MECHANICALLY PUMPED TWO-PHASE THERMAL CONTROL LOOPS

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ABSTRACT. The contents of this paper constitute the third part of the author's lectures on two-phase flow and heat transfer in terrestrial and aerospace-related thermal control system developments. These lectures were presented during the ICHMT Course on Passive Thermal Control (PTC-03) in Antalya, Turkey, October 22 - 25, 2003. This third part of the lecture focuses on mechanically pumped loop issues.

BACKGROUND

Already for more than a century two-phase heat transfer systems are frequently being applied in the power industry and the process industry. Novel mechanically pumped two-phase heat transfer system developments were started around 1980 for applications in the Space Station [1-5].

Multiphase flow, the simultaneous flow of the different phases (states of matter) -gas, liquid and solid- strongly depends on the level and direction of gravity, as these influence the spatial distribution of the phases which have different densities. Of major interest for aerospace applications are the more complicated liquid-vapour or liquid-gas flows, that are characteristic for aerospace thermal control systems, life sciences systems and propellant systems. Especially for liquid-vapour flow in aerospace two-phase thermal control systems, the phenomena are extremely complicated, because of heat and mass exchange between the two phases by evaporation, condensation or flashing. Though many publications discuss two-phase flow and heat transfer, publications on the impact of reduced gravity and super-gravity are scarce. This is the main driver to investigate the impact of various gravity levels.

The various heat and mass transfer research issues of two-phase heat transport technology for space applications are discussed in the next chapters. It is focused on the most complicated case: Liquid-vapour flow with heat and mass exchange. Simpler cases, adiabatic or isothermal liquid-vapour flow or liquid-gas flow, can be derived from this liquid-vapour case, by deleting terms in the constitutive equations.

A thermal utility or thermal bus is a pumped fluid, high-capacity heat transport system, serving as a common temperature controlled heat sink or source to more than one payload, usually to many payloads. Such thermal management systems for future large spacecraft have to transport large amounts of dissipated power (gathered at many dissipating stations) over large distances to the heat sinks, the radiator(s), where the heat is radiated to the cold space. Pumping pressures can be realised by mechanical pumps, capillary action or another means, like osmotic pumps or compressors [1-3].

Conventional single-phase thermal busses are mechanically pumped. They are based on the heat capacity of the working fluid, they are simple, well understood, easy to test, inexpensive and low risk. A very serious disadvantage is the required precise ordering of the modules in the thermal circuit. Changes in location or heat load of any individual module (station) will interfere with all other downstream stations. A prescribed, desired width of the isothermality band of the system (and its components) and the heat load determine the size of the pumping system [3]. Consequently, for proper thermal control with small end-to-end temperature differences to limit radiator size and mass, they require heavy thick walled, large diameter lines and noisy, heavy, high power pumps, hence leading to enlargement of solar arrays and radiators. Alternatives for mechanically pumped single-phase systems are mechanically pumped two-phase systems, pumped loops accepting heat by working fluid evaporation at heat dissipating stations and releasing heat by condensation at heat demanding stations and at radiators, for the heat rejection into space. Such systems, relying on the heat of vaporisation, have small end-to-end temperature differences (operate nearly isothermally) for large variations in direction and magnitude of the heat exchange with the individual payloads. The pumping power is reduced by orders of magnitude (as compared to single-phase systems), thus minimising radiator and solar array sizes. In mechanically pumped single-phase systems caloric the heat transport is by caloric heat of the liquid. In two-phase systems the transport is by the latent heat of evaporation and condensation. This implies, for dissipating stations in series in a single-phase system, a temperature increase in the downstream direction of the loop. For two-phase systems, with evaporators in series, it means an increase of the vapour quality in the downstream direction, accompanied by a (usually small) decrease of the saturation temperature. A two-phase thermal bus can serve several modules by, depending on operating conditions of any particular module, extracting heat from or dumping heat into it. Components can be coupled to the system to transfer heat from hot to cold regions. The ordering of modules in the circuit is hardly important, certainly not crucial. The stations can be arranged in a pure series (Fig. 1), a pure parallel (Fig. 2), or in a hybrid configuration, being a combination of parallel and series.

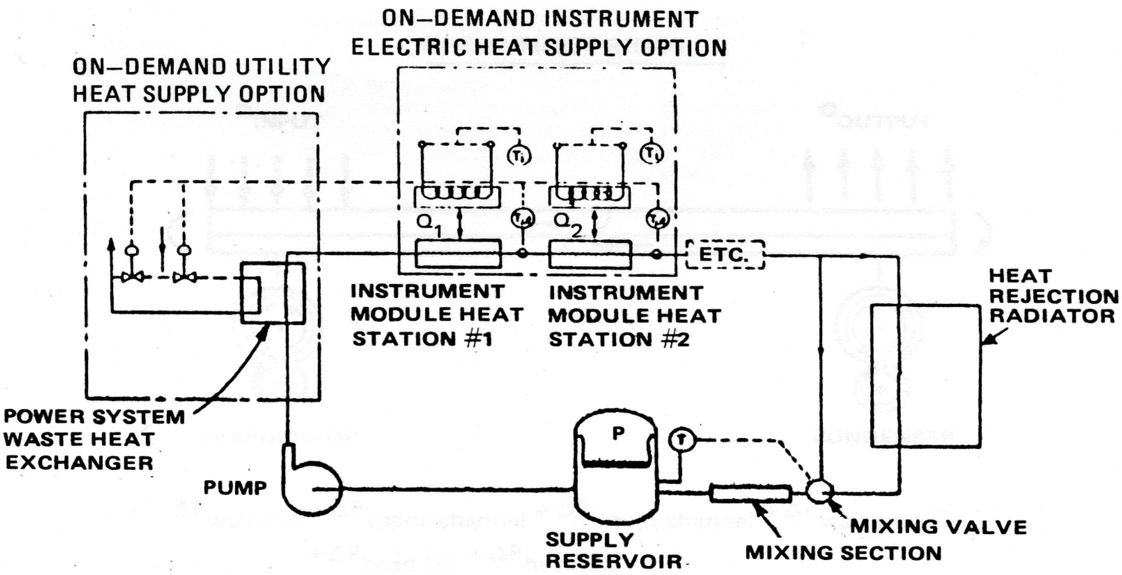


Figure 1. Schematic of a mechanically pumped two-phase thermal bus series configuration [1]

As compared to the parallel concept, the series concept (originally an ammonia, serial thermal bus was planned to be the central thermal management system of the Space Station) has the advantage of simplicity and shorter total piping length. But it has the disadvantage of a larger pressure drop (unless a larger piping diameter is chosen), some (minor) restrictions with respect to the sequence of stations in the loop, and a bit more complexity with respect to modularity. The advantage of the parallel concept is its modular approach, in which the branches with dissipating stations (evaporators/cold plates) or heat demanding stations (condensers/ radiators) simply can be added or deleted. But it also has the drawbacks of the tubing length, and of the complex feedback control system to adjust the vapour quality of the two-phase mixture in the exiting line of each cold plate. The latter control system is necessary to keep these mixture qualities close to a chosen value to guarantee the proper performance of the thermal bus, by preventing system instabilities and oscillations.

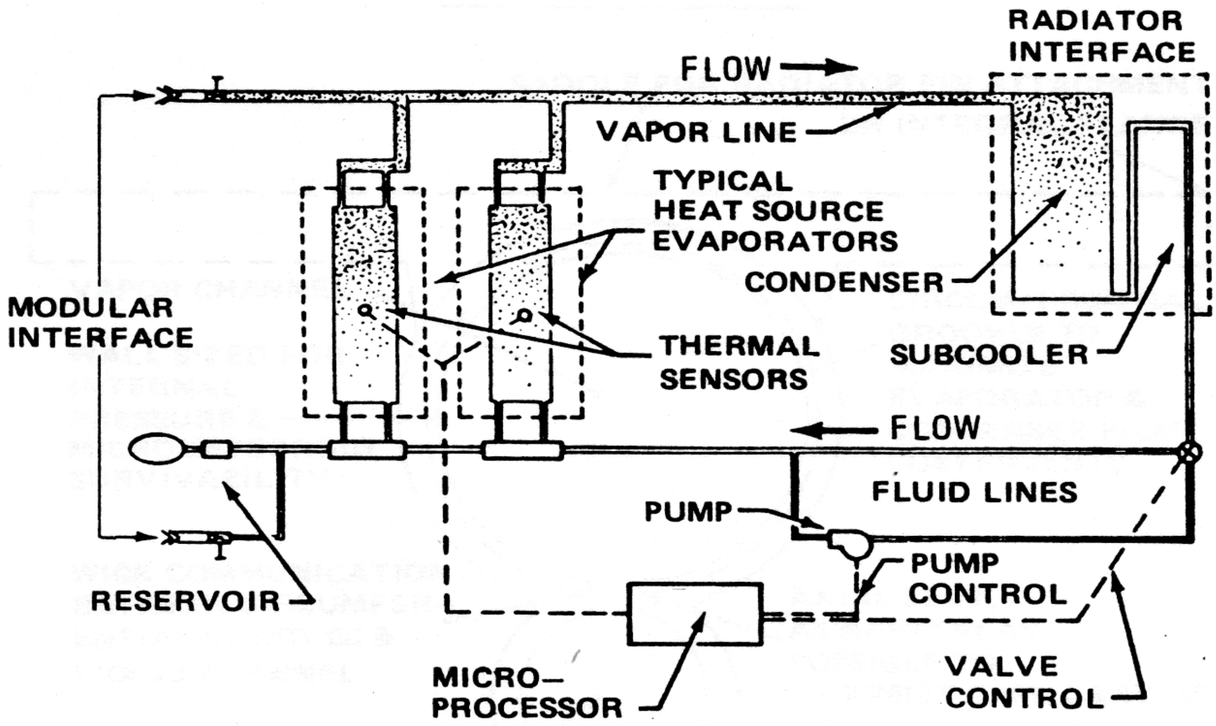


Figure 2. Schematic of mechanically pumped two-phase thermal bus, parallel configuration [4]

Most important issues in developing two-phase thermal busses were formulated in the early 1980’s [2-5]. Though they focused on developments for Space Station and other manned/unmanned space platforms, their outcomes can be usefully applied to develop other dedicated thermal control systems. The important general and more detailed issues can be summarised by:

- Evaluation of candidate techniques, identification and generation of promising thermal bus concepts.
- Comparison of promising concepts with respect to mass, sizing, complexity, reliability, required redundancy (to meet lifetime and maintenance specifications).
- Identification of critical items for the principle elements of two-phase thermal management systems.

The latter elements are:

- The transport system or thermal bus, which can be pure parallel, pure series, or hybrid.
- Radiators, which can be direct condensation radiators or indirect heat pipe radiators.
- Heat exchangers between the various instruments/modules and the thermal bus: Via a cold plate or a direct fluid coupling, via a temperature-controlled enclosure or via a self-contained instrument fluid loop/cold plate configuration.

Major critical items were the development of reliable mechanical and capillary pumps, and getting a better understanding of two-phase flow and heat transfer in micro-gravity. These two-phase technology development issues were investigated in the last 17 years, by NLR or with NLR involvement.

An overview [6], containing many references to relevant NLR publications, summarises these NLR activities that include research on:
- The impact of gravity level and direction on two-phase flow and heat transfer.
- Thermal/gravitational modelling and scaling of two-phase heat transport systems and system components, and modelling of the two-phase pressure drop, as a function of the vapour quality.
- Development of two-phase (R114, NH_3, ethanol, CO_2) test rigs for experimenting and calibration of components, vapour quality sensor, high-efficiency low-pressure drop condenser, (in-) direct radiators.
- Development, testing, in-orbit demonstration of two-phase technology, and evaluation of flight results of TPX [7] and the Loop Heat Pipe Flight eXperiment [8]. The latter was by a team led by Dynatherm-DTX, consisting of Hughes Space & Communications, the Naval Research Laboratory, two USAF Laboratories, BMDO, three NASA Institutes, and NLR.

