Heat Accumulators for Cryogenic In-Space Propulsion

Johannes Leiner, Jörg Riccius** and Oskar Haidn****
Deutsches Zentrum für Luft- und Raumfahrt (DLR), Institut für Raumfahrtantriebe,
Langer Grund 1, D-74239 Hardthausen, Germany
* Johannes.Leiner@dlr.de; ** Joerg.Riccius@dlr.de; *** Oskar.Haidn@dlr.de

Didier Vuillamy+
Snecma – Space Engines Division
Forêt de Vernon - BP 802, 27208 VERNON Cedex, France
+didier.vuillamy@snecma.fr

Abstract
In this work, the preparations for the experimental study of ISP-1 [1] work package 5 dealing with low temperature heat accumulators (LTA) for possible cryogenic in-space propulsion are presented. The functional principle of the LTA which is based on water and which is able to operate within the temperature range from -200°C to +80°C is quantitatively outlined. A prototype for the experimental validation of the numerical simulation of the phase change behaviour of both the heat transfer fluid and the heat storage material is shown.

1. Introduction

The "In Space Propulsion - 1" (ISP-1) project [1] investigates new concepts for launcher upper stages for space crafts. The propulsion techniques, which have been used for orbital transfer and space exploration up to now, rely on storable propellants which are toxic and only yield a poor propulsion performance. Therefore, there is the idea of using cryogenic propellants such as liquid oxygen in combination with liquid methane, or liquid hydrogen whose specific impulses are much better. The usage of electrical pumps might be a cost efficient way of operating such a system as well as enabling specific features of the new missions such as multiple restarts. However, this leads to additional challenges such as a long time lag between successive engine operations or multiple re-ignitions, thus, yielding among others a major interest in the thermal management of in space propulsion systems. The underlying concept is the Low Cost Cryogenic Propulsion (LCCP) concept [2,3]. Work package 5 (WP5) of ISP-1 deals with the thermal management, especially with heat accumulators which could be one of the major means for saving energy [1].

The idea of saving energy by heat accumulators stems from the fact that on the one hand a permanent cooling of the fuel cells, which provide electrical energy for the space craft and the electrical propellant pumps, is necessary and therefore also some sort of cooling device. On the other hand, thermal energy is necessary during the engine operation, especially at the start of the engines, in order to pressurise the tanks themselves by the help of vaporization of their corresponding propellant. It is possible to store the thermal energy coming from the cooling of the fuel cells and use it for the self-pressurisation of the liquid propellant tanks during the engine operation. In detail, it is actually intended to use two different heat accumulators, a High Temperature Accumulator (HTA) being based on lithium salt and a Low Temperature Accumulator (LTA) being based on water [1,2]. The HTA should provide the thermal energy for the self-pressurisation of the hydrogen tank (which is called the unloading process of the HTA) and the LTA should provide thermal energy for the self-pressurisation of the oxygen tank (which is called the unloading process of the LTA). During the free-flight (or ballistic) phases, which are the periods between the main engine hot run operations of the orbital transfer from LEO (Low Earth Orbit) to GEO (Geostationary Earth Orbit), both heat accumulators should be uploaded. This is carried out electrically for the HTA with energy provided by the fuel cells and simultaneously thermally for the LTA by the waste heat coming from the same fuel cells. The fuel cells have an operating temperature optimum at +80°C. The current study focuses on the development of a LTA being based on water and which is able to operate within the temperature range from approximately -200°C to +80°C. The heat accumulator prototype which is built for the first experiments is mainly designed for the experimental validation of numerical simulations of the heat transfer process inside the heat accumulator. These simulations will include both the phase change of the heat transfer fluid (propellant fluids) and the phase change of the heat storage material. The development of numerical simulations for the heat transfer inside the heat accumulator is carried out by Univèrsitat Politécnica de Catalunya (UPC) in Spain [4].

2. Tank pressurisation and heat balance

2.1. Mass flow rates necessary for tank pressurisation

The following calculations are based on an LOX/LH2 engine for the transfer from the LEO to the GEO. Typical characteristics for such an engine are a specific impulse of I_{sp} = 440 s and a thrust of F = 1000 N [2]. Using the effective exhaust velocity of the propellant, $\Delta v = I_{sp} \cdot g$ = 440 s 9.81 m/s^2 ~ 4316.4 m/s, the total propellant mass flow rate during the hot run of the engine can be determined to be $\dot{m}_{total} = F/\Delta v$ = 1000 N / 4316.4 m/s = 231.7 g/s. By assuming a propellant mass ratio of *LOX : LH2 = 5.8 : 1* we can determine the mass flow rate of liquid H_2 (LH2) to be $\dot{m}_{LH2}$ = 34.1 g/s and of liquid O_2 (LOX) to be $\dot{m}_{LOX}$ = 197.6 g/s. In order to keep the tank pressures which should be fixed at about p = 3 bar for the LOX tank and for the LH2 tank constant, the volume flow $\dot{V} = \rho_{liquid}\dot{m}_{liquid}$, where ρ_{liquid} is the density of the liquid propellant (ρ_{LH2} = 66 kg/m^3 at 3 bar and 24 K and ρ_{LOX} = 1080 kg/m^3 at 3 bar and 102 K [1]), of the draining liquid propellant is compensated by a gaseous volume flow in reversed direction having a pressure of 3 bar and a gas temperature T being selected for H_2 to be T_{H2} = 30 K and for O_2 to be T_{O2} = 273 K is suggested. Using the general equation of state for ideal gases, $pV = mRT$, with the gas constant R, with R_{H2} = 4126 J/kgK for H_2 and R_{O2} = 260 J/kgK for O_2 [2], we can calculate the H_2 – and the O_2 – gas mass flow rates $\dot{m}_{H2}$ and $\dot{m}_{O2}$, respectively, into the tanks for pressurisation purposes as follows,

$$\dot{m}_{H2} = \frac{p\ \dot{m}_{LH2}}{\rho_{LH2} R_{H2} T_{H2}} \sim 1.25 \text{ g/s} \tag{1}$$

and

$$\dot{m}_{O2} = \frac{p\ \dot{m}_{LOX}}{\rho_{LO2} R_{O2} T_{O2}} \sim 0.77 \text{ g/s.} \tag{2}$$

2.2 Heat balance of the system

The heat balance of the system will be calculated in the following for a hot run duration of 1800 s, which is meant to be an upper limit during the transfer from LEO to GEO, and a free-flight phase of 4500 s, which is meant to be a lower limit during the orbital transfer.

