

Heat exchangers in the ATLAS liquid argon calorimeter



Diploma Thesis
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Abstract

The Liquid Argon Calorimeter plays a central role in the future ATLAS detector at LHC. It is built from two end-caps and one barrel, housed in three separated cryostats. While operating, the cryostats will be filled with liquid argon at a temperature of approximately 87 K. Before the cryostats are filled with liquid argon, the calorimeter must be cooled down from ambient temperature to the temperature of filling. The Liquid Argon Calorimeter will be cooled down at least twice. First in the CERN West Area for cold tests, and a second time in its final position in the underground cavern of the ATLAS detector. Estimation of the cool-down time is therefore of significance.

This diploma thesis evaluates the cool-down time of the Barrel Liquid Argon Calorimeter, which is defined as the time needed to cool down from ambient temperature to the temperature in which the cryostat can be filled with liquid argon. A general model for cool-down simulations has been developed, and results from cool-down simulations based on this model are presented for three different modes. Two modes in which the cryostat is charged with argon gas, with and without condensation, and one mode in which the cryostat is charged with helium gas. The mechanical design of the cryostat and its calorimeter does not allow temperature gradients higher than 50 K between any two points, and this temperature gradient is the limitation of the cool-down.

The results from the cool-down simulations show 2 weeks as a reasonable estimate for the time needed to cool down the Barrel Liquid Argon Calorimeter and its cryostat when the cryostat is charged with argon. The cool-down can be done in less time if the cryostat is charged with helium, but this is not recommended. The reason is that helium can cause pollution of the liquid argon, and there by be detrimental for the functioning of the detector.

Sammen drag

Det Flytende Argon Kalorimeteret har en sentral rolle i den fremtidige ATLAS detektoren på LHC, og består av to "end-caps" og en "barrel" som alle er bygget inn i separate kryostater. Under eksperimentene vil kryostatene være fylt med flytende argon (ca. 87 K). Før fylling kan finne sted må kalorimeteret kjøles ned fra omgivelsestemperatur til kalorimeter temperaturen tillater fylling. Denne nedkjølingsprosessen vil finne sted minst to ganger. En gang i testområdet i hall 180, og en gang i sin endelige posisjon i ATLAS detektoren. Det er derfor viktig å ha et godt estimat av nedkjølingstiden.

Denne diplom oppgaven evaluerer nedkjølingstiden av "Barrel liquid Argon Calorimeter", som er definert som den tiden det tar å kjøle fra omgivelsestemperatur til fylling av flytende argon kan tillates. En generell nedkjølingsmodell er utviklet, og resultater fra simulering basert på denne modellen er presentert. To moduser med argon gas, med og uten kondensering, og en modus med helium gas. Det mekaniske designet tillater ikke temperaturforskjeller over 50 K mellom to vilkårlige punkter, og dette kriteriet er begrensningen for nedkjølingsfarten.

Basert på resultater fra nedkjølingssimuleringene ser det ut til at 2 uker er et rimelig estimat av nedkjølingstiden for "Barrel Liquid Argon Calorimeter" og dens kryostat. Ved bruk av helium gas vil nedkjølingstiden reduseres betydelig, men da helium kan medføre komplikasjoner er bruk av helium ikke anbefalt.

Preface

This diploma thesis was written at CERN autumn 1999/spring 2000 as a part of my studies at the Norwegian University of Science and Technology. It is the final part of the education leading towards a Master of Science Engineering degree.

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Chapter I

Introduction

What is CERN?

CERN is an European Organization for Nuclear Research, and is the world's largest particle physics centre. Founded in 1954, the laboratory was one of Europe's first joint ventures, and is today funded by 20 Member States. It is located on the border between France and Switzerland west of Geneva, and attract's about 6000 scientists from over 500 universities in more than 80 countries. Its location symbolizes the international spirit of collaboration which is the reason for the laboratory's success.

What is the mission of CERN?

The mission is pure science, finding out what our Universe is made of and how it is related. By accelerate particles into high energy collisions, it will take us back to the beginning of time, the Big Bang. The scientists at CERN study millions of these collisions to try to understand how the universe became the one we see today.

How do they do it?

The physicists can unravel the forces acting between particles by accelerate them to very high energies and smash them into targets, or into each other. CERN's biggest accelerator is the Large Electron Positron collider (LEP), which is a 27 kilometres circular accelerator buried over 100 meters under ground. LEP use powerful electric fields to push energy into a beam of particles, and magnetic fields are used to keep the beam tightly focused and to steer the particles around the ring. Particles go round and round again, collecting energy with each lap, and particles travelling near the speed of light lap it over 11000 times each second.

The collisions take place inside accelerators called colliders, producing new particles. Detectors placed around the collisions record all the information. Different layer of the detector measure different properties of the emerging particles.

Any new projects?

The Large Hadron Collider (LHC) is the next major project at CERN. As LEP shuts down in year 2000, LHC is designed to fit in the tunnel now housing LEP. The accelerator will take its place in 2005, and will allow proton-proton as well as heavy ions collisions.

To exploit the capabilities of the future LHC, four particle detectors are being designed and will be installed in the underground caverns. ATLAS will be the largest of these detectors, compromising a complex cryogenic system for superconductive magnets and liquid argon ionization sampling calorimetry. It will be more than 22 meter high and 42 meter long, with a total weight about 7000 tons. The ATLAS detector, from a cross-section view, is shown in Figure 1.1. The four major components of the detector are; the *inner tracker* which measures the momentum of each charged particle (yellow), the *calorimeter* which measures the energies carried by the particles (orange and green), the *muon spectrometer* which identifies

and measures muons (blue), and the *magnet system* which bends charged particles for momentum measurement (grey).

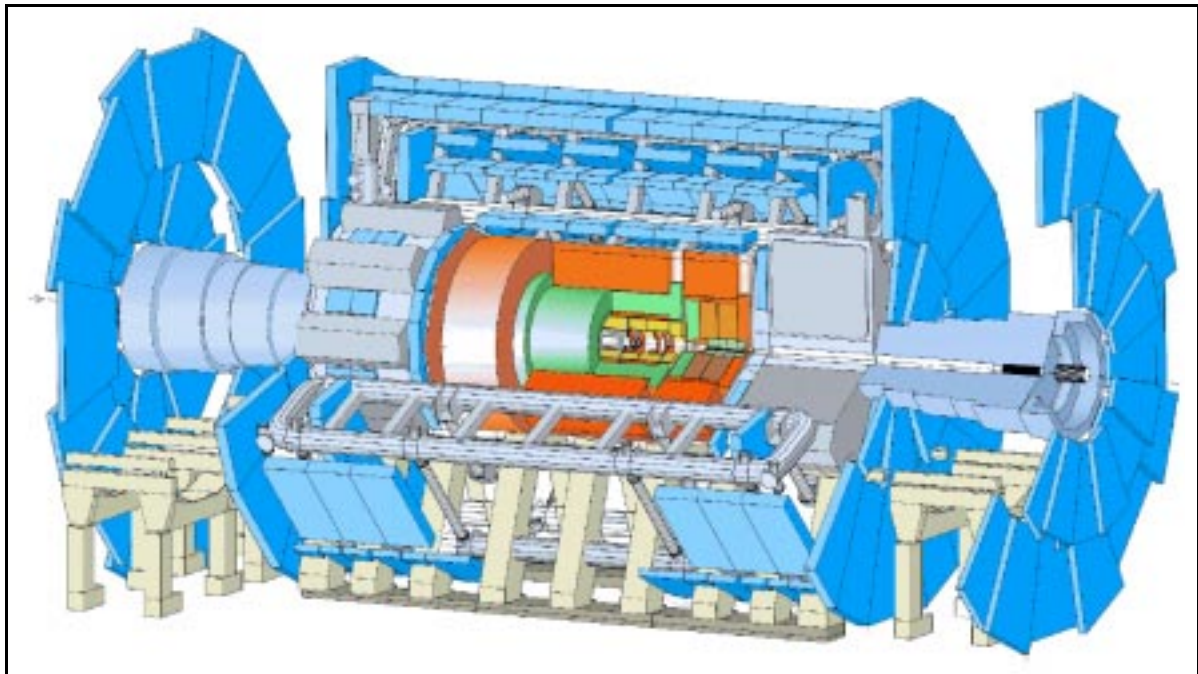


Figure 1.1 The ATLAS detector.

The Liquid Argon Calorimeter, a part of the *calorimeter*, plays a central role in ATLAS. It consists of two end-caps and one barrel, and will be charged with liquid argon while operating.

What is the scope of this thesis?

The main scopes of this thesis is to evaluate the cool-down time of the Barrel Liquid Argon Calorimeter from ambient to operating temperature, and to give an evaluation of temperature gradients during the cool-down. The thesis should as well give an evaluation of the heat transfer coefficient for tube heat exchangers cooled by two-phase flow of liquid nitrogen, and placed inside an argon/helium volume.

What is the content of the chapters?

- Chapter II : '**General Information**' contains the necessary background for this thesis. Words and phrases used in the following chapters will be explained.
- Chapter III : '**Heat transfer**' presents the theory behind the different modes of heat transfer appearing in this thesis.
- Chapter IV : '**The Barrel Liquid Argon Calorimeter**' takes a look at the true nature of the Barrel Liquid Argon Calorimeter.
- Chapter V : '**The simulation model**' presents a model for use in cool-down simulations of the Barrel Liquid Argon Calorimeter.
- Chapter VI : '**The cool-down simulation**' presents results from cool-down simulations based on the model presented in Chapter V.
- Chapter VII : '**Discussion**' discusses the validity of the cool-down model presented in Chapter V, and gives an evaluation of the results obtained from the cool-down simulations presented in Chapter VI.
- Chapter VIII : '**Conclusion and future**' concludes the work done.

Chapter II

General Information

The Barrel Liquid Argon Calorimeter will operate at a temperature of approximately 87 K. To gain/keep this temperature, energy must be removed from the barrel. The energy is moved by a *refrigeration plant*, which circulates a *process fluid* as an energy carrier to create the *refrigeration process*. The refrigeration costs are minimized by limiting the heat leak from the surroundings. Practically it is done by building a housing for the calorimeter, a *cryostat*. Another effort for optimization of the refrigeration process is done by keeping the cryogenic fluid in a *two-phase flow* inside the heat exchanger tubes. The boiling fluid will then ensure a good heat transfer all over the length of the heat exchanger tubes. This chapter takes a closer look at the general terms around this process.

2.1 Refrigeration

When energy is moved from a cold reservoir to one with a higher temperature, it is called a refrigeration process. A process fluid will absorb energy from the cold reservoir and emit it to the warm reservoir. The refrigeration process can be roughly divided into four steps.

- First, the process fluid absorbs energy from the cold reservoir. The energy exchange takes place in a heat exchanger, and the energy flow is driven by a lower temperature of the process fluid than the temperature of the cold reservoir.
- Second, the process fluid is compressed. The compression will not only increase the pressure of the fluid, but the fluid temperature as well. The temperature of the process fluid after the compression is higher than the temperature of the warm reservoir.
- Third, the process fluid heat exchanges with the warm reservoir. The energy exchange takes place in a heat exchanger, and it is caused by the temperature difference between the warm reservoir and the process fluid.
- Fourth, the process fluid is expanded. As for the compression, the process decreases the temperature as well as the pressure. The temperature after the expansion is lower than the temperature of the cold reservoir. A new heat exchange can now take place with the cold reservoir, and the refrigeration process is completed.

The expansion may take place either through a throttling device (Isenthalpic expansion) or in a work-producing device (Isentropic expansion).

2.2 Cryogenics

The word cryogenics usually refers to refrigeration processes taking place below 150 K. A cryogenic fluid is the process fluid and a cryo-plant is the hardware needed for the refrigeration process. This section takes a closer look at the cryogenic fluids used for production of the low temperature environment in the Barrel Liquid Argon Calorimeter.

The cryogenic fluid used by the cryo-plant is nitrogen, while the calorimeter itself will be charged with argon or helium during cool-down. Table 2.1 shows the main characteristics of these three fluids, and the following will give a short presentation of them.

<i>Characteristic</i>	<i>Argon</i>	<i>Helium</i>	<i>Nitrogen</i>
<i>Normal boiling point</i>	87.15 K	4.22 K	77.25 K
<i>Critical temperature</i>	150.86 K	5.20 K	126.26 K
<i>Critical pressure</i>	4.91 MPa	0.27 MPa	3.40 MPa
<i>Triple point temperature</i>	83.80 K	-	63.15 K

Table 2.1 Characteristics of cryogenic fluids.

2.2.1 Argon

Argon (Ar) is colourless and without an odour, both for the liquid phase and the gas phase. It is nontoxic and explosive proof. Argon has a normal boiling point of 87.15 K, the critical temperature is $T_c = 150.86$ K, the critical pressure is $P_c = 4.91$ MPa and the triple point temperature is $T_{\text{trippel}} = 83.80$ K.

2.2.2 Helium

Ordinary Helium (He) gas consists of two isotopes, He-3 and He-4. The existence of He-3 is very scarce, so when we speak of helium or liquid helium, we normally refer to helium-4. Liquid helium is odorless and colorless with a normal boiling point of 4.22 K. The critical temperature is $T_c = 5.20$ K and the critical pressure is $P_c = 0.27$ MPa, which makes helium the only efficient cryogenic fluid used for cooling below 20 K.

Helium has many unique properties. The solid phase do not exist in normal atmospheric pressure. In fact liquid helium must be compressed 2.53 MPa before it will solidify. Nor does helium has a triple point, and it has two liquid phases. The low-temperature liquid phase (He II) has properties exhibited by no other liquid [1].

2.2.3 Nitrogen

Liquid nitrogen (N) is a clear water like fluid which is only slightly reactive. Both the gas and the liquid state are odourless, colourless, nonflammable, nontoxic and non explosive. Nitrogen has a normal boiling point of 77.25 K, the critical temperature is $T_c = 126.26$ K, the critical pressure is $P_c = 3.40$ MPa and the triple point temperature is $T_{\text{trippel}} = 63.15$ K.

2.3 Cryostat

Cryostat is a generic term for an unit containing the necessary equipment to maintain/gain a requested cryogenic temperature. As an example the cryostat could be a storage tank for a cryogenic fluid or a housing for a detector. It is of primary importance to minimize the heat transfer from the surroundings to the cold mass/fluid. The cryostat is usually built from a thermal shield covered with insulation around the cold mass/fluid, and a vacuum chamber

surrounding the thermal shield. Vacuum will reduce the number of molecules available to transport energy from the ambient temperature. Depending of the degree of vacuum, it will reduce the heat transfer by conduction and convection in the residual gas. Heat transfer by radiation will therefore be the dominating mode of heat transfer from the surroundings to the thermal shield. The different modes of heat transfer are explained in Chapter III.

2.4 Two-phase flow

Addition of heat will change the nitrogen flow, in the heat exchanger tubes housed in the Barrel Liquid Argon Calorimeter, to a two-phase flow of liquid and vapour. The two-phase flow is very complex and many distinct flow patterns can be identified.

Characteristic forms of flows in a horizontal tube are; bubbly flow, plug flow, stratified flow, wavy flow, slug flow, and annular flow. Figure 2.2 shows sketches of the different flow patterns. The actual flow patterns encountered depends primarily of the mass flow rate, the vapour quality, and the void fraction. The void fraction is the ratio of the cross section area occupied by the gas and the total cross section area, and can only be determined by measurements.

The gravity force perpendicular to the flow influences the flow regime as well, especially at low flow rates. In bubbly and slug flows the bubbles tend to flow in the upper half of the pipe, and at low liquid and gas flows rates there is a possibility of stratified flow with the liquid flowing along the bottom of the tube. As the vapour velocity increases, waves forms on the liquid surface and can become large enough to form liquid slugs which wet the upper periphery of the pipe wall. A still higher vapour velocity results in annular flow, or the liquid can be dispersed into droplets entrained in the vapour flow. [2]

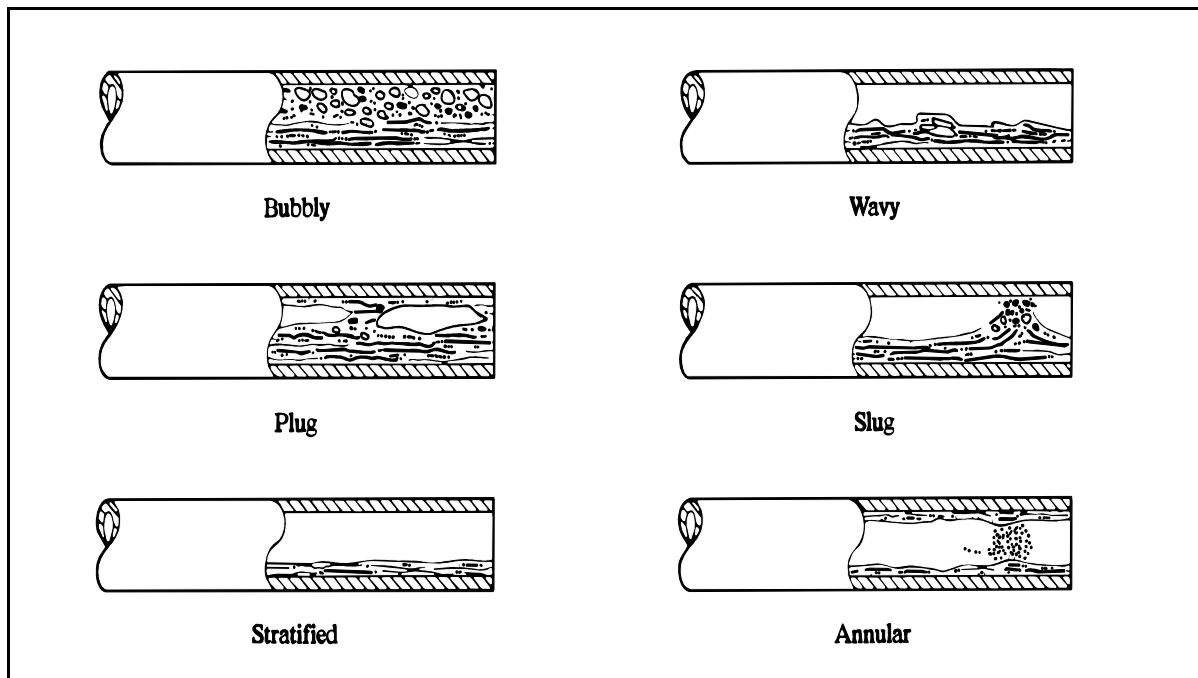


Figure 2.2 Flow patterns for horizontal cocurrent flow of a liquid and a gas or vapour. [2]

Chapter III

Heat transfer

This chapter presents the theory behind the different modes of heat transfer appearing in this thesis, and shows how the heat transfer can be calculated. All the heat transfer equations are for steady state conditions and radial temperature dependence only. Theory and formulas presented in this chapter are in accordance with [2]. The first section is about heat transfer in general, while the following sections describe in detail the different modes of heat transfer.

3.1 General

When an object is placed in surroundings with a different temperature than the object itself, the object and the surroundings will experience a change in the internal energy. The interaction is commonly described as heat transfer. The heat transfer is not a function of the temperature gradient only, it is as well a function of time, thermophysical properties, size, geometric shape, relative movement and flow conditions. Conduction, radiation and convection are the three main modes of heat transfer.

Heat transfer by conduction and radiation occur on a molecular or subatomic scale. In thermodynamics, heat is defined as energy transfer due to temperature gradients or differences. Consist with this viewpoint, thermodynamics recognize only these two modes of heat transfer, but convection is a third one.

Strictly speaking, convection is the transport of energy by motion of a medium. A fluid, by virtue of its mass and velocity, can transport momentum. In addition, by virtue of its temperature, it can transport energy. In the study of heat convection is dimensionless groups commonly used.

3.2 Dimensionless groups

Experimental data for convective heat transfer coefficients, as well as the results of analysis, can be conveniently and concisely organized as relationships between dimensionless groups of pertinent variables. In the case of natural convection flow, see Section 3.4, there can be found four independent dimensionless groups. Which are the Nusselt number, the Prandtl number, and two groups without individual names. This because both theory and experiments show that natural convection depends on the product of these two groups rather than on each group independently. The product is called the Grashof number. A further simplification is done by multiplying the Prandtl number with the Grashof number. The new dimensionless group is named the Rayleigh number. The following will show how these numbers can be calculated.

3.2.1 Nusselt number

The Nusselt number (Nu) is the expression for the dimensionless heat flux. It is expressed in terms of the convective heat transfer coefficient h_c , the characteristic length δ , and the thermal conductivity k , as:

$$Nu = \frac{h_c \times \delta}{k} \quad (3.1)$$

3.2.2 Prandtl number

The Prandtl number (Pr) is the dimensionless group of the fluid properties. Equation 3.2 shows how the number is calculated.

$$Pr = \frac{\mu \times C_p}{k} \quad (3.2)$$

Here is μ the dynamic viscosity coefficient, C_p is the heat capacity at constant pressure, and k is the thermal conductivity.

3.2.3 Grashof number

The Grashof number (Gr) is expressed in terms of free fall acceleration g , the coefficient of thermal expansion β , the density ρ , the characteristic length δ , the temperature difference ΔT , and the dynamic viscosity μ , as:

$$Gr = \frac{g \times \beta \times \rho^2 \times \delta^3 \times \Delta T}{\mu^2} \quad (3.3)$$

Since the Grashof number is a multiplication of two independent dimensionless groups, it is a convenient expression rather than an expression of a characteristic.

