Study of Cryogenic Losses in a Liquid Helium Transfer Line

Martin Magnuson

Technical Student from the Linköping Institute of Technology,
Linköping, Sweden.

Abstract

The cryogenic losses in a 30° bend of a transfer line manufactured by the company l'Air Liquide for the helium supply to superconducting cavities at interaction point 2 in the Large Electron Positron Collider (LEP) were studied as function of the surface temperature distribution. The possibility of using an infrared (IR) temperature-measurement system to estimate the losses was evaluated. Two different coatings of paint on the pipe were tested in a laboratory environment in order to minimize reflection of thermal radiation from the pipe surface. Colour images were recorded from an operating transfer-line bend in a LEP service tunnel. Heat transfer calculations of the losses from the bend surface were made and compared with calculations of internal conduction.

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1 INTRODUCTION

The installation of a new 6 kW helium refrigerator at interaction point 2 at the Large Electron Positron collider (LEP) is made to supply up to 32 superconducting cavities with liquid helium at 4.5 K in the straight tunnels on both sides of the L3 particle-detector experiment. The refrigerator, kept in a service tunnel in parallel with the LEP tunnel, has a two-branch distribution system of transfer lines (right and left). For reasons of limited space in the service tunnel, the transfer lines include several bends on the way between the refrigerator and the cavities. In each bend, the process pipe and the screen require additional supports to resist unbalanced forces which develop at high internal pressures, which leads to increased cryogenic losses.

As an example for the application of an infrared (IR) scanner, it was chosen to study the cryogenic losses of one of the 30° bends of the LHe delivery transfer line showing condensation on the outer surface during operation in the LEP service tunnel at interaction point 2.

2 THE CRYOGENIC HELIUM TRANSFER LINES

The transfer lines are made of a stainless steel alloy and manufactured by the company l'Air Liquide (France). Each branch consists of two separate lines with an independent vacuum enclosure, one with a 72 mm inner bore pipe for liquid helium (LHe) delivery and a second with a 100 mm inner bore pipe for cold helium gas (GHe) recovery. The delivery line was guaranteed with a heat input to the process pipe below 0.35 W/m and the recovery line below 0.5 W/m. In total losses of 436 W to the 4.5 K process pipes were guaranteed by the manufacturer of the 2 · (212.5+62.5) m long transfer lines installed for LEP 200 at Point 2, including higher local losses in bends, valves and flexibles.

![Schematic cut through a helium transfer line](image)

The transfer line is of a vacuum insulation type, with aluminized Mylar insulation (superinsulation) and screen cooling systems according to Fig. 1.
The LHe supply pipe contains the liquid helium at 4.5 K during normal operation. The screen cooling pipe has a diameter of 22 mm and contains helium gas at about 50-75 K. Both pipes are placed inside a vacuum jacket of 204 mm diameter, for the delivery line and 254 mm for the recovery line, which are evacuated under outbaking at temperatures up to 100°C, to achieve an insulation vacuum of up to $5 \times 10^{-7}$ Torr at cryogenic operating conditions. Vacuum provides effective insulation when the mean-free path of the molecules becomes larger than the dimensions of the vessel. In addition, the inner pipes are wrapped with many layers of reflecting material (aluminium-coated Mylar foil) to considerably reduce heat transfer by radiation.

The $30^\circ$ bends has a rather complicated internal structure, as can be seen from Fig. 2. The vacuum jackets are constructed such, that contraction due to accidental cool down can be absorbed by the mechanically reinforced bends. This avoids the use of compensation bellows but increases the local heat losses. The bends consist of a strong central ring to which are connected the vacuum jackets, absorbing the forces coming from the cold pipe. Thus, a strong support cage is fixed in the middle to the room temperature wall; 2 strong discs on each side are connected to the 50 - 75 K screen pipe, and from them are also supported the 4.5 K pipes using 2 concentric pipes in order to reduce heat conduction. The heat bridge from the central ring in the middle to the discs on each side results in a surface temperature profile at the bend, showing in its centre, temperatures up to 16°C below tunnel reference temperature. This profile is the result of the equilibrium between the internal heat conduction and heat leak from air convection and radiation from the environment.