Two-phase thermal control technology is the major thermal control innovation of the last decade [9]. Two-phase systems have reached a certain level of maturity and they are becoming more and more accepted as reliable heat transport systems. However, the design of a two-phase flow loop is still rather difficult and cumbersome due to the character of two-phase single-component flow dynamics and heat transfer. In the two-phase lines of mechanically pumped loops and in the condenser of any two-phase loop, the flow pattern dependent heat transfer is of great importance for the definition of a particular thermal management system.

The only two important near-future mechanically pumped two-phase heat transport applications are European, i.e.:
- The two-phase ammonia thermal control system of the Russian segment on ISS [10-12].
- The hybrid two-phase carbon dioxide thermal control loop of the AMS-2 Tracker Thermal Control System [13-19]. AMS-2, the Alpha Magnetic Spectrometer experiment [20] planned for a five years mission as attached payload on ISS, is an international experiment searching for anti-matter, dark and missing matter. AMS-2, an improved version of AMS-1, has flown on STS-91. It consists of several particle detector systems, the most crucial one being the Tracker.

On this Tracker Thermal Control System (TTCS), that constitutes the major contents of this lecture it is remarked that:
- In mechanically pumped two-phase loops, the flow pattern dependent heat transfer coefficient for convection flow boiling is reported to be between say 4 and 5 $kW/m^2.K$ [21]. This is not true for refrigerants (to be used in the TTCS) at qualities below 0.15 for which the value can increase to say 20 $kW/m^2.K$ at qualities of less than 0.03 [22]. Data from experiments with CO_2 in small diameter tubes confirm this [23]. The above implies that a mechanically pumped system has to be designed such that any evaporator exit quality is below 0.15 (or even much lower) for efficiency reasons.
- In the case of very lengthy lines in mechanically pumped two-phase loops the pressure (saturated temperature) gradient has to be kept small to guarantee a small end-to-end pressure (saturated temperature) difference to meet the requested isothermality, and to keep the evaporator exit vapour quality below 0.15. The latter is because in flowing refrigerants the vapour quality usually increases with pressure decay. This issue (called flashing) will be discussed later, since it one of the crucial differences between single- and two-component two-phase flow [24].
- A dedicated hybrid two-phase loop will guarantee the required isothermality and quality range.

Many development supporting, scientific experiments were also carried out in the last decade, within research programmes concentrating on the physics of microgravity two-phase flow and heat transfer. Experiments were done in drop towers, during Microgravity Science Laboratory missions on STS, and during reduced-gravity aircraft flights. But the usefulness of the results of most of these experiments is unfortunately only of limited use for two-phase heat transport systems developments, since they suffer from the severe restriction of short experiment duration, or as they pertain to two-component not to single-component two-phase flow.

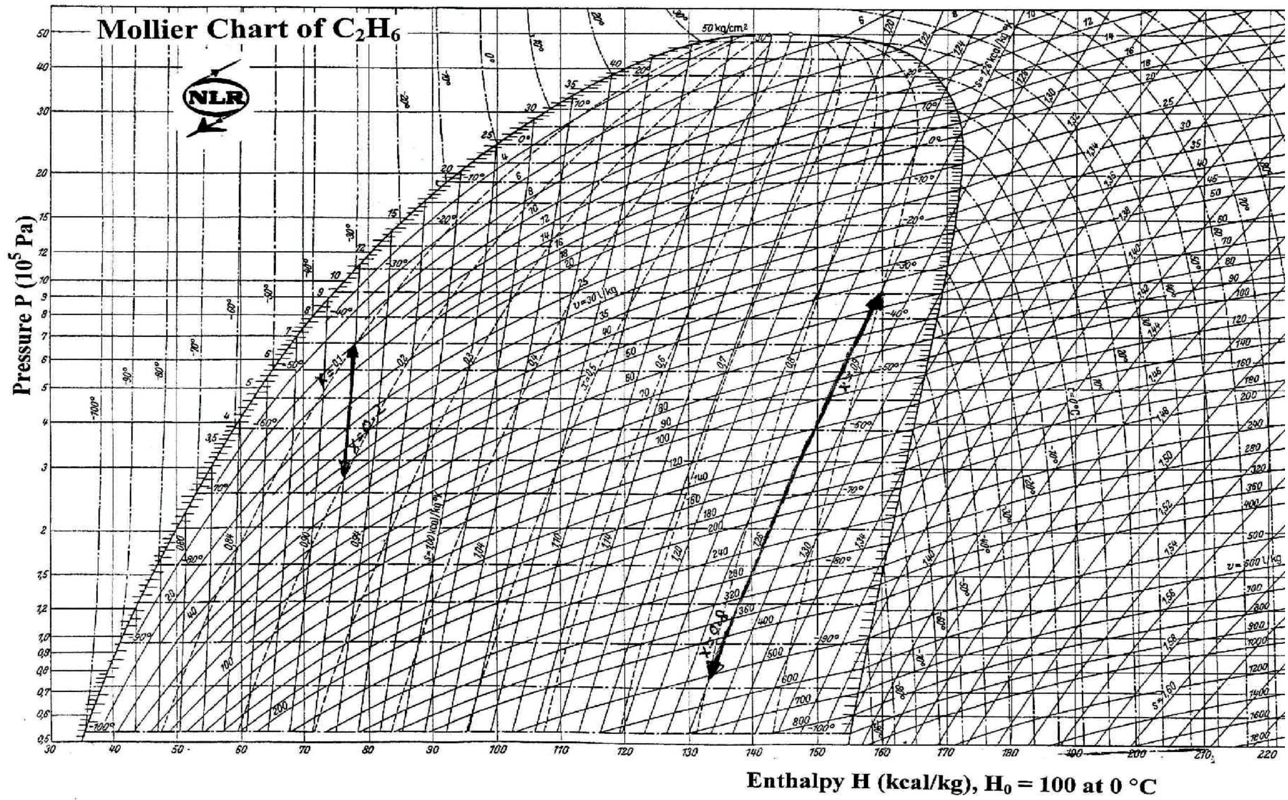


Figure 3. Mollier chart of ethane (1 kcal = 4.17 kJ)

The aforementioned effect of flashing, being mixture quality change by other mechanisms than heat addition or withdrawal, can be illustrated as follows:

- For steady state, adiabatic two-component flow through a tube, the gas flow rate remains constant in each cross-section hence the entering and exiting gas flow rates are equal. The same is valid for the liquid flow rate. Consequently the quality remains constant. The effect of the pressure gradient along the tube (needed to overcome frictional losses) is only an increase of the void fraction (the relative volume of the gas) in the down-flow direction.
- In case of steady state, adiabatic single-component flow through a tube, only the total mass flow rate remains constant in each cross-section, the quality changes along the flow path. For most fluids this means a quality increase. Ethane is an exception, as illustrated by the Mollier chart of ethane (Fig. 3). The isentropic (reversible) flow path indicated at the left side shows a quality increase from 0.1 to 0.2, caused by the pressure (temperature) decay. But the flow path at the right side shows a quality decrease from 0.9 to 0.8. Around 0.7 the quality remains constant. This effect, called flashing, is more pronounced in the more realistic case of non-reversible flow conditions. The latter can be explained for steady state single component two-phase flow (mass flow rate m) through a line or valve, by writing according to the 1st Law of Thermodynamics for a steady state process:

$$Q = m\,(H_e - H_i) + M + m\,(\Delta E_k + \Delta E_p)$$

The mechanical power M=0 in the line/valve. Reasonable hypotheses are negligible potential and kinetic energy change (($\Delta E_k = \Delta E_p = 0$), plus Q = 0 if the flow is adiabatic (no heat exchange with the surroundings). This means that enthalpy keeps constant. It can only change if you exchange heat between the fluid and surroundings.

One can write according to the 2nd Law of Thermodynamics, still for steady state, but with heat exchange Q at temperature T with the surroundings:

$$Q/T + S_{gen} = m\,(S_{out} - S_{in}) \qquad (2)$$

Hence for an adiabatic (Q = 0, hence irreversible)) real process the generated fluid entropy S_{gen} increases from inlet S_{in} to S_{out}. This process is absolutely irreversible (there is anyway a pressure drop due to friction) and cannot be idealised as isentropic. The above means that a real irreversible steady state adiabatic single-component two-phase process follows the vertical (isenthalpic) lines instead isentropic trajectories. This implies a larger quality increase as compared to reversible flow. Also for ethane it means that there is only vapour quality increase in the entire vapour-liquid co-existence region.

TWO-PHASE THERMAL CONTROL LOOP FOR THE AMS-02 TRACKER SYSTEM

The Alpha Magnetic Spectrometer AMS-02 [20] is an international experiment, led by Nobel Prize laureate Samuel Ting of MIT, searching for anti-matter, dark matter and lost matter. It is a particle detector for high-energy cosmic rays (Figs. 4, 5) consisting various sub-detectors, being the (Silicon) Tracker, Time of Flight (ToF) system, Veto Counters, Transition Radiation Detector (TRD), Synchrotron Radiation Detector (SRD), Ring Imaging Cherenkov Counter (RICH), Anti-Coincidence Counter (ACC), and the Electromagnetic Calorimeter (EC).

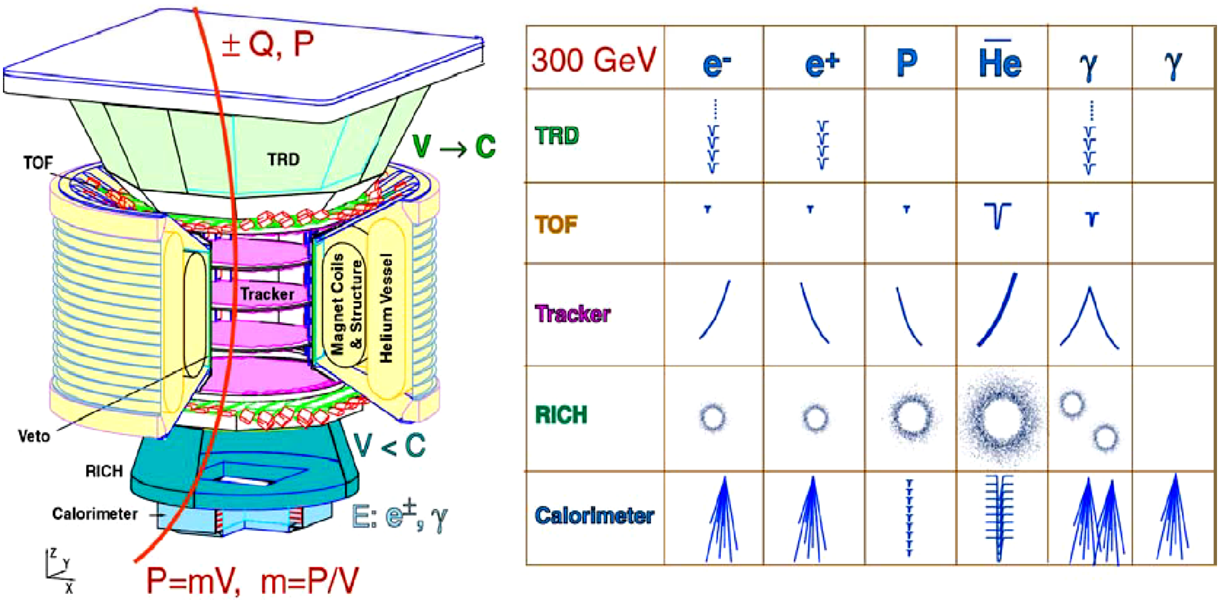


Figure 4. Alpha Magnetic Spectrometer AMS-2: Particles to be detected by signals of the different detectors (electrons, positrons, protons, Helium nuclei and gamma rays)

The AMS demonstration experiment, AMS-01, has successfully flown in June 1998 on the Space Shuttle Discovery STS-91 (Fig. 6a). AMS-02 is an improved (resolution) version of AMS-1. AMS-2 is manifested on Shuttle flight UF-4.1 for a 3 to 5 years mission as attached payload on the truss of the International Space Station ISS (Fig. 6b).

