Numerical Modeling of a 2 K J-T Heat Exchanger Used in Fermilab Vertical Test Stand VTS-1

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Abstract

Fermilab Vertical Test Stand-1 (VTS-1) is in operation since 2007 for testing the superconducting RF cavities at 2 K. This test stand has single layer coiled finned tubes heat exchanger before J-T valve. A finite difference based thermal model has been developed in Engineering Equation Solver (EES) to study its thermal performance during filling and refilling to maintain the constant liquid level of test stand. The model is also useful to predict its performance under other various operating conditions and will be useful to design the similar kind of heat exchanger for future needs. Present paper discusses the different operational modes of this heat exchanger and its thermal characteristics under these operational modes. Results of this model have also been compared with the experimental data gathered from the VTS-1 heat exchanger and they are in good agreement with the present model.

Keywords: 2 Kelvin J-T Heat Exchanger, Vertical Test Stand Operation, 2 Kelvin Refrigeration Systems

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1. Introduction

Vertical Test Stand-1 (VTS-1) [1] is used to test the superconducting RF cavities at 2 K liquid helium bath. Vertical Test Stand-1 (VTS-1) of Fermilab uses a single layer coiled finned tubing heat exchanger as a J-T heat exchanger. Other Fermilab test stand Vertical Magnet Test Facility (VMTF) [2] and DESY Tesla Test Facility (TTF) Vertical Cryostat [3] for SRF cavity testing have also used similar kind of single layer J-T heat exchanger.

A J-T heat exchanger usually used in 2 K refrigerator system to cool the incoming liquid helium to near 2.2 K before entering to J-T valve. The incoming liquid helium is cooled by exchanging the heat from the returning low pressure helium vapor and then expanded to cryostat pressure through J-T valve. This J-T heat exchanger is used in order to minimize the flashing losses and located prior to the J-T expansion valve as shown in Figure 1.

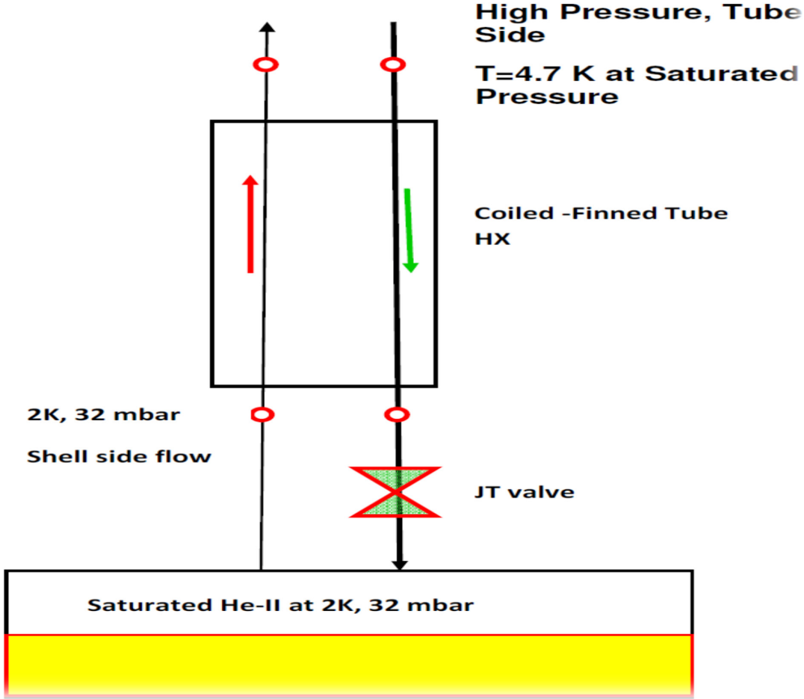

Figure 1. J-T heat exchanger flow scheme

In VTS-1, superconducting R.F. cavities are immersed in 2 Kelvin liquid helium in a vessel as shown in Figure 2.

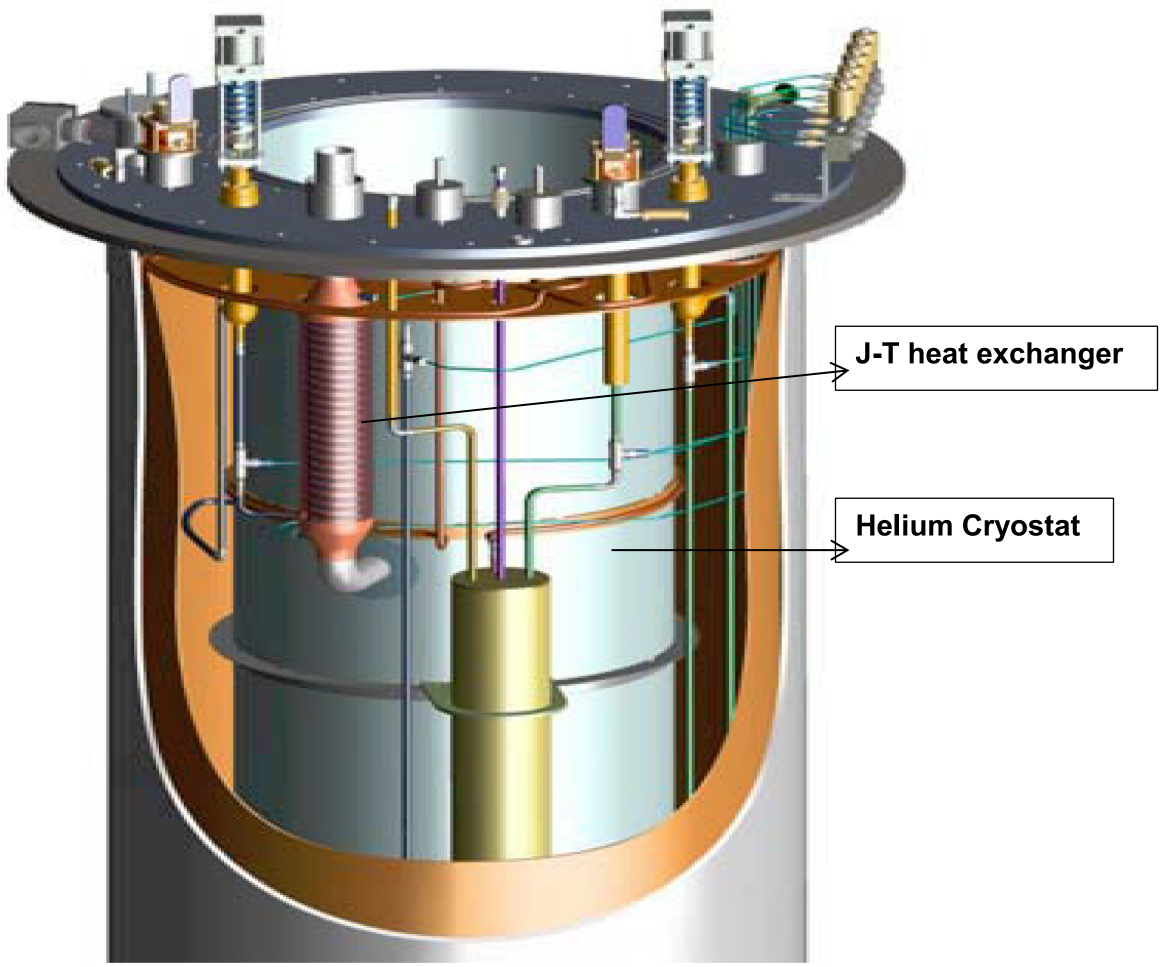

Figure 2. Vertical Test Stand (VTS-1) solid model showing J-T Heat Exchanger and Helium Cryostat

There may be different modes of filling of cryostat, and liquid flashing losses are dependent on these modes.