![Fig. 2. Cut-away picture of a 30° bend. The external diameter is 0.254 m in the bend area for the delivery line.](image-url)
3 HEAT RADIATION MEASUREMENTS

3.1 Basic laws of heat radiation

Thermal radiation is electromagnetic radiation emitted by a body as a result of its temperature. The wavelength of the thermal radiation lies in a broad range extending from 0.1 to 100 µm and its maximum is shifted to shorter wavelengths with increasing temperature according to Wiens displacement law. The Stefan–Boltzmann law gives the total specific power emitted by a black body of temperature T:

\[ E = \sigma \cdot T^4 \]

where:

- \( E \) = Emissive power of a black body [W/m²],
- \( \sigma = 5.669 \times 10^{-8} \text{ W/m}^2 \cdot \text{K}^4 \),
- \( T \) = degrees in Kelvin.

Black bodies appear black because they do not reflect any radiation and absorb all incident radiation. They are at the same time emitting the maximum possible thermal radiation for a given surface temperature T.

When radiant energy strikes a material surface, part of the radiation is reflected and part of it absorbed. One defines the reflectivity \( \rho \) by the fraction reflected and the absorptivity \( \alpha \) by the fraction absorbed. Thus,

\[ \rho + \alpha = 1 \]

The emissivity \( \varepsilon \) of a material is defined as:

\[ \varepsilon = \frac{E}{E_b} \]

where \( b \) denotes a black body. Thus, the emissivity is a measure of how close the object is to an ideal black body. For a black body \( \alpha = \varepsilon = 1 \) holds. Also for all other materials the law was established that absorptivity \( \alpha \) and emissivity \( \varepsilon \) are equal with a numerical value between 0 and 1.
3.2 The infrared measurement system

The IR heat measuring system available at CERN (AGEMA Thermovision 800) has a thermoelectrically cooled scanner operating in the wavelength range 2 to 5.6 µm. The IR scanning unit converts electromagnetic thermal energy received from different directions into a proportional electric signal and composes a temperature image of the object under study.

To determine the true object temperature is not a simple task; in general the conversion from signal to temperature is a function of several parameters. The radiation reaching the sensor is the sum of radiation from the object, from the surroundings reflected by the object, and from the atmosphere. In order to obtain the true temperature of the object, individual object emissivities must be known, either from measurement or from tables. They range, depending on object materials and surface coatings, from 0.05 to 0.95.

The radiation from the object and the reflections from the surroundings pass the atmosphere before they reach the scanner. The atmosphere attenuates the object radiation and radiates itself. The attenuation is mainly caused by water vapour (H₂O) and carbon dioxide (CO₂) which absorb infrared radiation in the spectral bands being used.

The AGEMA IR measuring system is capable of taking into account the object emissivity, ambient temperature, object distance, atmospheric temperature, and atmospheric transmission, but the quality of the temperature calculations depends largely on the reliability of the parameters introduced.

3.3 Preliminary experiments with the IR system

In order to evaluate the possibility of using an IR camera for measuring the temperature difference between the ambient tunnel reference and the surface of the transfer-line pipe, different surface coatings were tested in order to minimize thermal reflection and accomplish a well-defined surface emissivity, which can vary, in the case of stainless-steel surfaces, between 0.05 and 0.5. The experiment was carried out in a laboratory environment according to Fig. 3.
3.3.1 Transfer line uncoated

Reflection spots mainly due to thermal radiation from persons were found and made these spots appear 2°C warmer than the rest of the surface of the transfer line. Thus, this situation was not acceptable.

3.3.2 Transfer line coated with black non-shiny paint (Lucite 2441T)

Reflections were reduced to a large extent. The temperature of the transfer line could be measured perfectly all over the surface.

3.3.3 Transfer line coated with shiny uncoloured paint (Lucite 2438T)

The reflections were even much worse than in the untreated case. This paint could not be used for accurate measurements. This shows the importance of an appropriate paint coating.