3.2.4 Rayleigh number

The Rayleigh number (Ra) appear when multiplying the Grashof number with the Prandtl number. Equation 3.4 shows the Rayleigh number.

$$Ra = Gr \times Pr = \frac{g \times \beta \times \rho^2 \times \delta^3 \times \Delta T \times C_p}{\mu \times k} \quad (3.4)$$

Where g is the free fall acceleration, β is coefficient of thermal expansion, ρ is the density, δ is the characteristic length, ΔT is the temperature difference, C_p is the heat capacity at constant pressure, μ is the dynamic viscosity coefficient and k is the thermal conductivity. As for the Grashof number is the Rayleigh number not an expression of a characteristic.

3.3 Conduction

Conduction is heat transfer due to molecules travelling a very short distance before colliding with another molecule and exchanging energy. On a microscopic level, the physical mechanisms of conduction are complex, encompassing such varied phenomena as molecular collisions in gases, lattice vibrations in crystals, and flow of free electrons in metals. This section takes a closer look at conduction across cylindrical shells.

3.3.1 Conduction across cylindrical shells

For steady one-dimensional conduction across cylindrical shells is the temperature a function of the radial coordinate r only. The total heat transfer Q is calculated as shown in Equation 3.5. In the equation L is the length of a cylinder with an inner diameter D_i and an outer diameter D_o .

$$\dot{Q} = \frac{2\pi \times k \times L \times (T_i - T_o)}{\ln(D_o/D_i)} \quad (3.5)$$

The equation assumes the inner surface maintained at a temperature T_i , and the outer surface maintained at a temperature T_o , while k is the thermal conductivity of the shell.

3.4 Natural Convection

Convective heat transfer, convection, is heat transfer executed by the flow of a fluid. The fluid acts as carrier for the energy it exchanges with a solid wall. In general, the heat transfer by convection is expressed as Equation 3.6.

$$\dot{Q}_s = h_c \times A \times (T_s - T_e) \quad (3.6)$$

The heat transfer Q_s is expressed in terms of the convective heat transfer coefficient h_c , the surface area of the wall A , the surface temperature T_s , and the free-stream temperature T_e .

In natural convection the fluid flows “naturally” (by itself). The motion is caused by density variations, as it is driven by the effect of buoyancy. The effect is distributed throughout the fluid, and is associated with the general tendency of fluids to expand when heated. The convective heat transfer coefficient of natural convection flows is calculated by the way of Nusselt number. Rearranging the Nusselt number equation, Equation 3.1, gives an expression for the convective heat transfer coefficient h_c .

$$h_c = \frac{Nu \times k}{\delta} \quad (3.7)$$

Here Nu is the Nusselt number, k is the thermal conductivity and δ is the characteristic length. All thermal properties of Equation 3.7 should be calculated at the mean film temperature, an average between the surface temperature and the free-stream temperature, while the Nusselt number should be calculated from known correlations. This section shows theory behind natural convection flow on a horizontal cylinder, natural convection on objects of arbitrary shape, and natural convection between concentric cylinders.

3.4.1 Natural convection flow on a horizontal cylinder

Churchill and Chu correlated the average Nusselt number for the laminar natural convection flow around an isothermal cylinder positioned horizontally in a fluid reservoir as:

$$\overline{Nu}_D = 0.36 + \frac{0.518 \times Ra_D^{1/4}}{[1 + (0.559/Pr)^{9/16}]^{4/9}} ; Ra_D < 10^9 \quad (3.8)$$

The average Nusselt number and the Rayleigh number are both based on the diameter as characteristic length, and the equation is valid in the entire Prandtl number range. Given the average Nusselt number, the average convective heat transfer coefficient is calculated from Equation 3.7, and the total heat transfer from Equation 3.6.

3.4.2 Natural convection on objects of arbitrary shape

For a laminar natural convection boundary layer of any shape, in fluids other than those which $Pr \ll 1$, Lienhard proposed the average Nusselt number as:

$$\overline{Nu}_l \cong 0.52 \times Ra_l^{1/4} \quad (3.9)$$

Where the characteristic length l is the distance travelled by the boundary layer fluid while in contact with the object. As in the case of natural convection flow on a horizontal cylinder, the total heat transfer can now be calculated from Equation 3.6, by the way of Equation 3.7.

3.4.3 Natural convection between concentric cylinders

The correlation recommended by Raithby and Hollands for natural convection between concentric cylinders is in the form of an effective thermal conductivity for use in Equation 3.5. The expression for the effective thermal conductivity is:

$$\frac{k_{eff}}{k} = 0.386 \times \left(\frac{Pr}{0.861 + Pr} \right)^{1/4} \times Ra_{cyl}^{1/4} ; 10^2 < Ra_{cyl} < 10^7 \quad (3.10)$$

Where Pr is the Prandtl number and Ra_{cyl} is the Rayleigh number for cylinders, given from Equation 3.11:

$$Ra_{cyl} = \frac{[\ln(D_o/D_i)]^4}{L^3 \times (D_i^{-3/5} + D_o^{-3/5})^5} \times Ra_L \quad (3.11)$$

Where $L = (D_o - D_i)/2$ is the gap width.

3.5 Condensation

When a heat exchanger wall exposed to a vapour is cooled below the saturation temperature of the vapour, condensation will occur on the surface. In the case of pure vapour, the saturation temperature designates the temperature at which the phase change take place at a given pressure.

Two distinct modes of condensation are possible; film condensation, and drop by drop condensation. This section takes a closer look at laminar film condensation on horizontal tubes, so drop by drop condensation will not be explained further. Film condensation on the other hand is when the condensate wets the wall such that a complete film of liquid covers the wall, and the film thickness grows as it flows down the wall under the action of gravity.

3.5.1 Laminar Film Condensation on Horizontal Tubes

The average heat transfer coefficient for laminar condensation at a tube can be expressed as:

$$\bar{h} = 0.728 \times \left[\frac{(\rho_l - \rho_v) \times g \times h'_{fg} \times k_l^3}{\nu_l \times D \times (T_{sat} - T_w)} \right]^{1/4} \quad (3.12)$$

Where ρ is the density, g is the acceleration of gravity, h'_{fg} is the modified enthalpy of phase change (see Equation 3.13), k is the thermal conductivity, ν is the kinematic viscosity, D is the outer tube diameter, T_{sat} is the saturation temperature, and T_w is the outer wall temperature. The subscript l and v denotes liquid and vapour condition respectively.

The condensed liquid on the tube will be sub-cooled due to the temperature of the tube surface, and the surrounding gas will be superheated due to the saturation temperature. As a consequence the enthalpy of phase change is modified as:

$$h'_{fg} = h_{fg} + \left(0.683 - \frac{0.228}{Pr_l} \right) \times C_{pl} \times (T_{sat} - T_w) + C_{pv} \times (T_v - T_{sat}) \quad (3.13)$$

Where the first term is the enthalpy of phase change h_{fg} , the second term is the correction due to sub-cooling, and the third term is the correction due to superheating. In Equation 3.13 Pr is the Prandtl number, C_p is the specific heat at constant pressure, T_{sat} is the saturation temperature, T_w is the outer wall temperature, and T_v is the gas temperature. The subscript l and v denotes liquid and vapour condition respectively. All properties of Equation 3.12 and Equation 3.13 should be evaluated at the mean liquid film temperature, an average between the saturation temperature and the temperature of the wall, except h_{fg} which is evaluated at the saturation temperature, and C_{pv} which is evaluated at the mean vapour temperature.

3.6 Boiling

Boiling heat transfer occur when the temperature of a solid surface is sufficiently higher than the saturation temperature of the liquid with which it comes in contact. The solid-liquid heat transfer is accompanied by the transformation of some of the heated liquid into vapour, and by the formation of vapour bubbles, jets, and films. The vapour and the surrounding packet of heated liquid are carried away by the effect of buoyancy or by a combination of buoyancy and a forced flow. This section takes a closer look at internal forced convection boiling in an evaporation tube, which will be the case inside the heat exchanger tubes housed in the Barrel Liquid Argon Detector.

3.6.1 Internal forced convection boiling in an evaporation tube

In a horizontal evaporation tube, nucleate flow boiling dominates the overall process. The actual heat transfer is due to a combination of two closely interrelated effects. The first one is nucleate boiling from active sites on the heated wall, pool boiling, and the second is the direct sweeping of the heated surface by the liquid itself, the forced convection phenomenon. Under certain conditions a transition from nucleate boiling to film boiling may occur. A vapour film will blanket the wall, and the heat transfer coefficient will drop dramatically.

The heat transfer coefficient is strongly dependent of the heat flux, tube diameter, mass flow, flow regime, and fluid properties as density and saturation temperature. The heat transfer coefficient of internal forced convection boiling is therefore very hard to predict analytically, so experimental data should be produced.

3.7 Combined modes of heat transfer

In most cases the total heat transfer is a combination of several modes. For combined modes of heat transfer, the heat transfer is expressed in terms of the overall heat transfer coefficient U , as shown in Equation 3.14.

$$\dot{Q} = U \times A \times (T_i - T_o) \quad (3.14)$$

Here is A the surface area, T_i is the inner temperature and T_o is the outer temperature. In this thesis is the combined mode of heat transfer needed to evaluate the heat transfer over the heat exchanger tubes in the Barrel Liquid Argon Calorimeter.

3.7.1 Combined heat transfer over a heat exchanger tube

In the case of a heat exchanger tube cooled by a forced two-phase flow and placed inside a gas volume, there will be boiling heat transfer from the coolant to the wall, heat transfer by conduction through the tube wall, and heat transfer by natural convection or condensation from the heat exchanger surface to the surrounding gas (dependent of the temperature of the outer tube surface compared to the saturation temperature of the gas). For this combination of heat transfer modes, the overall heat transfer coefficient is expressed as:

$$\frac{1}{U \times A} = \frac{1}{(2\pi \times L \times r \times h)_i} + \frac{\ln(r_o/r_i)}{2\pi \times L \times k} + \frac{1}{(2\pi \times L \times r \times h)_o} \quad (3.15)$$

Where r is the radius, L is the tube length, k is the thermal conductivity of the tube wall and h is the heat transfer coefficient (natural convection, condensation and boiling). The subscript i and o denotes inner and outer respectively. For use of Equation 3.15 in combination with Equation 3.14, should the outer temperature be the free-stream temperature and the inner temperature should be the temperature of the boiling nitrogen flow.

Chapter IV

The Barrel Liquid Argon Calorimeter

In order to make a good model for cool-down simulations, it is of primary importance to get a good understanding of how the Barrel Liquid Argon Calorimeter is built and what functions the components have. This chapter will therefore take a closer look at the true nature of the calorimeter. The sections show details and dimensions, with figures and data taken from [3]. Additional figures of the barrel are shown in Appendix B.

The Barrel Liquid Argon Calorimeter has a cylindrical geometry built from two identical half-barrels. Each half-barrel is made of two distinct devices and their services. It is the Accordion Calorimeter and the Presampler.

Figure 4.1 shows one of the half-barrels, and gives a good view of the general layout. From the figure one can see that the accordion calorimeter is built from modules supported by rings on both sides; the external rings on the outer circumference, and the internal rings on the inner circumference. Each of these modules is built from accordion-shaped absorbers radially orientated around the horizontal axis of the calorimeter. The presampler will be divided into sections fixed to the inner circumference of the internal rings.

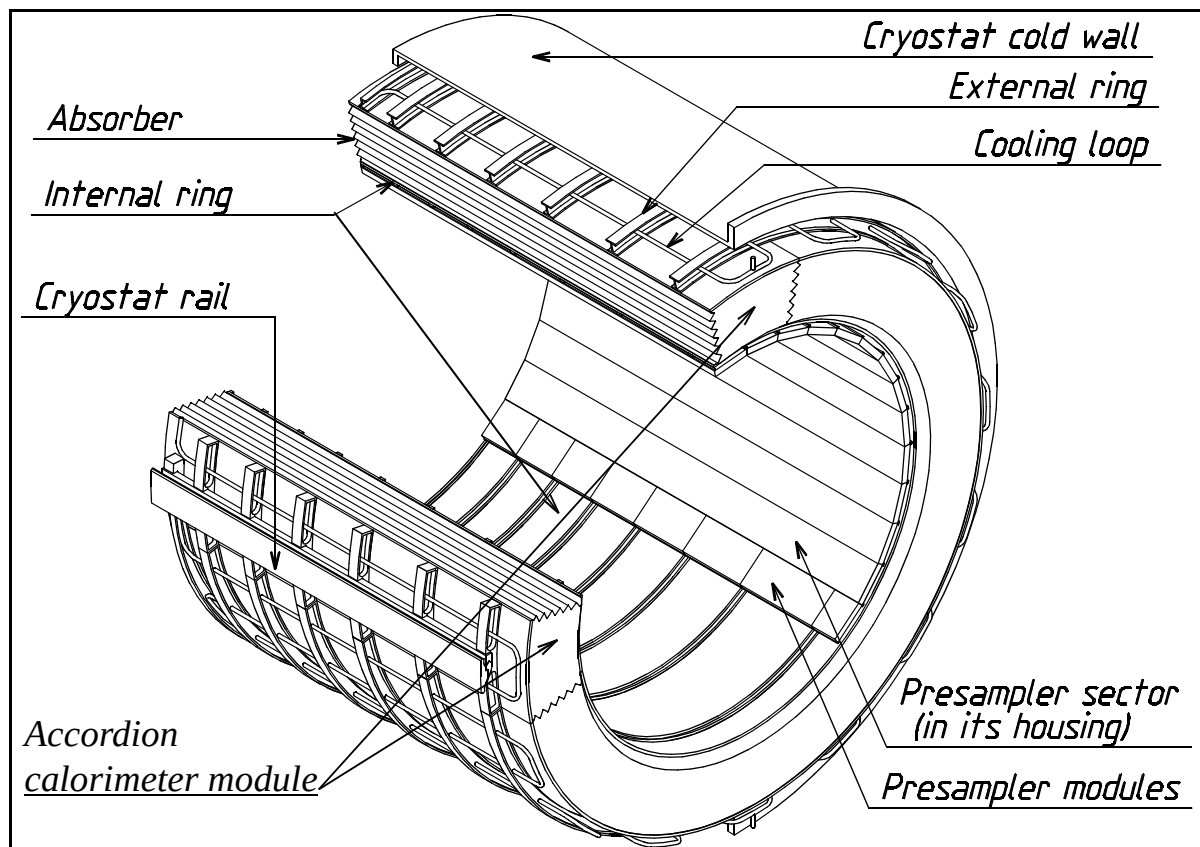


Figure 4.1 General layout of a half-barrel.

4.1 Cryostat

The barrel cryostat is built from two aluminium vessels with vacuum in between, and serves as the housing for the calorimeter. A warm vessel, which forms a hollow cylinder, contains the similarly shaped cold vessel. At each end of the cryostat there are symmetrically arranged signal feedthroughs, connecting the inside of the cold vessel to the outside. Figure 4.2 shows a layout of the barrel cryostat and the calorimeter, with the electronics indicated.

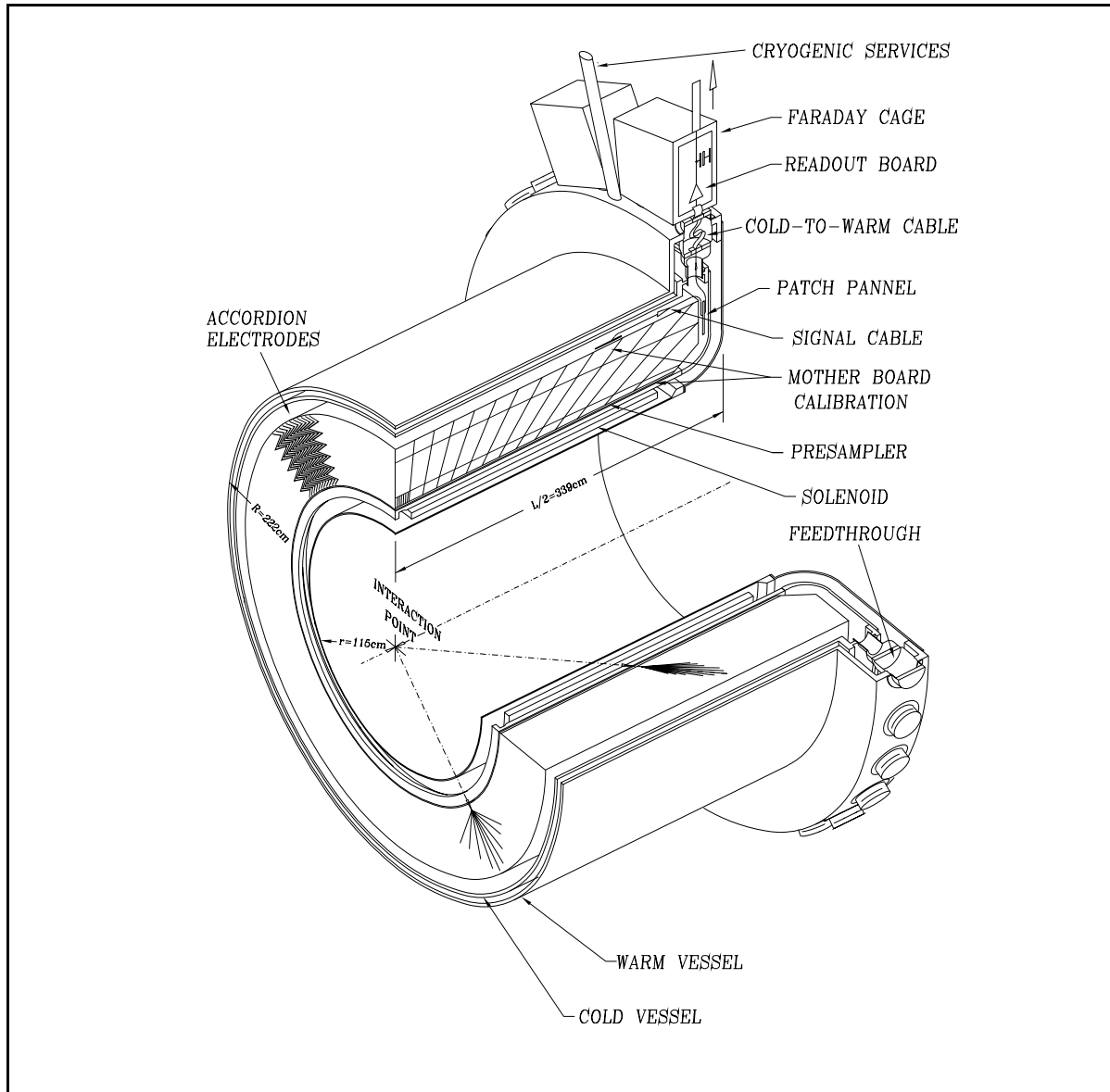


Figure 4.2 Perspective view of one half of the barrel cryostat.

Inside its vacuum the cryostat houses the Central Solenoid magnet, one of the magnets constituting the ATLAS Magnet System, which is not a part of the Barrel Liquid Argon Calorimeter. The solenoid is indicated in Figure 4.2, as well as in figures in Appendix B.

Because of fabrication reasons, both the cold and the warm vessel are built from an inner and an outer cylinder. An exploded view of the cryostat, identifying all its components, is shown in Appendix B.

4.2 Accordion calorimeter

Each half-barrel is built from 1024 radially arranged accordion-shaped absorbers. The absorbers are indicated in Figure 4.1 and Figure 4.2, while Figure 4.3 shows the geometry and the dimensions of one such absorber. The absorber is made of a lead sheet sandwiched between two stainless-steel plates (AISI 304), which are glued together by a layer of prepreg adhesive on each face. At the inner and the outer edge of the sandwich, it is encased in a precision bar made out of G10. The precision bars, referred to as the inner and the outer G10 bar, will be fixed to the support rings in order to hold the absorber in its place.

