Parametric study for use of stainless steel as a material for thermal shield in PIP2IT transferline at Fermilab

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Abstract. Transferline thermal shields are cooled by dedicated cooling lines welded/brazed to the shield at a single point along the circumference. Copper/Aluminium is widely used to fabricate thermal shields because of their higher thermal diffusivity. This causes uniformity of temperature along the surface of the shield thus reducing thermal stresses within allowable values. However, raw material price, the cost of fabrication depending on standard sizes of pipes/tubes, often drives up the final price of thermal shields. To reduce the cost by making use of easily available stock of standard pipe/tube, it is decided to use stainless steel as a material for thermal shields in the PIP2IT transferline. The present paper discusses the design approach, various factors affecting the design and data for conservative selection of thermal shield design.

1. Introduction
Proton Improvement Plan – II (PIP-II) has been planned at Fermilab for providing high-intensity proton beams to the laboratory’s experiments. Fermilab has undertaken the PIP-II Injector Test (PIP2IT) [1] for integrated systems testing of critical components comprising the PIP-II front end. PIP2IT includes two cryomodules, to be tested using a pre-existing supercritical helium refrigerator and a distribution box. The PIP2IT transferline connects the distribution box to the cryomodules of P12IT.

The PIP2IT transferline consist of five lines as follows—Sub atmospheric 2K return (Line B), 5K supply (line C), 8K return (Line D), 40K supply (Line E), 80K return (Line F). The thermal shield encloses Lines B, C, D, E and F and is actively cooled by Line F. During the transient cooldown, cooling starts form the point where Line F is welded to the shield. Temperature starts reducing at that point, which results in flow of heat from the surrounding body surface to that point in circumferential direction. Greater the thermal diffusivity of the shield material lesser are the thermal stresses and strains. Therefore, Copper or Aluminium are generally preferred materials for the thermal shield because of higher thermal diffusivity [2].

However, stainless steel has been selected for fabrication of PIP2IT thermal shield due to following advantages—

- Easy availability of seam welded 10inch Outer Diameter (OD) tube.
- Reduced cost as compared to Copper or Aluminium shield.
- Welding Stainless steel (SS) shield to SS pipe is easier than brazing of Copper/Aluminium to SS.
- Higher strength of SS.
This choice results in larger thermal stresses and strains because of low thermal conductivity of steel as compared to Copper and Aluminium.

Figure 1. CAD model of part of the PIP2IT transferline.

2. Problem Definition
The disadvantage of using stainless steel is its low thermal diffusivity which is two orders of magnitude lower than copper. This causes large temperature gradients along the shield surface, which eventually result in thermal stresses and strains, proportional to the temperature gradients. The axial temperature gradient will occur due to variation in temperature of the cooling fluid. However, the shield is free to move in the axial direction, because of the flexhoses at ends. It is the circumferential component of temperature gradient which induces thermal stresses. The cold end of the shield contracts more than the hot end (figure 2). Thermal stresses arise due to the hot area (shield length X $C_{\text{HOT}}$) (Figure 6) of the shield resisting the contraction of cold area (shield length X $C_{\text{COLD}}$) (Figure 6). It should be noted that increase in the ratio of $C_{\text{HOT}}$ to $C_{\text{COLD}}$ increases the stresses.

The differential contraction results in bowing of the thermal shield section, if it is unrestricted by the line F and the vacuum jacket through the spiders (Figure 2). It should be noted that all the shield sections are welded to Line F. Hence, independent bowing of each section would require discontinuous curvature of the Line F at all end points of the shield sections, which is not physically possible. Hence it is reasonable to assume that, all the shield sections will bow freely together on the same curve as shown in figure 3. This free bowing is prevented by the vacuum jacket of the transferline and the line F. This induces stresses in the thermal shield which are henceforth addressed as bowing stresses. It should be noted that these are a form of thermal stresses.

To summarize the above, the temperature distribution acts as a load causing thermal strains. Stresses arise due to restrictions to these strains, explained as follows –

i. The hot part of the shield resisting contraction of the cold part (thermal stresses).

ii. Resistance of the vacuum jacket and Line F (which form the support conditions) to the free bowing of the thermal shield (bowing stresses).

Following are the factors affecting the magnitude of the bowing and thermal stresses—

1. Rate of cooling flow: Thermal gradients increase with increase in cooling flow thus increasing the thermal stresses.

2. Diameter of the thermal shield: Larger diameter is expected to reduce flexibility and hence increase the thermal stresses.

3. Thickness of the thermal shield: Larger thickness reduces flexibility and hence is expected to increase the thermal stresses. A standard tube size of 10inch OD, 0.12inch thickness has been
selected for PIP2IT transferline based on the availability and the configuration of the internal process piping.