- A serial fill and pump-down achieves a low liquid level at 2 K. Approximately 50% of the accumulated 4.5 K liquid vaporizes during the pump-down to 2 K. In this mode of filling, the 2 K liquid is achieved at the cost of high flashing losses and cryostat can be filled with 2 K liquid up to half level of cryostat only. A J-T heat exchanger provides no benefit in this mode.

- A concurrent fill and pump-down achieves a very high liquid level at 2 K. Pump-down to 2 K is started when the cryostat is only filled partially with liquid helium, and the filling continues throughout the pump-down in order to achieve a high liquid level at 2 K. The J-T heat exchanger is used during this mode of filling to cool the liquid helium supplied to the cryostat thereby reducing the required 4.5 K liquid helium transfer and decreasing the time required to reach 2 K.
- A continuous fill maintains a steady liquid level during 2 K operation. Upon achieving 2 K, cavity tests are performed and liquid level begins to drop depending on the dissipating power to the liquid bath. Therefore during high continuous power test of superconducting RF cavities, constant liquid level has to be maintained and continuous liquid has to be supplied using J-T heat exchanger.

During a concurrent fill and pump-down, the J-T heat exchanger is operated in an unbalanced mass flow rate condition. Flashing losses are reduced while the tube-side flow is greater than the shell-side flow. During a continuous fill, there is equal flow rate in tube side and shell side of heat exchanger. In these different modes of operations, the total heat capacities of the streams are different due to mass unbalancing and strong variation of thermo-physical properties of helium in given operating temperature range. Therefore J-T heat exchanger will behave differently in both of operational modes. In the present study a finite difference based model is developed in Engineering Equation Solver (ESS) to analyze the temperature distribution of heat exchanger and its operational characteristics have been discussed in different operational modes. The results of this model are also compared with experimental results gathered from the J-T heat exchanger of Vertical Test Stand which is already in service.

2. Description of Vertical Test Stand J-T Heat Exchanger

Vertical Test Stand-1 (VTS) heat exchanger is a single layer coiled finned tube heat exchanger as shown in a cut view of VTS-1 solid model in Figure 2. It consists of single layer of finned copper tube helically wound on a polyethylene mandrel called inner core and then it is jacketed by stainless steel pipe called outer core. The dead space between two consecutive coils has to be filled by some cord. The complete picture of this heat exchanger is shown in Figure 3. Other geometric parameters of heat exchanger are given in Table 1.The 4.7 K saturated liquid passes through finned-tubes in spiral form from top to bottom. The outer shell of this heat exchanger is directly connected to the helium vessel as can be seen in test stand cut view (in Figure 2) and therefore sub-atmospheric cold helium gas passes over the finned-tubes in cross flow pattern and exchange the heat with incoming saturated liquid before exiting to pumping line of test stand.

Figure 3. Picture of J-T Heat Exchanger used in Test Stand

Table 1.Geometrical parameters of J-T heat exchanger

Geometry of helically wound brazed copper finned-tube	Inner tube diameter, di	7.74 mm
	Finned tube diameter, d_f	22.25 mm
	Height of finned tube, h_f	6.35 mm
	No. of fins per inch, n	8
Geometrical parameter of J-T heat exchanger	Total axial length, L	578.0 mm
	Actual length with finned tubes	488.0 mm
	Mean diameter, D_e	86.0 mm

3. J-T Heat Exchanger Modeling

To simulate the heat transfer characteristics of J-T heat exchanger, energy equations for the temperature profiles of hot and cold streams with no heat generation in the fluids and with no external heat-in-leaks have been formulated. These governing partial differential equations coupled with heat transfer process for the counter-flow heat exchangers can be written in the following form:

$$Hot\ fluid: \quad \dot{m}_h c_{ph} \frac{dT_h}{dx_h} + U\frac{A_h}{L}(T_h - T_c) = 0 \ldots\ldots (1)$$

$$Cold\ fluid: \quad \dot{m}_c c_{pc} \frac{dT_c}{dx_c} + U\frac{A_c}{L}\left(T_c - T_h\right) = 0 \ldots\ldots (2)$$

Where $\dot{m}_h$ and $\dot{m}_c$ are the mass flow rate of hot and cold streams. c_{ph} and c_{pc} are the specific heat of the respective fluids. U is the overall heat transfer coefficient between the two fluid streams. A_h and A_c are the heat transfer area for tube side and shell side. L is the axial length of heat exchanger. The T_h and T_c are the tube side (hot fluid) and shell side (cold fluid) temperatures.The x_h and x_c are the length of fluid paths for the respective fluid and both are different in the present heat exchanger configuration. In above equations, these fluid paths have to be normalized by considering the following relations between these two paths:

$$dx_h = l_h \frac{dx_c}{L} \quad \text{......(3)}$$

Where l_h is the total length of finned tubes wound on the inner core.

The perimeter of inner tube surface (surface area per unit axial length, $\frac{A_h}{L}$), s_i

$$s_i = \pi^2 \frac{D_e}{d_f} d_i \quad \text{......(4)}$$

The perimeter of outer finned surface (surface area per unit axial length, $\frac{A_c}{L}$), s_o

$$s_o = \pi^2 \left[\frac{n}{2}\left(d_f^{\,2} - d_o^{\,2}\right) + d_o(1 - tn) + d_f \cdot t \cdot n \right] \frac{D_e}{d_f} \quad \text{......(5)}$$

Free flow area offered by the fins cross section, A_{fc}

$$A_{fc} = \pi D_e \left[(d_f - d_o)(1 - nt) \right] \quad \text{......(6)}$$

The surface area offered by the outer finned surface in one coil; A_s

$$A_s = \pi^2 \left[\frac{n}{2}\left(d_f^{\,2} - d_o^{\,2}\right) + d_o(1 - tn) + d_f \cdot t \cdot n \right] D_e \quad \text{......(7)}$$

The shell side Reynolds Re_s is based on the total cross-section area available for shell side flow and can be calculated as follows:

$$\mathrm{Re}_s = \frac{\dot{m}_c D_h}{A_{fc}\mu} \quad \text{......(8)}$$

The characteristic dimension for the Reynolds number in Eq.(8) is the equivalent diameter, or the hydraulic diameter D_h

$$D_h = \frac{4A_{fc}}{A_s / L} \quad \text{....(9)}$$

$$G = \frac{\dot{m}_c}{A_{fc}} \quad \text{....(10)}$$

Where G is the mass flow rate per unit free-flow area and will be used for calculating the heat transfer coefficients.

In above geometrical formulae, $d_{i,} d_o$ and d_f are the inner, outer and finned tube diameter respectively. D_e is the mean diameter of heat exchanger. *t* and *n* are the fins thickness and number of fins per unit length.

The all formulae mentioned above for calculating the geometric parameters for this kind of heat exchanger have been taken from the design procedures developed at RRCAT [4,5].The overall heat transfer coefficient, U, has been calculated by the following formula using the fin efficiency as unity:

$$U = \frac{h_o h_i s_i}{h_o s_o + h_i s_i} \quad \text{....(11)}$$

Here h_i and h_o are the inner heat transfer coefficient and outer heat transfer coefficients and have been calculated using the following formulae [5]

$$h_i = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{1/3}\left(\frac{k}{d_i}\right) \quad \text{....(12)}$$

$$h_o = 0.19\left(\frac{k}{d_o}\right)\mathrm{Re}_s^{\;0.703}\,\mathrm{Pr}^{1/3} \quad \text{....(13)}$$

where Pr and Re is the Prandtl number and Reynolds number respectively. *k* is the thermal conductivity of fluid.