The AMS-2 thermal issues are far more demanding and critical than in AMS-1, because of the replacement of the original (heavy, high thermal capacitance) magnet by a liquid Helium II cooled super-conductive magnet, and by the long mission duration. Therefore a team consisting of NLR,

NIKHEF, Geneva University and IFN Perugia is developing a cooling system for the most critical part, the so-called Tracker Thermal Control System TTCS.
The TTCS involvement offers NLR the possibility to use two-phase thermal control expertise obtained in the past for the challenging task to develop and operate an advanced, demanding system like the TTCS, probably being the first full- size mechanically pumped two-phase thermal control system in space. NLR joined the AMS collaboration because it offers, in addition, the possibility to do scientific research with the two-phase cooling loop during the various dormant periods in the AMS experimentation. The to be gathered information is expected to yield a far better understanding of the physics of two-phase flow and heat transfer in a low-gravity environment, The latter is essential for the development of reliable two-phase thermal control systems for future spacecraft applications.

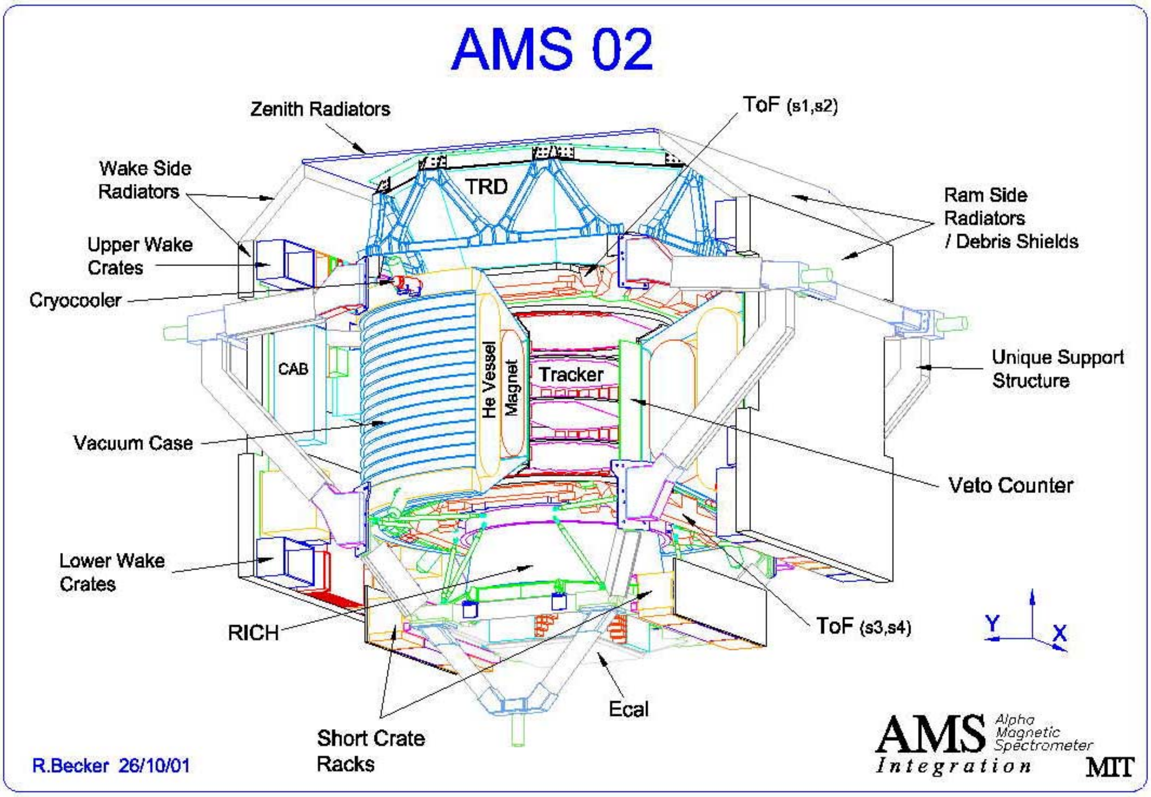


Figure 5. Alpha Magnetic Spectrometer AMS-2

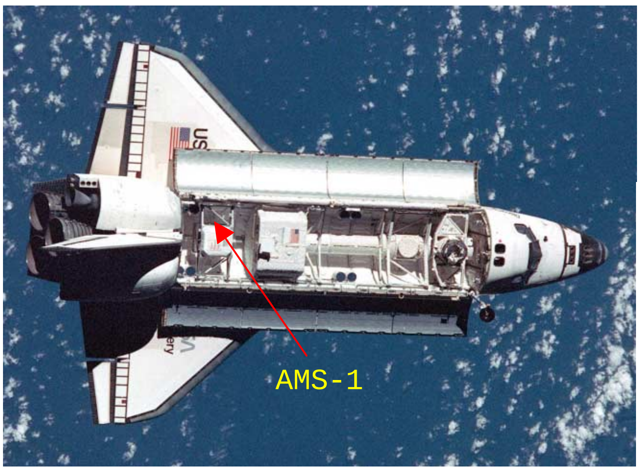


Figure 6a. AMS-1 payload aboard STS-91

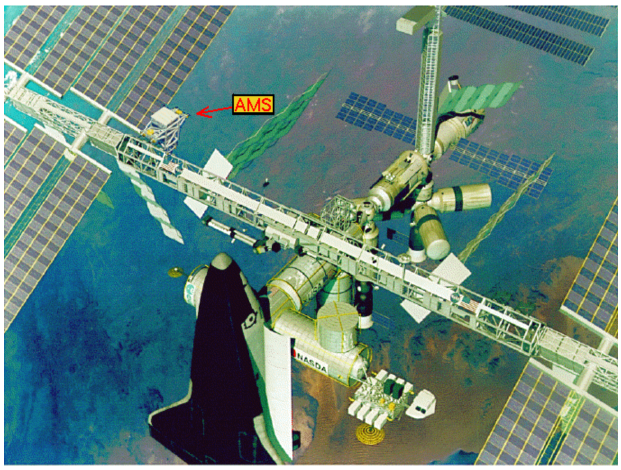
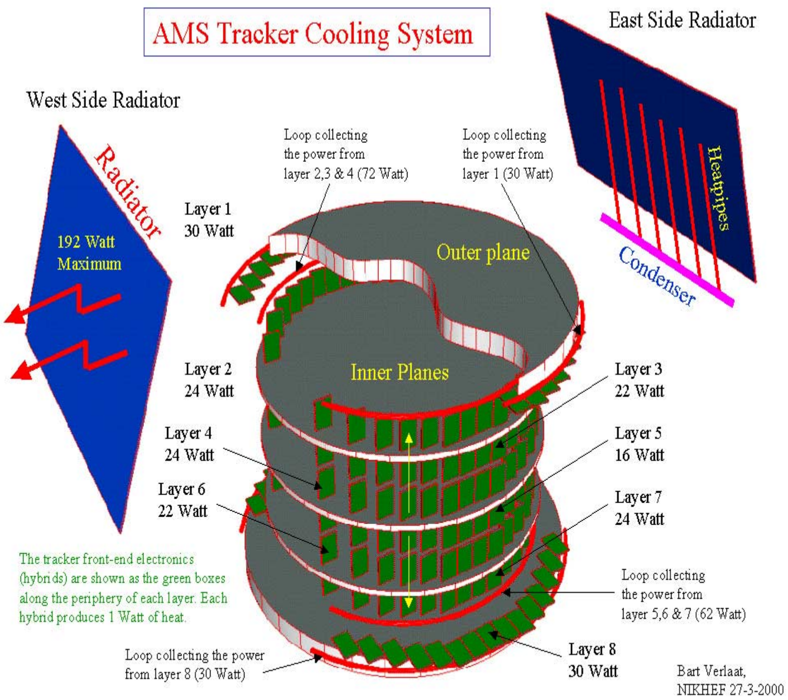


Figure 6b. AMS-2 location on ISS

The Tracker, located inside the vacuum case, is surrounded by the cryogenic magnet, which is not allowed to receive any heat from inside. Moreover the Tracker has severe requirements with respect to spatial and temporal temperature gradients. This and the existing complicated three-dimensional configuration, requires that the power dissipated inside the Tracker has to be removed to two thermally out of phase radiators (one in the RAM, one in the Wake direction) to be dumped into space. This task could be done by a mechanically pumped two-phase loop system, by a mechanically pumped liquid loop and by a capillary pumped loop system. The latter system requires heat collecting heat pipes to transport the dissipations from the silicon front-end electronics to the capillary system, as a capillary system can't properly handle evaporators (heat sources) in series. In addition, a parallel, capillary system [2, 3] leads to an unacceptable tubing length and mass, which can not be accommodated by the already existing 3-D Tracker configuration. To meet the isothermality requirements, the liquid loop needs large diameter, thick-walled tubing. Apart from its unacceptable mass, the existing AMS configuration does not offer enough spacing to accommodate large diameter liquid loop lines, because the chosen system has to be installed in two-fold to guarantee the full redundancy requirement.



Silicon wafer requirements:

- **Operating temperature:** -10 ºC / +25 ºC
- **Survival temperature:** -20 ºC / +40 ºC
- **Temperature stability:** 3 ºC per orbit
- **Maximum accepted gradient between any silicon:** 10.0 ºC
- **Dissipated heat:** 2.0 W EOL

Hybrid circuit requirements:

- **Operating temperature:** -10 ºC / +40 ºC
- **Survival temperature:** -20 ºC / +60 ºC
- **Dissipated heat:** 144 W, 0.75 W per hybrid pair

Figure 7. Silicon Tracker thermal issues

A schematic of the Tracker configuration and the requirements are depicted in figure 7. The currently valid requirements are: For the silicon wafer:, operating temperature 263-298 K, survival temperature 253-313 K, temperature stability 3 K per orbit, maximum gradient between any silicon: 10 K, and dissipated heat 1.5 W End of Life; for the hybrid circuit, operating temperature 263-313 K, survival temperature 253-333 K, dissipation 144 W total (± 10%), 0.75 W nominal per hybrid pair; for the two Star Trackers , operating temperature 263-313 K, survival temperature 253-333 K, dissipation 3.4 W each.

Keeping this in mind and following the earlier TTCS publications [3, 4], it can be said that:

- A series or hybrid two-phase Mechanically Pumped Loop (MPL) is well compatible with existing Tracker hardware. It is characterised by minimal material inside or near the tracker field of view. It is directly connected to the thermal bars, hence no additional heat collector needed. Multiple source heat input is possible, with minimum T-gradients (order of magnitude 1 K). It

has also the possibility to implement a fully redundant system. Costs and mass are relatively low. The only drawback is the mechanical pump.
- A Single-Phase (liquid) Mechanically Pumped Loop (SPL) has more or less the same layout as the MPL option, so it is relatively easy to fall back on the SPL solution, in case of unforeseen (serious) problems with the MPL development. It has the possibility of parallel and counter-current flow system set-up. It is a low-risk design, as there is sufficient experience in space with SPL's. Main drawbacks are the far larger temperature gradients (order of magnitude 10 K), as compared to the nearly isothermal MPL, and larger dimensions, mass, and the serious conflict with the full redundancy requirement.
- Any parallel two-phase system (MPL, LHP, CPL) can not to accommodate the existing Tracker hardware multiple location heat input, by it self in one stage, as of the huge mass and (not available) space needed, induced by redundancy. A two-stage approach needs an additional heat collector, heat pipe or TPG-flange, leading to serious mass increase and integration problems.