In order to determine the energy Q being necessary for the evaporation and for the temperature increase from the boiling temperature of the gas during the engine operation, the following formula is used,

$$Q = t_{hot}\ \dot{m}\,(q_l + \bar{c}_p\ \Delta T), \tag{3}$$

where t_{hot} = 1800 s is the time of the hot run of the engine, q_l is the specific latent heat, with $q_{l,H2}$ = 447 kJ/kg for H_2 and $q_{l,O2}$ = 213 kJ/kg for O_2 [2], ΔT is the temperature difference between the gas and the liquid, i.e. $\Delta T_{H2} = T_{H2} - T_{H2,boiling}$ = 6 K for H_2 with its boiling temperature $T_{H2,boiling}$ ~ 24 K (at 3 bar), and $\Delta T_{O2} = T_{O2} - T_{O2,boiling}$ = 171 K for O_2 with its boiling temperature $T_{O2,boiling}$ ~ 102 K (at 3 bar), and $\bar{c}_p$ is the intermediate specific heat capacity being estimated for the gas in the temperature range between the gas and the liquid, $\bar{c}_{p,H2}$ ~ 13 kJ/kgK for H_2 and $\bar{c}_{p,O2}$ ~ 0.93 kJ / kgK for O_2 [1]. Note that the averaging of $\bar{c}_p$ has only been done roughly between the value of c_p at the highest and the lowest temperature of the gas at p = 3 bar. Now, the thermal energy is obtained that has to be stored in the accumulators for a hot run of the engine of Q_{H2} ~ 1181 kJ (respectively the corresponding instant heat flow $\dot{Q}_{hot,H2} = Q_{H2}/\ t_{hot}$ ~ 656 W) for H_2 and Q_{O2} ~ 515 kJ (respectively the corresponding instant heat flow $\dot{Q}_{hot,O2}$ =

[1] NIST, *Reference Fluid Properties REFPROP*, URL: http://www.nist.gov/index.html [cited 14 June 2011]
[2] Engineeringtoolbox, URL: http://www.engineeringtoolbox.com/ [cited 14 June 2011]

$Q_{O2}/\ t_{hot} \sim 286$ W) for O_2. In order to heat and respectively charge the heat accumulators up during the free-flight phase of $t_{ff,} = 4500$ s, a constant heat flow of $\dot{Q}_{ff} = Q/t_{ff}$ could be used. Note that the free-flight phase actually increases from around 4500 s to 24 hours and 4500s is only the lower limit for the design calculations. $\dot{Q}_{ff,H2} \sim 262$ W for H_2 and $\dot{Q}_{ff,O2} = 115$ W for O_2 are obtained neglecting any heat losses between the uploading and unloading of the heat accumulator. Since the basic idea of the general concept using heat accumulators is to electrically heat up an HTA using the on-board fuel cells during the free-flight phase and to collect simultaneously the waste heat of the fuel cells in the LTA, which is approximated to be in the same magnitude assuming an electrical efficiency of the fuel cells of around 50 %, it can be seen from the calculations that there will be more energy stored in the LTA than is actually necessary. For this reason, the concept of the LTA is modified by a second cooling circuit coming from a heat exchanger at the main LOX line, which is able to cool away during the hot run of the engine the surplus of thermal energy being stored in the LTA. This is illustrated in the figure 1. However, the electrical efficiency of 50 % for the fuel cells is only valid at their maximum electrical load[3] so that during the free-flight phase the efficiency is probably better, thus less waste heat would be generated and therefore, the heat balance for the LTA might be better. Anyway, a second cooling channel as can be seen in figure 1 may solve this issue. Assuming a power consumption of around 2000W for the pumps of the main engine, additional thermal heat of the same magnitude has to be cooled from the fuel cells during the hot run of the engine. The cooling can also be carried out by an external heat exchanger, which can be shared with the same heat exchanger that may be used to cool away the possible surplus of thermal energy in the LTA. A summarizing graphical overview of the concept and the corresponding example of the heat balance is presented in figure 1. Note that the electrical power of the compressors that should circulate the heat transfer medium is not included in this heat balance, thus its waste heat input is disregarded. Nitrogen or Helium may be used as heat transfer medium for the LTA.

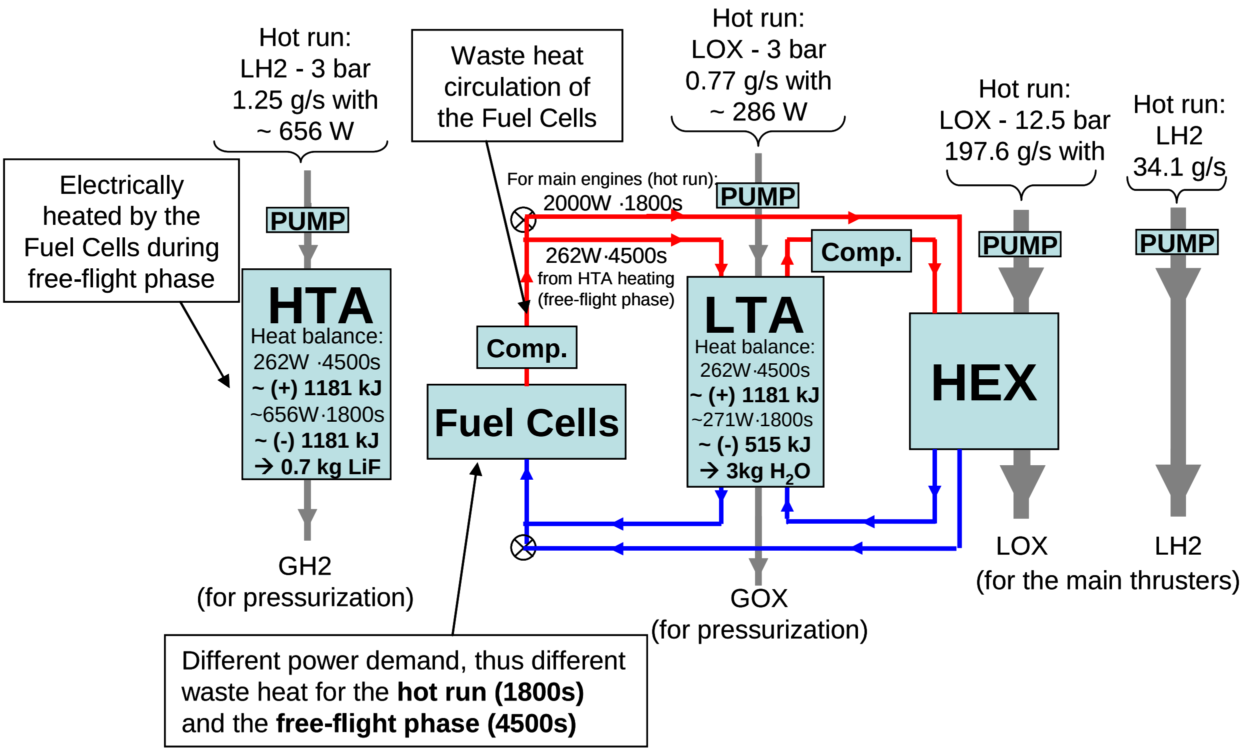

Figure 1: A graphical summary of the basic concept of both heat accumulators is shown. Abbreviations and Symbols: Comp. = Compressor (for the Helium or Nitrogen circulation), HEX = Heat Exchanger, ⊗= Control Valve, LTA = Low Temperature Accumulator, and HTA = High Temperature Accumulator.