To determine the temperature profiles across the heat exchanger, above described energy equations (Eqs.1 and 2) of heat exchanger have divided in to *n* elements (typically more than 200 nodes) using the finite difference method. These above equations are converted into linear algebraic equations. The Engineering Equation Solver (EES) has been used to obtain the solution. The program developed in ESS takes care of property variations along the length of heat exchanger.

4. Effectiveness of JT Heat Exchanger

The effectiveness of any heat exchanger is defined as the ratio of actual heat transfer to the maximum possible heat transfer. In conventional heat exchangers used at near room temperature applications, the properties of fluids do not vary much with the temperatures and pressures. Therefore the effectiveness of any heat exchanger operating near constant property zone can be expressed in terms of end temperatures of heat exchanger. However, the effectiveness of heat exchangers operating in the variable property zone should be expressed in terms of the relevant enthalpy differences.

Maximum possible heat transfer occurs when the temperatures of two fluids coincide at the end of heat exchangers. However, due to the variations in properties of helium fluid, these two temperature profiles may be coincided at any location within the heat exchanger. This location of minimum temperature difference within heat exchanger between stream to stream temperature differences depends on the operating temperature range and pressures of individual stream. For the convenience, effectiveness in these heat exchangers is also defined on the end conditions of heat exchangers and no matter where the minimum temperature difference occurs inside the heat exchangers.

Effectiveness of these heat exchangers operating in variable property zone can be defined on the basis of minimum capacity fluid. If hot fluid is minimum capacity fluid, the effectiveness of these heat exchangers is defined as follows:

$$\varepsilon_h = \frac{H_{h,in} - H_{h,out}}{H_{h,in} - H^*_{in}} \tag{14}$$

Where H^*_{in} is the hot fluid enthalpy at inlet temperature of cold fluid. Similarly if cold fluid is minimum capacity fluid, the effectiveness of heat exchangers is defined as follows:

$$\varepsilon_c = \frac{H_{c,out} - H_{c,in}}{H^*_{out} - H_{c,in}} \tag{15}$$

Where H^*_{out} is the cold fluid enthalpy at the inlet temperature of hot fluid. H_h and H_c are the hot fluid and cold fluid enthalpies at respective points.

5. Results and Discussions

5.1. Experimental validation of model

The present model has been validated by comparing the theoretical results with the experimental data gathered from Vertical Test Stand J-T heat exchanger. The experimental data gathered during VTS refill of cavity test Dewar have been used for this comparison [6]. Four data sets were gathered during VTS-1 refill. During obtaining of these data sets liquid were filled in test Dewar at the rate of 0.397, 0.293, 0.217 inch/min. In all these tests inlet liquid helium temperatures were 4.828 and 4.875 K. All four end temperatures were measured during these tests. The performance of J-T heat exchanger during a 2 K refill of VTS-1 is documented in reference 6.

During refilling of Dewar, there is mass imbalance in the J-T heat exchanger and there is always more liquid supplied mass flow rate. The accurate calculation of these different mass flow rates of both streams is an important parameter to determine the performance of J-T heat exchanger. For calculation of mass flow rate for tube side , $\dot{m}_h$, and shell side, $\dot{m}_c$, the flash rate have been calculated using the measured four end temperatures of heat exchanger and then using the liquid fill rate in test Dewar both mass flow rates have been calculated. The average mass flow rate for tube side flow rate is 13.29 g/s and for shell side flow rate is 4.75 g/s during 0.397 inch/min filling of VTS-1 cryostat.

Figure 4 shows the comparison between effectiveness calculated from measured data and predicted effectiveness for the existing heat exchanger in VTS-1. Figure shows that predicted effectiveness is within 5% of measured values. This figure also compares the predicted exit tube side temperature with the measured values and shows very good agreement with the measured values.

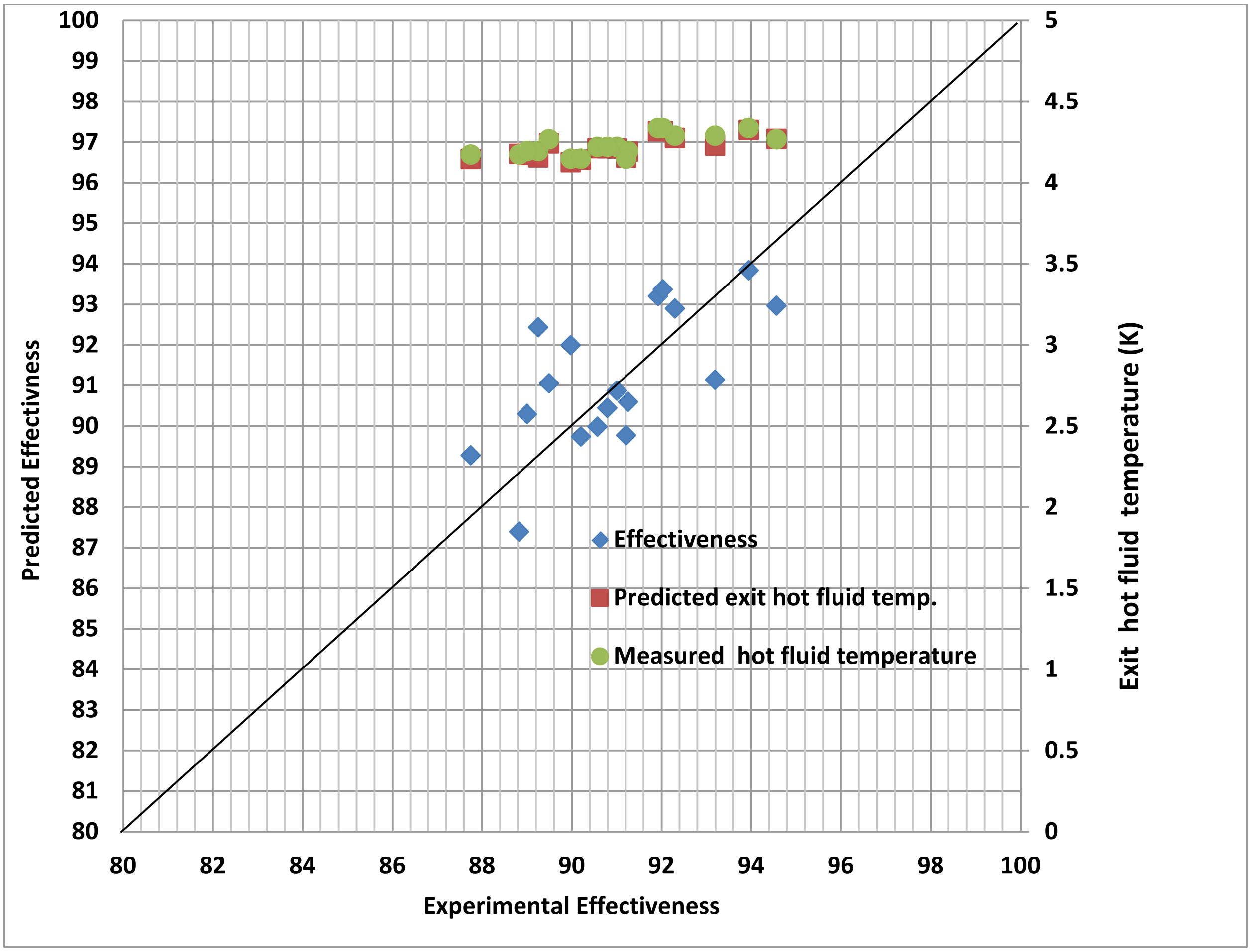

Figure 4. Comparison between experimental results and calculated results

5.2. Heat exchanger temperature profile studies under various operational modes

During filling of 2 K bath, the larger fraction of total mass flow rate will flow through the tubes and smaller fraction of total flow rate will flow through the shell side of heat exchanger. Here we can say that there is process driven mass imbalancing in the heat exchanger. Now we have to examine the impact of this mass imbalacing on the performance of this heat exchanger. Figure 5 shows that specific heat of shell side stream is lower than the tube side stream in most part of heat exchanger and crossed over each other at the cold end in prescribed temperature range of J-T heat exchanger.