The above makes obvious that by far the best solution is the series or hybrid two-phase MPL. A parallel or hybrid SPL is a possible back-up solution, but at the cost of more massy and lengthy lines and larger pumps. Parallel concepts are non-recommendable or impossible solutions. CO_2 has to be the working fluid because:
- It is considered to replace Freon-like refrigerants, as it is environment friendly and non-toxic. It is used for nuclear power plant cooling, as it is inert for radioactive radiation. For AMS-2 this means no ISS safety-related problems.
- It has a very low liquid/vapour density ratio, Order (1-10), being profitable for a series 2-phase system; its alternative, ammonia: Order (10^2-10^3).
- CO_2 experience was gained at NIKHEF, where tests have proven the concept feasibility of CO_2 cooling for the LHCb Vertex detector. For the Tracker this means small tube dimensions (3 mm OD) in case of 2 loops, low temperature drops (< 1 K) and low pumping power (< 10 W).

In addition it is remarked that:
- The basic difference between mechanically pumped single-phase (caloric heat transport by the liquid) and two-phase systems (transport by latent heat of evaporation/ condensation). This implies for dissipating stations in series in a single-phase system a temperature increase in the downstream direction of the loop. For two-phase systems, with evaporators in series, it means an increase of the vapour quality in the downstream direction, accompanied by a (usually small) decrease of the saturation temperature.
- In mechanically pumped two-phase loops, the flow pattern dependent heat transfer coefficient for convective flow boiling is reported [6] to be between say 4 and 5 $kW/m^2 \cdot K$. This is not true for refrigerants (to be used in the TTCS) at qualities below 0.15 for which the value can increase to say 20 $kW/m^2.K$ at qualities of less than 0.03 [6, 7]. Data from experiments with CO_2 in small diameter tubes confirm this [8]. The above implies that a mechanically pumped system has to be designed such that any evaporator exit quality is below 0.15 (preferably even much lower) for efficiency reasons.
- In the case of very lengthy mechanically pumped two-phase loop lines, the pressure (saturated temperature) gradient has to be kept small to guarantee a small end-to-end pressure (saturated temperature) difference. This is to meet the requested isothermality, and to keep the evaporator exit vapour quality below 0.15, as in flowing refrigerants the vapour quality usually increases with pressure decay (assuming isentropic flow [9]). Ethane is an exception: Quality increases below 0.7, decreases above. Real flow is isenthalpic [10]: The quality always increases also for ethane.

The general conclusion is that a dedicated hybrid two-phase TTCS loop, as it is schematically depicted in figure 8, will guarantee both "isothermal" specifications and the preferred quality range.

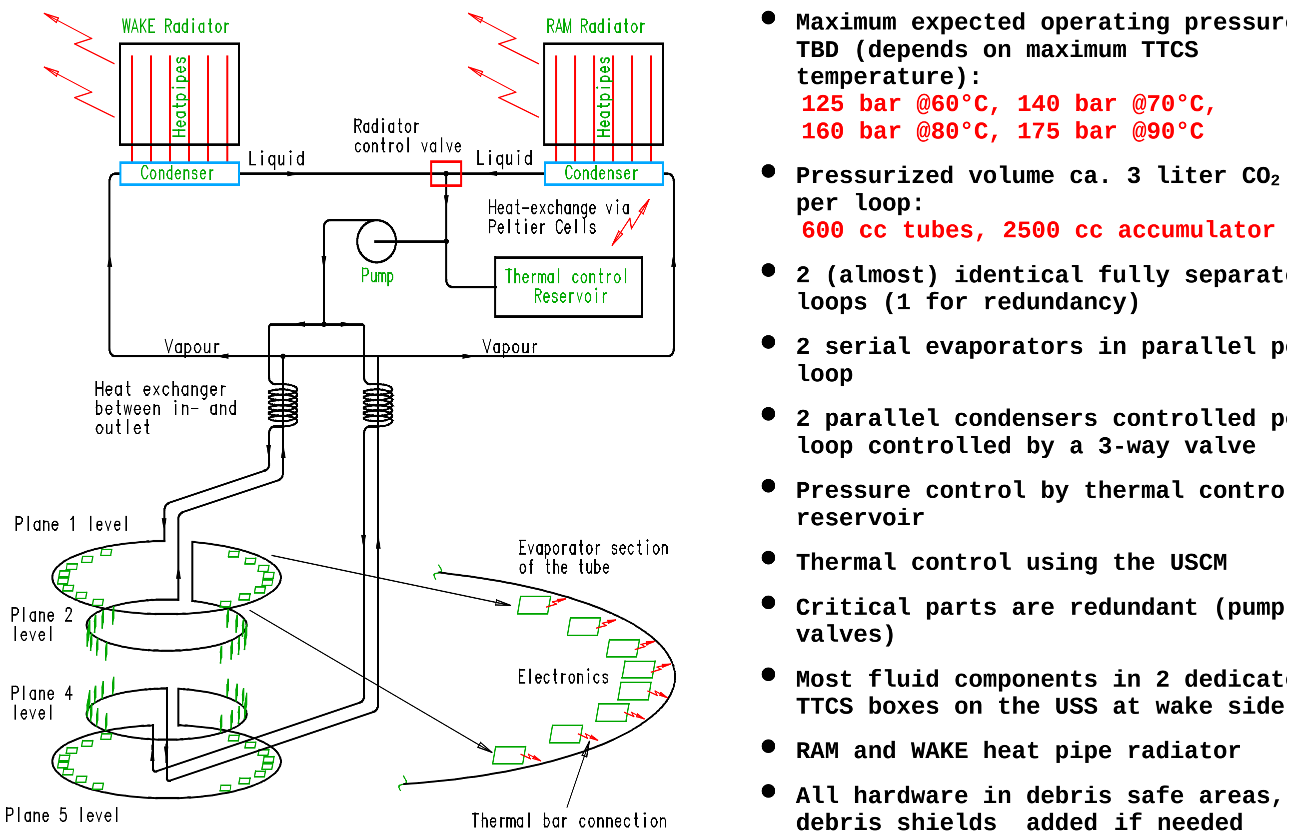


Figure 8. Hybrid MPL concept for the TTCS

CURRENT STATUS OF TTCS DESIGN

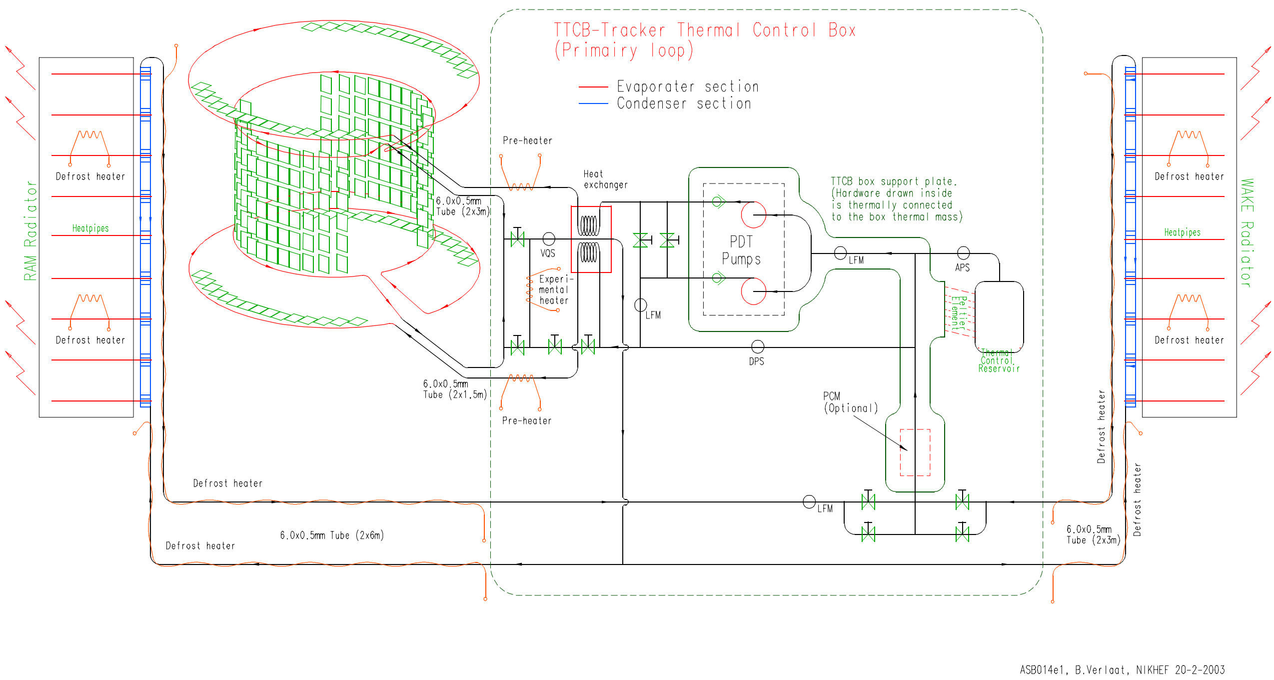


Figure 9. TTCS Primary loop (Secondary loop is almost identical, it only has less instrumentation)

The proposed TTCS primary loop is depicted in figure 9. It is a closed two-phase system: Heat is absorbed in the evaporators and withdrawn at the condensers, to be rejected to space the radiators. As the mechanical pump provides the liquid flow rate needed, it has to be located after the condensers, as it needs pure liquid to operate properly. Hence the condensers/radiators need not only to condense all vapour, but also to provide a certain amount of sub-cooling.

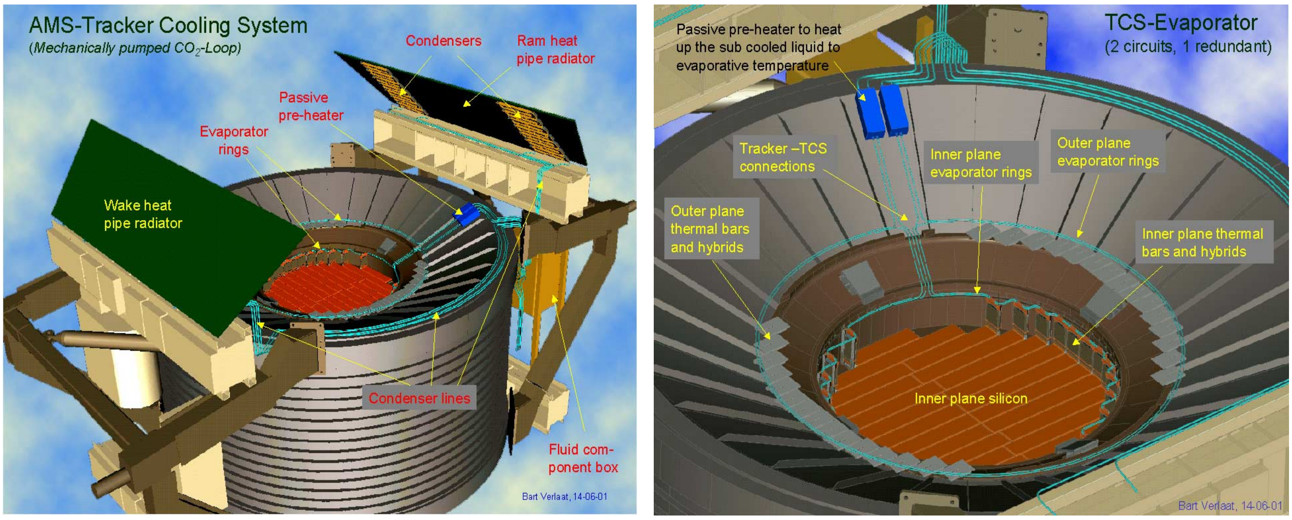


Figure 10. Impression of integrated TTCS (left) and TTCS evaporator (right)

The blue boxes on top (Fig. 10) are heat exchangers, thermally connecting inlet and outlet of the evaporator. In this way the absorbed heat can be used to heat the entering sub-cooled liquid from the pump so it gets close to the evaporative temperature needed in the Tracker. The evaporators consist of two parallel tubes each having an ID of 2.6 mm and a length of 10 metres. These two tubes are serially cooling the hybrid circuits, located on the outer periphery of the Tracker. The parallel evaporator branches (Figs. 8, 9). are routed as two rings following the widely distributed hybrids. The second branch is located similarly at the bottom of the Tracker. The evaporator tube is mounted with a copper connection bridge to the hybrid thermal support structure named thermal bars.

