

Development and Testing of a Two-Phase Mechanically Pumped Loop for Active Antennae

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The satellite telecommunications industry is currently undergoing significant evolutions. Future communication satellites need to accommodate a rapidly growing demand in data transfer, combined with more flexibility. For example, there is a strong need for Very High Throughput Satellites capable of delivering up to Terabyte per second over wide coverage areas. This is only possible when an active phased array antenna is used. However, cooling of active antennas requires the use of a highly efficient thermal control system because it has many heat sources (from hundreds to several thousands), high local heat fluxes (20 W/cm² at evaporator interface), high overall dissipation (around 10 kW), and isothermal requirements on the amplifier chain. These conditions are very difficult to meet with current thermal control solutions (e.g. heat pipes or loop heat pipes), but require a two-phase mechanically pumped fluid loop (MPL). In an MPL, a pump circulates a fluid which evaporates when it absorbs the waste heat from the active antenna. In the IMPACTA project, a demonstrator for such an MPL is being designed and build. This paper describes the test results for the IMPACTA demonstrator. The demonstrator is able to cool a total heat load of 9.8 kW divided over 10 parallel branches with a better than 2°C spatial temperature uniformity over the heat sources. In an active antenna application, the heat load can be unevenly distributed over the different branches. Tests show that even in the extreme case when half of the branches are turned off and the other half are set to full power, no sign of dry-out or too high temperatures is observed, demonstrating the ability of the MPL to cool imbalanced payloads. The demonstrator was tested in 3 different orientations and the test results are similar for all orientations, indicating that the system is not sensitive to gravity effects.

Nomenclature

IMPACTA	=	Innovative Mechanically Pumped loop for ACTIVE Antennae
MPL	=	mechanically pumped fluid loop
P	=	heat input (W)
SSPA	=	Solid State Power Amplifiers
T	=	temperature
T_{sat}	=	saturation temperature

I. Introduction

In the IMPACTA project (<https://impacta-mpl.com/>), which is led by AVS, a demonstrator of a two-phase mechanically pumped fluid loop (MPL) for an active antenna for a communication satellite is being developed.

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NLR is responsible for system analysis and testing and AVS for vibration analysis and building the demonstrator. Airbus provided the requirements and reviews of designs and system. The preliminary system analysis and fluid selection, and the preliminary evaporator design and tests on evaporator samples are described in a previous paper¹. This paper focuses on the test results with the system. In a two-phase MPL, a pump transports liquid to an evaporator that is mounted on e.g. an active antenna. In the evaporator, the heat from the active antenna is absorbed and the liquid (partly) turns into vapor. The vapor/liquid mixture then flows to the radiator/condenser. In the radiator, the absorbed heat from the active antenna is radiated into space, and the vapor is turned back into liquid. The pressure and temperature of a two-phase fluid are coupled. As a result, the temperature of the liquid/vapor mixture is the same in the entire system (assuming a negligible pressure drop), and independent of the heat input or heat sink temperature. The accumulator controls the saturation temperature in the system. Two-phase MPL have several advantages compared to other thermal control systems, such as Heat Pipes (HP), Loop Heat Pipes (LHP) or single-phase liquid MPL²:

- The fluid saturation temperature (which is set by the accumulator) is independent of the heat load or heat sink temperature. This results in a uniform temperature of the active antenna.
- The tubing diameter can be much smaller than for a single-phase liquid MPL, LHP or HP. This tubing can easily be routed along all the heat sources.
- The two-phase heat transfer coefficient is very high compared to single-phase liquid MPL, HP, or LHP. This high heat transfer coefficient results in a smaller temperature difference between the fluid and the active antenna. As a result, a higher fluid temperature can be used, which results in a smaller radiator surface.

A drawback of a pumped two-phase system is that it requires a mechanical pump. Furthermore, the design of a two-phase MPL is more complex than the design of a single-phase MPL or a system with heat pipes. As a result, the technology is not often used for space applications, despite the enormous advantages that this technology provides. A two-phase MPL was used for the tracker thermal control system of the Alpha Magnetic Spectrometer (AMS-02) on ISS to which the NLR contributed³. Recently, two commercial spacecraft from Thales Alenia Space have two-phase MPL operating in space⁴.