3.4 Temperature mapping on the transfer lines in the LEP service tunnel

During the temperature measurements on the transfer lines, they were fed with cold GHe from a 6 kW helium liquefier at operation temperature of the process pipe of 10 K and that of the screen pipe of 60 K.
Fig. 4. IR image of the right side of the transfer line bend

Firstly, a mapping of temperatures was made using an ordinary temperature probe for comparison with IR measurements. Secondly, the bend of the pipe was painted with black non-shiny paint (Lucite 2441T) to minimize thermal reflection. The IR measuring system was calibrated against the temperature values obtained simultaneously with an ordinary temperature probe. By this the emissivity of the object was measured to be 0.88 which is in accordance with values given in tables for black paint.

Pictures were taken with the IR camera; one on the right side of the bend and another on the left. Graphics processing was made by conversion of temperature into colours. A video printer was used for making hard copies of the images. As an example, an image of the right side is shown in Fig. 4.

4 CALCULATION OF THE HEAT INPUT TO THE BEND

In the following, an attempt is made to calculate the heat input to the bend area from the temperature map obtained, and to compare this with the analysis of the internal structure and heat conduction along its supports. Thus, the following relations for the heat transfer should approximately be valid:
\[ Q_{\text{cond.}} = Q_{\text{free conv.}} + Q_{\text{rad.}} \]

or

\[ Q_{\text{cond.}} = Q_{\text{forced conv.}} + Q_{\text{rad.}} \]

In order to calculate the convection heat transfer, an estimation of the areas with the different temperature zones had to be made using the IR images. Calculations were made both for free and forced convection. The air in the service tunnel is changed twice per hour which would correspond to an air velocity of about 1 m/s. The results are shown in Tables 1, 2 and 3. In addition, the heat absorbed by radiation coming from the environment is estimated in Table 4. Formulae and data are taken from Refs. [1-4].

4.1 Free convection

As an example, the dimensions of the LHe delivery transfer line are taken. The results would be very similar for the GHe recovery line. The characteristic dimension \( l \) for free convection around a horizontal pipe is one half of the circumference of the pipe:

\[ l = \frac{D \cdot \pi}{2} \]

The Grashoff number is defined as:

\[ \text{Gr} = \frac{g \cdot l^3 \cdot \beta \cdot (T_0 - T_\infty)}{\nu^2} \]

where:

- \( g \) = gravitational acceleration, 9.81 m/s\(^2\),
- \( D \) = diameter of pipe, 0.254 m,
- \( \beta = 1/293 \text{ K}^{-1} \),
- \( T_0 \) = surface temperature \([\text{K}]\),
- \( T_\infty = \text{ambient temperature} \([\text{K}]\), 295.4 K,
- \( \nu \) = kinematic viscosity, \( 14.5 \cdot 10^{-6} \text{ m}^2/\text{s} \)

The Reyleigh number is:

\[ \text{Ra} = \text{Gr} \cdot \text{Pr} \]

with Prandtl's number being:

\[ \text{Pr} = 0.7 \]

The Nusselt number for a horizontal pipe is calculated as:
\[ \text{Nu} = \left( 0.825 + \frac{0.387 \cdot \text{Ra}^{1/6}}{1.19} \right)^2 \]

which in this case is well within the valid interval \(10^{-3} < \text{Ra} < 10^{12}\)

The heat transfer coefficient finally is:

\[ \alpha = \frac{\text{Nu} \cdot \lambda}{l} \quad \text{[W/m}^2\text{K]} \]

using the thermal conductivity for air, \(\lambda = 25.5 \cdot 10^{-3} \quad \text{[W/m} \cdot \text{K]}\).

The heat transfer per unit length of transfer line is: \(Q = \pi \cdot D \cdot \alpha \cdot (T_0 - T_-) \quad \text{[W/m]}\). The calculated values, with free convection for the different temperature zones and their estimated corresponding areas as taken from the measured temperature map, are shown in Table 1.