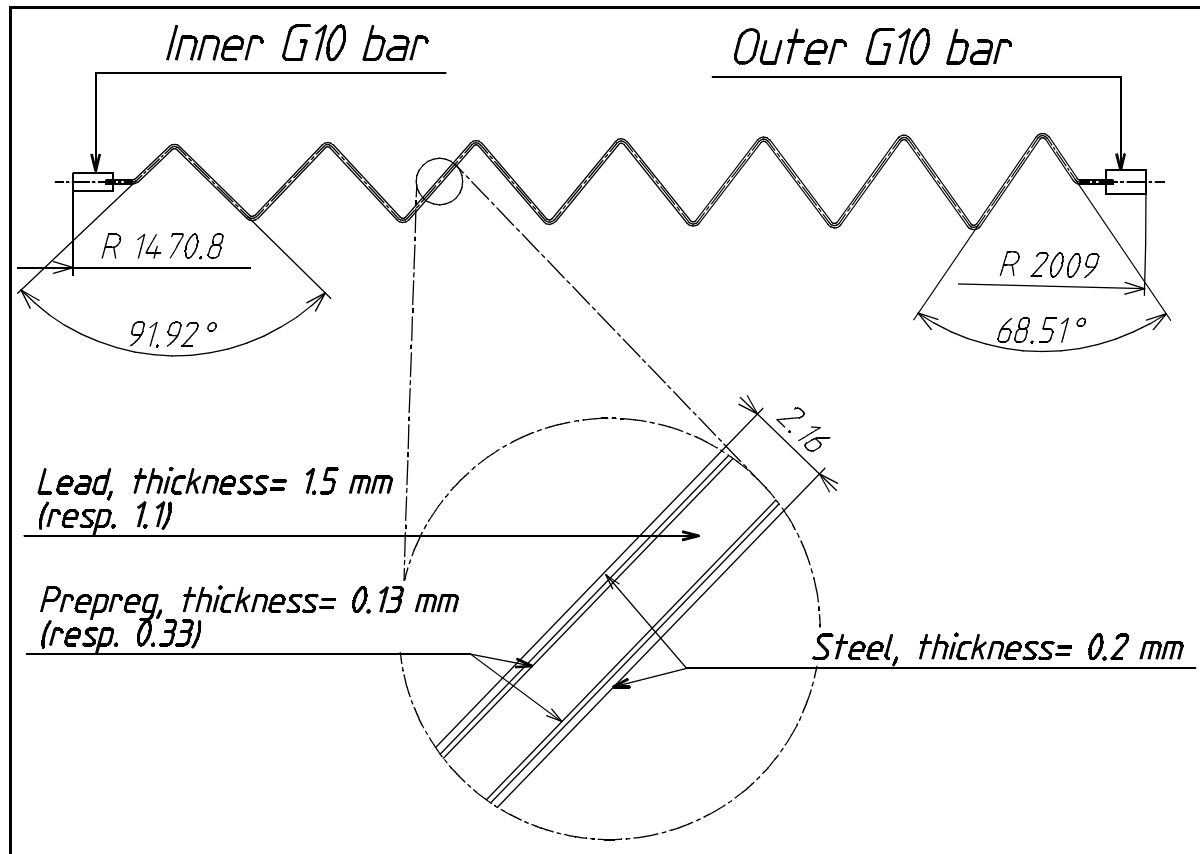


Figure 4.3 Accordion calorimeter absorber.

As already mentioned will the absorbers be built together as modules. Which is done for fabrication, supporting and connection purposes. Each half-barrel will consist of 16 such modules, making it a total of 32. Figure 4.4 shows one of the calorimeter modules, with two of its sixty four absorbers fixed to the rings.

4.3 Presampler

The presampler has a polygonal shape formed by 32 identical sectors per half-barrel. As seen from Figure 4.4, will two presampler sectors be fixed to the internal rings for each calorimeter module. Each sector is built from a thin glass-epoxy shell housing eight presampler modules of unequal length, which is indicated in Figure 4.1. Inside the presampler sectors, the modules are secured together by G10 bars.

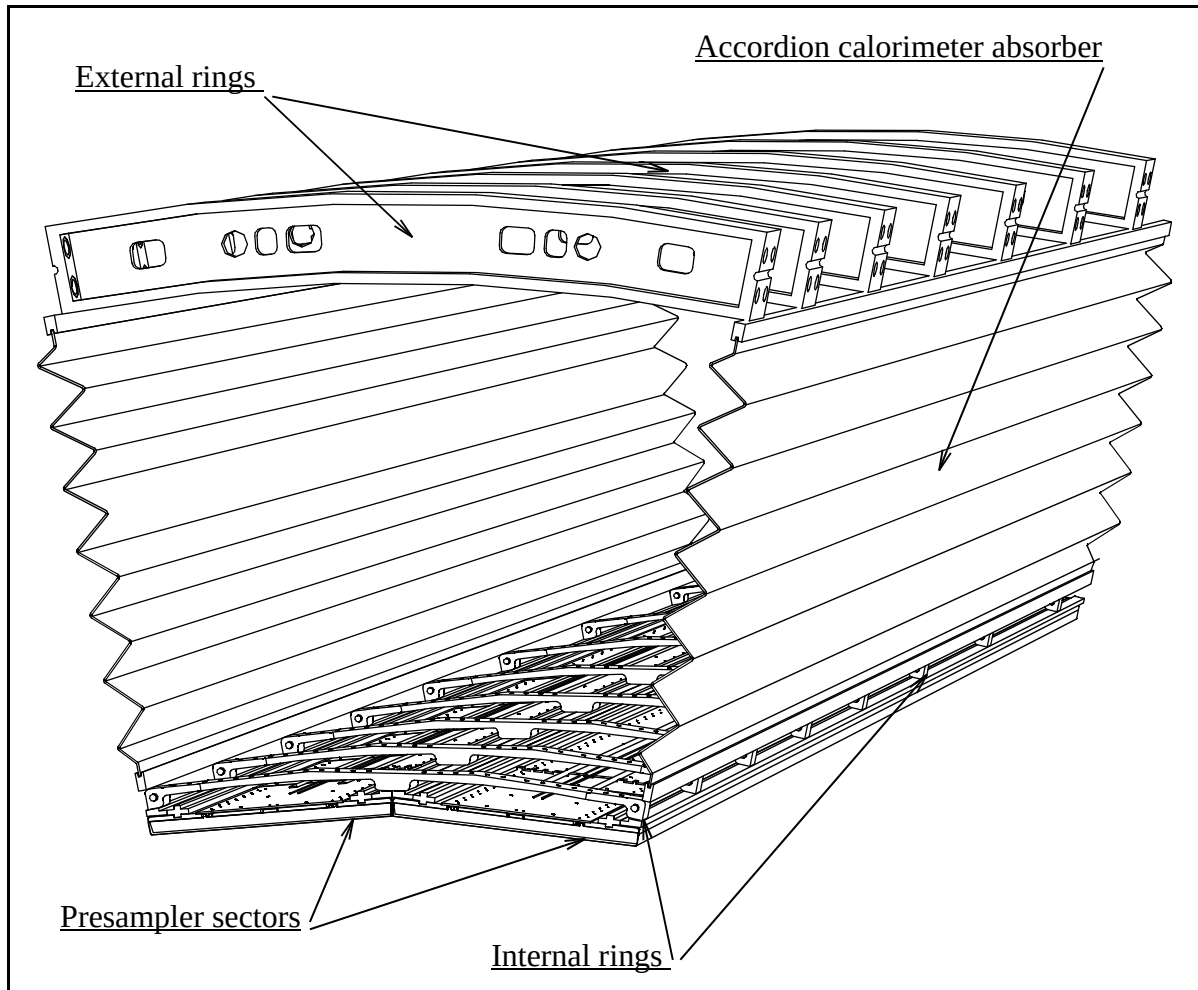


Figure 4.4 Accordion calorimeter module with two presampler sectors.

4.4 Rings

On both sides of the accordion calorimeter there are supporting rings; external rings on the outer circumference, and internal rings on the inner circumference. These rings are indicated in Figure 4.1 while Figure 4.4 shows the rings in detail.

4.4.1 External rings

The absorbers constituting one half-barrel are screwed onto seven stainless-steel rings (INOX 316LN). The aim of these supporting rings is to give the necessary rigidity to the whole half-calorimeter and to define precisely its geometry.

The external rings are made from 16 ring-pieces with an I-shaped beam. They will be separately mounted to the calorimeter modules, see Figure 4.4, and screwed together to form a complete circle. There are holes in the web of the ring-pieces to allow passage of cables and heat exchanger tubes.

4.4.2 Internal rings

The main purpose of the internal rings is to provide a small tension of the absorbers, such that all the absorbers are in extension at cold. An additional purpose is to fix the presampler to these rings.

There will be 8 rings, made of composite material, for each half-barrel. All the rings are built from 16 ring-pieces mounted separately to the calorimeter modules, as for the external rings.

4.5 Ancillary equipment

The most important service items of the Barrel Liquid Argon Calorimeter are the cooling circuits for operational temperature control and cool-down, and the cables and electronics for read out during the experiments. The cryostat rails are as well ancillary equipment of importance, transmitting the detector weight to the cryostat wall.

4.5.1 Cooling circuits

Six heat exchanger tubes will be placed inside the barrel cryostat. Their mission are temperature control of the liquid argon during operation and to allow cool-down. Four tubes are integrated with the calorimeter modules and runs through holes in the external rings, see Figure 4.1. The remaining two heat exchangers will be situated at each end of the barrel.

All the heat exchanger tubes will be equipped with a valve for control of the mass flow at the inlet, and a valve for control of the pressure at the outlet. The connection of the tubes to the outside will be made at each end of the cryostat, indicated as cryogenic services in Figure 4.2.

The cooling principle is based on evaporation of saturated liquid nitrogen in a closed circuit. In order to avoid a “dry-out”, the vapour quality of the nitrogen will be controlled to 10% at the exit of the heat exchanger tubes. The nitrogen will then be re-liquefied in a phase separator kept at 2.1 bar before pressurisation to 4.0 bar by a centrifugal pump. The nitrogen pressure in the heat exchanger tubes can then be controlled between 2.1 and 3.6 bar. A general flow sheet for the ATLAS Liquid Argon Calorimeter is presented in Appendix B.

4.5.2 Cables and electronics

The accordion calorimeter and the presampler will require ~120000 signal and calibration lines. The signal lines will be connected to the calorimeter with “mother boards” at the inner and the outer surface, see Figure 4.1. The same figure shows how the read-out and calibration signals are routed from the “mother boards” to the “patch panel” and then through the cold-to-warm feedthroughs located at each end of the cryostat.

4.5.3 Cryostat rails

In order to transmit the total weight of the calorimeter to the cryostat cold wall, there are built horizontal orientated cryostat rail on each side of the cryostat. They will connect the external rings with the wall, see Figure 4.1.

Chapter V

The simulation model

The cool-down of the Barrel Liquid Argon Calorimeter is a transient heat transfer problem of three dimensions. The heat is transferred by conduction, convection and radiation. A model for cool-down simulations which recreates the true nature of the process will be very complex, so a simplified model should be developed. This chapter presents such a model for use in cool-down simulations, and results from simulations based on this model will be presented in Chapter VI. The model is based on the detector dimensions and the data presented in Table 5.1.

<i>Material:</i>	<i>G10 and Kaptons</i>	<i>Cu</i>	<i>Pb</i>	<i>304</i>	<i>Total</i>
<i>Weight of detector (ton)</i>	14	5.0	71.0	20.0	110
<i>Density (ton/m³)</i>	1.7	9.0	11.7	7.9	-
<i>Delta H (kJ/kg)</i>	135.4	79.6	26.2	84.3	-
<i>Heat capacity (MJ)</i>	1896	398	1860	1686	5840
The heat capacity of the cryostat itself is approximately 1900 MJ					

Table 5.1 Barrel detector masses and heat capacities between 300 K and 80 K. [3]

5.1 Simplifications and assumptions

In order to make a model of the Barrel Liquid Argon Calorimeter for use in cool-down simulations, simplifications and assumptions are done. This section lists all simplifications and assumptions made for the cool-down model presented in this chapter, while Chapter VII discuss their validity.

- **Limited cooling capacity.** In order to not exceed a temperature gradient of 50 K between any parts of the detector structure, the energy output of the heat exchangers is regulated. Which means the energy output can theoretically be higher, but will be regulated with the nitrogen mass flow to stay within the 50 K limit.
- **Quasi time-independent.** To make the problem quasi time independent, the simulation is made step by step. Causing time/temperature dependent properties to be assumed constant over each time-step.
- **Uniform temperature in tangential direction.** The barrel is assumed uniform in the tangential direction. For the calculation purpose, the heat exchanger tubes, the cryostat walls, the G10 bars, the accordion calorimeter and the presampler, will have an uniform temperature in the tangential direction.
- **Uniform temperature in longitudinal direction.** The barrel is assumed uniform in the longitudinal direction as well. As for the tangential direction, the heat exchanger tubes,

the cryostat walls, the G10 bars, the accordion calorimeter and the presampler, have a uniform temperature in the longitudinal direction.

- **No end effects.** There are no end effects since the barrel is assumed uniform in the longitudinal direction.
- **Separated gas volumes.** The gas located in the Barrel Liquid Argon Calorimeter is assumed to be three separate regions without “communication” (no circulation of the gas between the separated volumes). The three regions are; the gas space between the outer G10 bar and the outer cryostat wall, the gas space between the presampler and the inner G10 bar, and the gas spaces between the cryostat wall and the presampler. The three volumes are indicated as "outer gas gap", "gap 1" and "gap 2" in Figure 5.2 and Figure 5.3.
- **Two separated parts of the cryostat.** The cold cryostat vessel is assumed to be built from two independent parts without “communication” (no conductive heat transfer between the two cryostat parts), see Appendix B for a view of the separated cryostat parts. The end walls of the cold vessel, the cold bulk head, are integrated as a part of the outer cold vessel, which means that the mass and the surface area of the end walls are added to this part of the cryostat.
- **No cryostat rails.** The cryostat rails connecting the outer cold cryostat wall with the external rings, see Figure 4.1 for a view of the location, are neglected. As a consequence, the heat transfer by conduction from the cryostat wall to the external rings is neglected, and likewise the heat capacity of the rails.
- **No internal rings.** The internal rings located in the gas space between the inner G10 bar and the presampler are neglected, see Figure 4.1 and Figure 4.4 for the location. As for the cryostat rails is the heat transfer by conduction through the rings ignored, and likewise the heat capacity.
- **Only heat transfer by convection in the "outer gas gap".** All parts located in the gas space between the outer cold cryostat wall and the outer G10 bar are assumed to be affected by convective heat transfer only. Which means that the heat transfer by conduction between the external rings and the outer G10 bar is neglected. Likewise the conduction in contact spots between cabling and other parts of the detector.
- **All cabling are located in the "outer gas gap".** All the cables and the electronics are assumed to be located in the gas space between the outer G10 bar and the cryostat wall.
- **Conduction in the “trapped” gas.** The gas “trapped” in the accordion calorimeter and the presampler is “solidified”. Indicating no natural convection in the “trapped” gas, only conduction.
- **No heat transfer by radiation.** The heat transfer by radiation inside the cryostat is neglected. Meaning no heat transfer by radiation between the cryostat parts.
- **Linear heat leaks.** The heat leaks into the barrel cryostat are assumed to have a linear temperature depends.
- **No heat leak to the inner cold cryostat wall.** There is assumed no heat leak into the inner cold cryostat vessel.

- **Constant gas pressure.** The pressure in the cryostat is assumed constant during cool-down. In concrete, this means gas refill at each time-step. The mass of the gas present in the cryostat is determined for each time-step of the cool-down, as a function of the density and the constant volume occupied by the gas.

5.2 Dimensions and symmetry of the model

Figure 5.2 and Figure 5.3 give a view of the simulation model. Figure 5.2 shows a sketch of the longitudinal direction, showing one half, while Figure 5.3 shows a cross section sketch. The scale of these sketches is not correct, but it gives a good overview of the model. The external rings, the cabling and electronics, and the cryostat feedthroughs, are not drawn in these sketches. The equipment is situated in the gap between the outer cold cryostat wall and the outer G10 bar, the "outer gas gap".

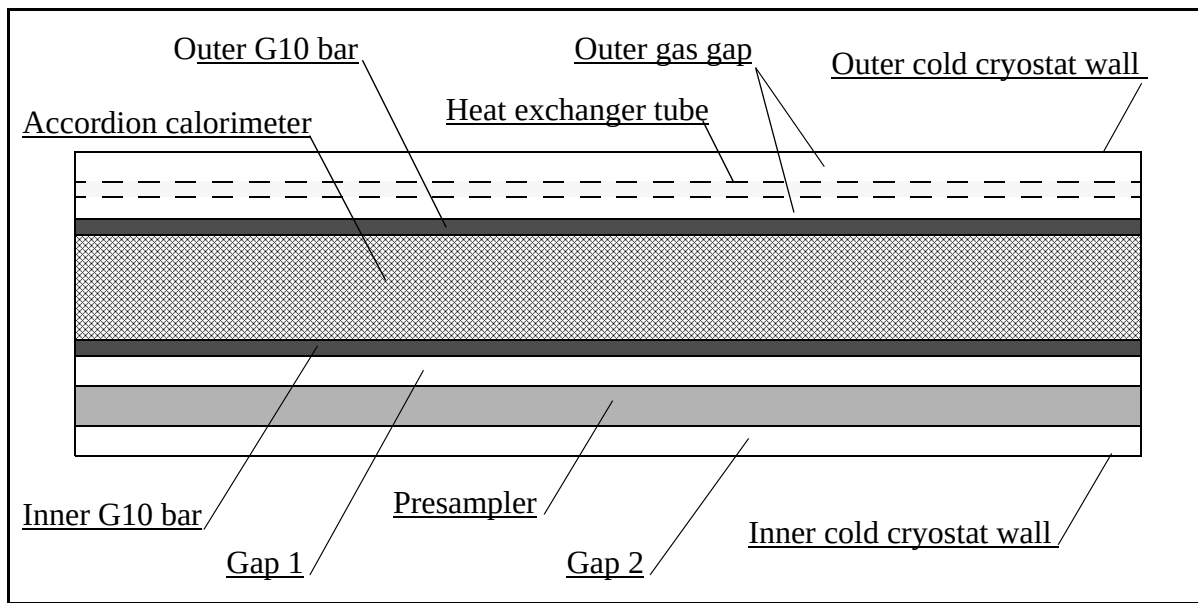


Figure 5.2 A cross section view in longitudinal direction of the barrel model, showing one half.

The barrel is now divided into nine regions, see Figure 5.2 and Figure 5.3. The radial dimensions of these regions are listed below. All lengths are for 293 Kelvin [3].

- | | |
|---|-----------------------------|
| 1. Outer cold cryostat wall. | (Inner radius = 2140.0 mm) |
| 2. Free space filled with gas (Outer gas gap). | (2140.0 mm > r > 2009.0 mm) |
| 3. Outer G10 bar. | (2009.0 mm > r > 1989.0 mm) |
| 4. Accordion calorimeter. | (1989.0 mm > r > 1490.8 mm) |
| 5. Inner G10 bar. | (1490.8 mm > r > 1470.8 mm) |
| 6. Free space filled with gas (Gap 1). | (1470.8 mm > r > 1448.3 mm) |
| 7. Presampler. | (1448.3 mm > r > 1413.3 mm) |
| 8. Free space filled with gas (Gap 2). | (1413.3 mm > r > 1390.0 mm) |
| 9. Inner cold cryostat wall. | (Outer radius = 1390.0 mm) |

The symmetry of revolution is broken in region 2 by:

- a. **External rings.** Seven rings are mounted to the "outer G10 bar" on each half-barrel.
- b. **Heat exchanger tubes.** The heat exchanger tubes are parallel to the beam axis, and runs through holes in the external rings.
- c. **Cables and electronics.** Mainly signal and calibration lines.
- d. **Feedthroughs.** At each end of the structure there are feedthroughs connecting the inside of the cold vessel with the outside.

The nine regions listed above, the external rings, the heat exchanger tubes, and cables and electronics, will be referred to as "detector parts". Making it a total of 12 "detector parts".

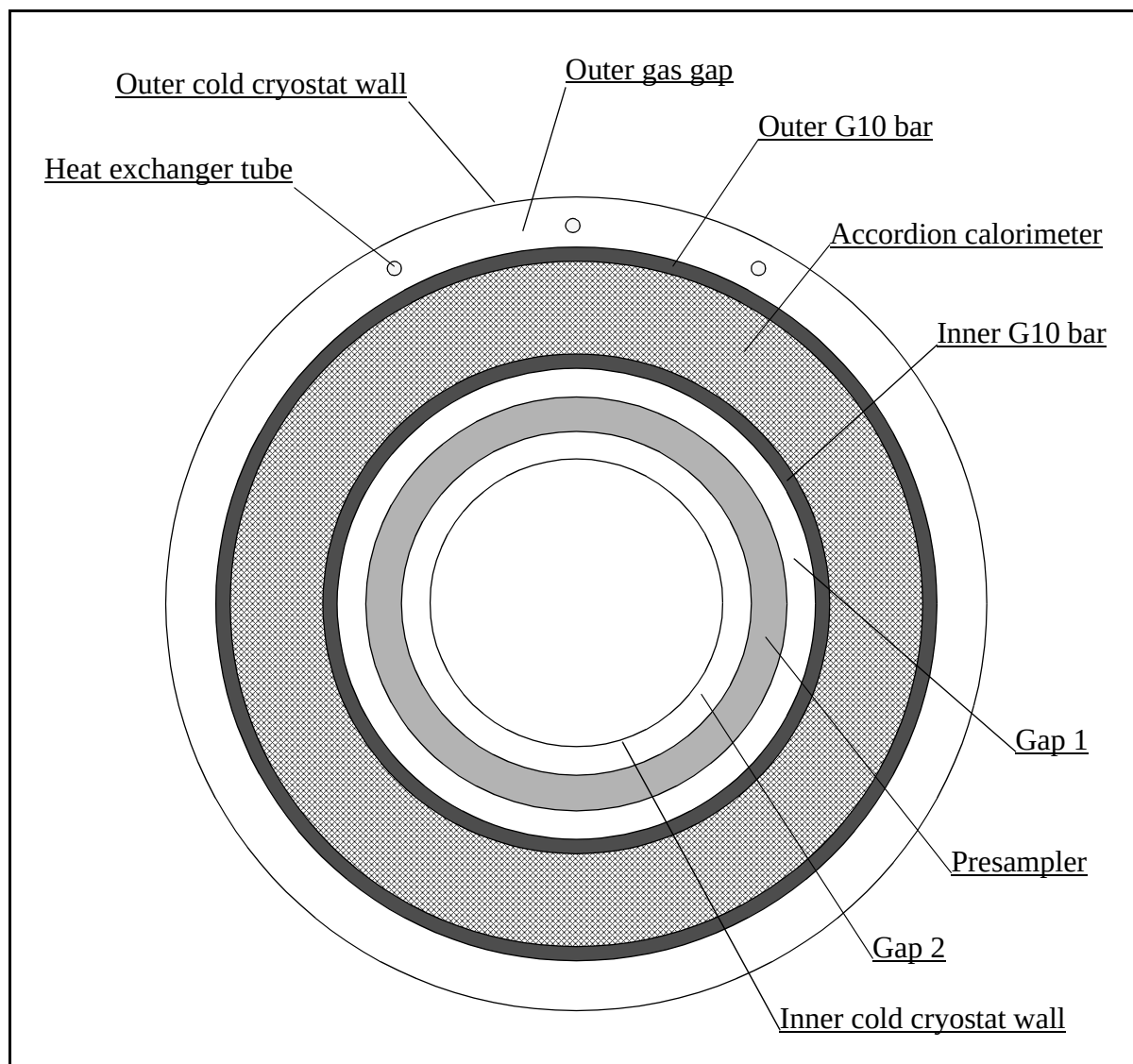


Figure 5.3 A cross section view in the radial-plane of the barrel model.

