can be attained for mass flow rate in the range of 2-3g/s. Figure 5 shows that the optimum heat exchanger length is around 1 m to 1.2 m for achieving effectiveness above 0.95. Overall heat transfer coefficient ( $U$ ) and the heat transfer area ( $A$ ) are crucial for achieving the desired heat transfer and hence the temperature drop/rise in hot/cold fluid streams of the heat exchanger. Figure 6 shows the plot of  $UA$  and effectiveness for different mass flow rates. It is observed from the figure that maximum effectiveness can be achieved for a mass flow rate in the range of 2 to 3 g/s with relatively lower  $UA$  values of 270-380 W/K. Figure 6 also shows the minimum requirement of  $UA$  to achieve the maximum effectiveness for a given mass flow rate.

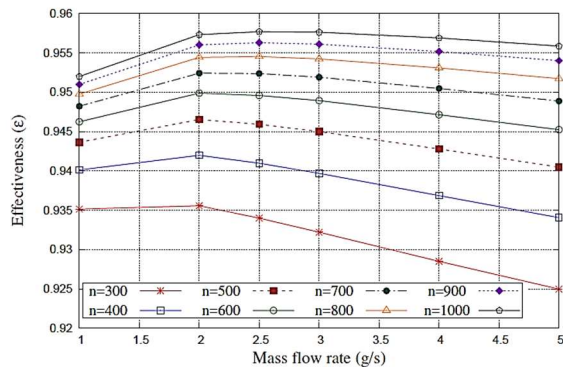


Figure 4: Effect of mass flow rate and fin density on heat exchanger effectiveness.

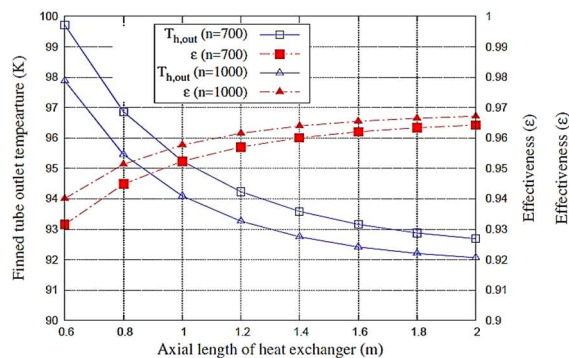


Figure 5: Heat exchanger length on temperature drop and effectiveness.

## 5. Experimentation

The schematic diagram of test facility of the coiled finned tube type heat exchanger is shown in Figure 7. Helium gas is used as a working medium. Hot high pressure gas flows through finned tube and cold gas flows through shell in cross-counter flow arrangement. Gas flow can be measured by mass flow meter calibrated for helium gas, connected directly with high pressure stream. PT 100 sensors are used to measure temperature at various points which are calibrated at liquid nitrogen temperature with uncertainty  $\pm 1\%$ . Temperature at entry and exit of hot and cold gas are measured. Helium gas is charged at 10 bar with 1.5 g/s mass flow rate. Temperature measurements are recorded until the steady state is reached. This experimental data is used to calculate effectiveness of heat exchanger. In all experimental runs the hot gas inlet temperature is varied between 300 K to 318 K. The lowest cold stream inlet temperature of 88 K is achieved with the liquid nitrogen bath. The drop in temperature of hot gas is around 140 K and the cold gas leaves the heat exchanger at 290 K. The numerical model is then used to predict the end temperatures of the fluid streams for the same experimental conditions. The numerical and experimental values of the hot gas out temperatures slightly differs due to losses.. An experimental value of effectiveness of heat exchanger which is 90% is near to value predicted by the numerical model.

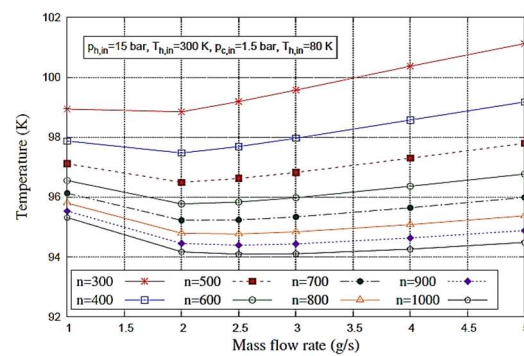


Figure 3: Effect of mass flow rate and fin density on hot side outlet temperature.

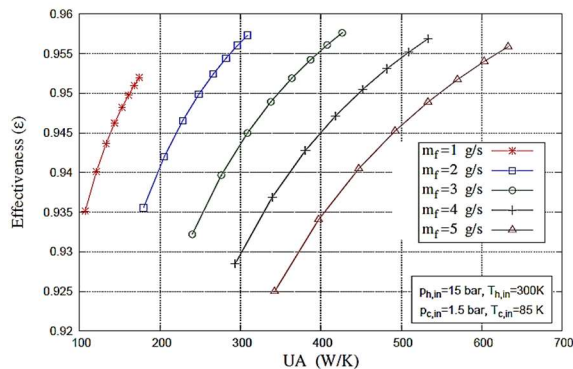


Figure 6: Effect of UA and mass flow rate on effectiveness.

