Experimental investigation on the effect of geometry on cryogenic transfer line chilldown

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Abstract. In order to analyze the effect of geometry on cryogenic chilldown, experiments were performed on straight, one twisted and twin twisted copper tubes using liquid nitrogen. Data presented in this paper is for the experiments conducted under terrestrial gravity conditions at supply pressures of 1.2 bar and 1.3 bar. Test sections considered were copper tubes (7.9375 mm OD, 0.8128 thick) of straight, one twisted and twin twisted geometry traversing a distance of 220mm under Polyurethane foam insulation. Pressure measurements at test section inlet and outlet was monitored using transducers. Temperature measurement at four equidistant sections along the length were tracked in real-time using a data acquisition system. Temperature and pressure profiles were obtained. The results of the experiment indicated that, for a given mass flux, the transition during chilldown occurs earlier in one twisted transfer lines compared to straight and twin twisted tubes. This indicates the earlier occurrence of film boiling for the one twisted channels. Thereby, changing the transfer line geometry from straight tubes to one twisted sections, the chilldown time can be reduced to a considerable extent further resulting in reduction of cryogen consumption during the initial stages of transfer process.

1. Introduction
Cryogenics, a scientific branch concerning with liquefaction of gases has outgrown conventional gas supply industry and has been instrumental to the extent that today we can find its applications over a wider horizon such as cryosprays, food preservation, space simulation chamber, cryonics, chemical and glass industries, superconducting magnets etc. Optimized transportation of cryogen from its storage to intended use point with minimum losses, still remain an area of concern, owing to the energy intensive nature of cryogen production. When the cryogen initially navigate through the transfer line which is in thermal stability with the ambient, it gets vaporized until the transfer line is brought to a steady state temperature near the saturation temperature of the fluid so that consequent flow would be in liquefied stage. This initial phase characterized as chilldown of cryogenic transfer line is accompanied with ravenous evaporation, pressure surges and temperature fluctuations. From chilldown studies it is inferred that an amount of 90% of the cryogen initially supplied to the transfer lines are consumed, [1] or rather wasted in the initial period of chilldown where, film boiling dominates. Any initiative in the way of reducing chilldown time in cryogenic transfer lines can result in energy conservation and thereby reducing the cost of overall performance of any cryogenic system. The first remarkable work in this field dates back to 1960s when Burke et al.,[2] probed pressure chilldown of cryogenic transfer lines using liquid nitrogen as the cryogen and proposed the existence of single phase convective heat transfer.
and film boiling. In 1962, Bronson et al.[3] observed the existence of flow patterns in two phase flow using liquid nitrogen while conducting visual assessment of chilldown. His conclusions validated the fact that flow stratification led to the oscillations in peripheral temperature. Chilldown experiments employing liquid hydrogen through copper tubes were conducted by Chi and Vetere[4] in 1963. They associated transient wall temperature with the visual inspections to identify the regimes in flow transitions. Demonstration of flow regimes grouping into single phase convective boiling, film boiling and nucleate boiling were reported by Chi [5] in 1965. He inferred that the film boiling dominates the total chilldown time. Klimenko [6] investigated the influence of channel orientation and geometry on heat transfer with two-phase forced flow of nitrogen and explained the need of conducting experiments in the same laboratory for identifying the variations associated with factors governing chilldown. Yuan et al.[7][8] studied the effect of mass flow rate and gravity on cryogenic chilldown and associated the former results concerning visual inspections and peripheral temperature fluctuations. He inferred that the driving force behind chilling of the bottom wall of transfer lines was liquid film-wall interaction while, forced convection of superheated vapor caused cool down in upper wall. Through their chilldown experiments at low and high mass flow rates to find its impact on various flow patterns, Hu et al. [9] studied the variation of heat flux and chilldown time through a vertical pipe under upward and downward flow. They found that chill down time was longer for upward flow while downward flow had higher critical heat flux. The existence of an optimum upward line inclination that can minimize chilldown time was reported by Johnson et al.[10] through their work on horizontal and inclined pipes. Inverse heat transfer technique to the transient chilldown period was employed to derive heat flux and local heat transfer coefficient. Darr et al.[11] established that optimal design of cryogen transfer lines can be accomplished by identifying methods for minimizing cryogen consumption. Effects of mass flux, pressure, flow direction, equilibrium quality and inlet subcooling in relation to gravity was investigated, enabling them to arrive at a correlation for the heat transfer coefficient in straight transfer lines. Apart from these parameters, geometry of transfer lines and the accompanying effect of other relevant factors are to be investigated into. Hence, optimization of chilldown parameters has been an area of research interest for decades.