Table 1. Calculated heat transfer from different temperature zones on half a helium transfer line bend with a diameter of 0.254 m. The total half bend surface is 0.24 \(m^2\). Ambient temperature 22.2°C.

<table>
<thead>
<tr>
<th>Temp [°C]</th>
<th>Length [mm]</th>
<th>Area [m²]</th>
<th>Ra ((10^7))</th>
<th>Nu</th>
<th>(\alpha) [W/m²K]</th>
<th>Heat transfer [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.5</td>
<td>3</td>
<td>0.0024</td>
<td>10.40</td>
<td>62.07</td>
<td>3.97</td>
<td>0.14</td>
</tr>
<tr>
<td>8.5</td>
<td>6</td>
<td>0.0048</td>
<td>9.70</td>
<td>60.78</td>
<td>3.88</td>
<td>0.25</td>
</tr>
<tr>
<td>9.5</td>
<td>6</td>
<td>0.0048</td>
<td>8.99</td>
<td>59.42</td>
<td>3.80</td>
<td>0.23</td>
</tr>
<tr>
<td>11.0</td>
<td>6</td>
<td>0.0048</td>
<td>7.93</td>
<td>57.24</td>
<td>3.66</td>
<td>0.20</td>
</tr>
<tr>
<td>12.0</td>
<td>7.5</td>
<td>0.0060</td>
<td>7.22</td>
<td>55.67</td>
<td>3.56</td>
<td>0.22</td>
</tr>
<tr>
<td>13.0</td>
<td>7.5</td>
<td>0.0060</td>
<td>6.51</td>
<td>54.00</td>
<td>3.45</td>
<td>0.19</td>
</tr>
<tr>
<td>14.0</td>
<td>9</td>
<td>0.0072</td>
<td>5.81</td>
<td>52.19</td>
<td>3.34</td>
<td>0.20</td>
</tr>
<tr>
<td>15.0</td>
<td>9</td>
<td>0.0072</td>
<td>5.10</td>
<td>50.23</td>
<td>3.21</td>
<td>0.17</td>
</tr>
<tr>
<td>16.0</td>
<td>9</td>
<td>0.0072</td>
<td>4.39</td>
<td>48.07</td>
<td>3.07</td>
<td>0.14</td>
</tr>
<tr>
<td>17.0</td>
<td>15</td>
<td>0.012</td>
<td>3.68</td>
<td>45.65</td>
<td>2.92</td>
<td>0.18</td>
</tr>
<tr>
<td>18.0</td>
<td>24</td>
<td>0.019</td>
<td>2.97</td>
<td>42.89</td>
<td>2.74</td>
<td>0.22</td>
</tr>
<tr>
<td>18.5</td>
<td>54</td>
<td>0.043</td>
<td>2.62</td>
<td>41.34</td>
<td>2.64</td>
<td>0.42</td>
</tr>
<tr>
<td>19.0</td>
<td>144</td>
<td>0.115</td>
<td>2.27</td>
<td>39.63</td>
<td>2.53</td>
<td>0.93</td>
</tr>
</tbody>
</table>

Calculation on the basis of free convection thus gives a total heat input of 7.0 W for the bend area.
4.2 Forced convection

Although imposed air movement in the tunnel is limited to 1–2 m/s, the resulting heat transfer values were calculated for comparison. Two different cases must be considered, either cross flow with air movement perpendicular to the transfer line axis or parallel flow.

4.2.1 Forced cross-flow convection.

In forced convection the heat transfer is largely dependent on the relevant Reynolds number.

The Reynolds number is defined as:

\[ Re = \frac{v \cdot D \cdot \rho}{\mu} \]

where:
- \( v \) = air flow velocity [m/s],
- \( D \) = diameter of the pipe, 0.254 m,
- \( \rho \) = air density at 22° C, 1.196 kg/m³,
- \( \mu \) = dynamic viscosity, 17.98 \( \cdot 10^{-6} \) kg/m \cdot s.