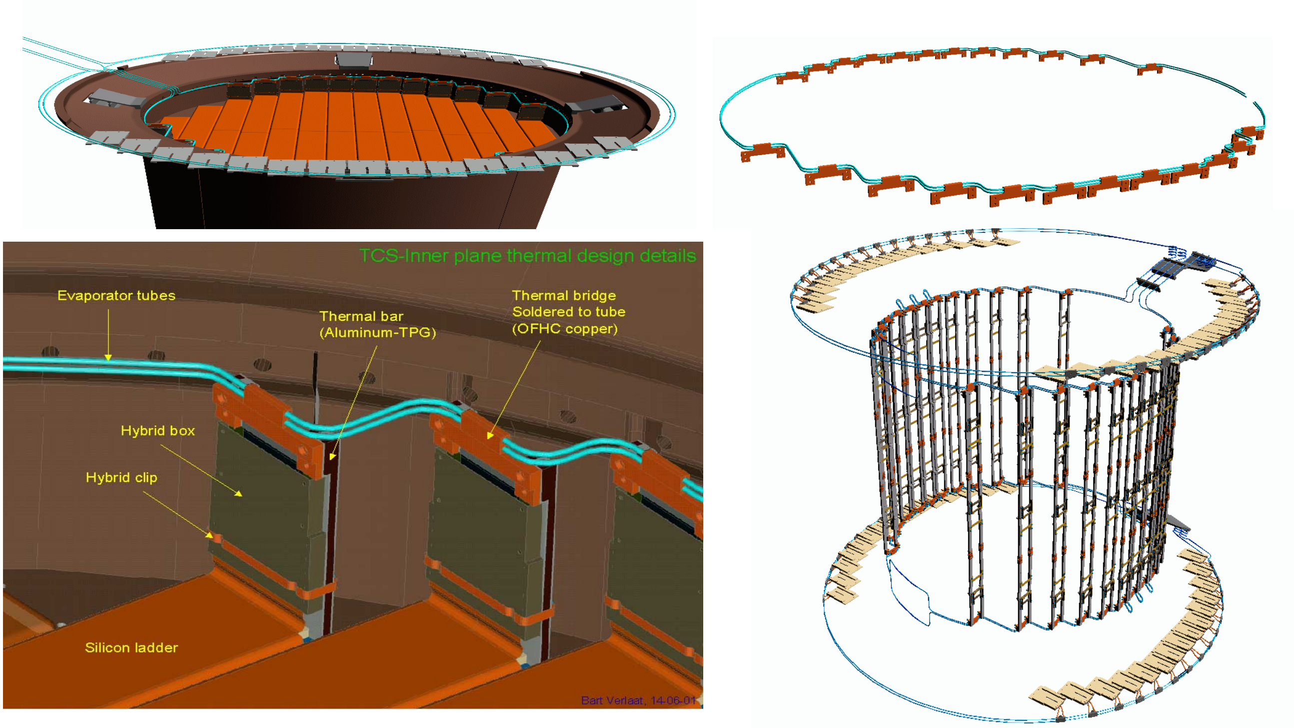


Figure 11. TTCS, inner ring evaporator and connections to bars in the real 3-D Tracker configuration

The figures 8 to 11 show the thermal connection from the inner thermal bars to evaporator. Clearly visible is the bent configuration of the evaporator tube; which is needed to follow the stepped orientation of the tracker hybrid boxes. This stepped orientation is one of the reasons that a small diameter evaporator tube was selected as the baseline, because it seemed to be the only design that was compatible with the already existing tracker hardware. There are two tubes, one acts as the redundant line in the case of a failure. The AMS-2 radiator panels are outside the experiment (Fig. 7). They are covered with high emissivity and low solar absorptivity coatings/paints. The two opposite radiator panels are thermally speaking out of phase, meaning that there is always one radiator shaded from the sun, hence able to radiate waste heat to space. The evaporation temperature is adjusted by the system pressure. This pressure is controlled via the accumulator, a small reservoir with a mixture of vapour and liquid. A Peltier element controls the reservoir temperature, hence the system pressure by condenser flooding. The majority of the TTCS hardware is in a box outside on the support structure. The evaporators, heat exchangers and condensers (Fig. 12) are outside this box.

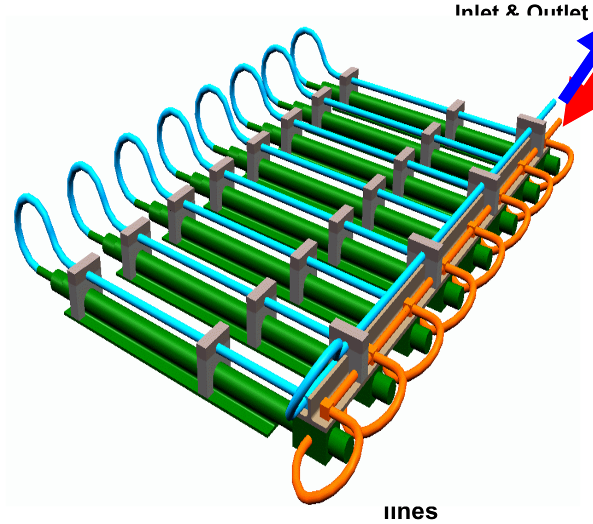
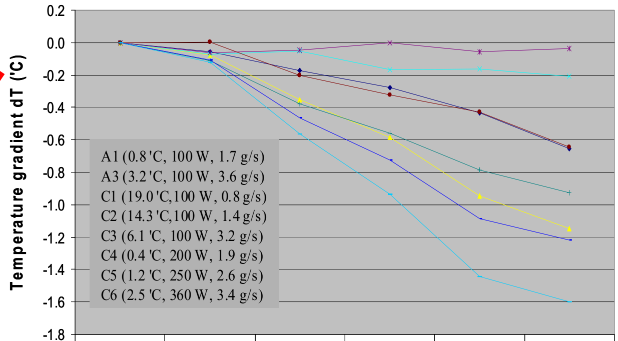


Figure 12. TTCS condenser



	dT inlet (0m)	dT tube 1 (1m)	dT tube 2 (3.4m)	dT tube 3 (5.8m)	dT tube 4 (8.3m)	dT outlet (9.3m)
A1	0.0	-0.1	-0.2	-0.3	-0.4	-0.7
A3	0.0	-0.1	-0.4	-0.6	-0.9	-1.1
C1	0.0	-0.1	-0.1	-0.2	-0.2	-0.2
C2	0.0	-0.1	0.0	0.0	-0.1	0.0
C3	0.0	0.0	-0.2	-0.3	-0.4	-0.6
C4	0.0	-0.1	-0.4	-0.6	-0.8	-0.9
C5	0.0	-0.1	-0.5	-0.7	-1.1	-1.2
C6	0.0	-0.1	-0.6	-0.9	-1.4	-1.6

Figure 13. Temperatures along the evaporator

TESTING ISSUES

An open loop test set-up, built at NIKHEF to prove the feasibility of the TTCS evaporator concept for CO_2, consisted of an evaporator section connected to a liquid CO_2 filled bottle. The CO_2 flow was adjusted by a needle valve, the pressure in the test tube by a spring-relieve valve (at the exit). In the real TTCS all thermal bridges are individually connected to the to evaporator tubes. In the feasibility test set-up heat is applied over the test section tube wall using the electric resistance of the tube as heater. Flow, pressure drop and temperatures along the tube were measured. Figure 13 shows some test results, which confirm that CO_2 is an adequate refrigerant for the TTCS loop.

More experiments were done next at NIKHEF to confirm this in a closed-loop test set-up, which more realistically simulates the TTCS. The goals of the experiments were:

- To measure the pressure drop characteristics and heat transfer coefficients at different flow rates, heat input and evaporation temperatures, using a 10 m long, 2.5 mm ID test evaporator, with helical sections between the long sections to simulate the multiple bends in the real Tracker.
- To compare the test outcomes to theoretical predictions and experimental data produced in a NIKHEF/SINTEF CO_2 test set-up.
- To prove the merits inserting a heat exchanger (as pre-heater) between evaporator in- and outlet.
- To yield recommendations for further TTCS development, on pumping rates and evaporators.

Though many experiments were executed, the results given here pertain only to the 10 m long, 2.5 mm ID evaporator performance, i.e.:
- Figure 14, showing the pressure and temperature drops, as a function of the mass flow, at 273 K.
- Figure 15, showing the heat transfer coefficients and observed flow patterns versus vapour quality and heat flux, at 278 K and nominal flow 2.7 g/s.

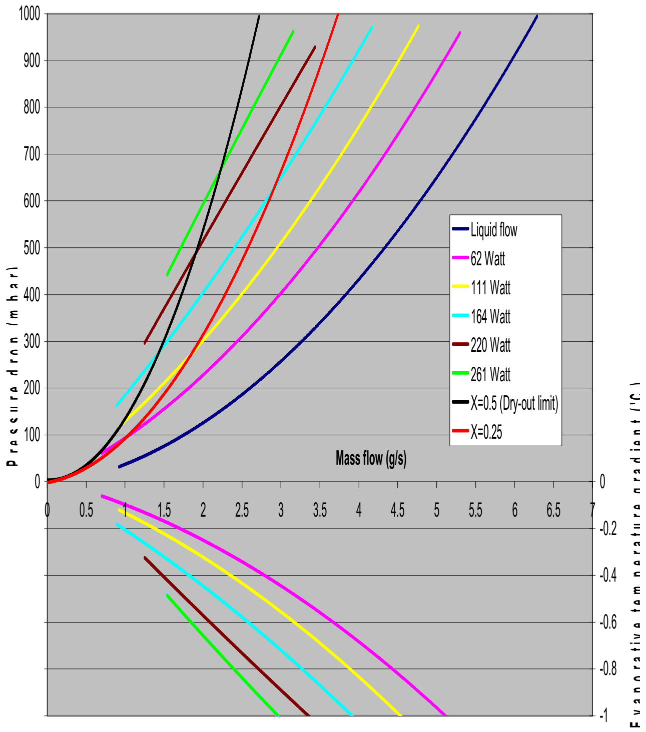


Figure 14. Power dependence of pressure and temperature drops at 273 K

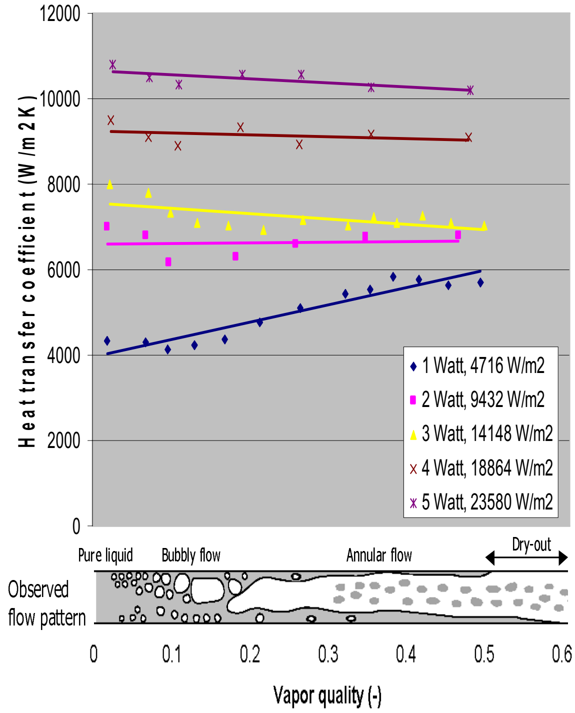


Figure 15. HTC and power (density) versus flow patterns and vapour quality at 2.7 g/s & 278 K

Finally it is remarked that preliminary test results confirm the usefulness of the presence of a heat exchanger as pre-heater between the in- and outlet of the evaporator. It was observed that up to say 90% of the heat collected in the evaporator could be reused for pre-heating the sub-cooled liquid coming from cold radiators. This amount of heat replaces part of the power to be added to the electric pre-heater that has to condition the liquid such that the fluid entering the evaporator is a pure liquid, close to saturation temperature as desired. It is obvious that the above yields a substantial power saving. Apart from this power saving impact, it can be said that the presence of the heat exchanger has also a stabilising effect on the temperature excursions of the evaporator during orbital radiator temperature variations.