4. Length of the thermal shield section: Increase in section length increases the bowing deflection thus increasing the stresses.

![Diagram](image)

**Figure 2.** Free bowing deflection of the thermal shield during transient cooldown

![Diagram](image)

**Figure 3.** Effect of vacuum jacket and physical coupling of the thermal shield sections through Line F on the bowing deflection.

The transient cooldown heat transfer analysis as described in section 3.1 for a prototype shield section, suggests that a flow control for flow of the order of 1-2g/s is necessary for safe cooldown. This
can be achieved only by using flow meters coupled with high resolution control valves which are absent in PIP2IT. The only variable out of the above, available to limit the thermal stresses, is the length of the individual thermal shield section(s).

Therefore, the problem at hand is to evaluate a suitable length of the thermal shield section(s) which causes reduction in thermal stresses within the allowable value. However, it should be noted that each thermal shield section has at least 2 spiders – one at each end, connecting it with the vacuum jacket(300K). Hence reducing the shield section length increases total number of spiders and thus the heat load to the 70K line. This puts a limitation on decreasing the section length.

3. Modelling of the problem

3.1. Conjugate heat transfer analysis
Modelling of the problem involves application of temperature distribution as a load. To derive this temperature distribution a conjugate heat transfer analysis has been carried out separately using Ansys® Wrokbench CFX solver.

- The problem consists of transient cooldown of a prototype thermal shield section.
- The section has outer diameter of 10inch with a thickness of 3mm and length of 10 ft.
- This involves convection cooling in line F and flow of heat from the thermal shield surface, circumferentially towards the line F.
- The cooling flow of 10g/s of helium at 80K (section 3.2) is at the flow pressure of Line F (12 bara).
- The isobaric (12 bara) properties of helium are imported in the solver, in a tabular format, which is derived from Hepak® [4] helium property software. Similarly, properties for stainless steel are obtained from NIST website [2].
- Results have been obtained by for different values of cooling flow, which are finally used to derive the load for worst case, with suitable approximations (section 3.5).

3.2. Assumptions:
- A flow of 10g/s of helium, at 12bar and 80K, through 1 inch NPS pipe, is assumed as the maximum possible cooling flow during a cooldown from 300K.
- Temperature gradient at the point of contact of Line F and thermal shield across the thickness is observed to be around 2.5K, as per section 3.1. Hence the thermal stresses at this point are neglected.
- Since the shield thickness is small as compared to the circumference, temperature is assumed constant along the thickness.
- Temperature is considered constant in axial direction, since the loads applied are already for the worst-case scenarios, and this assumption will only yield conservative results.
- The thermal strains occurring on the thermal shield do not have nature of a cyclic load. Hence, these are considered as primary loads in the criteria for allowable stresses.
- Half of the part length (figure 5) remains free at the ends. The curvature due to thermal strain in this 6-inch length is neglected as compared to the length of the shield section.

3.3. Development and approximation of support conditions:
As per the problem description, the shield sections are prevented from bowing freely due to the vacuum jacket and the Line F (Figure 3). A shield section to be analyzed is isolated and a free body diagram is described in figure 4. F1 and F2 are the forces exerted by the vacuum jacket on the shield through the spiders. V1, V2, M1 and M2 are the reaction forces and moments on Line F. R_F is the reaction force exerted by Line F on the thermal shield, the nature of which has not been evaluated. The problem is approximated as a case of pure bending wherein a cylindrical shield section of 10” OD and 0.12” thickness is restricted from bowing by, two frictionless roller-supports as shown in figure 4. M0 is the moment reaction due to the roller supports.
3.4. Geometric modelling:
CAD solid models are developed using NX® which are later imported in Ansys® Workbench 18 for modelling and solution. The roller supports are modelled by circular solid rings inside the shield (figure 5). Each ring models for one frictionless support, hence a total of 4 rings are required in the section. The rings have inner surface fixed and the outer surface in frictionless contact with the inner surface of the shield. The radial clearance between the contact surfaces is made zero for the purpose of solution. It is important to note that the rings are not modelling the spiders but only the support conditions as in figure 4.

To facilitate solving for different lengths, the CAD model is made as an assembly of parts which are cylinders 6inch long and having bonded contact in between them (Figure 5). Each part contains a solid ring for modelling the supports. To reduce/increase the length, the unwanted parts (including rings) are suppressed/unsuppressed in the solver (figure 5). The minimum distance between the frictionless supports is selected as 12 inches or the shield diameter (OD), whichever is less. For each diameter, the problem is solved for two thickness values, 3mm and 5mm. Four nodes at 300K (hot end – figure 2) at the center of the shield length on the outer surface are fixed to prevent rigid body motion. The stress concentration at that point is neglected.