[3] For the hot run of the engine the electrical load of the fuel cells is at maximum, since the propellant pumps which feed the engine chamber consume most of the electrical power. Their consumption may be approximated to be around 2000W.

In figure 1 also magnitudes for the mass of ice and respectively lithium fluoride are given (LiF) which are necessary for the heat accumulation inside the LTA and HTA, respectively. The mass of ice is calculated as follows

$$m_{ice} = Q_{H2} / [q_{l\text{-}water} + \bar{c}_{p-ice} (0°\mathrm{C} - T_{min}) + \bar{c}_{p-water} (T_{max}\text{-}0°\mathrm{C})], \quad (4)$$

with $q_{l\text{-}water}$ ~ 333 kJ/kg is the specific latent heat of water, $\bar{c}_{p-ice}$ ~ 2.0 kJ/kgK is an intermediate value of the specific latent heat of ice, $\bar{c}_{p-water}$ ~ 4.2 kJ/kgK is an intermediate value of the specific latent heat of water [4], and T_{min} / T_{max} is the minimum/maximum temperature. Requiring the LTA to be around its phase change at 0°C with T_{min} = -10°C and T_{max} = +10°C the mass of water can be approximated to be m_{ice} ~ 3 kg in case heat losses are neglected. Note that Q_{H2} is the thermal energy that is stored in both accumulators, the HTA and the LTA. The determination of the mass of LiF for the HTA can be carried out analogously as in equation (4). The specific latent heat of LiF is $q_{l\text{-}LiF}$ ~ 1044 kJ/kg, an approximate value for the specific heat capacity in the solid phase between 25°C and 848°C is $\bar{c}_{p-LiF,solid}$ ~ 2.0 kJ/kgK, and the specific heat capacity in the liquid phase (above 848°C) is $\bar{c}_{p-LiF,liquid}$ ~ 2.47 kJ/kgK [5]. For the temperature range with T_{min} ~ 500 °C and T_{max} ~ 900 °C a required LiF mass of about 0.7 kg can be derived in case heat losses are neglected.

3. Heat accumulator description

A literature study did not reveal any commonly used heat accumulators based on water as phase change material at operating temperatures between -200°C and +80°C. Therefore, different concepts for the LTA had to be developed from scratch. The simplest way of transferring heat to a heat storage material is by using ordinary tubes being either straight or coiled. There are some ways how to increase the heat transfer from the heat transferring tube to the heat storage material like the insertion of metal rings into the heat storage material [5]. However, for experiments which should validate numerical models for the phase change of the heat storage material water and the phase change of the heat transfer medium, the heat accumulator design is preferred to be as simple as possible. The chosen design which is optimized for validation experiments is presented. In the following subsection temperature sensing of the heat transfer tube, the heat storage material as well as the container walls are shown.

3.1. Heat accumulator design

The heat accumulator has a cylindrical shape with only a single straight tube for the heat exchange which is arranged in the axis of the cylinder. A corresponding CAD drawing is shown in figure 2. There are no other heat exchange tubes in the accumulator. In this arrangement the heat accumulator has two important characteristics: First, due to the straightness of the tube no centrifugal forces occur on the fluid dynamics and, second, the vessel has a rotational symmetric shape since the heat transferring tube is centred in the cylinder axis. This will be favourable for proceeding numerical analyses.

The heat accumulator container is not a flexible shell and does not have e.g. a bellow, so that the inner volume of the container cannot expand and contract with the volume change of the freezing water or the melting ice, respectively. This feature would be necessary for later in-space application, since air or water vapour bubbles inside the container would dramatically decrease the heat transfer from the heat transferring tube to the water when these bubbles accumulate around the tube. Nonetheless, for simulation purposes it is important to have an exactly defined shape of the container. Besides the exact shape of the container, it is important that the thermocouples will not be damaged or displaced by mechanical stresses of the ice when it presses against the container wall during the freezing process. For this reason, the non-flexible shell of the container will not be filled completely with water. The air on top of the water or ice, respectively, can be pressed into a separate balloon when the height level of the water increases during the freezing process.

The vessel and its lid with the corresponding fixing screws and fittings are made out of PTFE with a wall thickness of 25 mm. The only metal parts inside the vessel are the screw threads for the flange screws in order to be able to firmly fix the lid with PTFE screws. PTFE is applicable in the temperature range from -260°C till +260°C, which is

[4] Engineeringtoolbox, URL: http://www.engineeringtoolbox.com/ [cited 14 June 2011]
[5] NIST Webbook, URL: http://webbook.nist.gov [cited 14 June 2011]

within the intended operating temperature range from -196°C to + 80°C. The major advantage of PTFE for the current application is that it has a very low thermal conductivity, $\lambda_{PTFE} \sim 0.25$ W/Km (at 25°C) [6]. Therefore, it can be used as the wall material of the container as well as for thermal insolation purposes. In contrast to metal, for example, it is on the one hand a rather good insolation against the heat transfer along the interior walls of the vessel, so that the main heat transfer will be within the heat storage material and not along the container material. On the other hand PTFE can be used as well as insolation against the heat transfer through the container walls. For comparison, the heat conductivity is $\lambda_{steel,25°C} \sim 16$ W/Km for stainless steel (at 25°C), $\lambda_{water,25°C} \sim 0.58$ W/Km for water (at 25°C), and $\lambda_{ice,0°C} \sim 2.18$ W/Km for ice (at 0°C) [6]. The height of the inner volume of the container is 260 mm and its diameter is 180 mm.