The flow is defined as turbulent if the Reynolds number is larger than about 2300 and laminar below this value. In our case the flow is turbulent for air velocities above approximately 0.2 m/s and then the Nusselt number for cross flow is calculated as:

\[ Nu = 0.193 \cdot Re^{0.618} \cdot Pr^{(1/3)} \]

The heat transfer coefficient and the total heat transfer for the temperature profile given in Table 2 are calculated the same way as for the free convection.

<table>
<thead>
<tr>
<th>Velocity [m/s]</th>
<th>Re</th>
<th>Nu</th>
<th>( \alpha ) [W/m²K]</th>
<th>Total heat transfer [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.2</td>
<td>3378.5</td>
<td>25.98</td>
<td>2.61</td>
<td>6.03</td>
</tr>
<tr>
<td>0.6</td>
<td>10135.4</td>
<td>51.23</td>
<td>5.14</td>
<td>11.89</td>
</tr>
<tr>
<td>1.0</td>
<td>16892.4</td>
<td>70.25</td>
<td>7.05</td>
<td>16.31</td>
</tr>
<tr>
<td>2.0</td>
<td>33784.8</td>
<td>107.81</td>
<td>10.82</td>
<td>25.03</td>
</tr>
</tbody>
</table>
Table 2 gives the calculated heat transfer with an imposed air movement perpendicular to the transfer line and Table 3 that with an air movement along the transfer line.

4.2.2 Forced parallel-flow convection

With forced parallel-flow convection the Nusselt number is calculated as:

\[ \text{Nu} = 0.023 \cdot \text{Re}^{0.8} \cdot \text{Pr}^{(1/3)} \]

Table 3. Calculated heat losses at different air velocities with forced parallel-flow convection.

<table>
<thead>
<tr>
<th>Velocity [m/s]</th>
<th>Re</th>
<th>Nu</th>
<th>( \alpha ) [W/m²K]</th>
<th>Total heat transfer [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.2</td>
<td>3376.2</td>
<td>13.58</td>
<td>1.36</td>
<td>3.15</td>
</tr>
<tr>
<td>0.6</td>
<td>10128.6</td>
<td>32.70</td>
<td>3.28</td>
<td>7.59</td>
</tr>
<tr>
<td>1.0</td>
<td>16880.9</td>
<td>49.21</td>
<td>4.94</td>
<td>11.42</td>
</tr>
<tr>
<td>2.0</td>
<td>33761.9</td>
<td>85.67</td>
<td>8.60</td>
<td>19.89</td>
</tr>
</tbody>
</table>

4.3 Heat radiation

For surfaces of temperatures close to ambient, the radiative heat transfer is roughly proportional to the temperature difference between object and environment:

\[ \frac{q}{A_1} = \sigma \cdot \varepsilon_1 \cdot (T_1^4 - T_2^4) \sim \sigma \cdot \varepsilon_1 \cdot (T_1 - T_2) \]

where:

- \( q \) = heat flow [W],
- \( A_1 \) = object area [m²],
- \( \sigma \) = Stefan–Boltzmann constant = 5.669 \times 10⁻⁸ W/m²·K⁴,
- \( \varepsilon_1 \) = object emissivity, 0.88 for our paint,
- \( T_1 \) = object temperature [K],
- \( T_2 \) = ambient temperature [295.4 K]
Table 4. Calculated heat transfer from radiation.

<table>
<thead>
<tr>
<th>Temp. [°C]</th>
<th>q/A [W/m²]</th>
<th>q [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.5</td>
<td>70.13</td>
<td>0.17</td>
</tr>
<tr>
<td>8.5</td>
<td>65.70</td>
<td>0.32</td>
</tr>
<tr>
<td>9.5</td>
<td>61.21</td>
<td>0.29</td>
</tr>
<tr>
<td>11</td>
<td>54.39</td>
<td>0.26</td>
</tr>
<tr>
<td>12</td>
<td>49.80</td>
<td>0.30</td>
</tr>
<tr>
<td>13</td>
<td>45.14</td>
<td>0.27</td>
</tr>
<tr>
<td>14</td>
<td>40.44</td>
<td>0.29</td>
</tr>
<tr>
<td>15</td>
<td>35.69</td>
<td>0.26</td>
</tr>
<tr>
<td>16</td>
<td>30.89</td>
<td>0.22</td>
</tr>
<tr>
<td>17</td>
<td>26.04</td>
<td>0.31</td>
</tr>
<tr>
<td>18</td>
<td>21.14</td>
<td>0.40</td>
</tr>
<tr>
<td>18.5</td>
<td>18.68</td>
<td>0.80</td>
</tr>
<tr>
<td>19</td>
<td>16.19</td>
<td>1.86</td>
</tr>
</tbody>
</table>