The "heat exchanger tubes" are symmetrically arranged all around the barrel, not just at the top as indicated in Figure 5.3. The tubes in the figure are drawn to indicate the location related to the other "detector parts".

5.3 The calculation

This section gives an explanation of how the cool-down is simulated, and describes the calculation procedure. Figure 5.4, with the same cross section view as Figure 5.2, shows all temperatures and heat transfers appearing in the model. Explanation of the symbols follows.

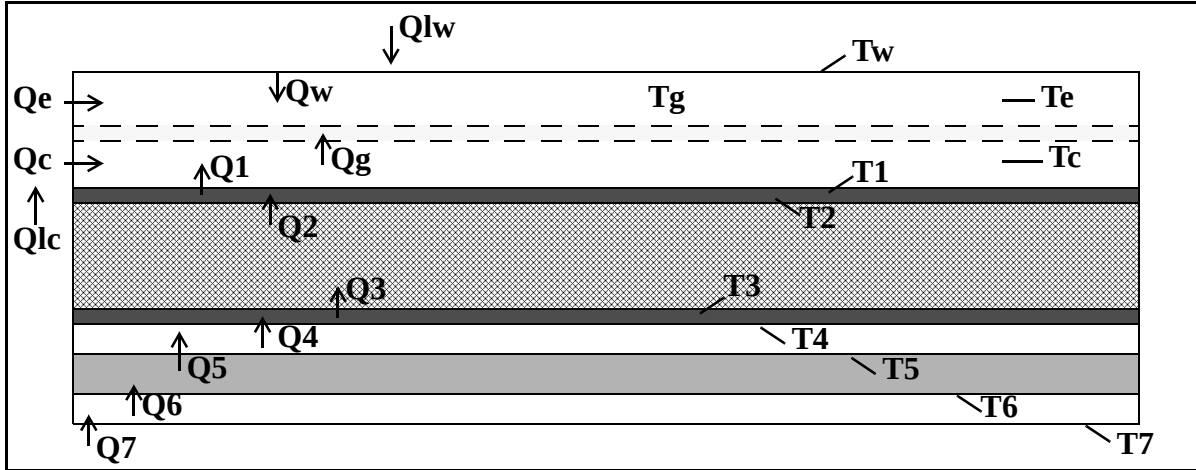


Figure 5.4 Heat transfers in the Barrel Liquid Argon Calorimeter model.

The heat transfers appearing in Figure 5.4 are:

- Q_g - The heat transfer from the "outer gas gap" to the "heat exchanger tubes".
- Q_w - The heat transfer from the "outer cold cryostat wall" to the "outer gas gap".
- Q_e - The heat transfer from the "external rings" to the "outer gas gap".
- Q_c - The heat transfer from the "cables and electronics" to the "outer gas gap".
- Q_1 - The heat transfer from the "outer G10 bar" to the "outer gas gap".
- Q_2 - The heat transfer from the "accordion calorimeter" to the "outer G10 bar".
- Q_3 - The heat transfer from the "inner G10 bar" to the "accordion calorimeter".
- Q_4 - The heat transfer from the "gap 1" to the "inner G10 bar".
- Q_5 - The heat transfer from the "presampler" to the "gap 1".
- Q_6 - The heat transfer from the "gap 2" to the "presampler".
- Q_7 - The heat transfer from the "inner cold cryostat wall" to the "gap 2".
- Q_{lw} - The heat leak to the "outer cold cryostat wall".
- Q_{lc} - The heat leak to the "cables and electronics".

The temperatures appearing in Figure 5.4 are:

- T_g - Average gas temperature in the "outer gas gap".
- T_w - Average temperature in the "outer cold cryostat wall".
- T_e - Average temperature in the "external rings".
- T_c - Average temperature in the "cables and electronics".
- T_1 - Surface temperature of the "outer G10 bar".
- T_2 - Outer surface temperature of the "accordion calorimeter".
- T_3 - Inner surface temperature of the "accordion calorimeter".
- T_4 - Surface temperature of the "inner G10 bar".
- T_5 - Outer surface temperature of the "presampler".
- T_6 - Inner surface temperature of the "presampler".
- T_7 - Average temperature of the "inner cold cryostat wall".

With the described heat transfers and temperatures, the cool-down simulation is implemented with the following procedure.

1. Initiation.

All temperatures are initiated to 300 K for calculation of the first time-step.

2. Calculation of heat transfers, Q .

Q_g - To calculate the heat transfer to the "heat exchanger tubes", the calculation method presented in Section 3.7.1 'Combined heat transfer over a heat exchanger tube' is used. In the case of non condensing mode, the outer convective heat transfer coefficient is calculated as showed in Section 3.4.1 'Natural convection flow on a horizontal cylinder'. In the case of condensation, the outer heat transfer coefficient is calculated in accordance with Section 3.5.1 'Laminar Film Condensation on Horizontal Tubes'. Based on the measurements done by D. Steiner and E. Ulrich Schlunder for boiling nitrogen flow in a horizontal tube [4], the inner heat transfer coefficient is set to 500 W/m-K for all modes. The heat transfer is then evaluated by iteration, using the inner and outer wall temperature in combination with the heat transfer. The heat transfer from the gas to the outer pipe wall must equal the heat transfer through the pipe wall which in turn must equal the heat transfer to the boiling nitrogen. The calculation is carried out with the average gas temperature T_g as the free-stream temperature.

Q_w , Q_e , Q_c and Q_1 - The heat transfer from the "detector parts" surrounded by the gas in region one, see list and figures in Section 5.2, is calculated with the method presented in Section 3.4.2 'Natural convection on objects of arbitrary shape'. It includes the heat transfer from the "outer cold cryostat wall", the "external rings", the "cables and electronics", and the "outer G10 bar".

Q_2 , Q_3 , Q_4 and Q_6 - The heat transfer through the "G10 bars", the "accordion calorimeter", and the "presampler", is calculated as showed in Section 3.3.1 'Conduction across cylindrical shells'.

Q_5 and Q_7 - From the "inner cold cryostat wall" to the "presampler" and from the "presampler" to the "inner G10 bar", the heat transfer is calculated as presented in Section 3.4.3 'Natural convection between concentric cylinders'. The gas properties are calculated at the characteristic temperature, which is the mean temperature of the cylinder surfaces.

Q_{lw} and Q_{lc} - As mentioned in Section 5.1 are the heat leaks assumed to be a linear function of the temperature in the temperature range of the cool-down. At operational temperature of the Barrel Liquid Argon Calorimeter (~ 90 K), the heat leaks into the barrel are 800 W to the "outer cold cryostat wall" (Q_{lw}), and 1000 W through the feedthroughs (Q_{lc}) [3]. The heat leaks are therefore calculated as a linear temperature dependent function between zero heat leak at 300 K, and the respectively heat leaks at 90 K. The heat leak from the feedthroughs is simulated to the "cables and electronics".

3. Calculation of average temperatures.

From the second step, all heat transfers are known. By using the total heat transfer from the “detector part”, the new average temperatures can be calculated. Equation 5.1 shows how.

$$T_{average_{new}} = T_{average} - \frac{\Sigma Q \times t}{C} \quad (5.1)$$

Here is $T_{average}$ the average temperature from the previous time-step, ΣQ is the sum of the heat fluxes of the “detector part” calculated, t is the time-step, and C is the thermal capacity. All thermal capacities are listed in Table 5.8, except from the thermal capacities of the three gas volumes. The thermal capacity of these “detector parts” is calculated using Equation 5.2.

$$C = C_p \times \rho \times V \quad (5.2)$$

Here is C_p the specific heat at constant pressure, ρ is the density, and V is the volume of the “gas gap”.

4. Calculation of absolute temperatures.

From the third step, the average temperature of all “detector parts” is known. The absolute temperatures can be calculated from these average temperatures. Which is done with the assumption of a linear temperature profile over the calculated “detector part”.

Tw, Te, Tc and T7 - For the "outer cold cryostat wall", the "external rings", the "cables and electronics", and for the "inner cold cryostat wall", the average temperature is used as the surface temperature as well.

T1, T2, T3, T4, T5 and T6 - Given the surface temperature of the "inner cold cryostat wall", the remaining six temperatures can be calculated. Equation 5.3 shows how these temperatures are calculated. The number I appearing in the equation indicates the calculated temperature (where I is a number from 1-6), and the calculation must be done in a decreasing order.

$$T_{new,I} = 2 \times T_{average_{new}} - T_{new,I+1} \quad (5.3)$$

$T_{average_{new}}$ is the new average temperature of the “detector part”, calculated from Equation 5.1, and $T_{new,I+1}$ is the new temperature of $I+1$.

5. Return to step 2.

All the new temperatures are now calculated, and the simulation can move on to the next time-step.

5.4 Characteristic lengths

The characteristic length appearing in the Nusselt number (Equation 3.1), the Grashof number (Equation 3.3), and the Rayleigh number (Equation 3.4), are of primary importance in heat transfer calculations. In the Grashof number and the Rayleigh number is the characteristic length of order three. Even a minor error could therefore affect the result. From the following explanations, the characteristic lengths of Table 5.5 can be derived. These are the numbers for use in the cool-down model.

<i>“Detector part”</i>	<i>Characteristic length</i>
<i>Heat exchanger tubes</i>	0.022 m
<i>Outer cold cryostat wall</i>	0.131 m
<i>Outer G10 bar</i>	0.131 m
<i>External rings</i>	0.100 m
<i>Cables and electronics</i>	0.050 m

Table 5.5 Characteristic lengths for use in the cool-down model.

5.4.1 Heat exchanger tubes

For natural convection flow on a horizontal cylinder, which is the case for the heat exchange between the gas and the "heat exchanger tubes", the characteristic length should be the tube diameter. This because the equation for calculation of the Nusselt number (Equation 3.8), under the given conditions, is correlated based on the diameter as characteristic length (see Section 3.4.1).

5.4.2 Outer cold cryostat wall and Outer G10 bar

The characteristic length of natural convection on objects of arbitrary shape should be the length of the boundary layer (see Section 3.4.2). Taking into account the small “gap” between the "outer cold cryostat wall" and the "outer G10 bar", compared to the diameter, will the convection loops most likely correspond to the dimension of the “gap”. The presence of different equipment in the “gap” (heat exchangers, cabling, electronics, etc.) and non-smooth surfaces, will create additional sources of mixing and enforce the process of loops separation, which will improve the heat transfer. Based on this, the characteristic length for calculation of the heat transfer from the gas to the "outer cold cryostat wall" and the "outer G10 bar" (Equation 3.9) is set to the “gap” width (0.131 m). [5]

5.4.3 External rings and Cables and electronics

The heat transfer from the "cables and electronics" and the "external rings" is also calculated with the equation for natural convection on objects of arbitrary shape (Equation 3.9). The characteristic length of the "external rings" is set to the height of the rings (0.10 m), while the characteristic length of the "cables and electronics" is set to the assumed diameter of the cable bundles (0.05 m).

5.5 Thermal Conductivities

The main masses in the Barrel Liquid Argon Calorimeter are Copper, Lead, Stainless Steel, G10 and Kaptons. The thermal conductivity of these materials shows small temperature dependence in the temperature range of the cool-down (300 K - 100 K). The thermal conductivity of these materials is therefore treated as constant in the simulation model. The estimated averages are presented in Table 5.6.

<i>Material:</i>	<i>G10 & Kaptons</i>	<i>Copper</i>	<i>Lead</i>	<i>Steel</i>
<i>Thermal conductivity</i>	0.35 W/m-K	420 W/m-K	35 W/m-K	13 W/m-K

Table 5.6 Average thermal conductivities between 300 K and 100 K. [6,7]

Based on the thermal conductivities given in Table 5.6, the thermal conductivity of the different parts of the Barrel Liquid Argon Calorimeter is obtained. Table 5.7 shows the thermal conductivities in the radial direction, and the following sections give an explanation of how these numbers are derived.

<i>“Detector part”</i>	<i>Thermal conductivity</i>
<i>Heat exchanger tubes</i>	13.00 W/m-K
<i>G10 bars</i>	0.350 W/m-K
<i>Accordion calorimeter</i>	3.300 W/m-K
<i>Presampler (Argon)</i>	0.097 W/m-K
<i>Presampler (Helium)</i>	0.174 W/m-K

Table 5.7 Thermal conductivities for use in the cool-down model.

5.5.1 Heat exchanger tubes

The "heat exchanger tubes" are made out of pure stainless steel 304, and will have the thermal conductivity of the steel.

5.5.2 G10 bars

The "outer G10 bar" and the "inner G10 bar" are made of massive G10, and have the same thermal conductivity as G10.

5.5.3 Accordion calorimeter

For calculation of the thermal conductivity in the radial direction of the "accordion calorimeter", there is a method suggested by Guy Le Meur [6]. First the equivalent conductivity in the direction parallel to the absorber plates is calculated, then the "accordion effect" is taken into account. The calculation is done at the mean absorber radius and the mean thickness of the lead and prepreg layers is used. At the mean radius of the absorber, the thickness of one sandwich element (Lead, Prepreg, Steel and gas) is:

$$t_{absorber} = (2\pi \times R_{mean})/1024 \quad (5.4)$$

The equivalent conductivity in the direction parallel to the plates is then:

$$k_{eq} = [(t \times k)_{lead} + (t \times k)_{prepreg} + (t \times k)_{steel} + (t \times k)_{argon}]/t_{absorber} \quad (5.5)$$

Where t is the mean thickness of the respective layers of the absorber. Taking into account the “accordion effect” (increasing of the plate length), the thermal conductivity of the absorber part is obtained:

$$k_{absorber} = k_{eq}/(\sqrt{2}) \quad (5.6)$$

5.5.4 Presampler

The "presampler" is built such that it “traps” gas. This gas will experience little convection during the cool-down, and the gas is therefore “solidified”. Assuming only conduction in the “trapped” gas. The mean thermal conductivity of argon in the temperature range of the cool-down is 0.012 W/m-K while helium has a mean thermal conductivity of 0.115 W/m-K in the same temperature range. The main material in the "presampler" is G10. Based on the "presampler" structure, the thermal conductivity is assumed to have a contribution of 75% from the gas and 25% from G10, as showed in Equation 5.7.

$$k_{presampler} = 0.75 \times k_{gas} + 0.25 \times k_{G10} \quad (5.7)$$

Where k is the thermal conductivity, and the subscript denotes the substance.

5.6 Thermal capacities

Based on the numbers listed in Table 5.1, and the simplifications and assumptions made for the model, the thermal capacities listed in Table 5.8 are derived. There is a minor uncertainty in these numbers, caused by uncertainty of the weight of the actual “detector part”.

“Detector part”	<i>Thermal capacity</i>
<i>Outer cold cryostat wall</i>	6.07 MJ/K
<i>Cables and electronics</i>	5.13 MJ/K
<i>External rings</i>	1.53 MJ/K
<i>Outer G10 bar</i>	1.75 MJ/K
<i>Accordion calorimeter</i>	15.56 MJ/K
<i>Inner G10 bar</i>	1.29 MJ/K
<i>Presampler</i>	0.80 MJ/K
<i>Inner cold cryostat wall</i>	3.05 MJ/K
<i>Total</i>	35.18 MJ/K

Table 5.8 Thermal capacities for use in the cool-down model.

5.7 Surface areas and gas volumes

Table 5.9 lists surface areas for use in estimation of the heat transfer coefficients between the gas in the "outer gas gap" and "detector parts" housed in the "gap".

<i>"Detector part"</i>	<i>Total surface area</i>
<i>Outer cryostat wall</i>	104.44 m ²
<i>Outer G10 bar</i>	80.03 m ²
<i>Heat exchanger tubes</i>	15.47 m ²
<i>External rings</i>	60 m ²
<i>Cables and electronics</i>	40 m ²

Table 5.9 Surface areas of "detector parts" housed in the "outer gas gap".

The surface area of the "cables and electronics" is just an estimation. This surface area is of minor importance as long as it gives a reasonable heat transfer to the surrounding gas.

The volumes of the "gaps" appearing in the model are listed in Table 5.10. The exact volume should be of minor importance since the heat capacity of the gas is small compared to the heat capacity of the detector.

<i>"Gap"</i>	<i>Volume</i>
<i>Outer gas gap</i>	15 m ³
<i>Gap 1</i>	1.5 m ³
<i>Gap 2</i>	1.5 m ³

Table 5.10 "Gap" volumes for use in the cool-down model.

5.8 Fluid properties

Properties data are taken from [8] at 1.3 bar, and a temperature dependent function is derived from interpolation. Depending of the property, the describing function has an order between one and four. The maximum deviation from the data taken from [8] is three percent.

Chapter VI

The cool-down simulation

The main scopes of the cool-down simulation are:

- Evaluation of heat transfer coefficients (natural convection and condensation) for tube heat exchangers cooled by two-phase flow of liquid nitrogen and placed inside an argon/helium volume.
- Evaluation of temperature gradients in the cryostat during cool-down.
- Evaluation of cool-down time from ambient to operating temperature.

The cool-down time is defined as the time needed to cool down the Barrel Liquid Argon Calorimeter and its cryostat from the ambient temperature (300 K) to the temperature at which the cryostat can be filled with liquid argon (120 K, [9]). Due to the cool-down model, as described in Chapter V, will the "inner cold cryostat wall" always be the "detector part" with the highest temperature. The cool-down can therefore stop when this "detector part" reaches a temperature of 120 K. This chapter presents results from cool-down simulations, based on the model presented in Chapter V, for three different cool-down modes. There is one section for each of the above mentioned scopes of the cool-down simulation.

6.1 Cool-down modes

The Barrel Liquid Argon Calorimeter model for simulation of the cool-down, as described in Chapter V, was programmed in Fortran. Simulations were then carried out for three different modes: "argon", "helium", and "condensing argon", all modes with a barrel pressure of 1.3 bar. Appendix A shows the Fortran program code for the three different cool-down modes, while the following gives a definition of them.

When argon is involved, it is of primary importance to keep the temperature higher than the argon triple point temperature (83.8 K) to avoid formation of solid argon. From Table 6.1, which shows corresponding saturation temperatures for given argon and nitrogen pressures, one can see that a nitrogen pressure of 2.1 bar corresponds to a saturation temperature of 84.1 K. The lowest pressure of the nitrogen cooling-circuit will be at this pressure to keep above the argon triple point.

<i>Fluid</i>	<i>Pressure</i>	<i>Saturation Temperature</i>
Argon	1.3 bar	89.7 K
Nitrogen	3.6 bar	90.0 K
Nitrogen	2.5 bar	85.9 K
Nitrogen	2.1 bar	84.1 K
Argon triple point temperature = 83.8 K		

Table 6.1 Saturation temperatures.

As mentioned in Section 4.5.1 can the pressure of the two-phase nitrogen flow in the tube heat exchangers be controlled between 2.1 - 3.6 bar, where the highest pressure is used to avoid argon condensation at the "heat exchanger tubes". The three different cool-down modes are therefore given the following specifications:

- **"Argon"** - The Barrel Liquid Argon Calorimeter is charged with argon gas. The temperature of the two-phase nitrogen flow in the tube heat exchangers is set to 90.0 K, barely higher than the saturation temperature of the argon gas (89.7 K), to prevent condensation of argon on the tubes. The corresponding nitrogen pressure is 3.6 bar. The simulation is done with a time-step of 0.75 second.
- **"Helium"** - The Barrel Liquid Argon Calorimeter is charged with helium gas. In order to get a good comparison of the heat transfer coefficient of "helium" and "argon", the cool-down simulation is done with the same nitrogen temperature (90.0 K). The corresponding nitrogen pressure is 3.6 bar. The simulation is done with a time-step of 0.5 second.
- **"Condensing argon"** is when the Barrel Liquid Argon Calorimeter is charged with argon gas, and the temperature of the two-phase nitrogen flow in the tube heat exchangers is lower than the saturation temperature at the given argon pressure. The simulation is done with a pressure of 2.5 bar for the two-phase nitrogen flow. The corresponding temperature is 85.7 K. The main scope of this mode is evaluation of the heat transfer coefficient on the "heat exchanger tubes". All other "detector parts" are therefore simulated as for the "argon" mode. In other words, the condensed argon is assumed to totally evaporate in the "outer gas gap" without dripping down on other "detector parts". The simulation is done with a time-step of 0.75 second.

6.2 Heat transfer coefficients for the tube heat exchangers

The heat transfer coefficients for the tube heat exchangers located in the Barrel Liquid Argon Calorimeter are evaluated for temperatures between 300 and 110 K, where the appearing temperature is the bulk temperature of the gas surrounding the "heat exchanger tubes". Table 6.2 shows the evaluated heat transfer coefficients for all three modes at 300 and 110 K, while the following sections present the coefficients for the whole temperature range of the cool-down.