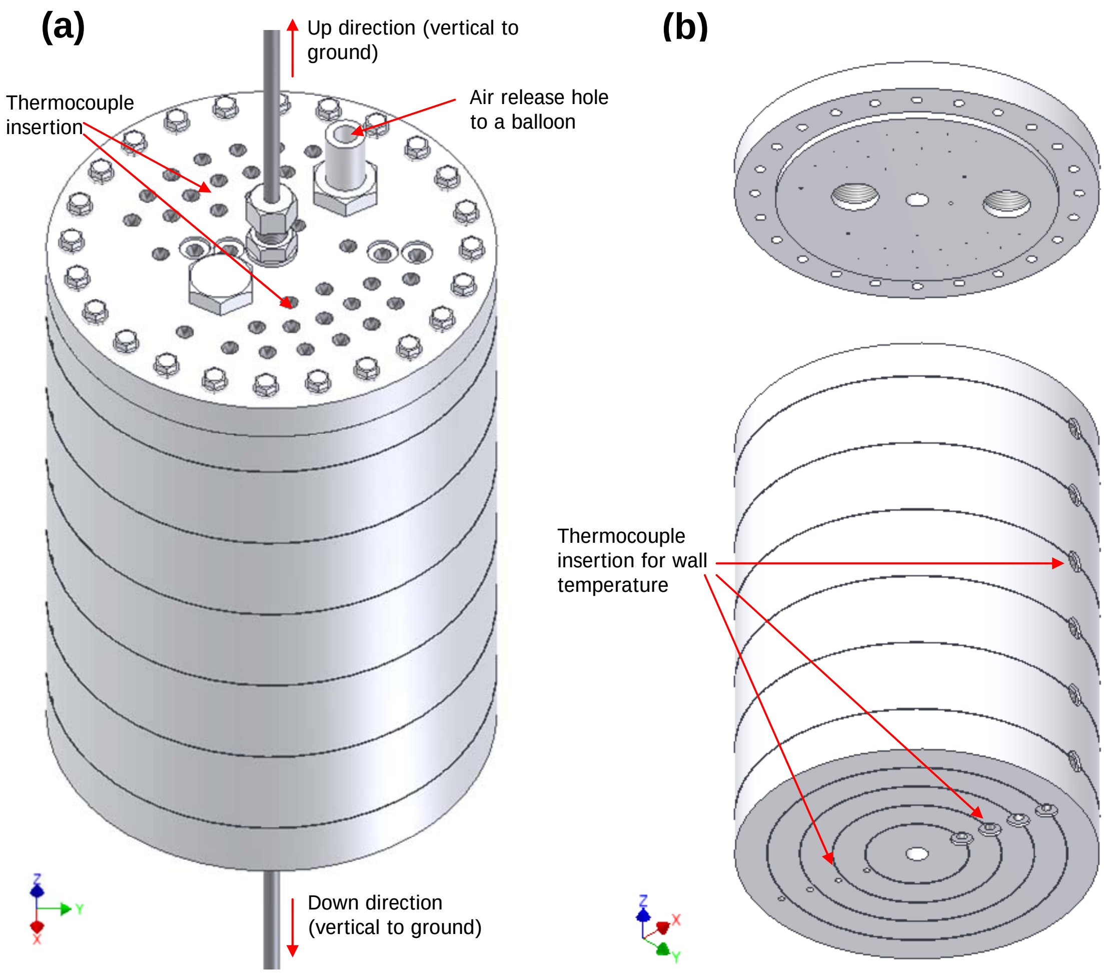

Figure 2: The heat accumulator design of the LTA for the experimental validation is shown as a CAD drawing. In (a) a diagonal view from the top with the heat transfer tube and in (b) a diagonal view from the bottom of the lid being raised from the vessel is shown. Note: The corresponding coordinate system is shown in each drawing on the left side at the bottom. Some details are labelled by red arrows.

3.2. Temperature measurement system

For validation of theoretical models of the heat transfer it is important to have enough temperature measurement points at the tube wall, at the container wall, and inside the water or ice, respectively. Measuring the tube temperature is necessary in order to determine indirectly the fluid temperature and the phase change inside the tube[7].

[6] Engineeringtoolbox, URL: http://www.engineeringtoolbox.com/ [cited 14 June 2011]

[7] Direct measurement of the fluid temperature would be much more accurate for the spatial determination of the fluid phase change. However, this method would generate turbulences in the fluid dynamics of the heat transfer which are too complex for the 1d numerical analysis models used by UPC.

Measuring a detailed spatial temperature field of the heat storage material is necessary in order to determine the spatial distribution of the temperature and the phase change. Measuring the wall temperature of the vessel wall is necessary in order to determine the heat flux through the walls, thus determining important boundary conditions for the numerical analyses.

Measurement of the wall temperature of the tube:

The thermocouples used for the measurement of the wall temperature of the tube and the container are of type K with a diameter of 0.5 mm. Figure 3 shows how and where the 28 thermocouples for the tube temperature measurement are inserted to the heat accumulator. They are inserted through the lid by PTFE seal glands as can be seen in 3 (a). The arrangement of the fittings is mirror-symmetric, so that 14 thermocouples are inserted from one quarter section and the others from the opposite quarter section. The thermocouples are fixed with rising height alternately from the one side and the other side as indicated in figure 3 (b) by the red arrows with the corresponding height levels. The black arrows indicate the radial positions of the two thermocouples which are soldered to the surface of the disc of the collar. Note that the surface of the disc is defined as the zero height level of the tube.