The next step in the development was the creation of a full-scale test set-up at NLR for a more realistic simulation of the TTCS. A preliminary rig was designed and built. Based on experimental results obtained with this rig, the full-scale test set-up was designed and manufactured. Figure 16 depicts the schematic of the set-up. Figure 17 shows a photograph of the current test set-up in the NLR climate chamber. Details are shown in figure 18 (the evaporator) and in figure 19 (a specimen of the baseline for the TTCS condensers, consisting of elements, which will interface the Ram and Wake heat pipe radiators).

The first experiments with this full-size test set-up yielded very encouraging results: The pressure drops across the system turned out to be even smaller than predicted: Almost ideal isothermality is approached.

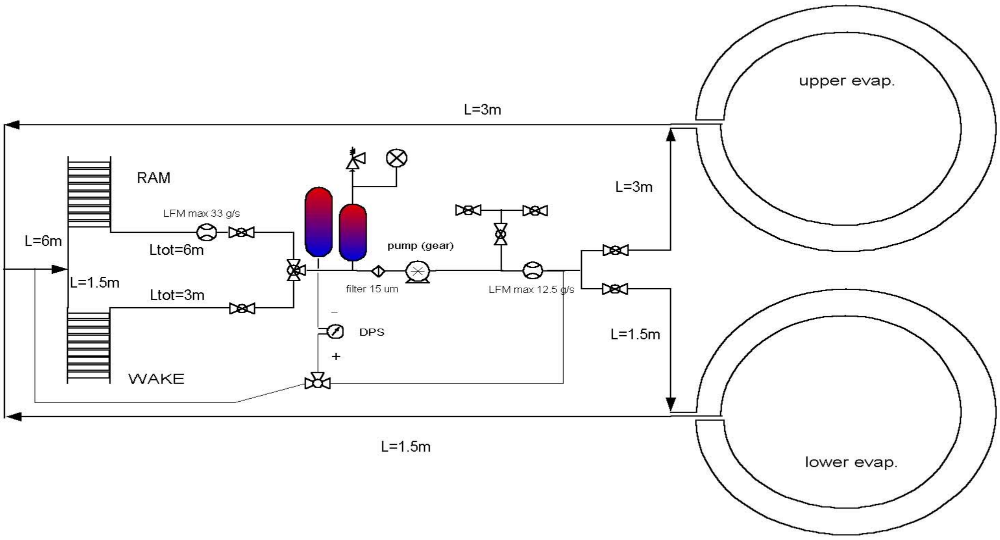


Figure 16. Schematic of NLR's full-size simulation test rig

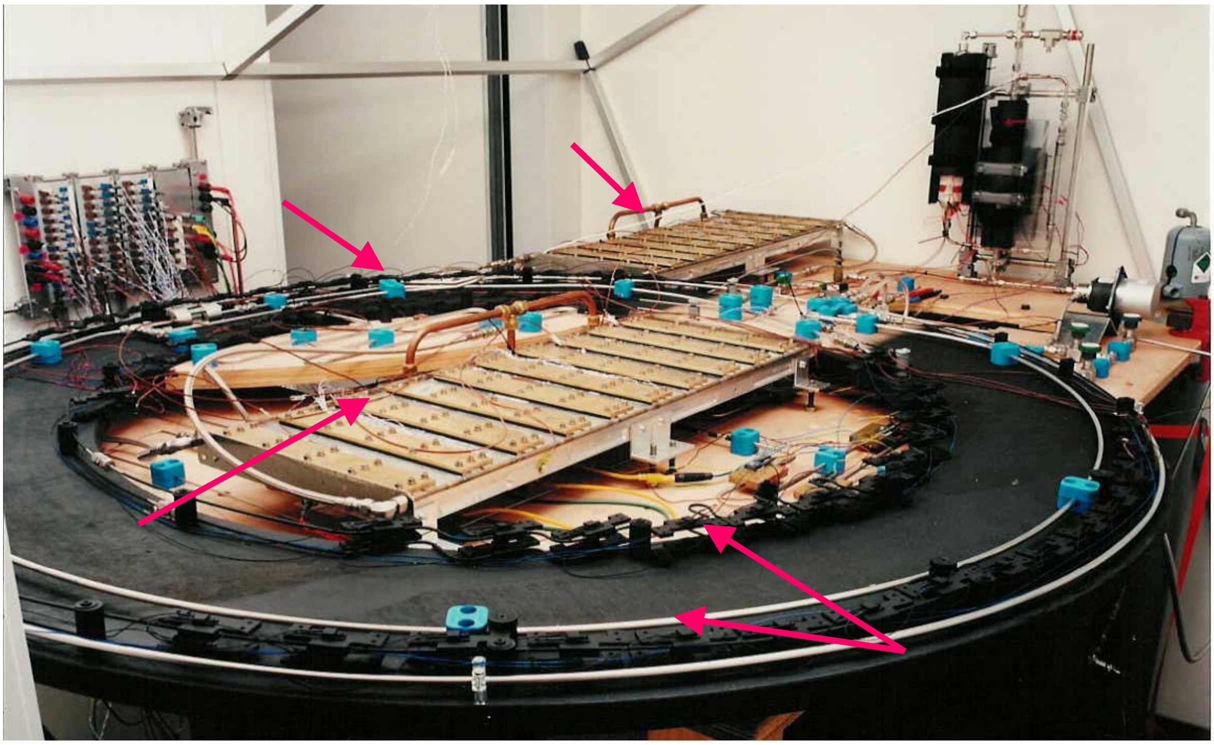


Figure 17. Full-size simulation test rig at NLR

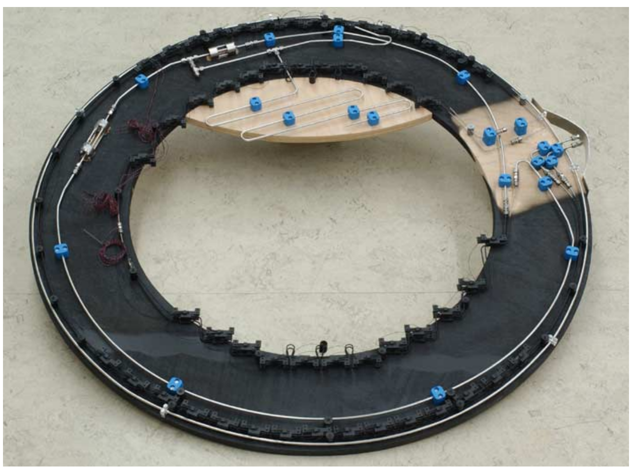


Figure 18. Full size evaporator

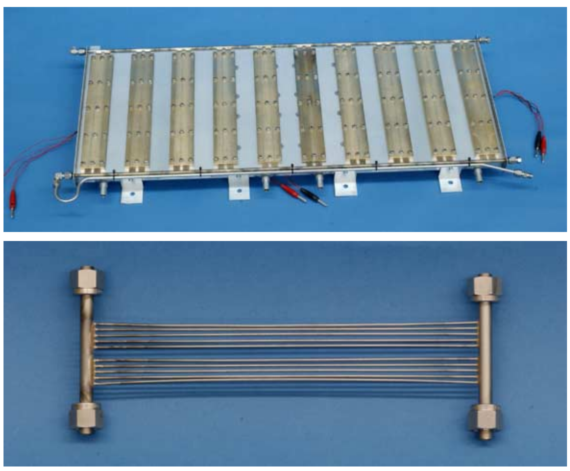


Figure 19. TTCS condenser & element

In order to study the heat transfer in vacuum, along the thermal bar itself and from the thermal bar to the loop evaporator, a test set-up has been built at NIKHEF (Figs. 20, 21). Some results will be presented in the next chapter, in order to compare these with outcomes from thermal modelling.

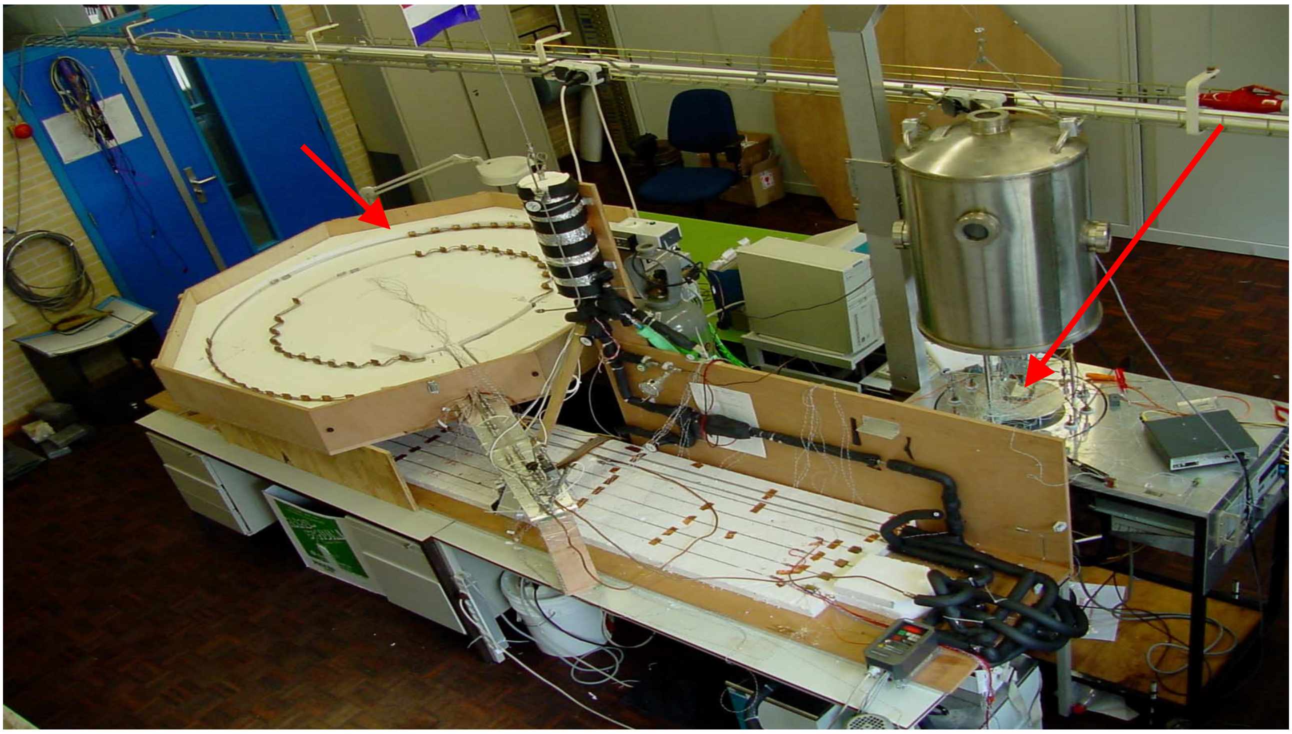
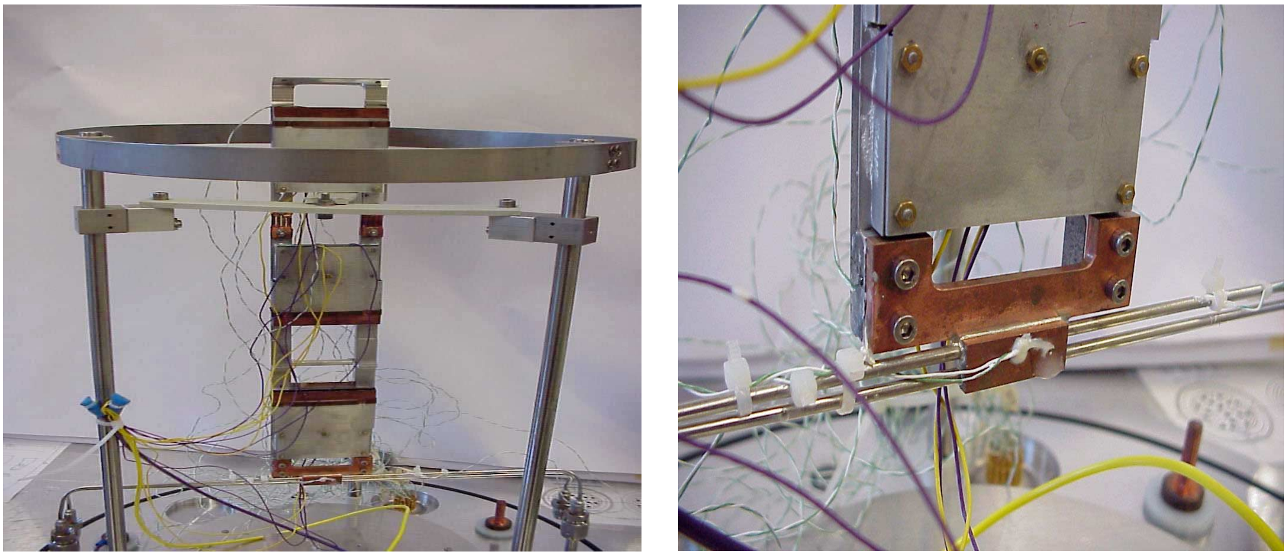