II. Overview of the IMPACTA system

Figure 1 shows a schematic drawing of the IMPACTA cooling system. The system consists of 4 sections:

- MPL equipment box: This box contains the pump, recuperator, accumulator and several heaters. The box has fluid connections for tubing to the evaporator and condenser sections. The box also contains the connections to the filling system and the rupture disc exhaust.
- Evaporator section: This section contains the evaporators that are used to cool the heat sources of the antenna. The evaporator section has a total heat dissipation of 9.8 kW.
- Condenser section: The waste heat from the antenna is released to the heat sink in the condenser section. In a space application, the condenser is connected to a radiator that radiates the heat into space. In this project, the focus is on the MPL equipment box and the evaporator section. The condenser is cooled with liquid water via a plate heat exchanger.
- Tubing between components box and evaporator and condenser: The length of the tubing in the IMPACTA system is representative for the length of tubing in an actual spacecraft.

The system contains a flow meter and several pressure sensors that are only used during ground testing and are therefore located outside the MPL equipment box. These flowmeter and pressure sensors are indicated with blue color in Figure 1. The location of a selection of the temperature sensors is also shown in this figure (in total, the system has 80 thermocouple temperature sensors).

In order to test the influence of gravity, the system can be rotated such that it can be positioned in three different orientations. Figure 2 shows a CAD drawing of the system in orientation 'C', while Figure 3 shows a photo of the complete system as built by AVS. In this orientation, the evaporators and the accumulator are in a horizontal orientation (see section III for a further discussion of the orientations) and this orientation is considered to be the most representative for the system behavior in micro-gravity. Figure 4 shows a photo where the system is rotated, while Figure 5 shows a CAD drawing of orientation A. The MPL component box, evaporator section, condenser section and transport tubes are discussed in the next sections.

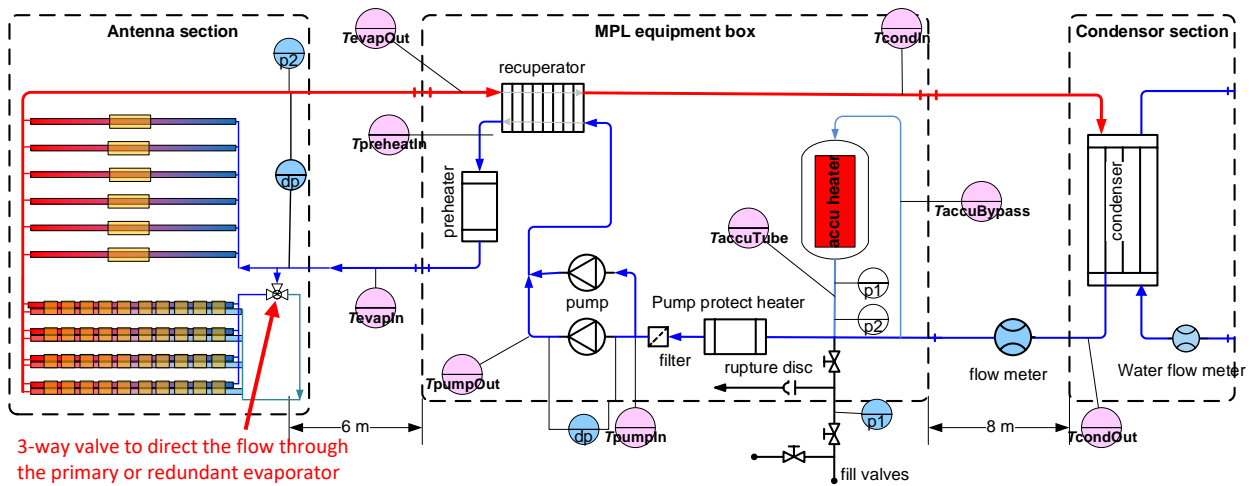


Figure 1. Schematic drawing of the layout of the IMPACTA cooling system

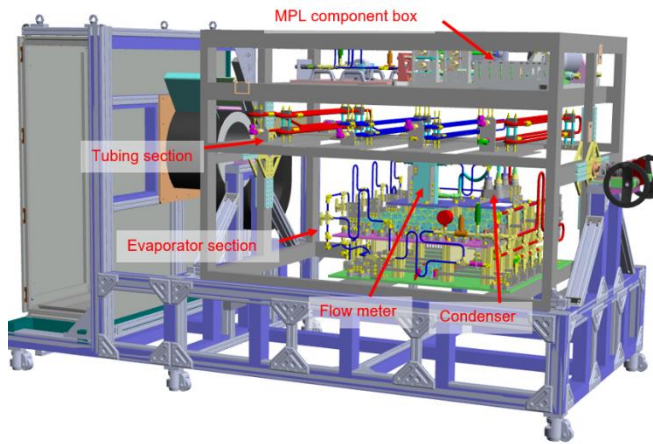


Figure 2. CAD drawing of the IMPACTA system in orientation C (horizontal accumulator and evaporators)