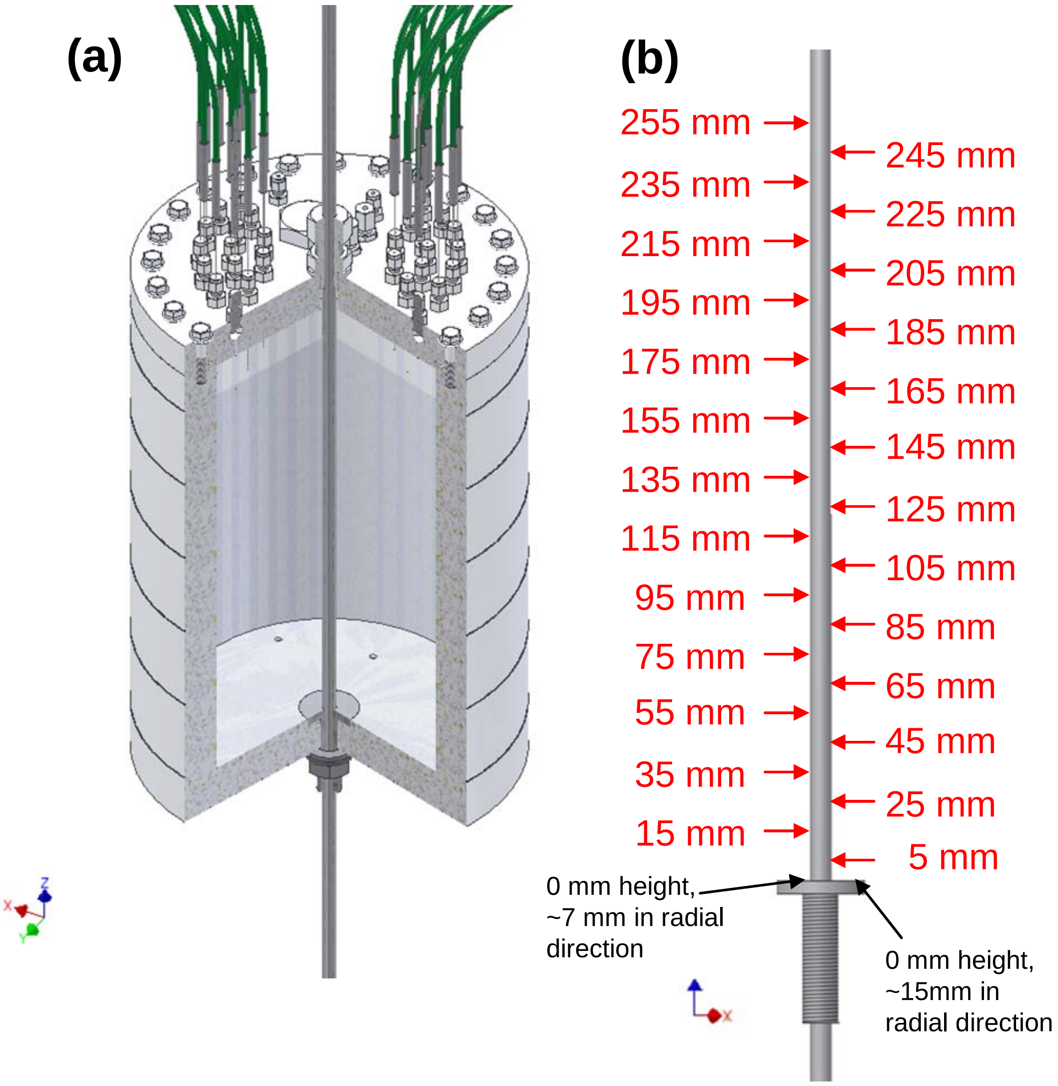

Figure 3: In (a) a three-Quarter section view of the LTA is shown. Water is filled into the vessel and 28 thermocouples are inserted from the top for the tube measurement. In (b) a diagonal view of the stainless steel heat transfer tube is shown. At the top surface of the disc of the weld-on collar two 0.5 mm thermocouples are placed (indicated by black arrows). As many as 26 thermocouples are attached to the tube above the disc (indicated by red arrows with corresponding height levels).

Measurement of the wall temperature of the container:

Measurement points in two different depths are foreseen, at 20mm (relative to the outer surface of the vessel wall) and at 0.5mm. For their attachment, two different methods are used.

The first method is shown in figure 4. The thermocouple tip is at 20 mm depth relative to the outer surface of the vessel wall. In figure 4 (a) it can be seen that there are 4 measurement points at the bottom and at the lid and 6 measurement points at the side wall of the vessel. In figure 4 (b) a diagonal cross-section through the fittings and the container wall is shown exposing how the method of fixing the thermocouples works. A spring softly presses onto a collar [6], which is welded up to the thermocouple about 8 mm behind the tip, and the tip is pressed into a hole which is of 0.6 mm in diameter and about 5.0 mm in length. The corresponding fitting is screwed into the wall and does not clamp the thermocouple.

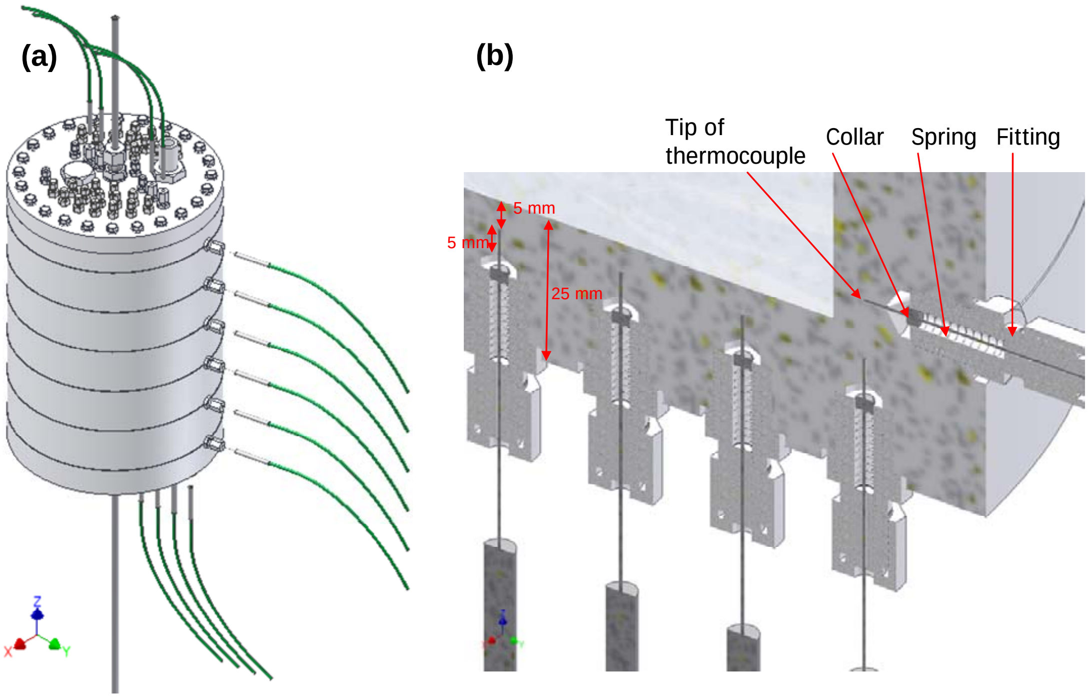

Figure 4: In (a) a diagonal view of the container with the thermocouples for the measurement of the inner wall temperature at 20 mm depth is shown. In (b) a diagonal cross-section view through the fitting is shown exposing how the thermocouples are attached into the container wall at 20 mm depth. A collar is welded onto the thermocouple and a spring presses the tip of the thermocouple into the small hole, which is of around the same diameter as the thermocouple.

The second method is shown in figure 5 which is intended for fixing the thermocouples at the outer surface of the vessel walls. The basic idea is to press the thermocouple into a milled groove of 0.5 mm in height and depth by flange screws which are placed in such a way that the screw head overlaps with the groove. The grooves have for the bottom and the lid the same radii and for the cylindrical part of the container the same heights as the thermocouple positions of method 1 shown in figure 4 with the difference that these screws are placed at the opposite side of the cylindrical housing of the heat accumulator as the ones of method 1. Assuming symmetry, it is therefore possible to determine the temperature gradient through the wall. In figure 5 (a) a general overview of the thermocouples for the surface measurement is shown. The thermocouples at the side and the bottom wall can be seen. Note that the thermocouples at the top are not shown for the sake of clarity. In figure 5 (b), the detailed view of one thermocouple being attached to the surface by method 2 is shown. The 0.5 mm thermocouple is attached in the way that it is pressed into a groove by the flange of a PTFE screw which overlaps the groove.