But due to this mass flow rate unbalancing in the heat exchanger (mass flow rate for tube side flow rate is 13.29 g/s and for shell side flow rate is 4.75 g/s), the heat capacity ($\dot{m}c_p$) of shell side becomes smaller than the tube side flow through out of heat exchanger length as can be seen in Figure 6. Therefore tube side stream will experience less temperature change as compared to the shell side flow.

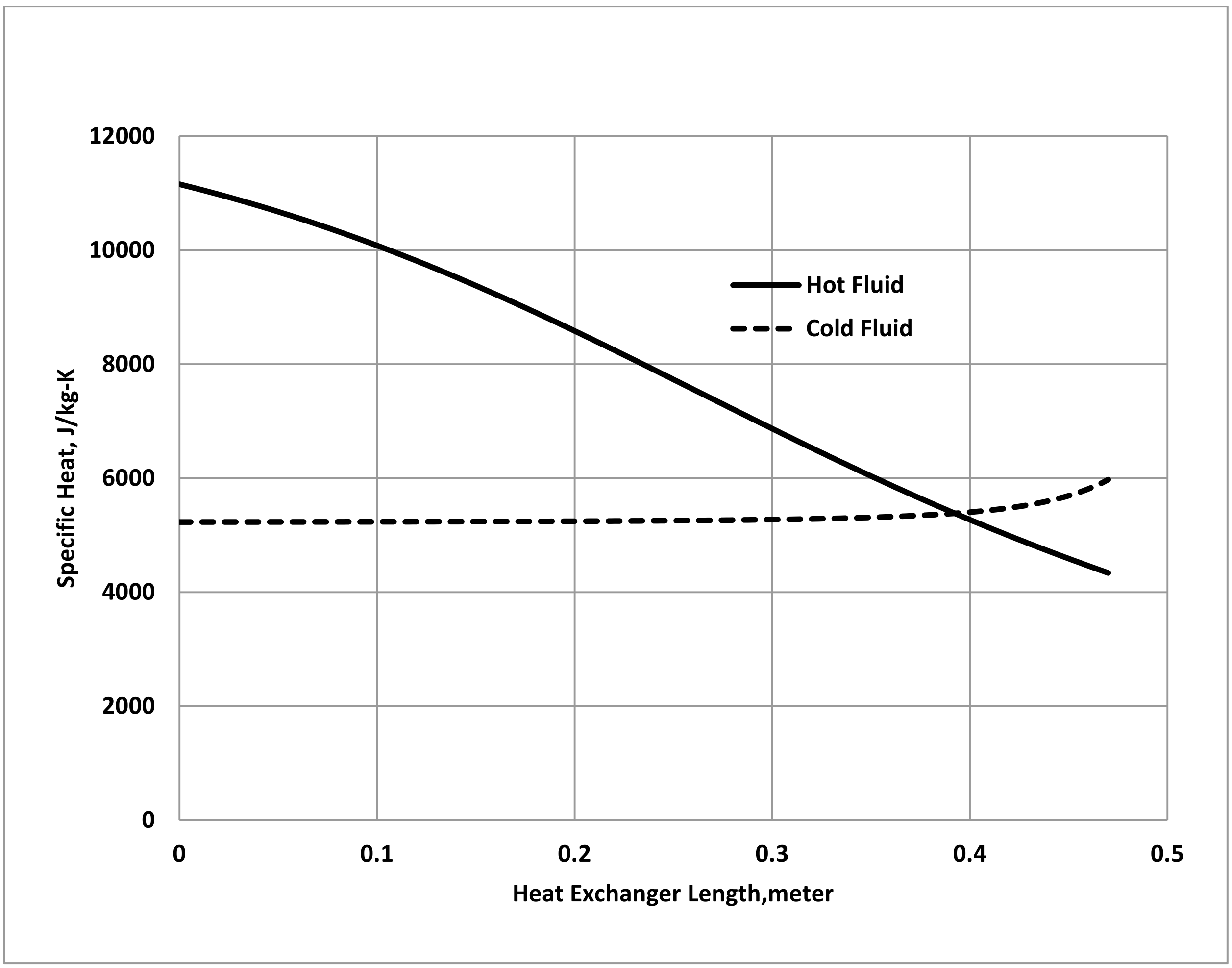

Figure 5. Variations in specific heat along the heat exchanger

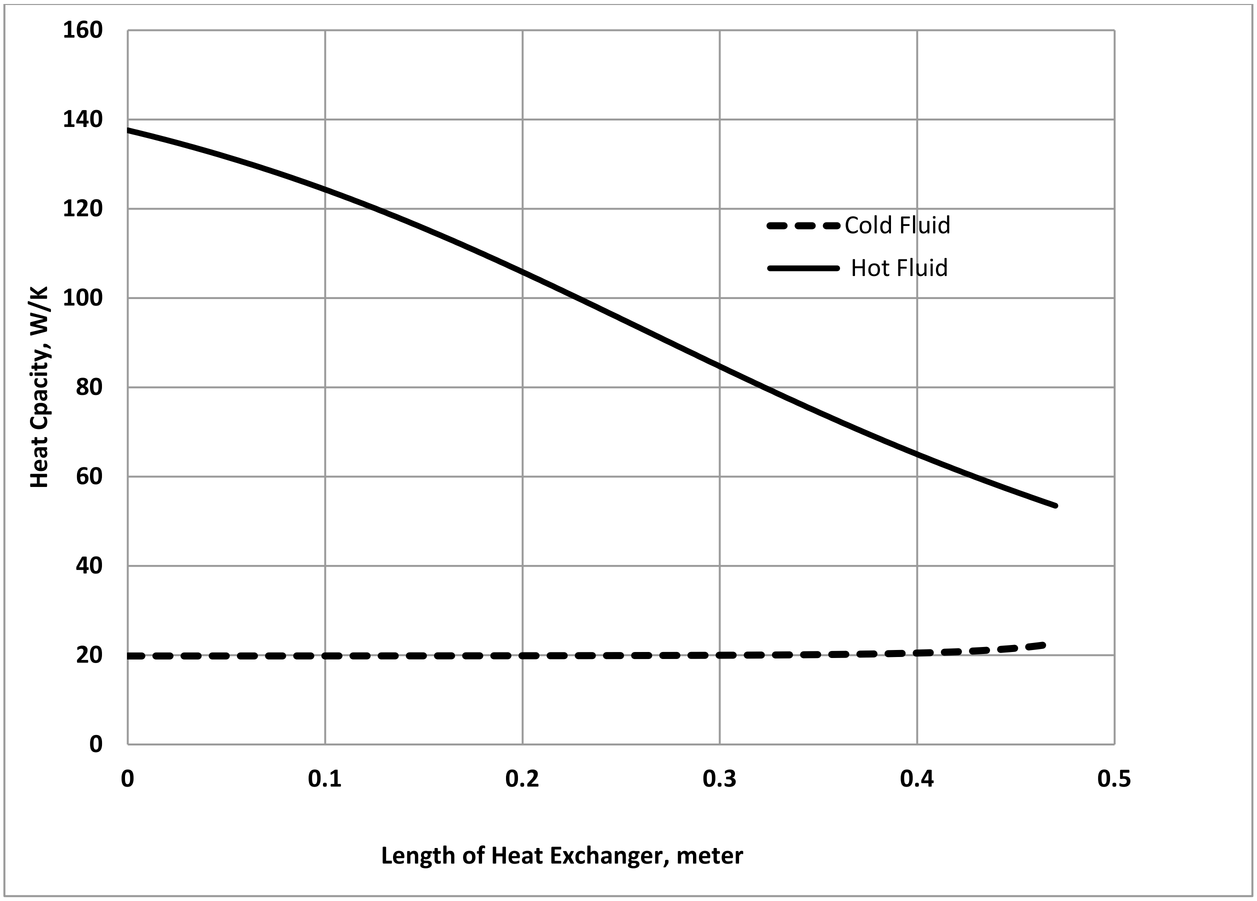

Figure 6. Variation in total heat capacity along the length of heat exchanger

Figure 7 shows the calculated temperature profiles of heat exchanger for the tube side mass flow rate 12.33 g/s and shell side mass flow rate 3.79 g/s conditions. This figure clearly shows that hot fluid experience less temperature change throughout the heat exchanger length. The irreversibility generated by the flow unbalancing and variable specific heats will inhibit the heat exchanger to achieve the lowest temperature no matter how big the heat exchanger as can be seen in Figure 8. Figure 8 shows the temperature profile of heat exchanger which is almost double in length with the same other geometric dimensions.