<i>Temperature</i>	<i>h - "argon"</i>	<i>h - "helium"</i>	<i>h - "condensing argon"</i>
300 K	9.2 W/m ² -K	29.6 W/m ² -K	3826.1 W/m ² -K
110 K	6.8 W/m ² -K	21.6 W/m ² -K	3415.5 W/m ² -K

Table 6.2 Evaluated heat transfer coefficients.

6.2.1 "Argon" and "helium"

The evaluated heat transfer coefficient for the "argon" and the "helium" mode can be seen from Figure 6.3. The coefficients are calculated in accordance with Section 3.4.1, and in the figure they are displayed as a function of the average gas temperature in the "outer gas gap" in the temperature range 300 - 110 K.

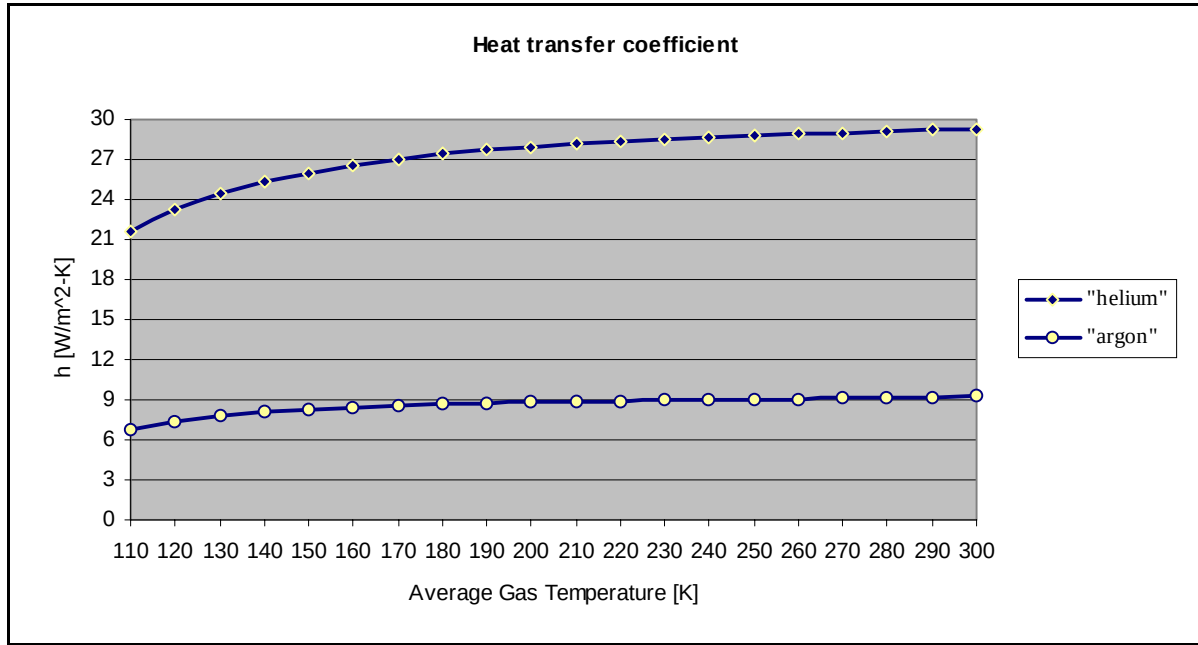


Figure 6.3 Heat transfer coefficients for natural convection flow on the "heat exchanger tubes".

As seen from Figure 6.3 is the heat transfer coefficient for "helium" higher than for "argon". This proves helium as a better fluid for heat transfer than argon. By comparison of the two heat transfer coefficients of "argon" and "helium", the ratio is found to be constant over the given temperature range. The ratio, as showed in Equation 6.1, is 3.2. Which means that helium is 3.2 times more efficient as a fluid for heat transfer than argon in this case of 'Natural convection flow on a horizontal cylinder'.

$$R_1 = \frac{h_{\text{helium}}}{h_{\text{argon}}} = 3.2 ; 300K > T_{\text{bulk}} > 110K \quad (6.1)$$

The heat capacity (C_p) and the thermal conductivity (k) of helium are approximately 10 times higher than for argon. The density (ρ) is on the other hand approximately 10 times less for helium than for argon. In total this makes the difference between the heat transfer coefficient of argon and helium, with helium as the best fluid for heat transfer.

6.2.2 "Condensing argon"

Figure 6.4 shows the evaluated heat transfer coefficient for the "condensing argon" mode. The coefficient is displayed as a function of the average gas temperature in the "outer gas gap" in the temperature range 300 - 110 K. The calculation is done in accordance with Section 3.5.1.

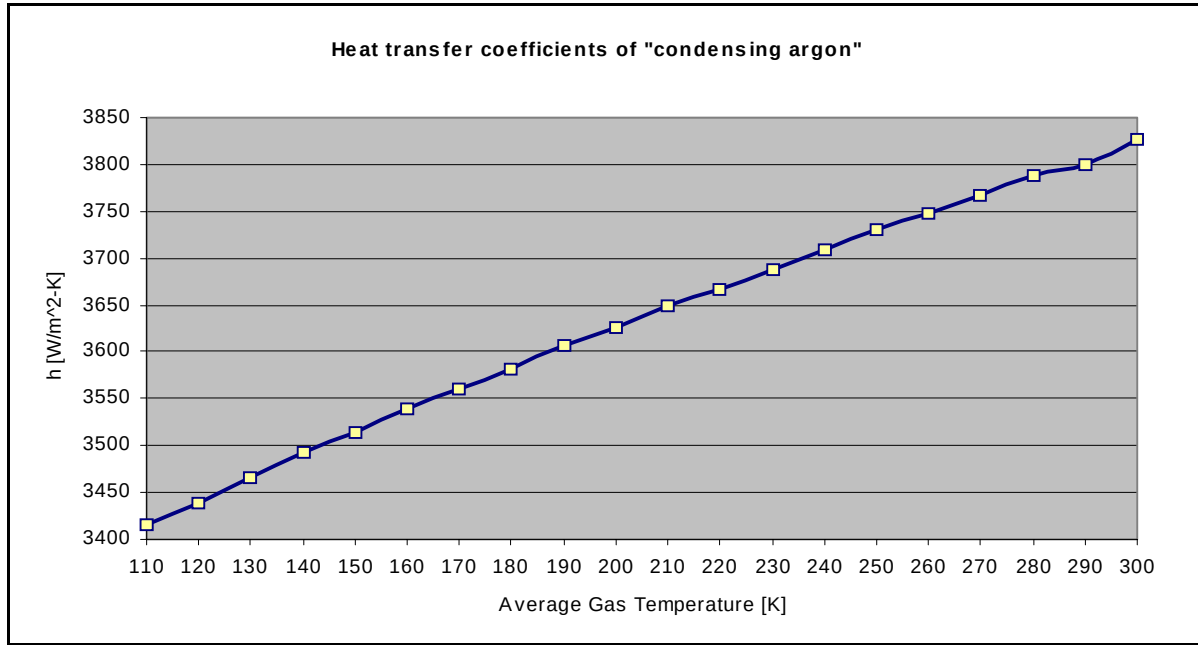


Figure 6.4 Heat transfer coefficient for laminar film condensation on the "heat exchanger tubes".

For the "condensing argon" mode the heat transfer coefficient shows a linear temperature dependence, and all over the temperature range is the heat transfer of the "heat exchanger tubes" much more efficient than for the "argon" mode. The ratio, as showed in Equation 6.2, between the heat transfer coefficient of "condensing argon" and "argon" has a value stretching from 415 (at 300 K) to 502 (at 110 K). This proves the heat transfer coefficient of the "condensing argon" mode to be 415 - 502 times higher than for the "argon" mode.

$$R_2 = \frac{h_{\text{"condensing argon"}}}{h_{\text{"argon"}}} \quad (6.2)$$

6.3 Temperature gradients

As mentioned in Section 5.1 are the cool-down simulations done with a limited cooling capacity of the "heat exchanger tubes". Which means that the energy output is regulated, in order to not exceed a temperature gradient of 50 K in any parts of the detector structure. In the model used for the cool-down simulation, see Chapter V, does the largest temperature gradient occur in the cold cryostat vessel. In the Barrel Liquid Argon model, T_w is the temperature of the "outer cold cryostat wall" and T_7 is the temperature of the "inner cold cryostat wall". The difference between these two temperatures is therefore the limiting factor of the cool-down simulation.

This section takes a closer look at the temperature gradients, and the limitation of the heat exchangers for the three cool-down modes. As already mentioned are not only the gradients an important output parameter, but the limited capacity of the "heat exchanger tubes" as well. Figure 6.6, Figure 6.7, and Figure 6.8, shows the temperature gradients of the "cryostat", the "detector surface", the "accordion" and the "sampler", for "argon", "helium" and "condensing argon" respectively. These gradients are defined as:

- The "**cryostat**" gradient is the temperature difference between the average temperature in the "inner cold cryostat wall" and the average temperature in the "outer cold cryostat wall", $T7 - Tw$.
- The gradient of the "**detector surface**" is the difference between the surface temperature of the "outer G10 bar" and the average temperature in the "outer gas gap", $T1 - Tg$.
- The "**accordion**" gradient is the temperature difference between the surface temperature of the "inner G10 bar" and the surface temperature of the "outer G10 bar", $T4 - T1$.
- The "**sampler**" gradient is the temperature difference between the inner surface temperature of the "presampler" and the outer surface temperature of the "presampler", $T6 - T5$.

The maximum temperature gradients and the limitation of the cooling capacity for the three cool-down modes can be seen from Table 6.5.

<i>Gradient</i>	<i>"argon"</i>	<i>"helium"</i>	<i>"condensing argon"</i>
<i>Cryostat</i>	48.0 K	48.9 K	48.0 K
<i>Detector surface</i>	16.9 K	15.5 K	16.9 K
<i>Accordion</i>	6.9 K	19.0 K	6.9 K
<i>Sampler</i>	3.7 K	5.5 K	3.7 K
<i>Limitation:</i>	<i>9 kW</i>	<i>24 kW</i>	<i>9 kW</i>
<i>Total nitrogen flow:</i>	<i>60 g/s</i>	<i>150 g/s</i>	<i>60 g/s</i>

Table 6.5 Maximum gradients during cool-down and the limited cooling capacity.

As a result of better thermal conductivity of helium than for argon, the capacity of the "heat exchanger tubes" can be regulated higher for the "helium" mode than for the "argon" mode. The energy output for the "helium" mode is 2.7 times higher than for the "argon" mode, and the cool-down is therefore approximately 3 times faster.

Assuming a 10% vapour quality of the nitrogen flow at the exit of the "heat exchanger tubes", the total nitrogen mass flow is approximately 60 g/s in the "argon" and "condensing argon" mode, and 150 g/s in the "helium" mode.

6.3.1 "Argon"

For the "argon" mode was the limitation of the "heat exchanger tubes" set to 9 kW. The gradients of the cool-down simulation, based on the limitation, can be seen from Figure 6.6. From the same figure one can see that the maximum temperature gradient of the "cryostat" will be 48 K, for the "accordion" the maximum temperature gradient will reach approximately 7 K, while the gradient of the "sampler" reaches a maximum of approximately 4 K.

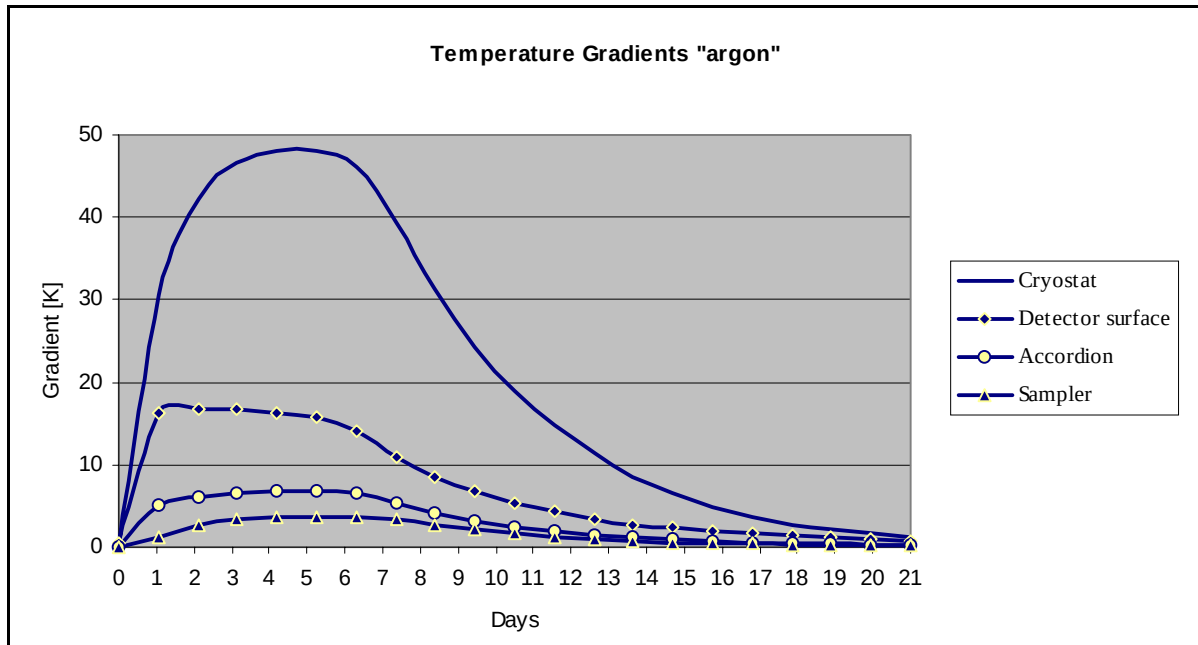


Figure 6.6 Temperature gradients during cool-down in "argon" mode.

6.3.2 "Helium"

Figure 6.7 shows the temperature gradients for the cool-down simulation in "helium" mode, with the limitation of the "heat exchanger tubes" at 24 kW.

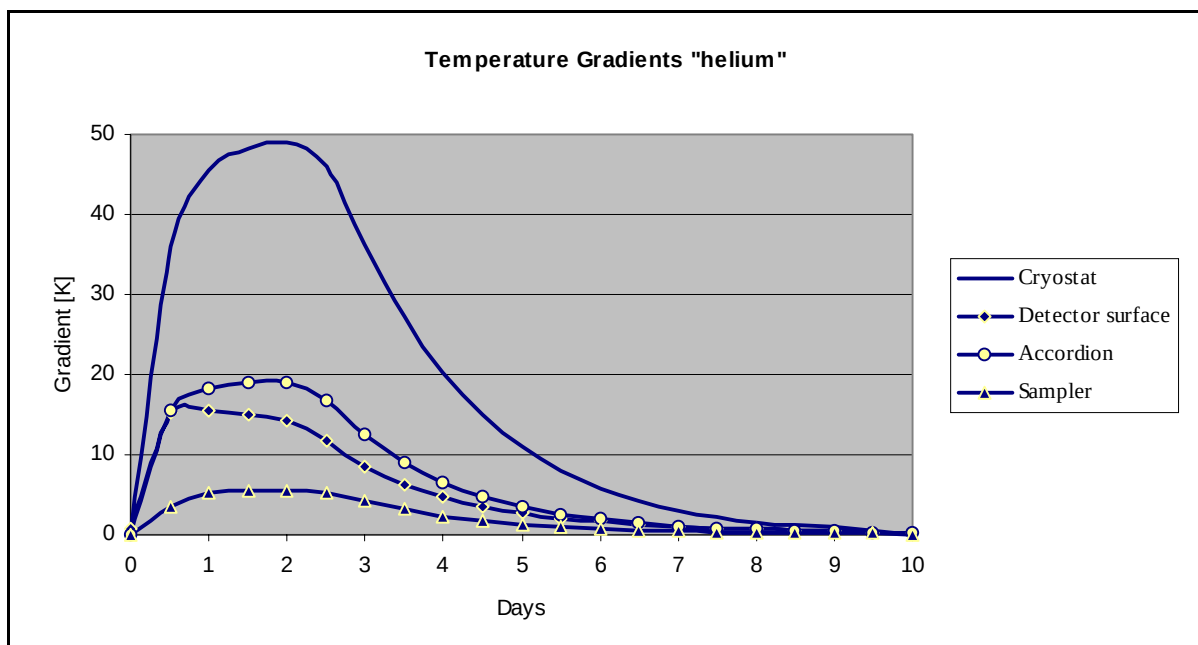


Figure 6.7 Temperature gradients during cool-down in "helium" mode.

As seen from the figure, will the "cryostat" have a maximum temperature gradient of approximately 49 K, the "accordion" will have a maximum gradient of 19 K, while the "sampler" will have a maximum gradient of approximately 6 K.

6.3.3 "Condensing argon"

The definition of the "condensing argon" causes this mode to have the exact same limitation and temperature gradients as for the "argon" mode. The limitation of the "heat exchanger tubes" was therefore set to 9 kW. The gradients of the cool-down simulation can be seen from Figure 6.8.

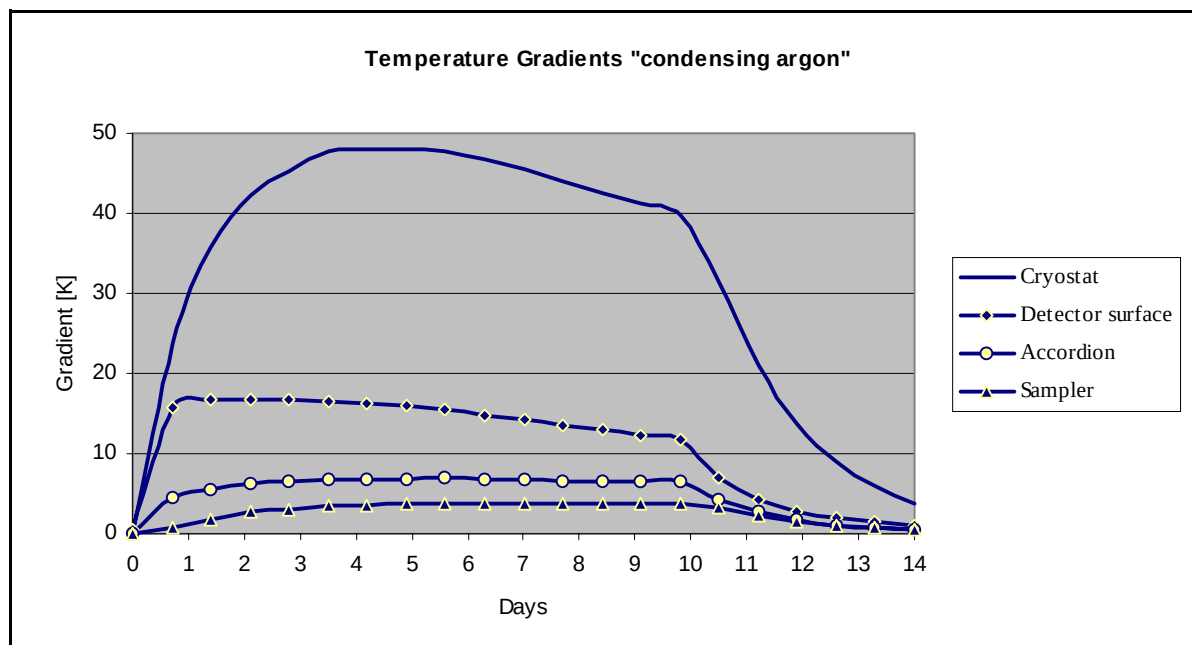


Figure 6.8 Temperature gradients during cool-down in "condensing argon" mode.

The maximum temperature gradient of the "cryostat" will be 48 K, for the "accordion" the maximum temperature gradient will reach approximately 7 K, while the gradient of the "sampler" reaches a maximum of approximately 4 K.

6.4 Cool-down time

This section presents charts showing certain temperatures during cool-down for the three simulated cool-down modes. The cool-down time, as defined in the introduction to this chapter, is an important output from these charts. The appearing temperatures in the cool-down charts are:

- **T_g** , the average gas temperature in the "outer gas gap".
- **$T1$** , the surface temperature of the "outer G10 bar".
- **$T4$** , the surface temperature of the "inner G10 bar".
- **$T7$** , the average temperature of the "inner cold cryostat wall".

When the temperature $T7$ reaches 120 K the Barrel Liquid Argon Calorimeter is cold enough to start filling of liquid argon. Based on this, the cool-down time for the three different simulation modes is presented in Table 6.9.

Mode:	"argon"	"helium"	"condensing argon"
<i>Cool-down time</i>	16 days	5 days	11 days
<i>"Plateau" temperature</i>	110 K	100 K	-

Table 6.9 Evaluated cool-down time and temperature "plateaus".

When the heat moved by the "heat exchanger tubes" in the Barrel Liquid Argon Calorimeter balances the heat leaks, the temperatures will reach a "plateau". As a result, the "plateau" is the coldest temperature which it is possible to gain for the given cool-down mode. The "plateau" temperature for the three different simulation modes is presented in Table 6.9 as well.

6.4.1 "Argon"

Figure 6.10 shows temperatures during cool-down simulation in "argon" mode. As seen from the figure is the "plateau" approximately at 110 K, and the cool-down time is 16 days.