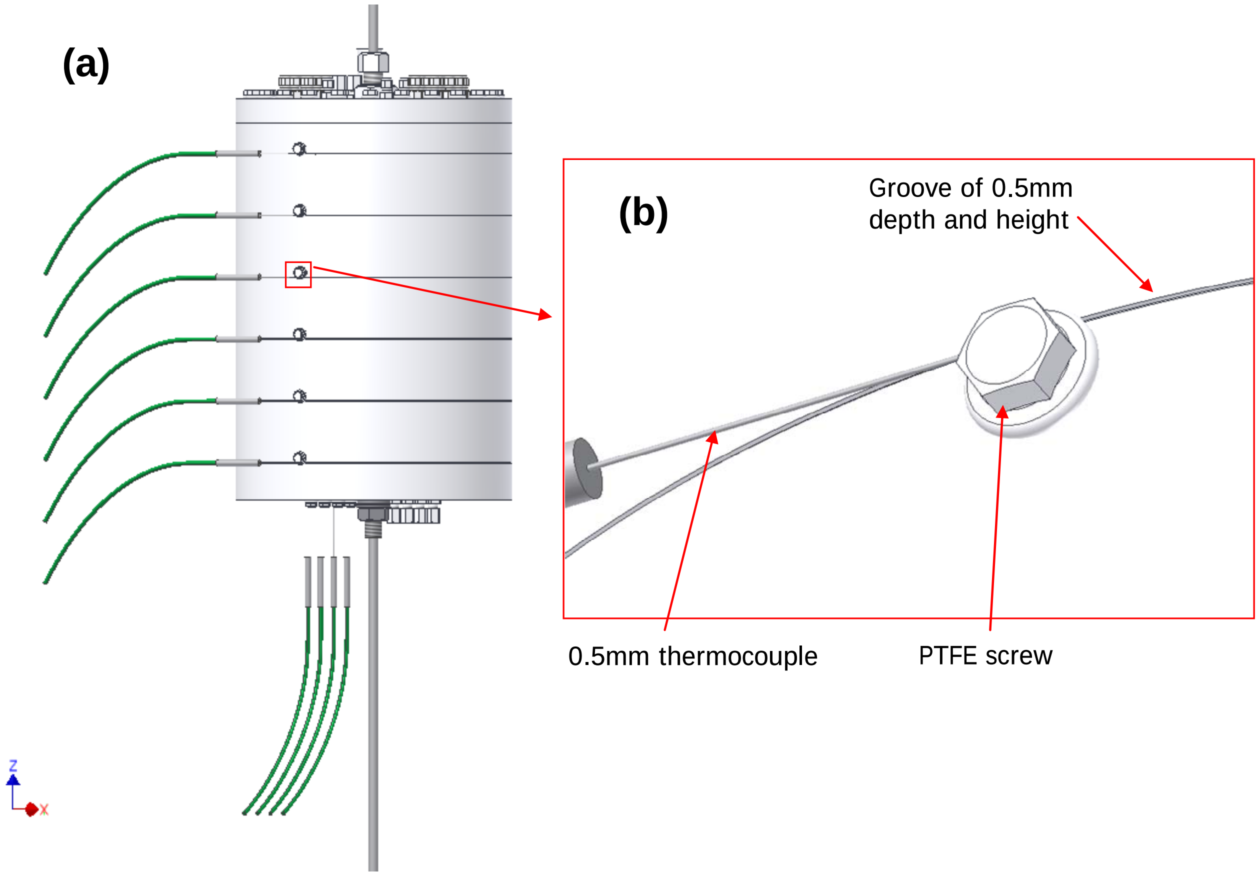

Figure 5: In (a) a side view of the container with the thermocouples for the measurement of the outer surface temperature of the container walls is shown. In (b) a detailed view of the surface temperature measurement is shown: A PTFE screw presses the thermocouple into a groove of 0.5mm depth and height. The positions of the temperature measurement points are for the bottom and the lid at the same radii and for the cylindrical part of the container at the same heights as in figure 4 for the 20 mm deep thermocouples.

Measurement of the temperature of the heat storage material:

The measurement of the spatial temperature field of the volume of the heat storage material is carried out by four 6-level thermocouple rods which are inserted from the top of the LTA parallel to the cylinder axis at different radial positions. The PTFE seal glands are screwed into the lid. The arrangement of the thermocouple rods within the vessel is shown in a CAD three-quarter cross-section view in figure 6 (a). In figure 6 (b) the arrangement of the measurement points M1 to M6 along the 6-level thermocouple rod is sketched. The measurement points are 50 mm away from each other starting 10 mm behind the tip of the rod. The tip of the rod is inserted into 5 mm deep holes at the bottom of the vessel which are of the same diameter as the rod. The vertical heights of the measurement points in relation to the zero height level of the tube are h = 5 mm, 55 mm, 105 mm, 155 mm, 205 mm, and 255 mm. Note that the total height of the inner volume is 260 mm. In figure 6 (c) a cross-section view of a sketch of the rod between M6 and the sleeve is shown. It is illustrated that one rod is made out of six single 0.5 mm thermocouples. The outer diameter of the rod is 2.0 mm.

Note that the heat accumulator is axisymmetric, so that for the numerical analyses it is not important to know the angle in which the four 6-level thermocouple rods are arranged (only the radius and the height of the measurement points). However, in order to avoid any interference with the temperature field distortion caused by the single thermocouples, which are inserted at two opposite quarter sections from the top and which are attached along these two sides of the tube, the four thermocouple rods are placed into the two other opposite quarter sections where no single thermocouple is placed. Furthermore, the angle between the two neighbouring rods in one quarter section is large enough (50°) in order to avoid any interference of the temperature field due to the neighbouring thermocouple rod.