Vashisth et.al.[12] in their review paper on the applications of curved geometries in process industries, demonstrates that helical geometries exhibit similar and sometimes better performance at lower energy consumption and reduced maintenance requirements over conventional configurations. Mohammed et al. [13] identified the earlier occurrence of film boiling for the helical channels when compared to straight tubes. In spite of the aforementioned works, experimental studies focused on the effect of geometry variations on chilldown are minimal. Those available are limited to straight transfer lines. With the added advantage of secondary flows in curved geometry, the authors have investigated the effect of providing slight twist to straight transfer lines and henceforth, investigate the heat transfer characteristics during chilldown of one twisted and twin twisted transfer lines. Moreover, literature lacks information concerning data on direct comparison of heat transfer intensity in different geometries obtained in the same laboratory.

2. Experiment work
A systematic layout of the experimental setup can be shown in figure 1. The entire setup consists of liquid nitrogen storage and supply system, test sections and instrumentation with Data Acquisition system.

Cryogen for this experimental study was Liquid nitrogen. It was selected because of its non-corrosive, relatively inexpensive, chemically inert, easily available, non-flammable characteristics and it does not pose any major hazards. The liquid nitrogen was stored in CRYOCYL 120LP which is a low pressure Dewar vessel with 120 liters of capacity. It expels liquid nitrogen by self-pressurization. LN2 is forced through the test section by using the pressure exerted by the gaseous nitrogen generated. Flow rate of liquid nitrogen was controlled manually by controlling the Dewar pressure by employing a pressure regulator.

Copper tubes (specific heat capacity of 401 J/kgK and thermal conductivity of 0.14 W/mK at 300 K) were selected for the study in regard with the low flow rates from the Dewar. Test section under
consideration were 5/16" Copper tubes (7.9375 mm OD 0.8128 mm Thick) of for straight channel (220 mm) and copper tubes of one twisted (coil length 275 mm) and twin twisted (coil length 385 mm). Details of the test section used are shown in figure.2. Liquid nitrogen was supplied to the test section through 1/2” SS 304 grade pipes and brass fittings. A bypass line was provided before the test section to bypass the initial vapour produced during the initial cool down of the supply system. A transparent line provided at the bypass line ensures a steady state of liquid being supplied and only after that the line to the test section is opened. Heat in leak at the inlet section was minimized by employing a combination of asbestos rope, nitrile rubber and urethane modified poly-isocyanurate foam (thermal conductivity of 0.14 W/mK).

The wall temperatures of each geometry was measured by employing16 T-type thermocouples. Temperature measurements were taken from four equidistant sections till a distance of 220 mm (along the tube) from inlet. Each section was attached with four equidistant thermocouples along its circumference. Two calibrated piezo-electric pressure transducers were used to measure the inlet and outlet pressure of the test section. An insulation was provided with 140 mm OD with urethane modified poly-isocyanurate foam to reduce the heat in leak at the section. Keysight 34972A Data Acquisition / Data Logger Switch Unit was adopted for logging data. Mass flux was measured by a single phase mass flow measuring Rota meter at the exit line kept after a hot water bath maintained at 100°C.

Figure 1. Layout of Experimental setup for chilldown studies.