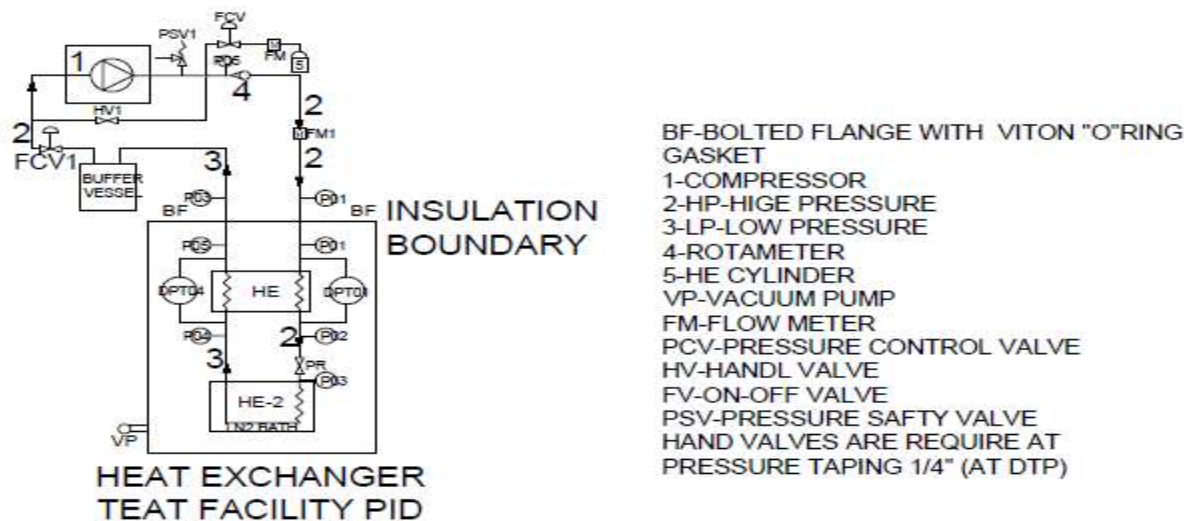


Figure 7 Schematic diagram of experimental set-up

## 6. Conclusion

In this work a transient numerical model has been employed to optimize the performance of a coiled finned tube heat exchanger for a helium liquefier. The numerically predicted values of outlet temperature of gas in finned tube are in good agreement with experimental results published in literature. Parametric study of the heat exchanger shows that there is an optimum mass flow rate, between 2g/s to 2.5g/s, for which effectiveness reaches a maximum value of 0.95. Minimum requirement of UA value to achieve the required effectiveness can be predicted with different combinations of fins density, tube diameter, axial length, coil diameter and mass flow. Finally, for a fabricated prototype of the coiled heat exchanger, experimental value of end temperatures and effectiveness which is 90-92 % also agree well with the numerical predictions

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## References

- [1] Bhusan J Helium Purification by Gas Adsorption method. Dissertation, NIT Raurkela.
- [2] Barron R F 1985 *Cryogenics Systems*. 2nd ed. Oxford: Oxford University Press
- [3] Atrey M D 1998 *Cryogenics* **38**: 1199-1206
- [4] Collins S C (1947) A helium cryostat. *Rev Sci Instr* **18**:157-67
- [5] Xue H, Ng K C and Wang J B 2001 *Appl Therm Eng* **21**:1829-44
- [6] Ng K C, Xue H and Wang J B 2002 *Int J Heat Mass Transfer* **45**:609-18
- [7] Chua H T, Wang X and Teo HY 2006 *Int J Heat Mass Transfer* **49**:582-93
- [8] Hong YJ, Park S J and Choi YD 2009 *Cryocoolers* **15**:379-86.
- [9] Ardhapurkar P M and Atrey M D 2014 *Cryogenics* **63**:94-101
- [10] Damle R M and Atrey MD 2015 *Cryogenics* **65**:49-58.
- [11] Jayakumar J S, Mahajani S M, Mandal, J C, Vijayan P K and Bhoi R 2008 *Chemical Engg Research and Design* **86**(3): 221-232.
- [12] Gupta P K, Kush P K and Tiwari A 2009 *Int. J. of Refrigeration* **32**:960-972
- [13] Gupta PK, Kush PK, and Tiwari A 2010 *Cryogenics* **50**:257-265
- [14] Gupta PK, Kush PK, Tiwari A 2007 *Cryogenics* **47**:322-332.
- [15] Haskins D A and El-Genk M S 2016 *Nuclear Engineering and Design* **305**:531-546.
- [16] Timmerhaus K D and Flynn T M 1989 *Cryogenic Process engineering*. Plenum Press USA
- [17] aspenONE 7.1 (2009) Aspen Technology Inc, Burlington, MA 01803 USA.