3.5. Application of loads (temperature distribution)
The Ansys® solver project file consists of a thermal module and a structural module. The temperature distribution (load) is obtained separately as described in section 3.1. Axial variation of temperatures is neglected and an average value of the end temperatures is considered for simplification. Also, it is impossible to solve for all time steps. Hence a worst-case load condition is derived such that the thermal gradient is high along the circumference with a lowest possible cold end temperature.

This worst-case load is further approximated by a steady state temperature distribution as shown in figure 6. It is observed that the cold end temperature becomes asymptotic at 105K. Hence a temperature distribution corresponding to the boundary temperatures as shown in figure 6 is evaluated by the thermal module and further imported in the structural module as load.
Figure 5. Geometric model of thermal shield for structural analysis (unwanted rings and parts suppressed).

Figure 6. Thermal shield cross section, showing approximation of the transient temperature distribution with a steady state distribution for application as a load in structural analysis.

4. Criteria for evaluation of stresses
The allowable stress value is calculated as per ASME Section VIII, Div 2 [3]. Accordingly, for components to be safe,

\[ P_t + P_b \leq S_{pl} \] .........(1)

Where,
- \( P_t \) = Primary local membrane stress
- \( P_b \) = Primary local bending stress
- \( S_{pl} \) = Allowable stress value
Therefore, the Von-Mises equivalent stress induced shall be less than 2.07e8N/m² \((S_y)\) for the allowable section length (Ls) to be acceptable. As mentioned previously, the length of the model can be varied only by 6 inches (Part length). Hence the safe length is interpolated for the allowable stress value using the higher and the lower stress values for two corresponding lengths.

5. Solution for thermal shields of different diameter sizes

An attempt has been made to study the effect of parameters such as length, diameter and thickness on the thermal stresses induced in the thermal shield during cooldown. For this purpose, allowable lengths (Ls) for different diameters (6” to 16” OD) and thickness of 3mm and 5mm have been evaluated and listed in Table 2. The support conditions remain the same for all sizes. Regarding the load, \(\theta_0\) has been modified proportional to the diameter for values higher than 10 inch OD. Smaller diameters have the same \(\theta_0\), which will yield conservative results.

6. Results and Discussion

The allowable lengths (Ls) of the thermal shield sections are plotted for different diameter values on X-axis for thickness 3mm and 5mm as shown in Figure 7. This is also listed in Table 2. An increasing – decreasing trend is observed with a maximum at \(\phi_{10}\). It should be noted that increase in length of the shield section causes an increase in the induced stresses and is observed for all diameters.

Considering the data for 3mm thickness, for diameters greater than 10inch, Ls decreases with increase in diameter. This can be due to decrease in flexibility with increase in diameter. As opposed to this, Ls decreases with decrease in diameter for values less than 10 inches. This large decrease in Ls for diameters below \(\phi_{10}\) inch may be due to the conservative temperature load applied, as mentioned in section 5. Also, the decrease in distance between the roller supports for smaller diameters, as mentioned in 3.4 may have also resulted in increase of the induced stresses and thus reduction of Ls.

Considering the data points for 5mm thickness, the values of Ls are lower than those for 3mm thickness for all diameters. This is due to decrease in flexibility with increase in thickness. The unexpected drop in Ls for 5mm thick, \(\phi_{12}\)inch shield has not been understood yet.

In case of 5mm thick sections, the heat flow improves because of increased thickness, thus reducing the stresses. However, this advantage has not been considered in present analysis / results, and may be investigated later.

It is possible to select a length for thickness values less than or equal to the values for which the results are plotted. Interpolation for values other than the plotted diameters may not be possible because of the non-uniform pattern or behaviour of the points plotted. However, it is possible to select the lower of the lengths corresponding to the plotted diameters on either side of the required diameter value.

| SR No. | Diameter | Thickness | Allowable length | Element / part length |
|--------|----------|-----------|------------------|-----------------------|
|        | (inches) | (mm)      | (inches)         |                       |
| 1      | 6        | 3         | 26.9             | 6                     |
| 2      | 8        | 3         | 35.8             | 6                     |
| 3      | 10       | 3         | 60.6             | 6                     |
| 4      | 12       | 3         | 53.8             | 6                     |
| 5      | 14       | 3         | 46.3             | 4                     |
| 6      | 16       | 3         | 33.4             | 4                     |
| 7      | 6        | 5         | 24.8             | 6                     |
| 8      | 8        | 5         | 29.6             | 4                     |
| 9      | 10       | 5         | 47.1             | 6                     |
| 10     | 12       | 5         | 33.2             | 6                     |
| 11     | 14       | 5         | 40.7             | 4                     |
| 12     | 16       | 5         | 25.9             | 6                     |
Figure 7. Allowable section lengths ($L_s$) for different diameter values of thermal shield.

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