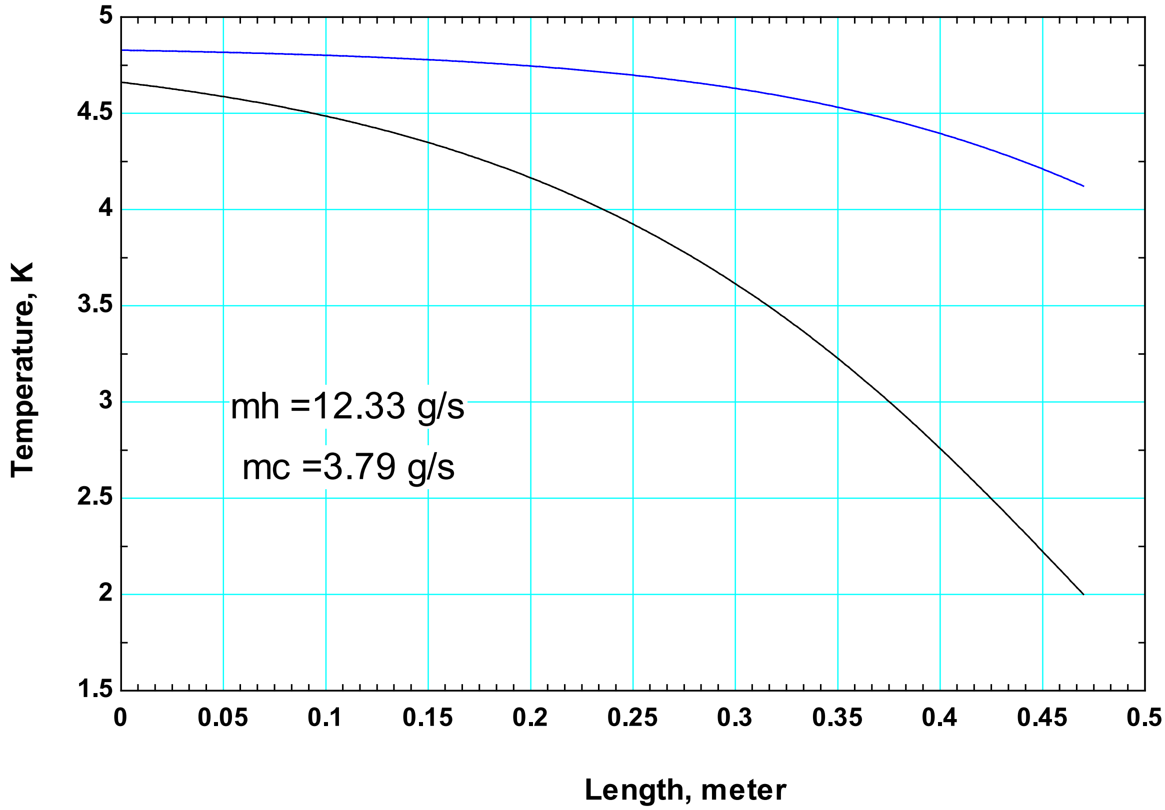

Figure 7. Temperature profiles of heat exchanger during filling operation of VTS

This figure shows that there is temperature pinch at the hot end. But hot fluid temperature is lowered only to 4.064 K from 4.121 K however the length of heat exchanger becomes almost doubled. Therefore increasing the length of heat exchanger is not providing any extra advantage to get the lower tube side fluid temperature because of this flow rate unbalancing in the heat exchanger. Here it could be concluded that even bigger heat exchanger (larger surface area) would not be beneficial to increase the 2K liquid fraction in this mode of operation.

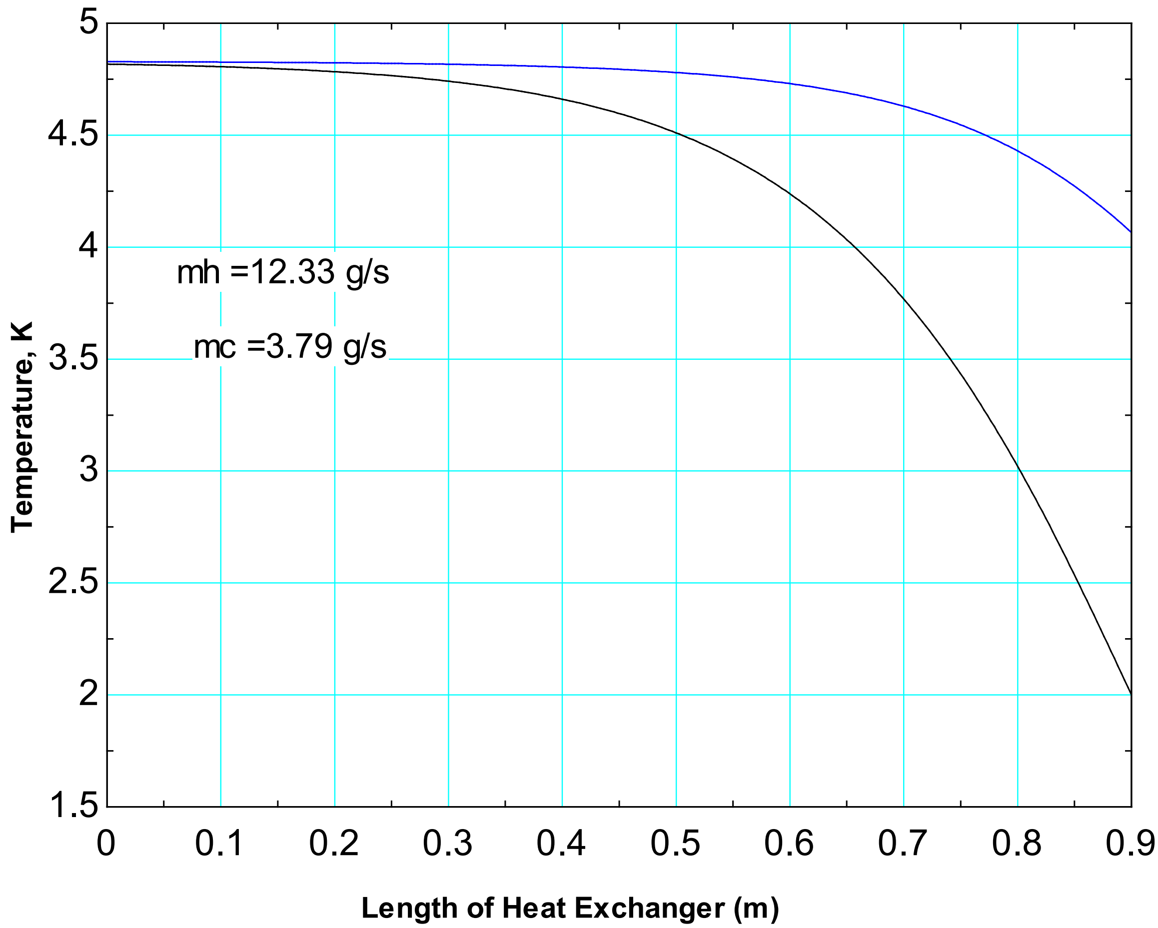

Figure 8. Heat exchanger length effect on temperature profiles during filling mode