Figure 20. NIKHEF's evaporator & thermal bar (in vacuum) test loop



THERMAL MODELLING ISSUES

Calculations with a very detailed transient TTCS model (Fig. 22) have been done for many possible orbital (environmental loading) cases. The outcomes clearly indicate that:

- The TTCS will operate without problems at the nominal loop set-point temperature 273 K, for the nominal case and most other thermal loading cases (Fig. 23).

- In some hot orbital cases, the set-point temperature of the loop has to be increased by approximately 10 K (Fig. 24).
- The incorporation of the heat exchanger between evaporator in- and outlet considerably reduces the pre-heater power needed (Fig. 25). This is important, since the power available for pre-heating is very limited.
- Figure 26 proves that the outcomes of the measured thermal bar temperature gradients and the thermal modelling predictions are in reasonably agreement.

The modelling was refined when more accurate environmental loading conditions were provided by CGS, the "AMS Overall Thermal" main contractor. Using these new boundary conditions, new calculation runs were executed for various orbital environments and loop temperature set-points. The results shown in the figure 27 confirm the must of including a heat exchanger: Considerable reduction of pre-heater power, though the pre-heat power needed is still far higher than the power available.

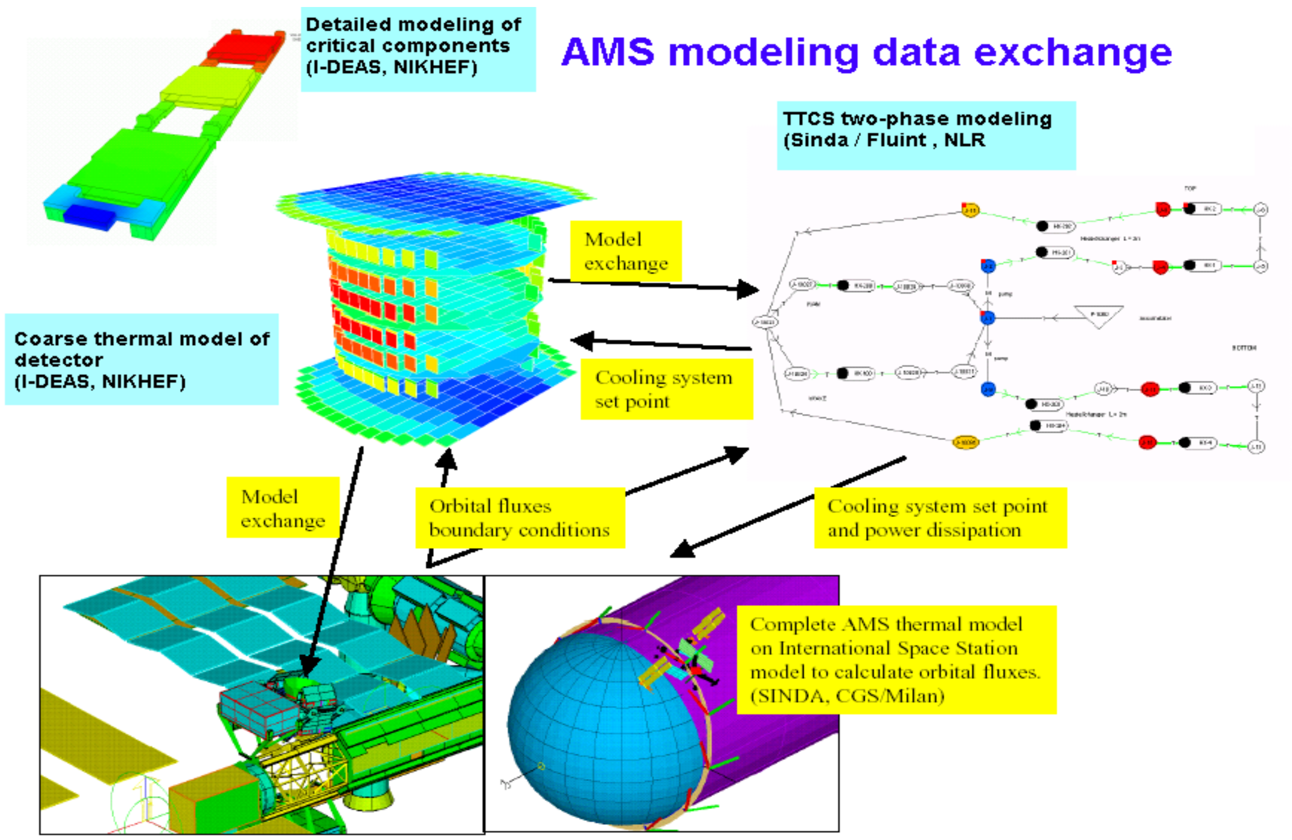


Figure 22. AMS-2 Modelling data exchange diagram

The above results are indicative only, because up to this point all calculations were done for a total hybrid dissipation of 192 W (1 W per hybrid pair) for 1.6 m^2 radiators. But the measured dissipation of recently delivered hybrids turned out to be only 0.75 W (±10 %). Consequently all the following design calculations were to be done for a total hybrid dissipation of 144 W (+10 % for the hot orbits, -10 % for the cold orbits).

In addition, the door dimensions of the Boeing 747 (the carrier to transport AMS-2 from Europe to NASA-JSC and KSC) limit the radiator size appreciably. Therefore the calculations were to be done for the maximum radiator sizes possible, being 1.25 m^2, 11 kg for a flat radiator option, 1.43 m^2, respectively 12 kg, for a curved radiator option (both radiator options are shown in figure 28).

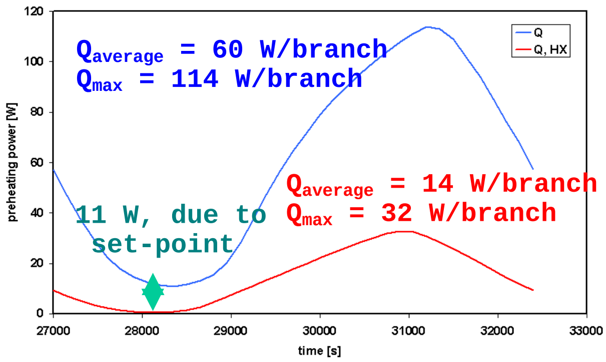


Figure 23. Influence of heat exchanger presence

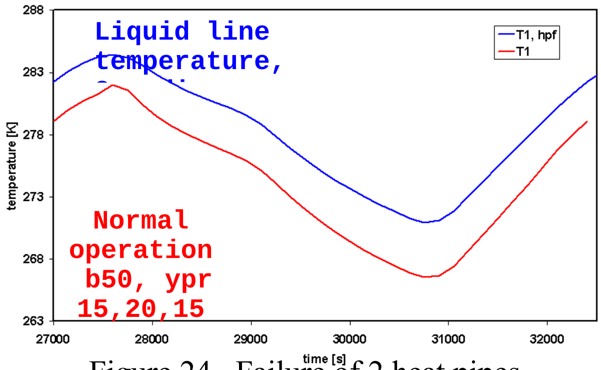


Figure 24. Failure of 2 heat pipes

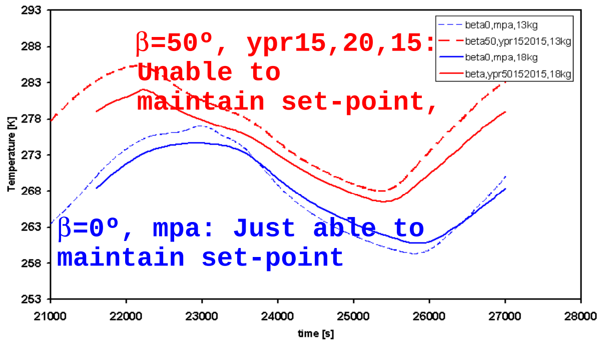


Figure 25. Liquid temperatures entering pump: Effect of total mass of radiators (2x13 kg /2x18 kg & orbit. (β=0º/50º)