This gives a total heat radiation to the transfer line bend of 5.75 W.

4.4 Thermal conduction

The main part of the support structure in the 30° bend, as shown in Fig. 2 is a heat bridge from 279 K at the external surface of the bend to the strong discs on each side and is calculated by using the Fourier’s law of heat conduction. In this temperature region it is necessary to treat the thermal conductivity of stainless steel as a function of the temperature. Assuming a temperature T of each of the strong discs gives the following equations:

$$q_o = -\frac{A_o}{l_o} \int_T^{279} \lambda_o(T)dT$$

$$q_i = -\frac{A_i}{l_i} \int_T^{160} \lambda_i(T)dT$$

where:

- $A_o =$ cross section area of the outer heat bridge, 13.8 cm²,
- $A_i =$ cross section area of the inner heat bridge, 1.61 cm²,
- $l_o =$ length of conduction along the outer heat bridge, 22 cm,
- $l_i =$ length of conduction along the inner heat bridge, 1.8 cm,
- $\lambda =$ thermal conductivity of stainless steel [W/cm·K].
This calculation gives with $q_i = q_o$ a thermal conduction of 8.0 W for each side, thus a total of about 16 W for the bend and an intermediate temperature, $T$, of 153 K for the strong disks. The heat conduction from the 60 K screen pipe down to the 4.5 K process pipe is minimized by increasing the length of conduction using the two concentric pipes. A similar calculation in this temperature region shows that only about 0.5 W of the 16 W heat input is transmitted to the process pipe.
5 CONCLUSIONS AND COMPARISONS OF THE RESULTS

The IR temperature-measuring system is useful to determine heat transfer from surfaces (i.e. localization of heat leaks), where simple temperature probes can not be used easily (i.e. surface areas with difficult access) or complex shapes of isotherms. Modern systems, such as the AGEMA scanner, available at CERN, provide good images with varying colours for different temperature zones, and other useful options are available with the IR graphics processing system. A non-shiny black paint coating must be used on shiny metal surfaces to eliminate thermal reflection from heat sources in the environment, and to establish a well-defined emissivity value. Radiation shields can be used, in addition, to eliminate reflection from the environment.

The study shows that an evaluation of the heat transfer from a known temperature map on the external pipe is not precise and can only give an estimation of the cryogenic losses since it is difficult to know well enough the emissivity of the surfaces, the radiation from the environment, and the presence of forced air flow.

Calculations assuming free convection together with radiation on the outside of the transfer-line bend areas studied at LEP point 2 (30° bends) give a loss of about 13 W. The calculations assuming forced convection are probably more realistic, even though it is difficult to estimate the magnitude of the air velocity. The forced cross flow together with radiation give a total loss of about 22 W, while the parallel flow gives a value of about 17 W, assuming in both cases an air velocity of 1 m/s. The parallel air flow is probably more realistic than the cross flow since the transfer lines are mainly installed in a horizontal tunnel with forced ventilation.

A more precise calculation of heat conduction in the internal support structure of the 30° bends can indeed explain heat transfer losses of about 16 W (most of it is going to the 50–70 K screen pipe). This corresponds well to heat transferred to the surface by radiation from the environment (about 6 W) and by an ambient air velocity of 0.9 m/s along the transfer line (about 10 W), when using the observed temperature map on the bend. However, each bend has its own characteristics depending on the thermal resistance of individual weldings in the internal structure.
6. REFERENCES