On the contrary while maintaining the constant liquid level, 2 K liquid has to be filled constantly in test Dewar during dissipation of power in test Dewar. The equal mass flow rate will flow through the tube and shell side of heat exchanger to maintain the constant liquid level in the 2 K bath. Therefore in this mode of operation, there would not be any mass flow imbalance and only variations in specific heat along the heat exchanger as shown in Figure 5 will govern the performance of heat exchanger. Figure 9 shows the temperature profiles of same heat exchanger during this mode of operation. Here it can be noted that the same heat exchanger is capable to bring down the liquid helium temperature to 2.48 K against the 4.121 K (Figure 7) calculated while operating in filling mode.

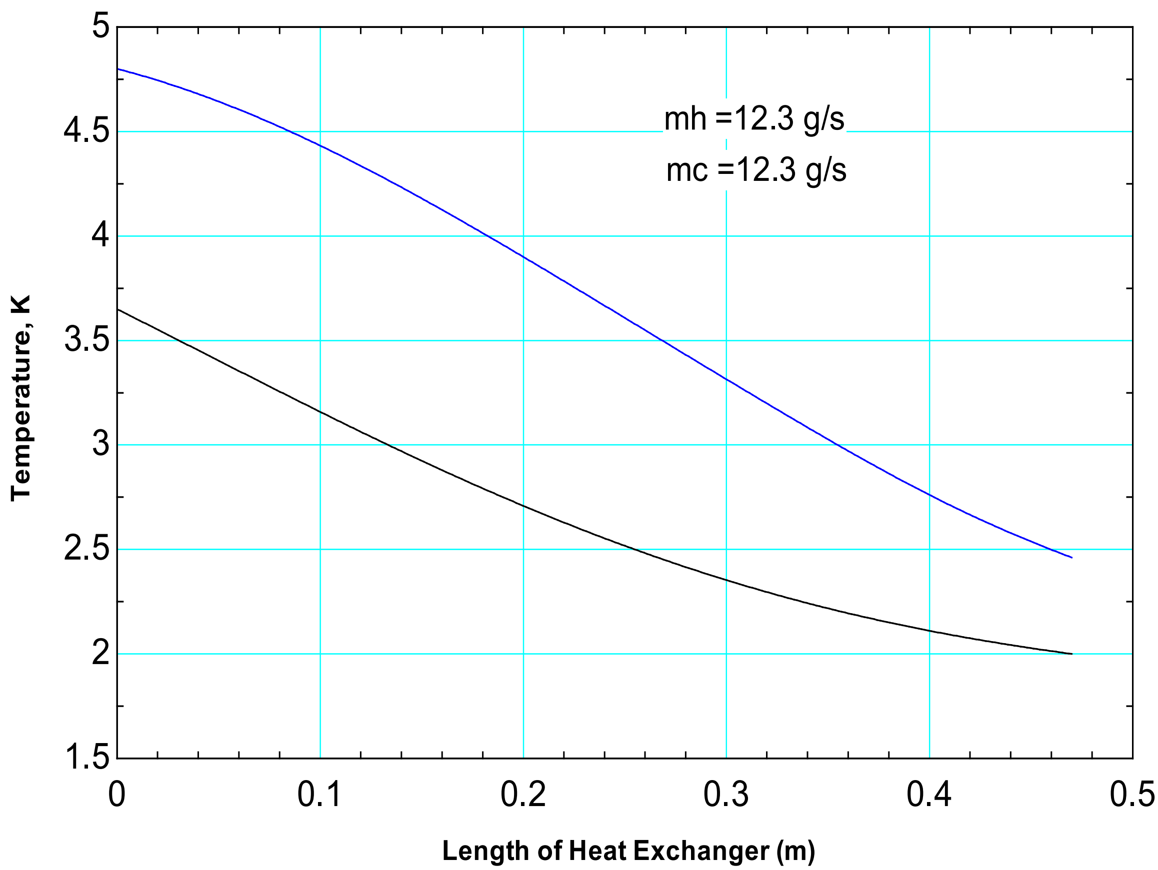

Figure 9. Temperature profiles of heat exchanger during refilling to maintain the constant liquid level in VTS

Figure 10 shows the temperature profiles for the bigger heat exchanger (0.7 m). It could be seen here that liquid helium temperature is dropped to 2.189 K as the length of heat exchanger increased. Hence in this mode of operation; liquid helium temperature would be closer to the bath temperature as the length of heat exchanger increases and pressure drop would be the only limiting factor to optimize the length of heat exchanger. During this mode of operation, use of heat exchanger will significantly increase the 2 K liquid yield as the temperature of liquid helium would be much lower before J-T valve.

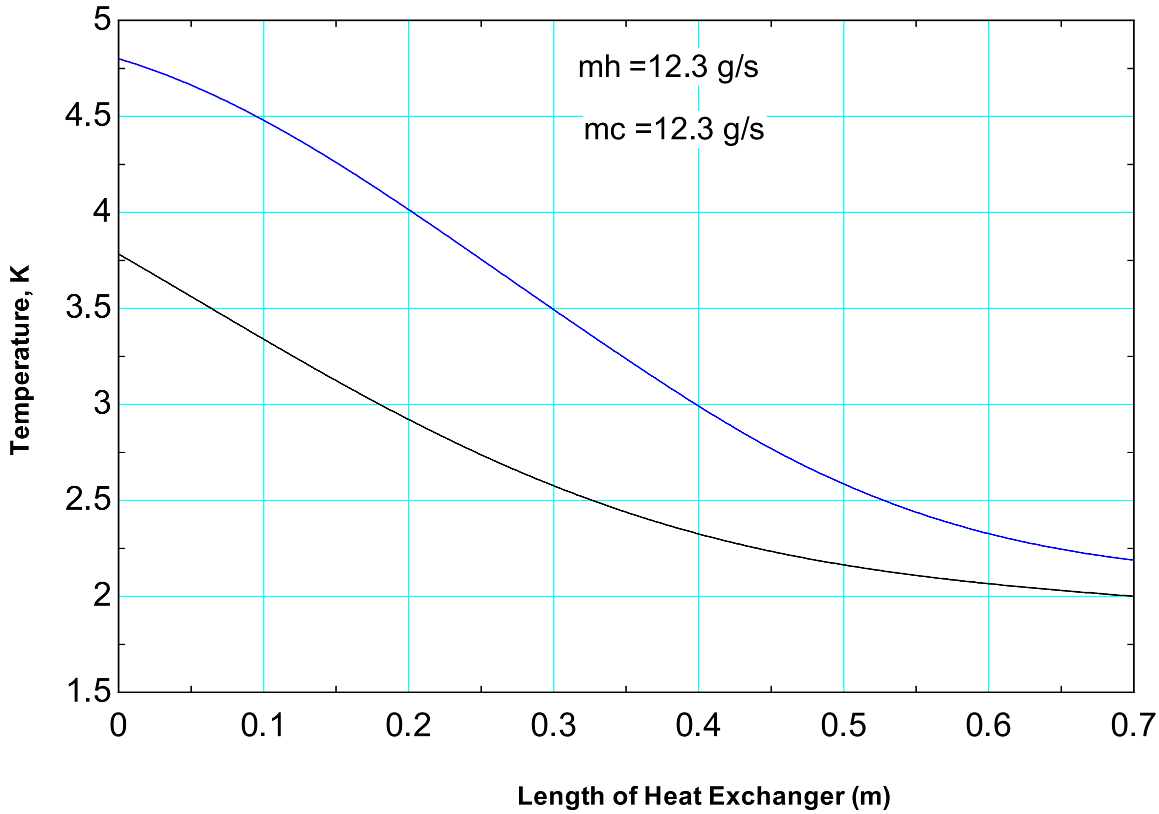

Figure 10. Temperature profiles of larger heat exchanger during refilling to maintain the constant liquid level in VTS