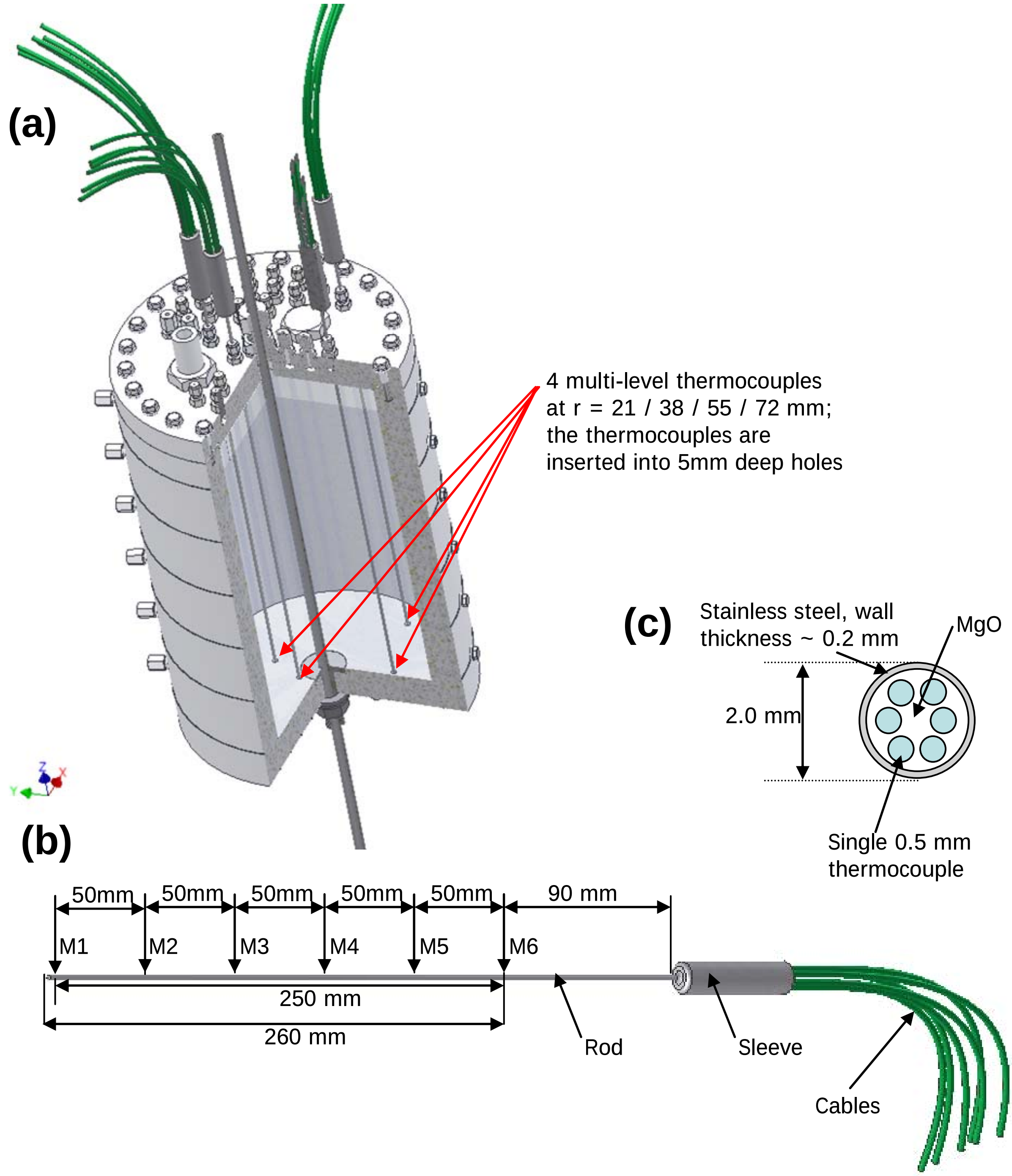

Figure 6: In (a) a three-quarter section view of the heat accumulator with the 6-level measurement rods being arranged inside the heat accumulator is shown. The red arrows indicate where the four rods are inserted to the 5 mm deep holes at the bottom. In (b) a CAD drawing of one 6-level thermocouple rod is shown. M1 to M6 label the six measurement points. The positions of the temperature measurement points are indicated by black arrows. In (c) a cross-section view of the thermocouple sheath between M6 and the sleeve – containing six single thermocouples - is shown. The main dimensions and components are specified in the sketch by black arrows.

Measurement of the properties of the heat-transfer-fluid:

The direct measurement of the properties of the heat-transfer-fluid is carried out at the two ends of the heat-transfer tube. The heat-transfer tube has a total length of around 1000 mm. For this reason, insulation foam is installed around the tube below and above the heat accumulator. It has an appropriate thickness of 45 mm and a thermal conductivity of $\lambda_{insulation}$ ~ 0.03 W/mK at 0°C and is applicable for cryogenic operation. In figure 7, the dimensions of the heat transfer tube and the LTA are illustrated. The length between the bottom of the vessel and the liquid Nitrogen bath, in which the tubes and instruments are placed in order to have an exact boundary condition for the flow of liquid Nitrogen (LN2), is around 0.5 m. The reason for the long distance is to guarantee a constant flow type of the fluid, as e.g. a fully developed laminar flow, before it enters the vessel.

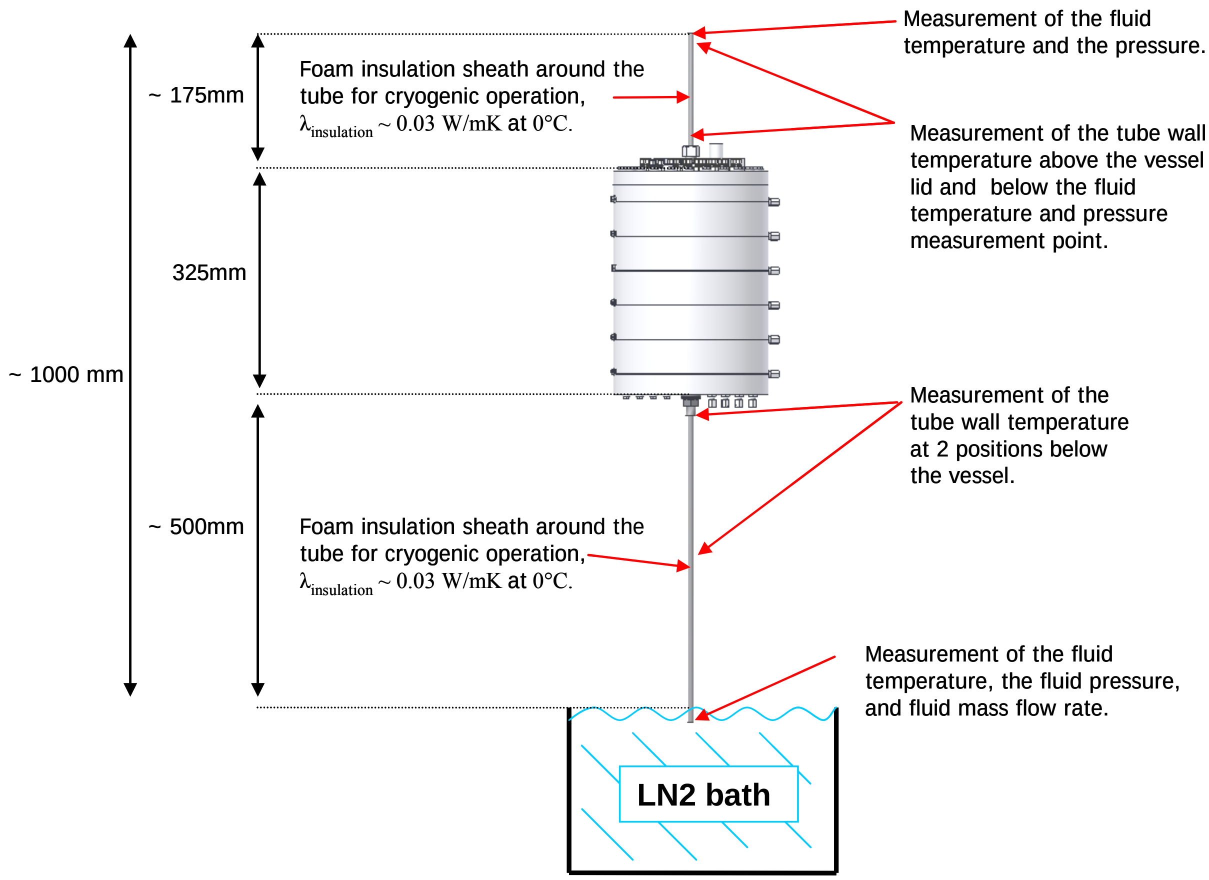

Figure 7: Side view of the LTA with the corresponding heat transfer tube. The fluid temperature and the fluid pressure are directly measured at the inlet of the heat transfer tube at around -0.5 m and at its outlet at around +0.5 m, the mass flow rate only at the inlet. The inlet of the tube is placed into liquid Nitrogen. The heat transfer tube before and after the LTA container is insulated by foam for cryogenic operation having a thermal conductivity of around $\lambda_{insulation}$ ~ 0.03 W/mK (at 0°C) and a wall thickness of around 45 mm. Red arrows indicate the details of the arrangement and black arrows label the dimensions of the tube.