Figure 2. Test sections (a) Twin Twisted (b) One Twisted (c) Straight.
3. Experiment Procedure

Entire turning along with the test section was purged with gaseous nitrogen to avoid moisture condensation inside the experimentation setup. The inlet lines before the test section were allowed to chilldown before commencement of the experiment and the vapor generated were vented to the atmosphere via bypass line. The data acquisition program was initiated to record data as soon as activated. The required supply pressure was obtained by regulating the pressure regulator connected to the Dewar vessel. Then the flow towards bypass line is closed and the flow is introduced to the test section, by that time the DAQ should also started for recording various thermocouple readings. Liquid nitrogen was allowed to flow through the test section until all the thermocouples that connected on the wall of the test section read a steady value corresponding to the saturation temperature of liquid nitrogen. By that time we can conclude that chilldown process was completed. Wall temperature and Flow rate and supply pressure were recorded. The above steps were repeated with the different supply pressures and at various flow direction so that a wide parameter range may be investigated.

4. Results and Discussion

For single sampled experiment, Kline and McClintock [14] method is widely used for uncertainty measurement. Simple equation relating the measured data to the heat flux does not exist. For this reason, this section contains only the uncertainty of the instruments used for the experiment. ‘T’ type thermocouples have an uncertainty of ±0.50 C. for obtaining a fast response time the wire of the thermocouple used is of 0.25 mm and the response time corresponding to it is less than 0.2 seconds. Data Acquisition Systems have an accuracy of ±1°C between -100°C to 100°C and ±1.5°C between -200°C to -100°C. Mass flow rate have an accuracy of 0.5L/min. the uncertainty from third metal is avoided by using the same materials for connection and attention was provided to avoid large temperature gradients to avoid feed through errors.

4.1 Time temperature profile of straight, one twisted and twin twisted transfer lines

Experiments were conducted for straight, one twisted and twin twisted channels under two different supply pressures of 1.2 bar and 1.3 bar amounting to mass fluxes of 162 Kg/m²s and 176 Kg/m²s respectively. The results obtained are provided here. Wall temperature readings obtained were used for predicting the heat transfer and flow characteristics and was subjected to comparison. Jackson et al. [15] proposed three different phases during chilldown phenomena, namely film boiling regime, where the flow structure can be either stratified flow or inverted annular film flow. Transition boiling regime,
where a sudden drop in temperature is observed as liquid droplets begin to wet the wall. Nucleate boiling regime, where flow can be either bubbly or slug flow. Figure 3 depicts a chilldown curve with the three regimes demarked. As observed from the works of researchers in this field, the present study also regards the initiation of transition regime as the basis for the comparison of chilldown parameters. This is because the temperature is found to be almost steady after the transition regime. Three regimes observed during previous studies of chilldown in straight tube was also observed in curved geometries. However, the observed chilldown transition temperature and time were different from the straight geometry.

![Figure 3. Chilldown curve with three regimes demarked.](image)

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Variation of outer wall temperature of the section with respect to time is provided in figure 4. The different flow regimes can be identified by comparing with the chilldown curve given in figure 3. It is evident that even though, straight channels completed the film boiling and transition regimes at higher wall temperatures, time taken for onset of transition and nucleate boiling is lesser for one twisted tube whereas twin twisted lagged behind straight channel. This leads to the existence of optimized curved geometries that can perform better than straight transfer lines. Another observation from the results is that, as supply pressure increased, the chilldown time reduced. While comparing the twisted tubes separately, it can be seen that, under all flow conditions, one twisted tube performed better than twin twisted tubes in terms of chilldown time. In one twisted tubes the liquid phase comes in contact with the tube wall earlier than that in twin twisted tubes. This can be credited to the longer dwelling time of the liquid phase in one twisted tubes.

### 4.2 Heat Flux variation

In the present study outer wall temperature measurements were the only direct temperature measurements. In similar studies conducted by Reid Shaeffer et al. [1] and Hong Hu et al. [9] used Burggraf correlations [16]. Burgraff’s method is applicable to unsteady heat conduction with a single boundary condition. Burgraff developed a method that allows the estimation of the inner wall surface temperature by the measured outer wall temperature history through the solution of the inverse heat conduction problem (IHCP).