5.3. Heat exchanger sizing effect on flash reduction

This section presents the heat exchanger sizing effect on vapor flash reduction during filling of test Dewar. It is assumed that test Dewar is filling at the rate of 0.397 inch/minute and liquid helium supplied temperature is 4.8 K. To calculate the relative reduction in flashed vapor flow rate, vapor fraction is calculated with and without heat exchanger in the system. Therefore, % relative flash reduction can be expressed by the following formula:

$$\frac{\Delta m_v}{m_h} = (X_{noHX} - X_{HX}) \times 100 \quad (16)$$

where X_{noHX} is the quality entering the VTS-1 cryostat without a J-T heat exchanger and X_{HX} is the quality entering the VTS-1 cryostat with a J-T heat exchanger. Δm_v is the change in vapor flash.

Figure 11 shows the % relative flash reduction vs. heat exchanger length. It could be noted here that there is 21.5 % reduction in flashed vapor as compared to if there is no heat exchanger in the system. It can also be seen from figure if length of heat exchanger is increased after 0.7 meter; there is no gain in flash reduction. This is because of unbalanced operation of heat exchanger as described in previous section. This figure also shows that effectiveness of heat exchanger increases with the length of heat exchanger. This increases because the hot end of heat exchanger gets pinched as the length of heat exchanger increases due to its unbalanced operation. Here it can be stated that high effectiveness of this heat exchanger is not a true performance parameter for this mode of operation.

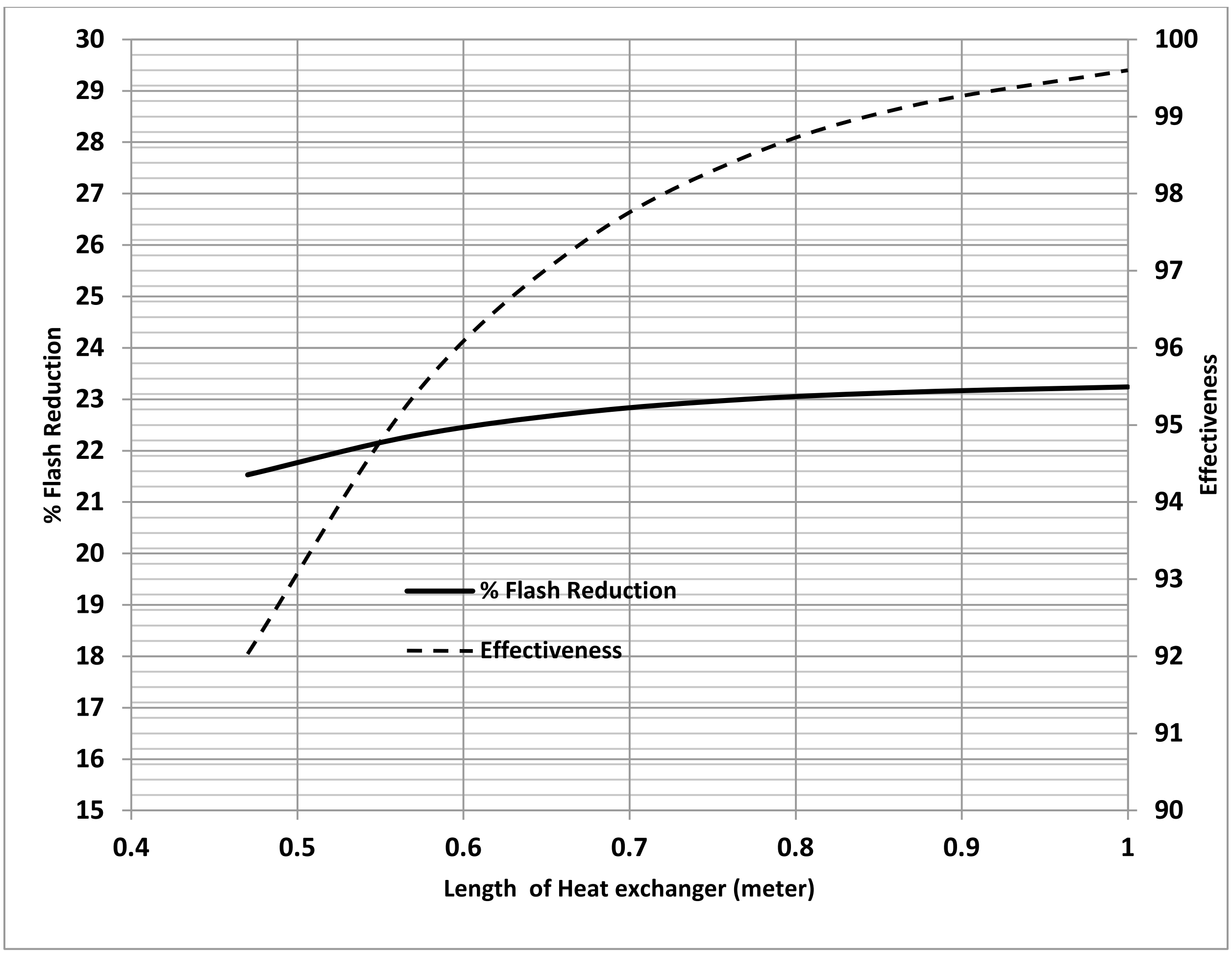

Figure 11. % relative flash reduction vs. heat exchanger length during filling mode

5.4. Shell side inlet temperature effect

Figure 12 shows the effect of inlet shell side temperature of heat exchanger to relative vapor flash reduction for the filling rate of 0.397 inch/minute. Figure shows that if inlet temperature is 2 K, there is 21.5 % reduction in vapor flashing and if inlet temperature rises to 2.8 K there is only 15.4 % reduction in the vapor flashing. This happens due to rise in temperature before JT valve from 3.978 to 4.3 K.