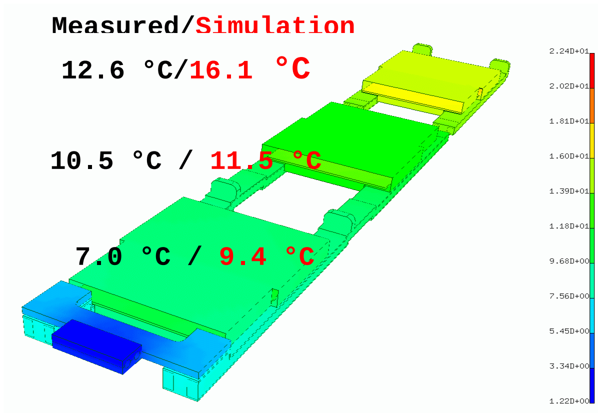


Figure 26. Outcomes of thermal bar modelling versus results of experiments

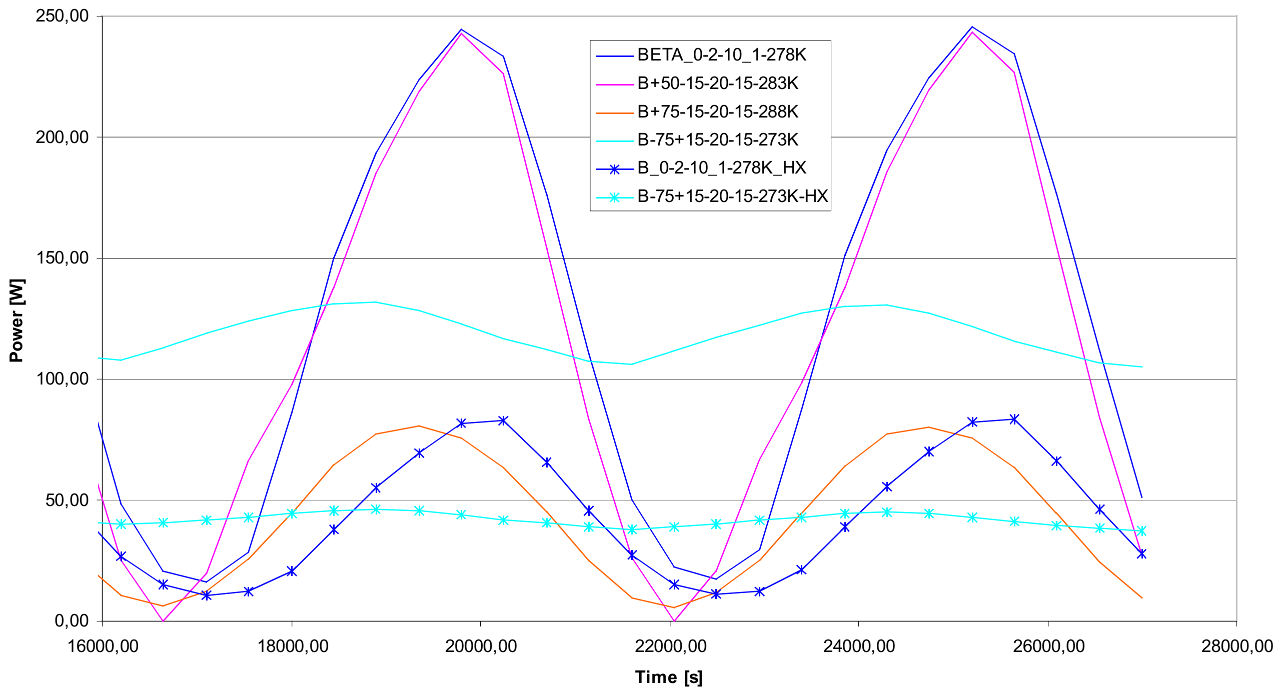


Figure 27. Pre-heater power curves for six different orbits

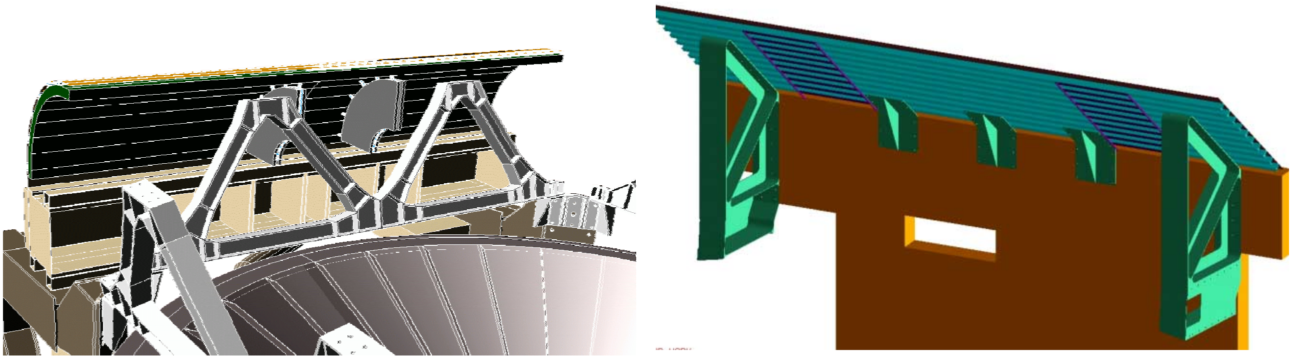


Figure 28. Curved (left) and flat (right) radiators (incl. the condenser configurations).

Results of calculated have shown that:
- For cold orbits, like B-75+15-20-15, both radiators show almost identical performances, for a flow rate of 2 g/s . The need for pre-heater power is less than 12 W maximum, 6 W average, at a set-point of 258 K for the flat radiator, 257 K for the curved one.
- For an average (nominal) orbit, like B_0-2-10_1, the pre-heater power needed is for both options (for a flow rate of 2 g/s and a hybrid pair dissipation of 0.75 W) the same: 33 W maximum, 17 W average. But the set-point in the case of the flat radiator is 278 K, being 6 K higher than for the curved one. Set-points close to or somewhat below 273 K are preferred ones.
- For the hottest case, B+75-15-20-15 and a hybrid dissipation of 0.825 W per pair (and a flow rate of 2 g/s), about 10 W maximum and 6 W average pre-heater power is needed for both options. However, the set-point for the curved radiator is acceptable (285 K), the flat radiator option yields a set-point slightly above the maximum value permitted (290 K).

The above results suggest that the curved radiator option is (thermally seen) the better one. It is also structurally the stronger one and is easier transportable (with respect to the B747 envelope limitations). It also can accommodate easier producible condensers, which can be designed such that the chance of condenser penetration by micro-meteorites etc., hence loss of the loop, is extremely close to zero (probability of non-penetration pnp = 99.999 %) for a 5 years mission. The only drawback of the curved radiators is the price, twice the flat radiator price.

The above mentioned pre-heater power values are encouraging, but still too high. A further reduction will be realised by incorporating a PCM (Phase Change Material) device. This thermal capacitor or energy storage device will dampen the temperature excursions of the loop and thus further reducing the pre-heater power needed, by a melting and solidification cycle of the PCM, a paraffin or mixture of paraffins (to create a melting trajectory instead of a fixed melting point).

The TTCS loop model (Fig. 22) was recently extended by including a PCM device. Because of the loop operating temperature requirements, three different PCM's were chosen for the calculations and for experimenting: n-dodecane (melting point 263.5 K, melting heat 210.5 kJ/kg), n-tetradecane (melting point 279.0 K, melting heat 229.9 kJ/kg), and and n-eicosane (melting point 307.5 K, melting heat 247.3 kJ/kg). Preliminary calculation results confirm that the presence of the PCM device dampens the temperature excursions of the loop and substantially reduces the pre-heater power needed, for various orbital cases. The outcomes suggest that an optimal profit of a PCM device incorporation will be reached by either creating a melting trajectory (if realisable) by a mixture of different PCM's, or by a PCM device consisting of sections, each section containing an optimised amount of a specific PCM.

IN-ORBIT EXPERIMENTS AND FINAL REMARKS

Apart from the challenge to develop a novel two-phase thermal control system for such an advanced experiment as AMS-2, NLR interest also pertains to the acquiring of in-orbit experience with real two-phase thermal control systems. NLR joined the AMS Collaboration, as it was guaranteed that the AMS-2 dormant (non-operation) periods could be used by NLR to execute dedicated experiments to study in-orbit two-phase heat transport system technology issues. Therefore the TTCS will be equipped with some extra heaters, sensors, and meters.

The baseline philosophy will be that:

- There is minimum risk for Tracker and AMS-2.
- Any period AMS is not active can be used for thermal experiments
- There is at least one week of thermal experiments during the first six months
- Minimum power and mass will be added.
- The TTCS loop will, in principle, not be intruded.

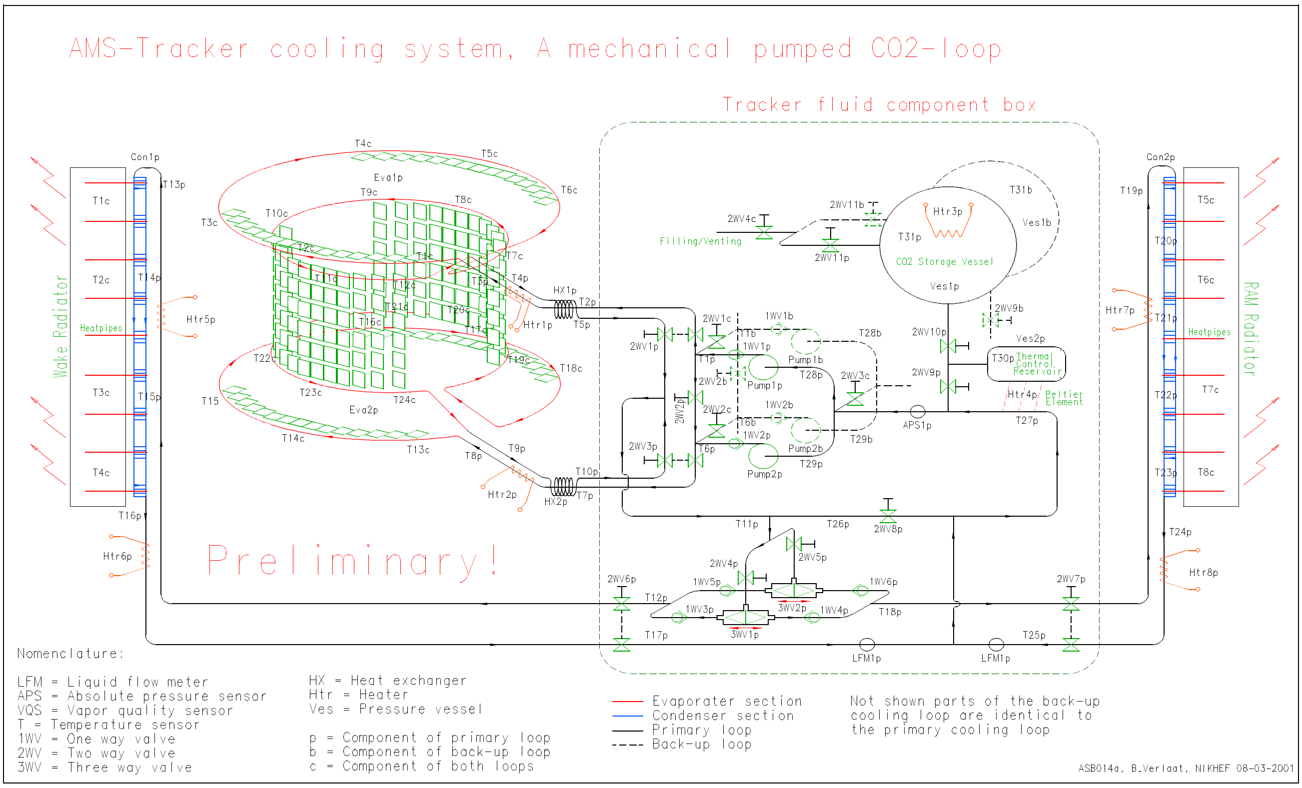


Figure. 29. Preliminary fully redundant TTCS, equipped with extra experiment components

Figure 29 depicts how complicated such a fully redundant, for extra NLR experimentation equipped, TTCS can look like. However, it can already be said now that AMS-2 overall mass reduction requirements certainly will lead to a less complicated system. This will realised by partly reducing the redundancy level required and by the deleting of some components.

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- Any period AMS is not active can be used for thermal experiments
- There is at least one week of thermal experiments during the first six months
- Minimum power and mass will be added.
- The TTCS loop will, in principle, not be intruded.

The reported TTCS status already reflects the AMS-2 overall mass reduction requirements. It is developing straightforwardly to the status required at its Critical Design Review to be held in October/November 2003. Critical issues like the development of the pumps, minimising of mass and the required power, etc. are more or less solved. But it should be stressed that the results of the many experiments to be done with the full-size test set-up, is expected to lead to some substantial changes.

NOMENCLATURE

ACC	Anti-Coincidence Counter
AMS	Alpha Magnetic Spectrometer
APS	Absolute Pressure Sensor
CGS	Carlo Gavazzi Space
CPL	Capillary Pumped Loop
DAC	Data Acquisition and Control System
DPS	Differential Pressure Sensor
DP	Pressure Difference (Pa or mBar)
EC	Electromagnetic Calorimeter
HTC	Heat Transfer Coefficient ($W/m^2.K$)
INFN	Italian Institute for Nuclear Physics
ISS	International Space Station
LFM	Liquid Flow Meter
LHP	Loop Heat Pipe
MPL	Mechanically Pumped Loop
NIKHEF	Dutch Inst. for Nuclear & Particle Physics
NLR	Dutch National Aerospace Laboratory
RICH	Ring Imaging Cherenkov Counter
SPL	Single-Phase Loop
SINTEF	Norwegian Foundation for Scientific and Industrial Research
SRD	Synchrotron Radiation Detector
STS	Space Transportation System (Space Shuttle)
TC	Thermal control
TM	Thermal Model(ling)
ToF	Time of Flight
TPG	Thermal Pyrolytic Graphite
TPHTS	Two-Phase Heat Transport System
TRD	Transition Radiation Detector
TTCS	Tracker Thermal Control System
VQS	Vapour Quality (Mass Fraction in %) Sensor

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