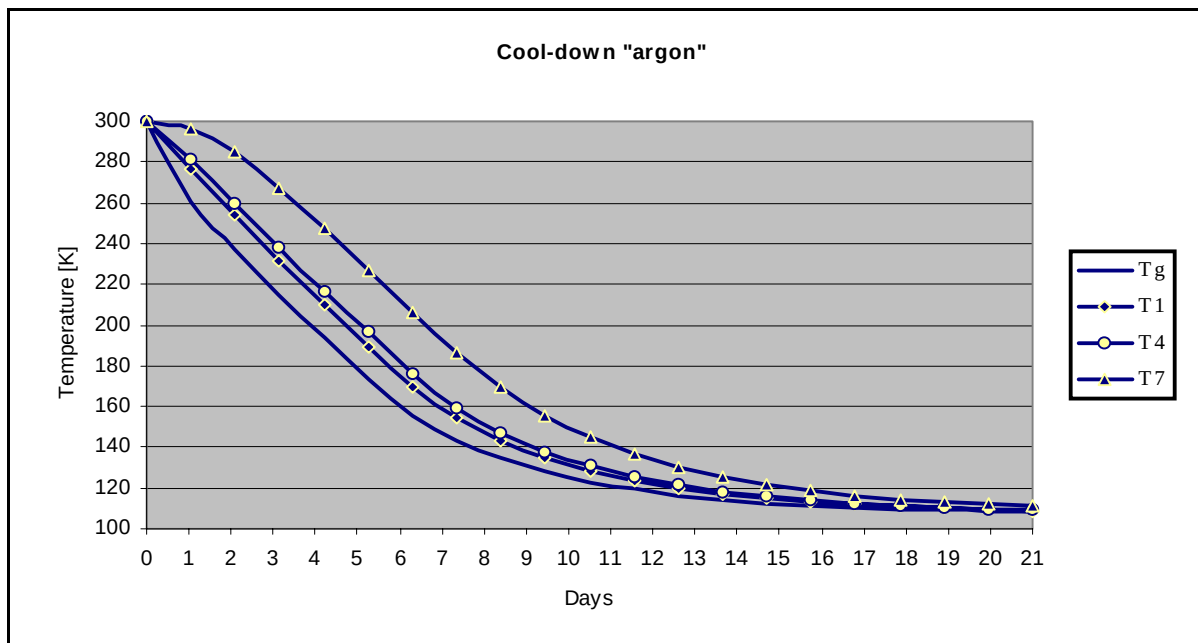


Figure 6.10 Temperatures during cool-down in "argon" mode.

6.4.2 "Helium"

Figure 6.11 shows temperatures during cool-down simulation in "helium" mode. As seen from the figure is the "plateau" approximately at 100 K, and the cool-down time is 5 days.

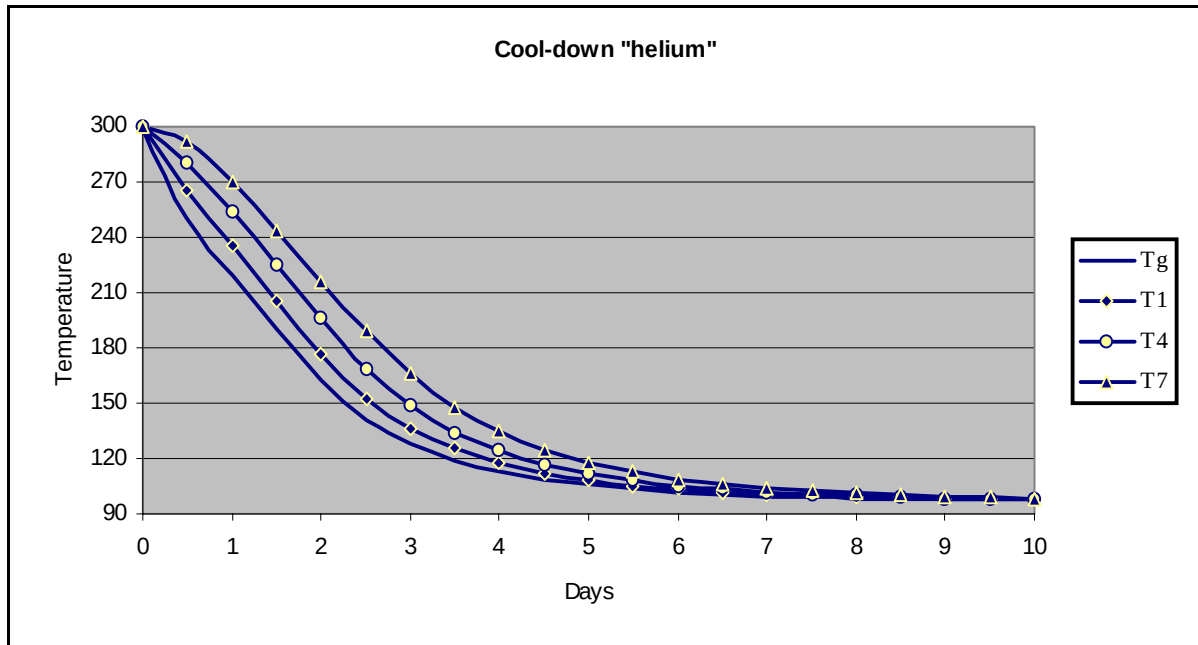


Figure 6.11 Temperatures during cool-down in "helium" mode.

6.4.3 "Condensing argon"

Figure 6.12 shows temperatures during cool-down simulation in "condensing argon" mode. As seen from the figure is the cool-down time 11 days. For the "condensing argon" mode it does not exist a "plateau" for the cool-down the same way as it does for the "argon" and the "helium" mode. The "plateau" in this mode will be the saturation temperature of the argon (89.7 K).

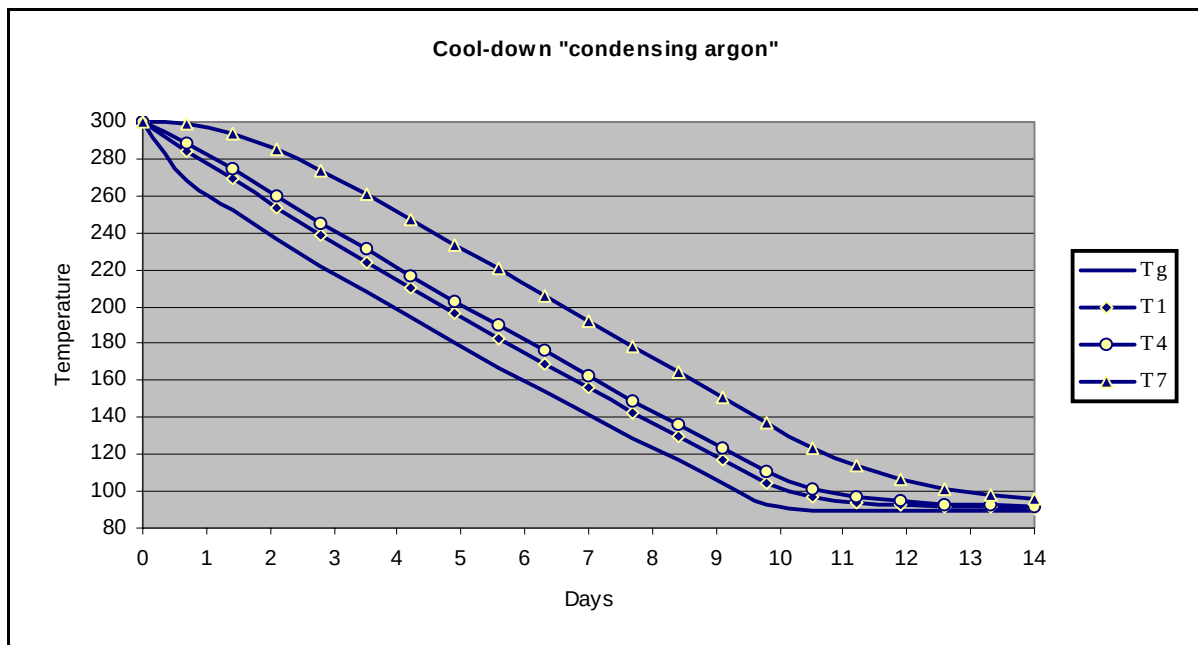


Figure 6.12 Temperatures during cool-down in "condensing argon" mode.

Chapter VII

Discussion

This chapter takes a closer look at the validity of the cool-down model presented in Chapter V, and discusses the results from the cool-down simulations presented in Chapter VI. All simplifications and assumptions which the model is based upon will be discussed separately, followed by a validity evaluation of the heat transfer coefficients for the tube heat exchangers, and an evaluation of the estimated cool-down times.

7.1 Simplifications and assumptions

This section discuss the validity of all simplifications and assumptions of the Barrel Liquid Argon Calorimeter model presented in Section 5.1. The simplifications and assumptions are written with a smaller font with a bullet ahead, followed by the discussion.

- **Limited cooling capacity.** In order to not exceed a temperature gradient of 50 K between any parts of the detector structure, the energy output of the heat exchangers is regulated. Which means the energy output can theoretically be higher, but will be regulated with the nitrogen mass flow to stay within the 50 K limit.

As mentioned in Section 4.5.1 are the "heat exchanger tubes" equipped with a valve for control of the mass flow at the inlet, and a valve for control of the pressure at the outlet. The energy output can therefore easily be controlled, and there by the critical temperature gradient avoided.

- **Quasi time-independent.** To make the problem quasi time independent, the simulation is made step by step. Causing time/temperature dependent properties to be assumed constant over each time-step.

For such small time-steps as used for the cool-down simulation (0.75 second for "argon" and "condensing argon", and 0.5 second for "helium"), there are very small variations of the fluid properties. The simplification makes therefore a negligible error.

- **Uniform temperature in tangential direction.** The barrel is assumed uniform in the tangential direction. For the calculation purpose, the heat exchanger tubes, the cryostat walls, the G10 bars, the accordion calorimeter and the presampler, will have an uniform temperature in the tangential direction.

The four heat exchangers tubes which perform the cool-down of the Barrel Liquid Argon Calorimeter are symmetrically mounted, giving the calorimeter an approximate uniform tangential geometry. The tangential geometry gives the surrounding gas of the "heat exchanger tubes" an uniform temperature during the cool-down, which implies that the convective heat transfer to the "outer G10 bar" is approximately constant on the entire circumference. This in turn implies that the tangential temperature variations can be neglected in the model. The simplification leaves out evaluation of temperature gradients in the tangential direction, but will not affect the estimation of the cool-down time since the tangential temperature gradients are much less than the radial ones.

- **Uniform temperature in longitudinal direction.** The barrel is assumed uniform in the longitudinal direction as well. As for the tangential direction, the heat exchanger tubes, the cryostat walls, the G10 bars, the accordion calorimeter and the presampler, have an uniform temperature in the longitudinal direction.
- **No end effects.** There are no end effects since the barrel is assumed uniform in the longitudinal direction.

The Barrel Liquid Argon Calorimeter has an uniform geometry in the longitudinal direction except from the ends. The total length of the barrel is large compared to the regions affected by the “end effect”, so the temperature variations in longitudinal direction can be ignored in the model with a certain degree of accuracy.

The "heat exchanger tubes" will not necessarily have an uniform temperature in the longitudinal direction in the region where the energy output is limited by the nitrogen mass flow, in order to not exceed the critical temperature gradient. In this region, the nitrogen will most likely completely vaporize, and even superheat. As a consequence, the "heat exchanger tubes" will not have an uniform temperature in the longitudinal direction. The way the "heat exchanger tubes" are mounted in the barrel, the natural convection and mixing of the surrounding gas should still give an approximately uniform temperature of the gas. Since the cool-down model takes into account the total amount of energy transferred from the gas to the "heat exchanger tubes", will the simplification not affect the result of the simulations. In the region where the heat transfer to the nitrogen flow not is limited by the nitrogen flow itself, will the "heat exchanger tubes" have an uniform temperature in the longitudinal direction, and the assumption is correct.

- **Separated gas volumes.** The gas located in the Barrel Liquid Argon Calorimeter is assumed to be three separate regions without “communication” (no circulation of the gas between the separated volumes). The three regions are; the gas space between the outer G10 bar and the outer cryostat wall, the gas space between the presampler and the inner G10 bar, and the gas spaces between the cryostat wall and the presampler. The three volumes are indicated as "outer gas gap", "gap 1" and "gap 2" in Figure 5.2 and Figure 5.3.

There will be mixing of the gas between the three separated volumes due to density variations. However, the mixing will most likely be moderate in the longitudinal direction, and therefore take place mainly as an end-effect. The end regions will be partly blocked by cable bundles and patch panels, see Figure 4.2, which prevents natural convection flow in this regions. In addition, considered the low thickness of the gas “gaps” between the "inner cryostat wall" and the "presampler" and the "presampler" and the "inner G10 bar", the natural convection flow in the longitudinal direction is negligible. The mixing between the three gas “gaps” is therefore limited, and the cool-down model should still give a trustworthy estimate of the cool-down time and the gradients in the detector structure.

- **Two separated parts of the cryostat.** The cold cryostat vessel is assumed to be built from two independent parts without “communication” (no conductive heat transfer between the two cryostat parts), see Appendix B for a view of the separated cryostat parts. The end walls of the cold vessel, the cold bulk head, are integrated as a part of the outer cold vessel, which means that the mass and the surface area of the end walls are added to this part of the cryostat.

Since the barrel is simplified into radial temperature dependence only, with three separate gas volumes, it is necessary to assume two separate cryostat parts as well. The simplification assumes no conductive heat transfer between the two cryostat parts, which it will be. The estimated temperature gradient in the cold cryostat vessel could therefore be higher than the real one, and as a consequence, the energy output of the "heat exchanger tubes", in order to

not exceed the dimensioning temperature gradient of 50 K, can be regulated to a higher value. This would lower the cool-down time.

- **No cryostat rails.** The cryostat rails connecting the outer cold cryostat wall with the external rings, see Figure 4.1 for a view of the location, are neglected. As a consequence, the heat transfer by conduction from the cryostat wall to the external rings is neglected, and likewise the heat capacity of the rails.

The connection surface between the cryostat rails and the "external rings" is small compared to the total surface area of the rings. The heat transfer by conduction is therefore small compared to the convective heat transfer. This implies that the rails can be neglected without making a considerable error. In addition, the heat capacity of the cryostat rails is very small compared to the total heat capacity of the calorimeter, so a trifling error is caused by this simplification.

- **No internal rings.** The internal rings located in the gas space between the inner G10 bar and the presampler are neglected, see Figure 4.1 and Figure 4.4 for the location. As for the cryostat rails is the heat transfer by conduction through the rings ignored, and likewise the heat capacity.

The internal rings are connected to the "presampler" and the "inner G10 bar", but the connection surface is small compared to the total surface area of these "detector parts". The heat transfer from the "presampler" to the "inner G10 bar" due to conductive heat transfer in the internal rings can therefore be neglected. The heat capacity of these rings is small compared to the total heat capacity of the Barrel Liquid Argon Calorimeter, so the heat capacity can be neglected as well.

- **Only heat transfer by convection in the "outer gas gap".** All parts located in the gas space between the outer cold cryostat wall and the outer G10 bar are assumed to be affected by convective heat transfer only. Which means that the heat transfer by conduction between the external rings and the outer G10 bar is neglected. Likewise the conduction in contact spots between cabling and other parts of the detector.

As in the case of the internal rings, the contact surfaces are very small compared to the total surface of the equipment. To leave out the heat transfer by conduction in these parts makes therefore a negligible error.

- **All cabling are located in the "outer gas gap".** All the cables and the electronics are assumed to be located in the gas space between the outer G10 bar and the cryostat wall.

Most of the cabling is located in this gas space, so this will make a minor error.

- **Conduction in the "trapped" gas.** The gas "trapped" in the accordion calorimeter and the presampler is "solidified". Indicating no natural convection in the "trapped" gas, only conduction.

The "trapped" gas in the "accordion calorimeter" makes a minor contribution to the mean conductive heat transfer coefficient, see Section 5.5.3, independent of natural convection or not. The simplification makes therefore a negligible difference of the heat transfer through the "accordion calorimeter". For the "presampler", which is mainly built from G10, could natural convection of the "trapped" gas make a contribution to the mean thermal conductivity. However, the separated volumes of "trapped" gas in the "presampler" are small, so the natural convection should make a minor contribution to the total heat transfer through the "presampler".

- **No heat transfer by radiation.** The heat transfer by radiation inside the cryostat is neglected. Meaning no heat transfer by radiation between the cryostat parts.

With the appearing temperature differences during the cool-down, and the small surface of the "heat exchanger tubes", will the heat transfer by radiation be negligible compared to the heat transfer by natural convection.

- **Linear heat leaks.** The heat leaks into the barrel cryostat are assumed to have a linear temperature dependence.

The heat leaks into the barrel come mainly through the external surfaces. It comes by radiation across the vacuum insulation and by conduction along the cables and bellows of the feedthroughs [3], the feedthroughs are best seen from Figure 4.2. Heat leak by conduction has a linear temperature dependence, while heat leak by radiation has a temperature dependence of order four. Which means that the heat leak by radiation is counted to high in the cool-down simulations, but it is small compared to the heat transfer from the surface affected by the radiative heat leak in the region where the largest error occur. All in all, the simplification should therefore make a minor error.

- **No heat leak to the inner cold cryostat wall.** There is assumed no heat leak into the inner cold cryostat vessel.

During operation of the Barrel Liquid Argon Calorimeter will the inner cold cryostat vessel not receive radiation heat because of the presence of the solenoid (actually about 5 W is radiated towards the solenoid) [3]. It is not finally decided if the solenoid will be cooled down before, after or at the same time as the barrel [9], so the heat leak through the inner cryostat vessel is assumed to be of the same magnitude as for the operation mode, which is approximately zero.

- **Constant gas pressure.** The pressure in the cryostat is assumed constant during cool-down. In concrete, this means gas refill at each time-step. The mass of the gas present in the cryostat is determined for each time-step of the cool-down, as a function of the density and the constant volume occupied by the gas.

In order to keep a constant pressure, there will be refill of gas during the cool-down of the Barrel Liquid Argon Calorimeter. However, not as often as in the cool-down simulation. The refill will be done through a pressure controlled valve, releasing the gas when the barrel pressure reaches a certain level [9]. The cool-down model assumes the gas to be refilled at the same temperature as the gas in the cryostat, which is not the case. It assumes as well refill of the gas directly into the three separated gas "gaps", which nor is the case. The heat capacity of the gas itself is however negligible compared to the heat capacity the calorimeter, so this assumption will still not affect the result of the cool-down simulation.

7.2 Fluid properties and average gas temperature

In the cool-down simulations is the free-stream temperature of natural convection heat transfer, see Section 3.4, substituted with the average gas temperature of the surrounding gas, T_g . The substitution is done for calculation of the heat transfers from the "outer gas gap", and estimation of the fluid properties, in order to calculate the heat transfer coefficients. This section discusses the validity of the substitution and the validity of the fluid properties in general.

7.2.1 Average gas temperature

According to the theory presented in Chapter III, shall fluid properties for estimation of heat transfer coefficients be evaluated at the mean film temperature, and the total heat transfers from the temperature gradient over the film. The outer temperature of the film is the free-stream temperature and the inner temperature is the surface temperature. It leads to a discussion whether the average gas temperature can substitute the free-stream temperature in the cool-down simulations, and its validity.

Fluid properties: Figure 6.3 shows the heat transfer coefficients for natural convection flow on the "heat exchanger tubes" for "argon" and "helium" mode. The coefficients are displayed as a function of the average gas temperature. Figure 6.4 shows the heat transfer coefficient of "condensing argon" as a function of the same temperature. As seen from the figures are the rate of change in the heat transfer coefficient very small all over the temperature range of the cool-down. The error made by using the average gas temperature to calculate the outer heat transfer coefficient is therefore not likely to be more than a few percent. Using the average gas temperature to determine the average temperature of the boundary layer should therefore be a good simplification taken into account the simplicity of the cool-down model. The estimated heat transfer coefficients for the heat exchange between the gas located in the "outer gas gap" and the "outer G10 bar", the "cables and electronics", the "external rings" and the "outer cryostat wall" shows an equal tendency. Therefore, it can be concluded that the average gas temperature can be used for calculation of the fluid properties.

Natural convection heat transfers: The free-stream temperature is most likely to be higher than the average temperature of the gas. Which means that the temperature gradients used for calculation of the total heat transfers from the gas housed in the "outer gas gap" to the "detector parts" surrounded by the gas, probably are lower than the real ones. The assumption that the average gas temperature can substitute the free-stream temperature during the cool-down simulation, has its best validity when the fluid has an approximately uniform temperature.

7.2.2 Fluid properties

All fluid properties are based on data taken from [8] at a pressure of 1.3 bar. The maximum deviation of the calculated fluid properties are 3%. The gas pressure in the barrel will be constant during the cool-down simulation, while there actually will be some small changes as explained in Section 7.1. The gas properties shows however minor changes in this pressure range, so all in all the validity of the fluid properties should be very good.

7.3 Heat transfer coefficients of the heat exchanger tubes

This section discusses the heat transfer coefficients evaluated in the cool-down simulation versus the theory presented in Chapter III. Their validity in the Barrel Liquid Argon Calorimeter model, as well as for general cases with similar conditions and geometry will be evaluated. The discussion covers all three simulation modes, and these three heat transfer coefficients:

- The **overall heat transfer coefficient**, which is the coefficient for calculation of the heat transfer from the surrounding gas of the "heat exchanger tubes" to the boiling nitrogen flow.
- The **inner heat transfer coefficient**, which is the coefficient for calculation of the heat transfer between the inner wall of the "heat exchanger tubes" and the boiling nitrogen flow.
- The **outer heat transfer coefficient**, which is the coefficient for calculation of the heat transfer between the outer wall of the "heat exchanger tubes" and the surrounding gas.