The method uses lumped capacitance analysis and can produce accurate results with fewer terms. Inside wall temperature can be given by the correlation[16]
\[ T_i = T_0 + \left( \frac{r_i^2}{4a} \left( \left( \frac{r_i}{r_0} \right)^2 - 1 - 2 \ln \frac{r_i}{r_0} \right) \right) \frac{dT_0}{dt} + \left( \frac{1}{64a^2} \left( r_i^4 - 5r_0^4 \right) \right) \frac{r_i^2}{8a^2} \ln \frac{r_i}{r_0} - \frac{r_i^4}{16a^2} \ln \frac{r_i}{r_0} \right) \frac{d^2T_0}{dt^2} + \ldots \] (1)

Inner wall surface heat can be calculated with the equation[16]

\[ q''_i = \rho c \left( r_i^2 - r_0^2 \right) \frac{dT_0}{dt} + \left( \rho c \right)^2 \left( \frac{r_i^2}{16} - \frac{r_0^2}{4} \ln \frac{r_i}{r_0} \right) \frac{d^2T_0}{dt^2} + \frac{\rho c}{k} \left( \frac{r_i^7}{384} - \frac{3r_0^3r_i}{128} + \frac{3r_0^2r_i^3}{128} - \frac{r_0^6}{384r_i} \right) \frac{d^3T_0}{dt^3} + \ldots \] (2)

Heat transfer coefficient can be found using the equation

\[ h_i = \frac{q'}{(T_i - T_{sat})} \] (3)

The average temperature was used for finding the inner wall surface heat flux. Heat flux is plotted as a function of wall superheat of straight tube at different mass flux. The curve is similar to boiling curve in pool boiling conditions. Critical heat flux and Leidenfrost point are clearly identified and time for reaching these points are also determined. Figure 5 represents the variation of heat flux with time for the test sections considered. Critical heat flux (CHF) and Leidenfrost point are usually recognized as key points in a boiling curve, dividing it into three distinct flow regimes. During a quenching process, heat flux diminishes to a minimum value at the Leidenfrost point and it increases drastically in transition boiling regime until the CHF point.

In laminar flow, previous studies showed that curved geometries generally have superior heat transfer characteristics than straight channels. But this may not be true in turbulent flow to which our study belongs.

![Figure 5. Variation of heat flux with time](image)
Table 1. Critical heat flux values.

| Geometry       | Critical Heat Flux (W/m²) | 1.2 bar Supply pressure | 1.3bar Supply pressure |
|----------------|---------------------------|-------------------------|------------------------|
| Straight tube  | 2272                      | 2319.864                |                        |
| One twisted    | 4299.66                   | 4433.437                |                        |
| Twin Twisted   | 3921.358                  | 4261.197                |                        |

The variation may be due to the combined effect of added turbulence due to the curvature and the effect of body forces present in the system. This is also coincident with the results got through this work which suggests that, CHF was consistently higher in twisted tubes but, its occurrence was delayed in twin twisted tubes when compared with straight channel. Table 1 provides the Critical Heat Flux values for the test sections considered. Gravitational effect must have played a significant role in chilldown of a horizontal straight line. In case of straight geometry, the gravity forces and pressure force are perpendicular to each other. In the present scenario the pressure force changes the direction depending on the alignment of the twisted tube. Pressure force was at a positive inclination with the horizontal during the first half turn and was in a negative inclination in the second half turn. With increase in mass flux, twin twisted tubes had an increase of 7.8% in its CHF while, the increment in one twisted was limited to 3%. This leads to an expectation that with further increase in mass flux, twin twisted tubes might over perform one twisted tubes.