The fluid properties that are measured at the lower end of the heat transfer tube are the pressure, the temperature, and the mass flow rate. At the upper end of the heat transfer tube, only the pressure and the temperature of the fluid is measured. Note that the temperature of the fluid is measured by immersing the tip of the thermocouple directly into the fluid. This method causes turbulences in the fluid and is therefore not applicable for the fluid temperature measurement at other positions of the heat-transfer-fluid.

4. Fluid system

The technical requirements for the fluid system for unloading the heat accumulator in the validation experiments can be derived from section 2. For the hot run of the engine it is necessary that for a duration of 1800 s liquid Nitrogen (acting as a validation experiment replacement for LOX) flows at a constant mass flow rate of 0.7 g/s through the heat transfer tube at a pressure of 3 bar. For the uploading phase of the heat accumulator, a heat transfer fluid flow is necessary for a duration of 4500 s, which transports the heat $\dot{Q}_{ff,H2}$ from the fuel cells to the heat accumulator having a maximum temperature of around +80°C. The heat transfer medium used for the uploading phase is gaseous Nitrogen since it does not freeze within the temperature range from -196°C till +80°C, which is the most important criterion for cooling the fuel cells. The lower temperature limit of the heat transfer fluid for the uploading phase is because of the theoretical lowest temperature of the heat accumulator when it is completely unloaded. As the temperature of the heat accumulator is changing during its uploading process, the mass flow rate would have to be adapted in order to keep $\dot{Q}_{ff,H2}$ constant. However, an active control of the mass flow rate could lead to some mass flow rate oscillations and therefore, to potential inaccuracies. For this reason, a constant mass flow rate is chosen instead which can be maintained easily by means of a sonic nozzle. The most important interval for the uploading process is the time of the phase change from ice to water. For this interval, a constant mass flow rate of around 5 g/s can be assumed for $\dot{Q}_{ff,H2}$ =262 W as will be shown in the following.

Cooling down mode:

Liquid Nitrogen having an inlet temperature of around -196°C is flowing with a constant mass flow rate of 0.7 g/s through the heat-transfer tube of the heat accumulator. The Liquid Nitrogen will heat up and vaporize during its way through the heat accumulator. Therefore, the water or, respectively, the ice inside the LTA will be cooled down at the same time. The pressure at the exit of the heat transfer tube is kept constantly at 3 bar. For example, if the outlet temperature of the Nitrogen (gaseous) is 0°C, the heat flux to the Nitrogen is around 300 W when it is flowing through the accumulator at a constant mass flow rate of 0.7 g/s and at a pressure level of 3 bar [8]. This operation mode must be possible for a time interval of at least 1800 s.

Heating up mode:

Gaseous Nitrogen having an inlet temperature of +80°C is flowing with a constant mass flow rate of around 5.0 g/s through the heat-transfer tube of the heat accumulator. The pressure at the exit of the heat transfer tube is kept constantly at 3 bar. The gaseous Nitrogen will cool down during its passage through the heat accumulator and at the same time the ice or, respectively, the water inside the LTA will heat up. The magnitude of $\dot{m} =$ 5.0 g/s is derived by using the time derivative of the thermal energy equation $\dot{Q}_{ff,H2} = \dot{m} c_{p,gas} \Delta T_{gas}$ with $c_{p,gas}$=1.04J/gK and ΔT_{gas}~50K (temperature drop from +80°C to +30°C) at a fluid pressure of 3 bar.

Conclusions

The basic ideas for using heat accumulators for in-space propulsion for the transfer from LEO to GEO have been reviewed and a quantitative example of the functional principle for the method was shown. It has been demonstrated how to determine the size of the heat accumulator for this operation. Furthermore, detailed information of the prototype heat accumulator being optimized for experimental validation has been presented. This setup makes it possible to study the efficiency of such complex thermal systems with fluid flowing through one tube that undergoes phase change from liquid to vapour and of a heat accumulator material that changes between liquid and solid states.

Acknowledgement

The research leading to the results shown in this publication has received funding from the European Union Seventh Framework Programme under grant agreement n° 218849.

[8] $\dot{Q} = \dot{m}(c_{p,liquid} \cdot \Delta T_{liquid} + q_{latent} + c_{p,gas} \cdot \Delta T_{gas}) \sim 0.7\ \text{g/s} \cdot (2.1\ \text{J/gK} \cdot 11\ \text{K} + 199\ \text{J/g} + 1.1\ \text{J/gK} \cdot 185\ \text{K})$ with a boiling temperature of -185°C at 3 bar, the latent heat q_{latent}, and average values for the heat capacities c_p for the gaseous and the liquid phase.

References

[1] M. Muszynski, P. Alliot. 2010. The In-Space Propulsion (ISP-1) project. In: *61st International Aeronautical Congress*. IAC-10.C4.3.10

[2] D. Valentian, J. M. Conrardy, D. Vuillamy, H. Goislot, P. Garceau, R. Bec. 2006. Low Cost Cryogenic PropulsionModule with Fuel Cell Power Supply. *42nd AIAA/ASME/SAE/ASEE Joint Propulsion Conference & Exhibit.* AIAA 2006-4906

[3] F. Caramelli, M. van Pelt, A. Fesidis, G. Chirulli, A. Sirbi, A. Atzei. 2006. Long Term Cryogenic Propulsion for Interplanetary Missions. *42nd AIAA Joint Propulsion Conference* AIAA 2006-5129.

[4] S. Morales-Ruiz, J. Castro, J. Rigola, C. D. Pérez-Segarra, and A. Oliva. 2011. Numerical resolution of the heat accumulator in a Low Cost Cryogenic Propulsion (LCCP) system. *4th European Conference for Aerospace Sciences*. ID 185

[5] R. Velraj, R. V. Seeniraj, B. Hafner, C. Faber and K. Schwarzer. 1998. Heat transfer enhancement in a latent heat storage system. *Solar Energy* Vol. 65, No. 3, pp. 171–180, 1999

[6] D.Suslov, A.Woschnak, J.Sender, M.Oschwald. 2003. Test specimen design and measurement technique for investigation of heat transfer processes in cooling channels of rocket engines under real thermal condidtions. *39th AIAA/ASME/SAE/ASEE/ JPC Conference and Exhibit*. AIAA 2003-4613