7.3.1 Theory versus simulation

Overall heat transfer coefficient: The overall heat transfer coefficient was calculated from Equation 3.15 which is based on the theory of Section 3.7.1. The equation is derived from an energy balance over the tube; the heat transfer from the surrounding gas to the outer tube wall equals the heat transfer through the wall, which in turn equals the heat transfer from the inner tube wall to the boiling nitrogen flow. The uncertainty is therefore not in the equation itself, but in the estimation of the convective heat transfer coefficients.

The thermal conductivity of the "heat exchanger tubes" was set constant over the temperature range of the cool-down. The true nature of the thermal conductivity of stainless steel 304 shows a variation between 17 W/m-K and 9 W/m-K with a linear trend. The thermal conductivity was therefore set to 13 W/m-K for simulation of the cool-down, which is an estimate with minor influence of the results.

Inner heat transfer coefficient: For all three simulation modes were the heat transfer coefficient between the inner wall of the "heat exchanger tubes" and the boiling nitrogen flow set to 500 W/m²-K. The number was set by comparing the given conditions of the heat exchange in "argon" mode with the experiment 'Heat transfer for boiling nitrogen flow in a horizontal tube' performed by D. Steiner and E. U. Schlunder [4]. The heat transfer coefficient was found a bit higher, but finally set to 500 W/m²-K as a moderate estimate.

Outer heat transfer coefficient: The theory behind the estimation of the heat transfer coefficient between the "heat exchanger tubes" and the surrounding gas was presented in Section 3.4.1 and Section 3.5.1. The natural convection flow takes place in "argon" and "helium" mode, while condensation will take place in "condensing argon" mode.

The correlation used to estimate the average Nusselt number in natural convection flow on a horizontal cylinder, Equation 3.8, uses the diameter as the characteristic length. The estimation of the characteristic length which could involve a certain degree of uncertainty is therefore avoided. The equation used to estimate the heat transfer coefficient for laminar film condensation on horizontal tubes, Equation 3.12, is derived from common theory of force and energy balance and should give a very good estimate of the coefficient. The theory used in the cool-down simulation should therefore be very well fitted for estimation of the outer heat transfer coefficient.

All fluid properties should be evaluated at the mean film temperature, which is the mean temperature of the tube wall and the free-stream temperature. For cool-down simulation of

barrel was the average gas temperature set to substitute the free-stream temperature. As discussed in Section 7.2.2 is this a good approximation.

7.3.2 Validity

Overall heat transfer coefficient: The uncertainty mainly comes from the inner heat transfer coefficient, but it is negligible compared to the outer heat transfer coefficient in the case of "argon" and "helium". This can easily be seen from Equation 3.15, which is the equation for calculation of the overall heat transfer coefficient for 'Combined heat transfer over a heat exchanger tube'. When the heat transfer from the "heat exchanger tubes" is in the form of 'Natural convection flow on a horizontal cylinder', which is the case in "argon" mode and "helium" mode, is the heat transfer coefficient between the inner wall of the "heat exchanger tubes" and the boiling nitrogen flow large compared to the heat transfer coefficient between the outer wall of the "heat exchanger tubes" and the surrounding gas. The inner heat transfer coefficient is therefore negligible. For the "condensing argon" mode is the estimated heat transfer coefficient of more importance.

Inner heat transfer coefficient: The "condensing argon" mode will experience the same heat transfer coefficient as "argon" since the coefficient is mainly limited by the nitrogen mass flow, which is equal. In "helium" mode will the heat transfer coefficient be higher since the nitrogen mass flow is almost three times the mass flow of the "argon" mode. As mentioned is the inner heat transfer coefficient negligible in the case of "argon" and "helium". The exact estimation of the inner heat transfer coefficient for these two modes is therefore of minor importance, so the number used for the cool-down simulation should be valid. For the "condensing argon" mode is the estimated heat transfer coefficient of more importance. However, it seems as a trustworthy estimate by comparing the given conditions with [4].

Outer heat transfer coefficient: Since the heat transfer coefficient between the gas in the "outer gas gap" and the outer wall of the "heat exchanger tubes" is estimated as a function of the average gas temperature, the result from the estimation is independent of the case. The result can therefore be used for all "heat exchanger tubes" with the same dimensions and boiling nitrogen mass flow. As mentioned is the boiling heat transfer coefficient negligible compared to the outer heat transfer coefficient for "argon" and "helium" mode, so the estimated coefficients can be used almost independent of the boiling nitrogen mass flow in the case of heat transfer by natural convection on the outer tube surface. The cool-down rate will not affect the estimation of the heat transfer coefficients.

7.4 Cool-down time

In order to not exceed the critical temperature gradient of 50 K between any two "detector parts", the energy output of the "heat exchanger tubes" is regulated during the cool-down simulation, see Section 5.1. The capacity of the "heat exchanger tubes" is determined by the nitrogen mass flow, and the critical temperature gradient occur in the cold cryostat vessel, which then becomes the dimensioning factor. The cool-down time is directly related to the energy output, so the validity of the cool-down time is dependent of the validity of the estimated temperature gradient in the cold cryostat vessel. The cool-down model is based on the assumptions and simplifications listed in Section 5.1. Their validity is already discussed separately, so this section will give an evaluation of the entirety of the cool-down model.

All simplifications and assumptions cause inaccuracy in the cool-down simulation, some with a bigger influence than others. The main uncertainties of the estimated temperature gradient in the cold cryostat vessel is because;

- there will be circulation of the gas between the three separated gas volumes,
- there will be conduction between inner and outer cold cryostat vessel,
- and there will be natural convection of the “trapped” gas in the "presampler".

In the cool-down model these three effects were ignored, and it could cause the temperature gradient of the cold cryostat vessel to be estimated too high. The cool-down model assumes uniform temperature in the longitudinal direction, which implies no end effects. The real cool-down is off course a heat transfer problem of three dimensions with end-effects. These end-effects occur as a consequence of circulation of the gas in the “gap” between the "accordion calorimeter" and the cold cryostat wall, and conduction in the vessel itself. The estimated temperature gradient in the cold cryostat vessel could therefore be higher than the real one. However, the length of the vessel is long compared to the regions affected by the end effects. The cold cryostat vessel will therefore most likely have an approximately uniform temperature in the longitudinal direction, and the end effect should have a minor influence of the estimated temperature gradient in the cold cryostat vessel.

Natural convection of “trapped” gas in the "presampler" will improve the heat transfer from the inner parts of the detector structure, and the improved heat transfer will lower the temperature gradient of the cold cryostat vessel. By changing the thermal conductivity of the "presampler" in the cool-down simulation of "argon", it seems like the uncertainty of the estimated temperature gradient is not likely to be more than 2 K.

In addition there are uncertainties in the characteristic lengths presented in Section 5.4.2. These characteristic lengths are based on assumptions of how the fluid will circulate. Which is very hard to predict, and is mainly based on experience. However, the error is not critical. If the characteristic length is twice as big, the temperature gradient of the cold cryostat vessel raises 1.5 K, and the critical temperature gradient is still within the 50 K limit. On the other hand, if the characteristic length is less, the critical temperature gradient is less as well, and the cool-down can be performed in less time.

The above discussion proves the estimated cool-down times as “worst case” estimates, which means that the real cool-down times probably are lower than the estimated. New simulations, based on the discussion covered by this section, indicates the estimated cool-down times as not likely to be of with more than 15%.

Chapter VIII

Conclusion and future

The Barrel Liquid Argon Calorimeter must be cooled down from ambient temperature before the cryostat can be filled with liquid argon. In this thesis the cool-down time is defined as the time needed to cool down the barrel and its cryostat to the temperature at which the calorimeter can be filled. The liquid argon filling can start when the warmest component reaches a temperature of 120 K. The warmest component being, at any point of the cool-down, the "inner cold cryostat wall".

This diploma thesis shows how the cool-down time can be estimated for three different modes. Two modes in which the cryostat is charged with argon gas, one with condensation of the argon gas on the "heat exchanger tubes" and one without, and one mode in which the cryostat is charged with helium gas. The mechanical design of the cryostat and its calorimeter does not allow temperature gradients higher than 50 K between any two points, which becomes the limitation of the cool-down. The limiting temperature gradient occurs in the cold cryostat vessel, with the "outer cold cryostat wall" as the coldest part of the barrel and the "inner cold cryostat wall" as the warmest part of the barrel. The energy output of the "heat exchanger tubes" was regulated during the cool-down simulations, to make sure the critical temperature gradient never was exceeded. Results from the cool-down simulation of the three above mentioned cool-down modes are presented in Chapter VI, while Table 8.1 summarizes the most important results. The mode presented as "mixed" suggests how the cool-down should be performed. It is a combination of the "argon" and the "condensing argon" mode, and will be explained further in the following of this chapter.

<i>Mode:</i>	<i>"helium"</i>	<i>"condensing argon"</i>	<i>"argon"</i>	<i>"mixed"</i>
<i>Cool-down time</i>	5 days	11 days	16 days	13 days

Table 8.1 Estimated cool-down times.

When the cool-down is performed in "helium" mode, it is of major importance to evacuate all the helium from the barrel before liquid argon filling is allowed. This because helium pollution in the liquid argon could be detrimental for the functioning of the detector [9]. There will always be a certain risk of pockets of helium gas left behind in the barrel, even with very good vacuum pumps, and the remaining helium gas could cause such detrimental pollution. Since cool-down in the two argon modes can be performed within "reasonable time", it is recommended to avoid use of helium in the cool-down.

There is a certain risk of the dripping liquid argon causing excessive stresses in the detector structure when the barrel is cooled in "condensing argon" mode. The use of the "condensing argon" mode should therefore be avoided or at least used moderately if possible. It is recommended to avoid use of the "condensing argon" mode until the warmest part of the Barrel Liquid Argon Calorimeter, the "inner cold cryostat wall", reaches a temperature of 140

K [5]. The temperature gradient between the warmest part and the dripping liquid argon is then approximately 50 K.

To use the advantages of the "condensing argon" mode, but taking as little risk as possible, the cool-down of the Barrel Liquid Argon Calorimeter can be performed as suggested for the "mixed" mode.

- **"Mixed"** - The cool-down is performed in "argon" mode until the temperature of the "inner cold cryostat wall" reaches 140 K. When the temperature is reached, the cool-down is continued in "condensing argon" mode.

Cool-down performed in "mixed" mode implies 11 days of cooling in "argon" mode and 2 days of cooling in "condensing argon" mode before the liquid argon filling can start (the warmest temperature of the barrel is below 120 K). Making it a total of 13 days to perform the cool-down in "mixed" mode. See Table 8.1 for comparison with the three other cool-down modes.

The discussion of Chapter VII shows that the estimated cool-down times are "worst case" estimates with a maximum deviation of 15%. In other words, the real cool-down time will not be higher than the estimated, but can be of with 15%. Therefore, as a final conclusion, the estimated cool-down times are good indications of the time needed to cool down the Barrel Liquid Argon Calorimeter from ambient temperature to the temperature at which the calorimeter can be filled with liquid argon.

This thesis has estimated the cool-down time of the Barrel Liquid Argon Calorimeter, but an estimate of the time needed to warm up the barrel from operation temperature to ambient temperature is of significance as well. Simulation of the warm-up time, based on the model presented in this thesis, has already started, and results from these simulations will be published in a future paper.

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Appendix

Appendix A

Fortran program codes

A.1 "Argon".

A.2 "Condensing argon".

A.3 "Helium".

A.1 "Argon" presents the Fortran program code used for cool-down simulation in "argon" mode, except from the initiation. For the other two cool-down modes are only the code parts different from "argon" presented.

A.1 "Argon"

! Program Dippen: Simulation of the liquid argon calorimeter barrel

PROGRAM dippen

IMPLICIT NONE

! Declare all variables

REAL x,y,z,A,B,C	!Variables for everything
REAL density,prandtl,conductivity,rayleigh	!Declaration of the properties functions
REAL heat_capacity,expansion,viscosity	!Declaration of the properties functions
REAL Tw,Te,Tc,T1,T2,T3,T4,T5,T6,T7,Tg,Tn	!Absolute temperatures
REAL TAO,TAd,TAi,TA1,TAp,TA2	!Average temperatures
REAL TCw,TCe,TCc,TCg,TCo,TC1,TC2	!Characteristic temperatures
REAL DTw,DTe,DTc,DTg,DT1,DT2,DT3,DT4,DT5,DT6,DT7	!Temperature differences
REAL Qw,Qe,Qc,Qg,Q1,Q2,Q3,Q4,Q5,Q6,Q7,QA,QLw,QLc	!Heat transfers
REAL days,houres,seconds	!Time
REAL LCw,LCE,LCc,LCP,LCo	!Characteristic lengths
REAL Vg,V1,V2	!Volumes
REAL k_G10,k_detector,k_presampler,k_steel	!Thermal conductivities
REAL timestep,Qlimit	!Times-step & heat transfer limit
REAL Ra,Pr,Nu,hc,UA,ke1,ke2,k,dens,Cp,Ra1,Ra2	!Properties
REAL hcP,hcD,hcN	!Convective heat transfer coefficients
REAL Qw1,Qw2,DTw1,DTw2	!Inner and outer pipe wall (heat transfer/delta T)
INTEGER N,sim,Pipes	!Counter & simulation time in days & number of pipes

! Initiate all variables

!Program structure

10 IF (days<sim) THEN	!Start of calculation loop
N=N+1	!counter of loops
seconds=N*timestep	!Time
houres=(N*timestep)/3600	!Time
days=(N*timestep)/86400	!Time

!Calculation of the heat transfer between the gas and the heat exchanger tubes

Qw1=1000	!Initiate, heat transfer nitrogen/inner pipe wall
Qw2=1000	!Initiate, heat transfer pipe wall

DO WHILE (ABS(Qw1-Qg)>1)

Qw1=Qg	!
Qw2=Qg	!
DTw1=Qw1/(hcN*14.07)	!Delta T nitrogen/inner pipe wall
DTw2=Qw2*LOG(0.022/0.02)/((6.28*k_steel*56*Pipes)	!Delta T pipe wall
DTg=Tg-Tn-DTw1-DTw2	!Delta T argon gas/outer pipe wall
TCg=Tg-DTg/2	!Characteristic temperature gas
Ra=rayleigh(TCg)*DTg*Lcp**3	!Rayleigh number
Pr=prandtl(TCg)	!Prandtl number
Nu=0.36+(0.518*(Ra**0.25))/((1+(0.559/Pr)**0.5625)**0.444444)	!Ra<10E9, Laminar
k=conductivity(TCg)	!Thermal conductivity
hcP=(Nu*k)/Lcp	!Heat transfer coefficient outer pipe wall
A=1/(6.28*0.011*hcP*56*Pipes)	!
B=LOG(0.011/0.010)/(6.28*k_steel*56*Pipes)	!
C=1/(6.28*0.010*hcN*56*Pipes)	!
UA=1/(A+B+C)	!Total heat transfer coefficient
QA=UA*(Tg-Tn)	!Max heat transfer
Qg=MIN(QA,Qlimit)	!Real heat transfer

END DO

!Calculation of the heat transfer between the gas and the outer cryostat wall

DTw=Tw-Tg	!Delta T
TCw=(Tw+Tg)/2	!Characteristic temperature
Ra=rayleigh(TCw)*DTw*Lcw**3	!Rayleigh number
Nu=0.52*(Ra**0.25)	!Nusselt number
k=conductivity(TCw)	!Thermal conductivity
hc=(Nu*k)/Lcw	!Heat transfer coefficient
Qw=hc*DTw*104.44	!Surface area of the outer cryostat wall = 104.44 m^2

!Calculation of the heat transfer between the gas and the external ring

DTe=Te-Tg	!Delta T
TcE=(Te+Tg)/2	!Characteristic temperature
Ra=rayleigh(TcE)*DTe*Lce**3	!Rayleigh number
Nu=0.52*(Ra**0.25)	!Nusselt number
k=conductivity(TcE)	!Thermal conductivity
hc=(Nu*k)/Lce	!Heat transfer coefficient
Qe=hc*DTe*60	!Surface area of the external ring = 60 m^2

!Calculation of the heat transfer between the gas and the cables & electronics

DTc=Tc-Tg	!Delta T
TcC=(Tc+Tg)/2	!Characteristic temperature
Ra=rayleigh(TcC)*DTc*Lcc**3	!Rayleigh number
Nu=0.52*Ra**0.25	!Nusselt number
k=conductivity(TcC)	!Thermal conductivity
hc=(Nu*k)/Lcc	!Heat transfer coefficient
Qc=hc*DTc*40	!Surface area of the cables & electronics = 40 m^2

!Calculation of the heat transfer between the gas and the outer G10 bar

DT1=T1-Tg	!Delta T
TCo=(T1+Tg)/2	!Characteristic temperature
Ra=rayleigh(TCo)*DT1*Lco**3	!Rayleigh number
Nu=0.52*Ra**0.25	!Nusselt number
k=conductivity(TCo)	!Thermal conductivity
hcD=(Nu*k)/Lco	!Heat transfer coefficient
Q1=hcD*DT1*80.03	!Surface area of the outer G10 bar = 80.03 m^2

!Calculation of the heat transfer in the detector parts

DT2=T2-T1	!Delta T outer G10 bar
DT3=T3-T2	!Delta T accordion calorimeter
DT4=T4-T3	!Delta T inner G10 bar
DT5=T5-T4	!Delta T gap 1
DT6=T6-T5	!Delta T presampler
DT7=T7-T6	!Delta T gap 2
Tc1=(T5+T4)/2	!Characteristic temperature
Tc2=(T7+T6)/2	!Characteristic temperature
Q2=3981.38*k_G10*DT2	!Heat transfer to outer G10 bar
Q3=138.16*k_detector*DT3	!Heat transfer from inner G10 bar
Q4=2949.27*k_G10*DT4	!Heat transfer to inner G10 bar
Ra1=rayleigh(Tc1)*0.000000043896*DT5	!Rayleigh number
Pr=prandtl(Tc1)	!Prandtl number
k=conductivity(Tc1)	!Thermal conductivity
ke1=0.386*((Pr/(0.861+Pr))**0.25)*(Ra1**0.25)	!Effective conductivity in gap 1
IF (ke1.LT.1) ke1 = 1	!
Q5=2583.94*DT5*ke1*k	!Heat transfer from presampler
Q6=1983.20*k_presampler*DT6	!Heat transfer to presampler
Ra2=rayleigh(Tc2)*0.000000052564*DT7	!Rayleigh number
Pr=prandtl(Tc2)	!Prandtl number
k=conductivity(Tc2)	!Thermal conductivity
ke2=0.386*((Pr/(0.861+Pr))**0.25)*(Ra2**0.25)	!Effective conductivity in gap 2
IF (ke2.LT.1) ke2 = 1	!
Q7=2396.23*DT7*ke2*k	!Heat transfer from inner cryostat wall

!calculation of the heat leaks

Qlw=800*((300-Tw)/210)	!Outer cryostat cylinder
Qlc=1000*((300-Tc)/210)	!Cables & electronics

!calculation of temperatures and heat transfer for next time step**!temperature gas, outer cryostat wall, external ring and cables & electronics**

dens=density(Tg)	!Density
Cp=heat_capacity(Tg)	!Heat capacity
Tg=Tg-(((Qg-Qw-Qe-Qc-Q1)*timestep)/(dens*Cp*Vg))	!Gas Temperature
Tw=Tw-(((Qw-Qlw)*timestep)/(6.08*1000000))	!Thermal capacity = 6.08 MJ/K
Te=Te-(((Qe)*timestep)/(1.53*1000000))	!Thermal capacity = 1.53 MJ/K
Tc=Tc-(((Qc-Qlc)*timestep)/(5.13*1000000))	!Thermal capacity = 5.13 MJ/K

```
!average temperatures outer G10 bar, detector, inner G10 bar, gap 1, presampler, gap 2 and inner cryostat wall
TAo=TAo-(((Q1-Q2)*timestep)/(1.75*1000000))      !Thermal capacity outer G10 bar = 1.75 MJ/K
TAd=TAd-(((Q2-Q3)*timestep)/(15.56*1000000))     !Thermal capacity detector = 15.56 MJ/K
TAi=TAi-(((Q3-Q4)*timestep)/(1.29*1000000))     !Thermal capacity inner G10 bar = 1.29 MJ/K
dens=density(Tc1)                                !Density
Cp=heat_capacity(Tc1)                            !Heat capacity
TA1=TA1-(((Q4-Q5)*timestep)/(dens*Cp*V1))        !Gap 1
TAp=TAp-(((Q5-Q6)*timestep)/(0.80*1000000))     !effective heat capacity presampler = 0.80 MJ/K
dens=density(Tc2)                                !Density
Cp=heat_capacity(Tc2)                            !Heat capacity
TA2=TA2-(((Q6-Q7)*timestep)/(dens*Cp*V2))        !Gap 2
T7=T7-((Q7*timestep)/(3.05*1000000))            !Thermal capacity inner cryostat wall = 3.05 MJ/K

!absolute temperatures in the detector
T6=2*TA2-T7                                     !Inner presampler surface
T5=2*TAp-T6                                     !Outer presampler surface
T4=2*TA1-T5                                     !Inner inner G10 bar surface
T3=2*TAi-T4                                     !Inner detector surface
T2=2*TAd-T3                                     !Outer detector surface
T1=2*TAo-T2                                     !Outer outer G10 bar surface

GOTO 10
ENDIF
```