4.3 Heat Flux variation at a section

![Figure 6. Heat Flux variation at a point 220mm from inlet.](image-url)
The variation in heat flux at a distance 220mm from inlet is provided in figure 6. It can be seen that the twisted tubes complete their transition regime earlier than the straight section considered. However, the critical heat flux values for one twisted is nearly 15% higher than twin twisted, and has an increment of 5% when compared to straight tubes. It can be said that, by providing means to increase swirl resulting in secondary flow during the initial phases of cryogen transfer, time and energy saving can be envisaged. From the results it can also be inferred that there exists an optimum condition for transfer lines that can lead to faster chilldown with minimum time consumed for film boiling.

4.4 Variation of fluid pressure and heat flux with time
Figure. 7 (a), (b) and (c) shows the variation of fluid pressure and corresponding variations in heat flux along with time during the chilldown process for twin twisted, one twisted and straight tubes respectively. It can be seen that in all the three cases there is a pressure reduction during the initial phases of film boiling where the vapor phase dominates. As the vapor fraction decreases, the pressure difference reduces and shows a deviation from inlet conditions again during the onset of nucleate boiling regime. The fluid pressure levels back to the inlet conditions as the flow proceeds to single phase nucleate boiling. Thus, in twisted tubes the swirl flow created has of course a cost in terms of pressure loss. Pressure losses in the tubes are produced by the additional turbulence resulting from swirling. However, experiments show that the increased pressure loss is relatively small compared to the critical heat flux.

![Figure 7](image-url)
gain. From figure 7(c), it can be realized that the deviation of inlet and outlet fluid pressure follows the same pattern but the percentage deviation is not much evident during initial flow regimes and unlike in twisted tubes, the pressure fluctuations are severely varying through the onset of nucleate boiling and turns more rigorous as the flow takes on to single phase convection.

**Conclusion**

This paper presents the results of experiments performed on straight, one twisted and twin twisted copper tubes using liquid nitrogen at supply pressures of 1.2 bar and 1.3 bar. For analyzing heat flow characteristics during chilldown process the transient temperatures for different mass flux conditions were measured. It was identified that the chilldown characteristics of copper transfer lines with a simple insulation like Polyurethane foam were similar to that of previously published studies. The chilldown characteristics of twisted tubes were found to be different from the straight tube due to the variation in direction vectors of pressure force with positions in them. There is an inverse relation existing between the mass flux and chill down time for each geometry. Straight channels completed the film boiling and transition regimes at higher wall temperatures, time taken for onset of transition and nucleate boiling is lesser for one twisted tube whereas twin twisted lagged behind straight channel. This leads to the existence of optimized curved geometries that can perform better than straight transfer lines. In straight tubes, deviation of inlet and outlet fluid pressure follows the same pattern but unlike in twisted tubes, the percentage deviation is not much evident during initial flow regimes and the pressure fluctuations are severely varying through the onset of nucleate boiling and turns more rigorous as the flow takes on to single phase convection. At an instance during the process, the critical heat flux values for one twisted is nearly 15% higher than twin twisted tube, and has an increment of 5% when compared to straight tubes. From the results it can also be inferred that there exists an optimum condition for transfer lines that can lead to faster chilldown with minimum time consumed for film boiling. Unlike the temperature difference between upper and lower portion of the straight pipe, there is uniform distribution of wall temperatures at each sections in twisted tubes. This may be due to additional turbulence caused by the curvature. Future works on identifying the avenues for reduction in cryogen consumption by varying flow parameters can be investigated into.

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**Nomenclature**

- \( d \) Diameter, m
- \( q' \) Heat flux, W/m²
- \( r \) Radius, mm
- \( T \) Temperature, K
- \( t \) Time, s
- \( \Delta T \) Wall super heat, K
- \( \rho \) Density, kg/m³
- \( C \) Specific heat, J/(kgK)
- \( h_i \) Heat transfer coefficient, W/m²K
- \( \alpha \) Thermal Diffusivity

**Abbreviations**

- CHF Critical heat flux
- HTC Heat transfer coefficient
- FB Film boiling
- NB Nucleate boiling
- TB Transition boiling
Subscripts

i  inner surface of wall
o  outer surface of wall
sat  saturation conditions

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