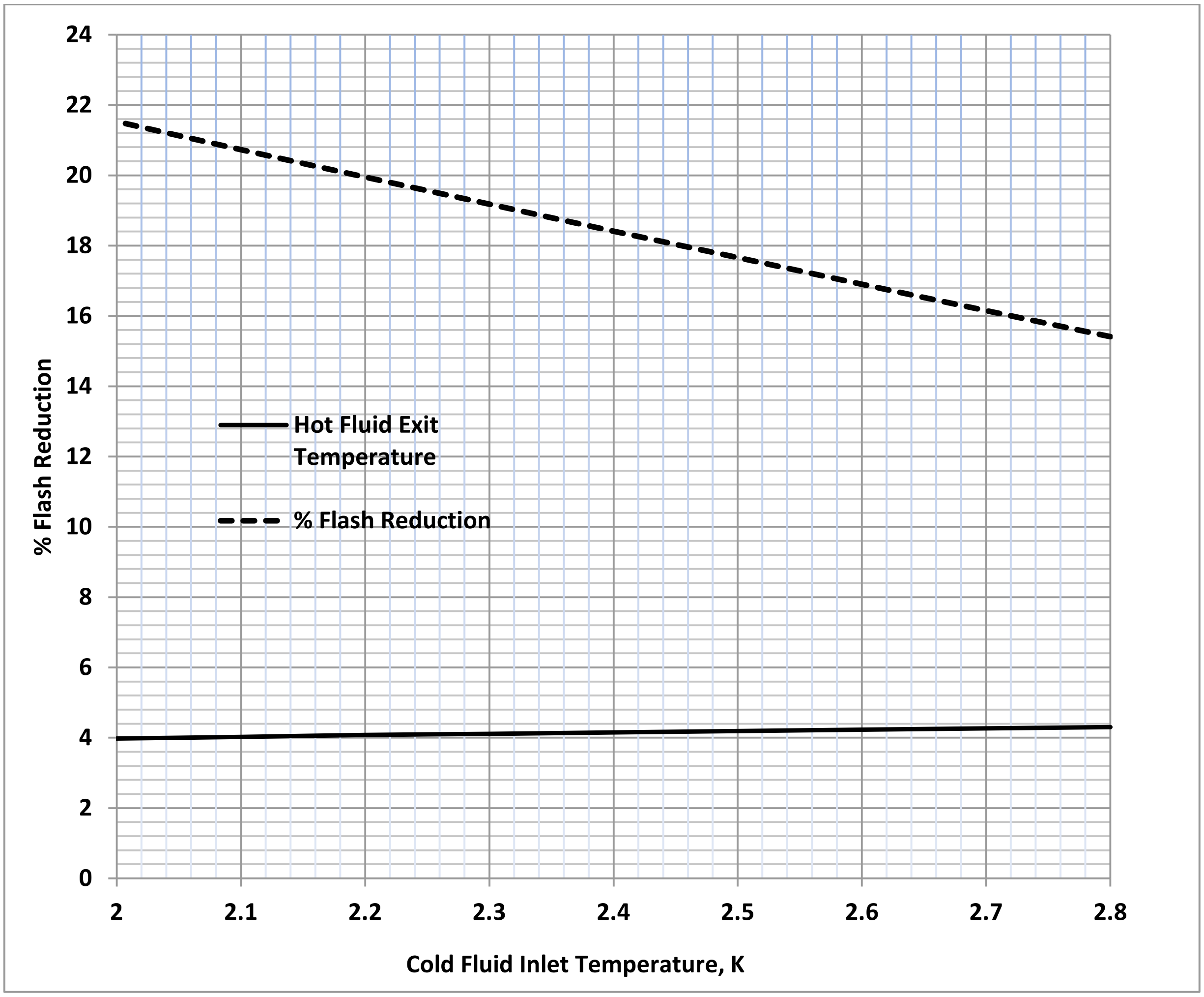

Figure 12. Effect of inlet shell side temperature of heat exchanger to relative vapor flash reduction

5.5. Effect of liquid supplied temperature

Figure 13 shows the effect of supplied liquid helium temperature on vapor flash reduction during filling mode of VTS. Figure shows that if supplied liquid temperature is

4.3 K, there is 32.41% reduction in vapor flashing as compared to if there is no heat exchanger in the system. As this supplied liquid saturated temperature increases, % flash reduction decreases due to rise in temperature before J-T valve. There is only 21.5% flash reduction if supplied temperature of liquid is 4.82K and temperature of supplied helium will drop to only 3.978 K in J-T heat exchanger as can be seen in Figure 13.

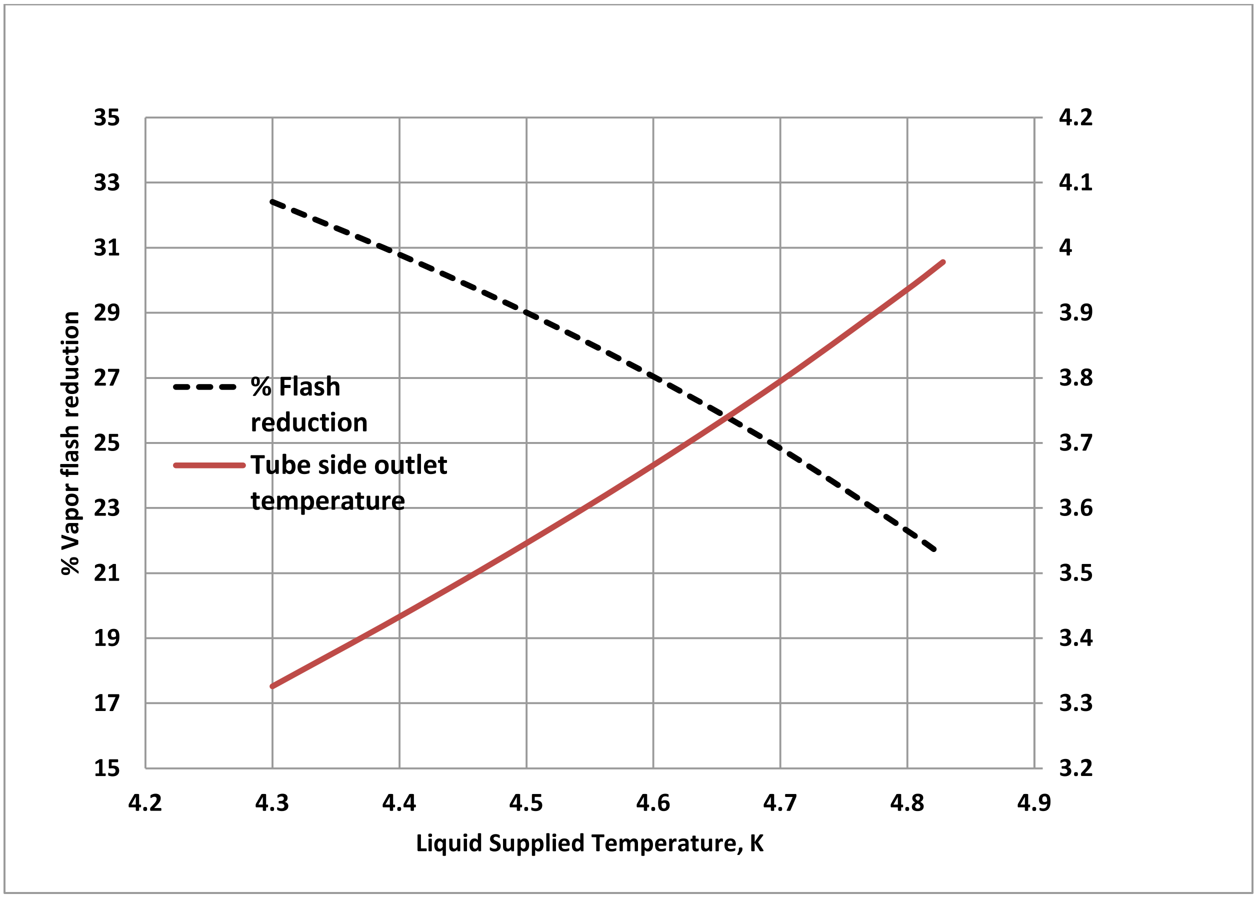

Figure 13. Effect of supplied liquid helium temperature on vapor flash reduction

6.0. Conclusions

Finite difference based J-T heat exchanger model has been developed. Results obtained from the model are in good agreement with the experimental results. Present study shows that J-T heat exchanger performance characteristics are different in different modes of operations and plays an important role in vapor flash reduction. Study also brings interesting facts that sizing of heat exchanger can play an important role while operating in maintaining the constant liquid level in test stand and can play the major role in vapor flash reduction during this mode of operation. However, length of heat exchanger does not play much role while operating in filling mode of test stand due to unbalanced operation of this heat exchanger.

This study also quantifies the effect of liquid supplied temperature and shell side inlet temperature of heat exchanger on the % vapor flash reduction. Present developed model will serve as a useful tool to design such kind of heat exchangers for future needs.

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