END PROGRAM dippen

! Functions for calculation of argon properties at 1.3 bar with an error less than 3%. The read in variable temp is the temperature.

```
REAL FUNCTION rayleigh (temp)                    !Rayleigh number
  y=9.81*expansion(temp)*density(temp)*density(temp)*heat_capacity(temp)
  z=viscosity(temp)*conductivity(temp)
  rayleigh=y/z
END FUNCTION

REAL FUNCTION density (temp)                     !Density
  x=(temp-75)/15
  density=-0.0026*x**3+0.0914*x**2-1.198*x+8.2313
END FUNCTION

REAL FUNCTION prandtl (temp)                     !Prandtl number
  x=(315-temp)/5
  prandtl=(0.001*x**4-0.0714*x**3+1.7042*x**2-13.556*x+6696.2)/10000
END FUNCTION

REAL FUNCTION conductivity (temp)                !Thermal conductivity
  conductivity=(0.0569*temp+1.0555)/1000
END FUNCTION

REAL FUNCTION heat_capacity (temp)               !Heat capacity
  x=(320-temp)/10
  heat_capacity=0.0018*x**4-0.0678*x**3+0.8588*x**2-3.7634*x+526.2
END FUNCTION

REAL FUNCTION expansion (temp)                  !Expansion
  x=(320-temp)/10
  expansion=(0.0015*x**3-0.0307*x**2+0.34*x+2.739)/1000
END FUNCTION

REAL FUNCTION viscosity (temp)                  !Viscosity
  viscosity=(0.0739*temp+1.0484)*0.000001
END FUNCTION
```

A.2 "Condensing argon"

!Calculation of the heat transfer between the gas and the heat exchanger tubes

```

Qw1=1000                                !Initiate, heat transfer nitrogen/inner pipe wall
Qw2=1000                                !Initiate, heat transfer pipe wall

DO WHILE (ABS(Qw1-Qg)>1)
  Qw1=Qg                                !
  Qw2=Qg                                !
  DTw1=Qw1/(hcN*14.07)                  !Delta T nitrogen/inner pipe wall
  DTw2=Qw2*LOG(0.011/0.010)/(6.28*k_steel*56*Pipes) !Delta T pipe wall
  DTl=Tsat-Tn-DTw1-DTw2                !Delta T liquid film
  TCl=Tsat-DTl/2                        !Characteristic temperature liquid film
  DTg=Tg-Tsat                           !Delta T gas/liquid film surface
  TCg=Tg-DTg/2                          !Characteristic temperature gas
  Cpg=heat_capacity(TCg)                !Heat capacity gas
  Pr=prandtl(TCl)                       !Prandtl number liquid
  k=conductivityL(TCl)                  !Thermal conductivity liquid
  dens=densityL(TCl)                   !Density liquid
  Cp=heat_capacityL(TCl)                !Heat capacity liquid
  Pr=Prandtl(TCl)                      !Prandtl number liquid
  v=viscosityL(TCl)/densityL(TCl)       !Viscosity liquid
  hfgS=hfg+((0.683-(0.228/Pr))*Cp*DTl)+(Cpg*DTg) !Modified enthalpy of phase change
  x=(dens-dens_v)*9.81*hfgS*(k**3)      !
  y=v*0.022*DTl                        !
  hcP=0.728*((x/y)**0.25)               !Heat transfer coefficient outer pipe wall
  A=1/(6.28*0.011*hcP*56*Pipes)         !
  B=LOG(0.011/0.010)/(6.28*k_steel*56*Pipes) !
  C=1/(6.28*0.010*hcN*56*Pipes)        !
  UA=1/(A+B+C)                          !Total heat transfer coefficient
  QA=UA*(Tsat-Tn)                       !Max heat transfer
  Qg=MIN(QA,Qlimit)                     !True heat transfer
END DO

```

! Functions for calculation of liquid argon properties at 1.3 bar, error less than 3%, the read in variable temp is the temperature

```

REAL FUNCTION densityL (temp)            !Density liquid phase
  x=temp
  densityL=-6.2133*x+1936.3
END FUNCTION

REAL FUNCTION prandtlL (temp)            !Prandtl number liquid phase
  x=temp
  prandtlL=(-3.6249*x+543.99)/100
END FUNCTION

REAL FUNCTION conductivityL (temp)       !Conductivity liquid phase
  x=temp
  conductivityL=(-1.7807*x+283.78)/1000
END FUNCTION

REAL FUNCTION heat_capacityL (temp)       !Heat capacity liquid phase
  x=temp
  heat_capacityL=0.3349*x**2-57.768*x+3610.3
END FUNCTION

REAL FUNCTION viscosityL (temp)           !Viscosity liquid phase
  x=temp
  viscosityL=(-7.9165*x+951.85)/1000000
END FUNCTION

```

A.3 "Helium"

! Functions for calculation of helium properties at 1.3 bar with an error less than 3%. The read in variable temp is the temperature

REAL FUNCTION density (temp) !Density

 x=(320-temp)/10

 density=(0.5926*x**3-10.518*x**2+146.26*x+1818)/10000

END FUNCTION

REAL FUNCTION prandtl (temp) !Prandtl number

 x=temp

 prandtl=0.0000007*x**2-0.0004*x+0.7176

END FUNCTION

REAL FUNCTION conductivity (temp) !Thermal conductivity

 x=temp

 conductivity=(0.4139*x+34.036)/1000

END FUNCTION

REAL FUNCTION heat_capacity (temp) !Heat capacity

 heat_capacity=5193

END FUNCTION

REAL FUNCTION viscosity (temp) !Viscosity

 x=temp

 viscosity=0.00000005*x+0.000005

END FUNCTION

REAL FUNCTION expansion (temp) !Expansion

 x=temp

 expansion=1/x

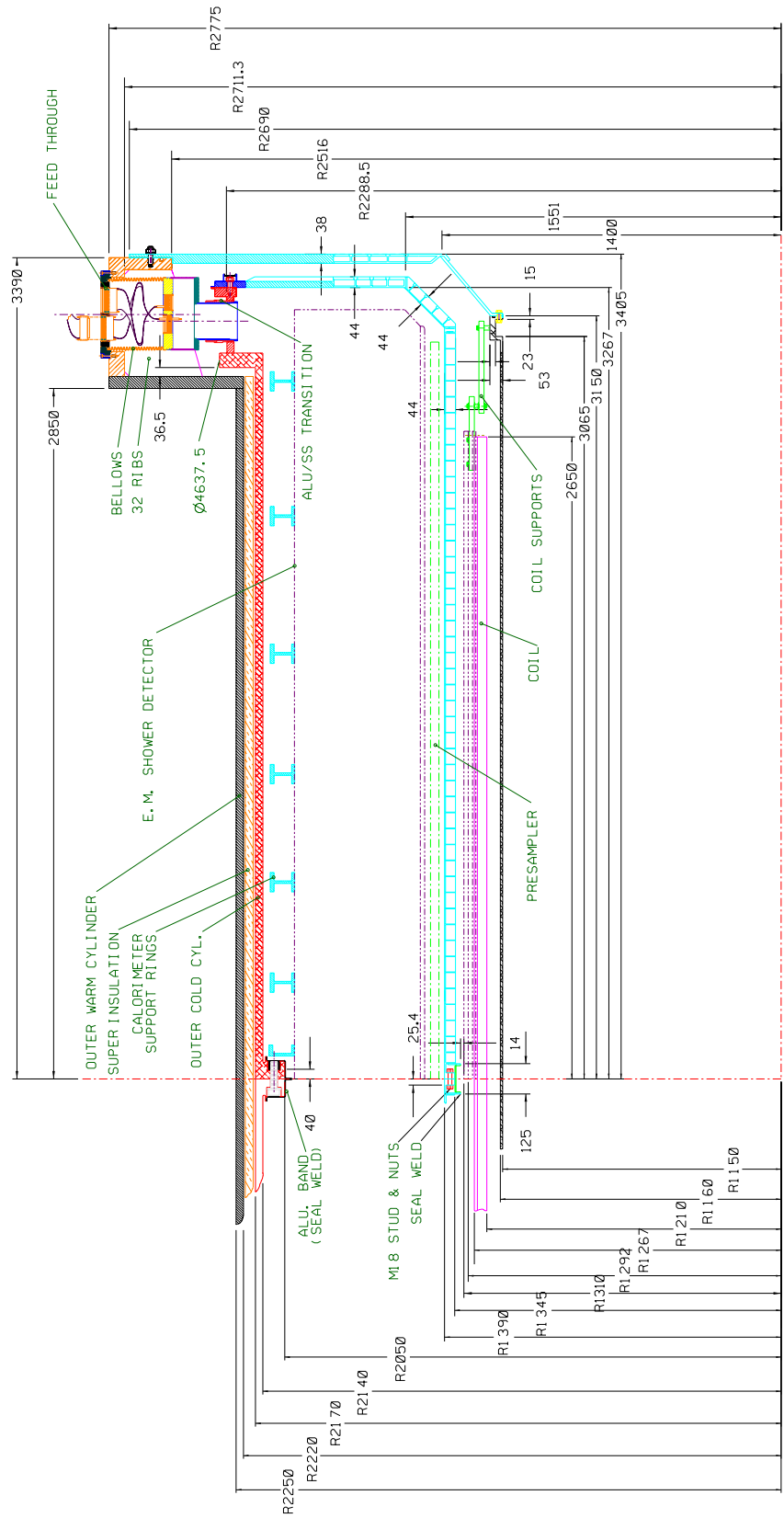
END FUNCTION

Appendix B

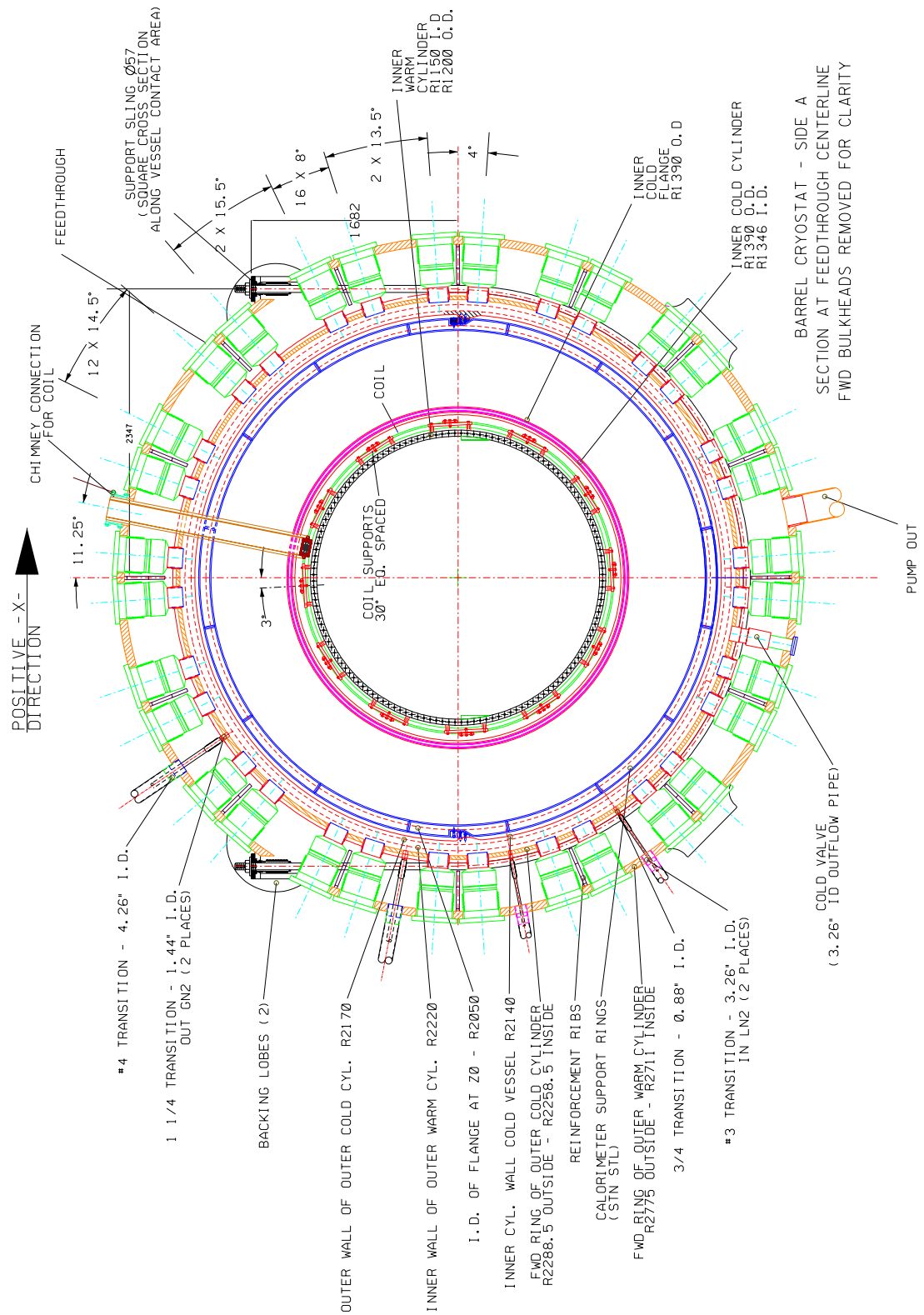
Figures and pictures

- B.1 Side view of the barrel cryostat, showing a "quarter view".
- B.2 End view of the barrel cryostat.
- B.3 An exploded view of the barrel cryostat, identifying the individual components.
- B.4 Side view of a feedthrough.
- B.5 General flow sheet.

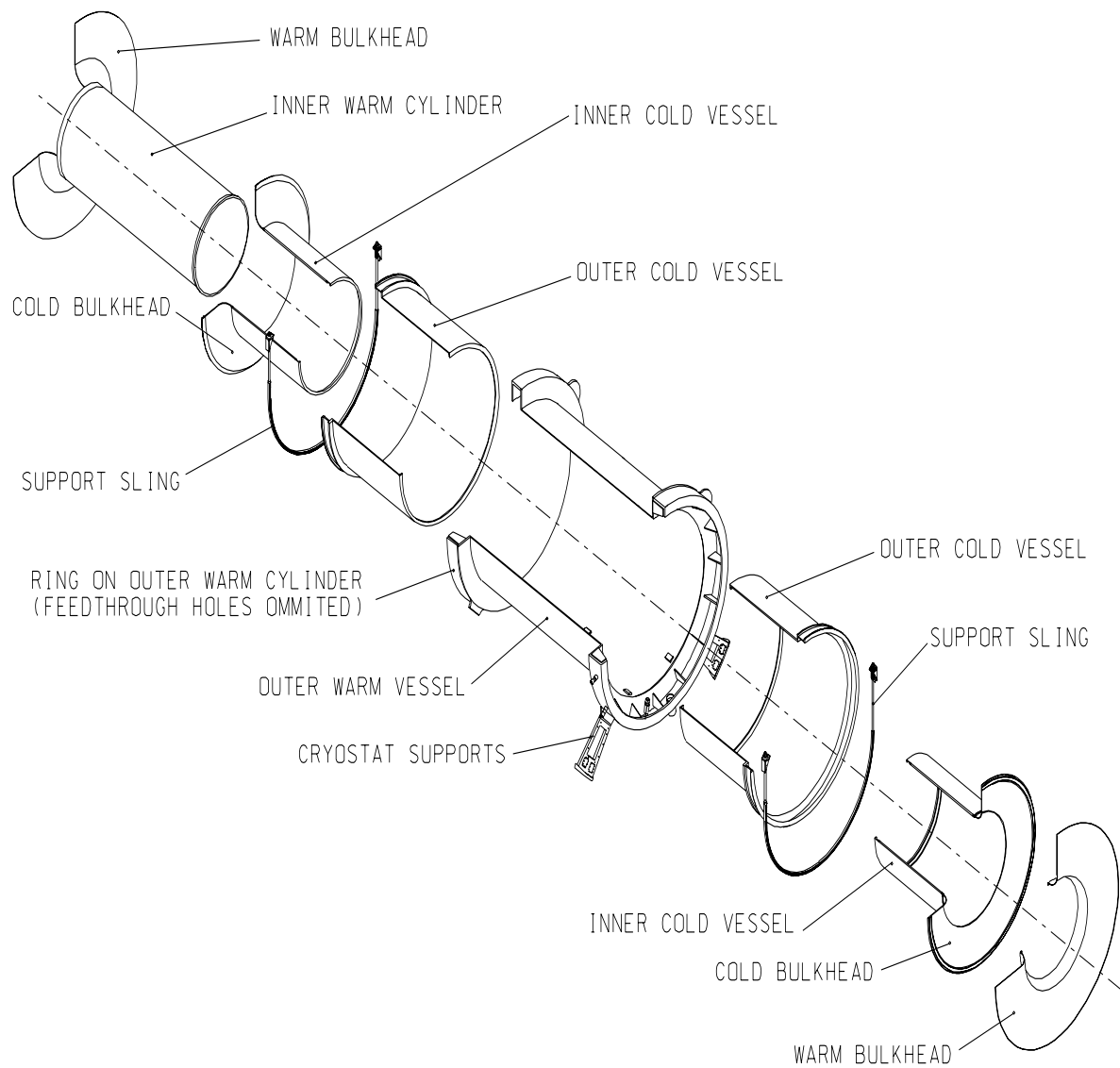
B.1 Side view of the barrel cryostat, showing a "quarter view".

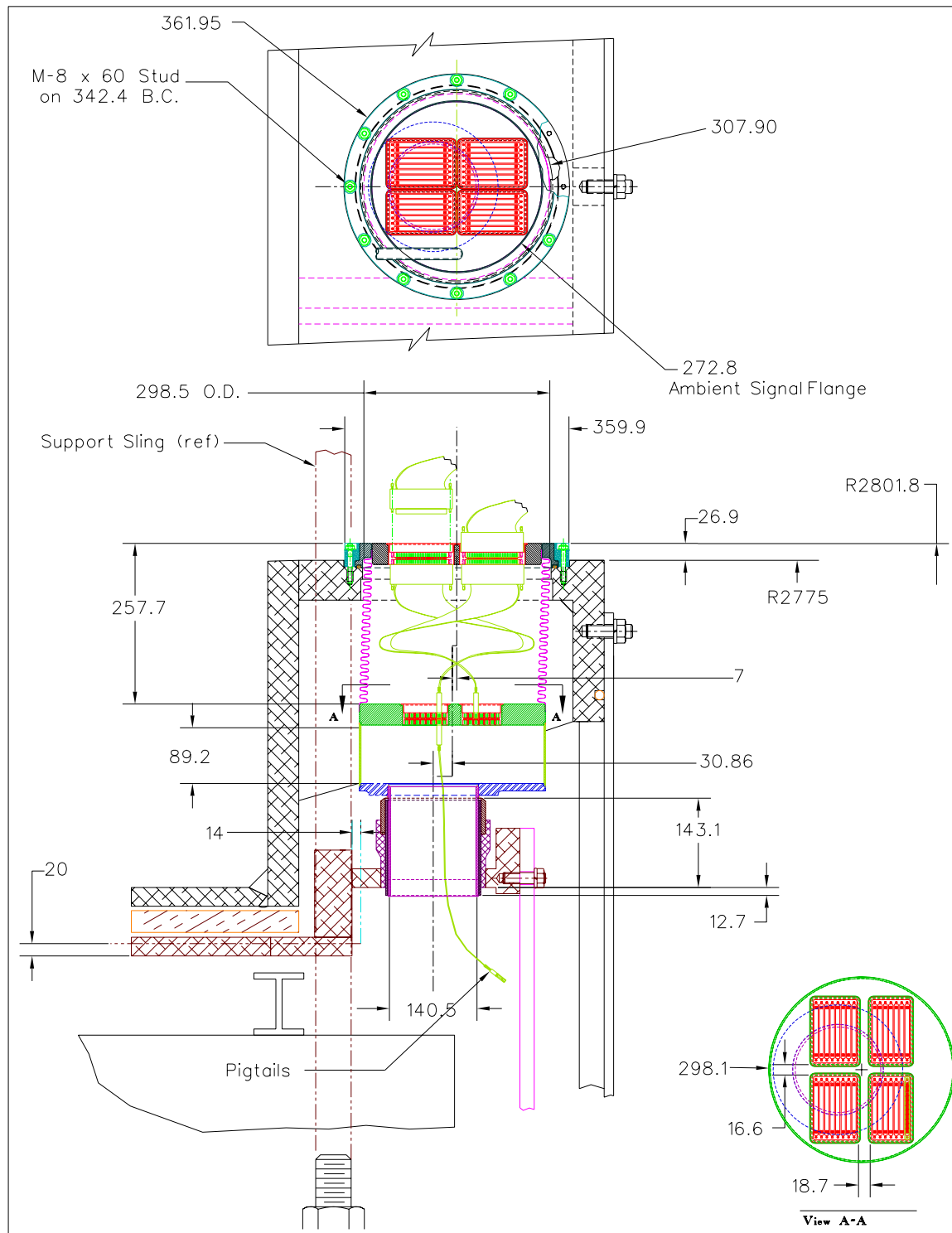


B.2 End view of the barrel cryostat.



B.3 An exploded view of the barrel cryostat, identifying the individual components.



B.4 Side view of a feedthrough.

B.5 General flow sheet.