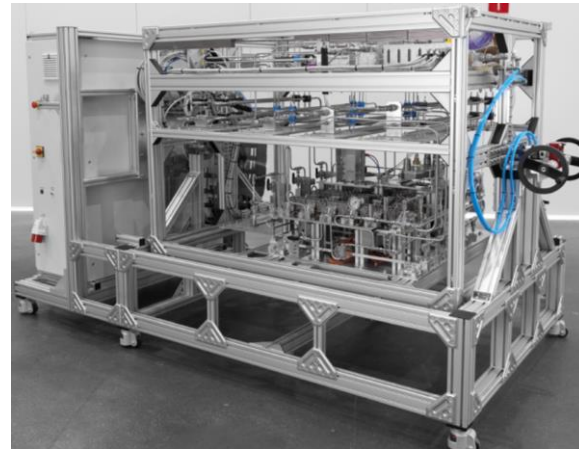


Figure 3. Photo of the IMPACTA system in orientation C

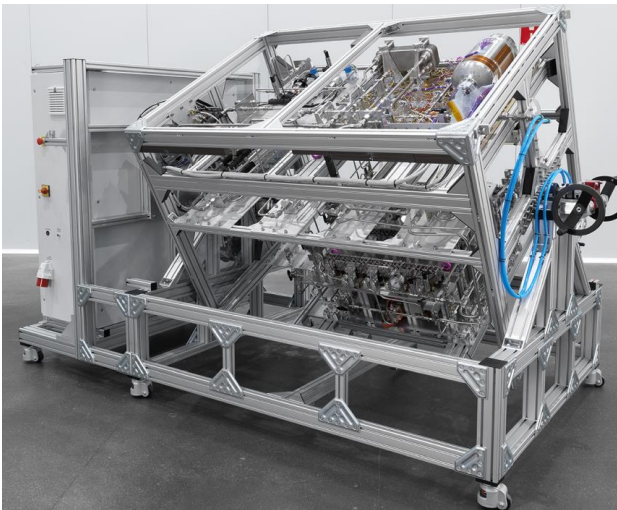


Figure 4. Photo of the IMPACTA system partly rotated

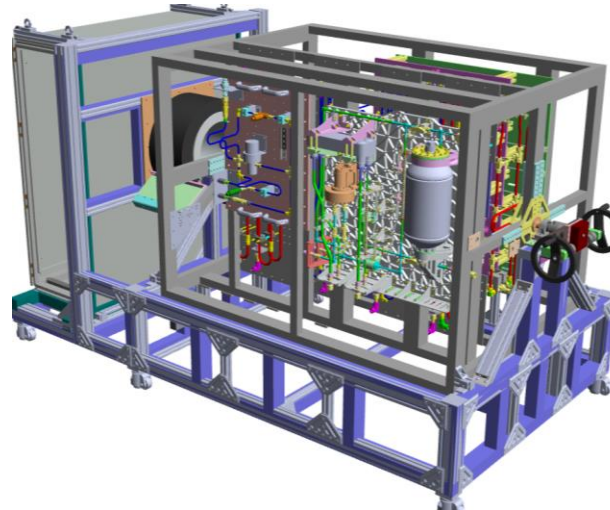


Figure 5. CAD drawing of the system in orientation A (vertical accumulator and evaporator section)

A. Equipment box

Figure 6 shows a CAD drawing of the MPL equipment box. This box contains the main flight components of the system: Accumulator, recuperator, pump, particle filter and non-condensable gasses blocker, pump protect heater, preheater and pressure sensors. Unfortunately, the newly developed AVS Positive-Displacement (PD) pump was not fully ready for assembly and integration in the system. For this reason, a commercially available positive displacement gear pump (Gather series 1, type 12/11) was installed on a separate plate (see Figure 7) and connected to the rest of the system. This pump was used for the test described in section III. The AVS PD pump was later tested in a separate setup⁵. No significant influence on the system behavior is expected by using this alternative pump.

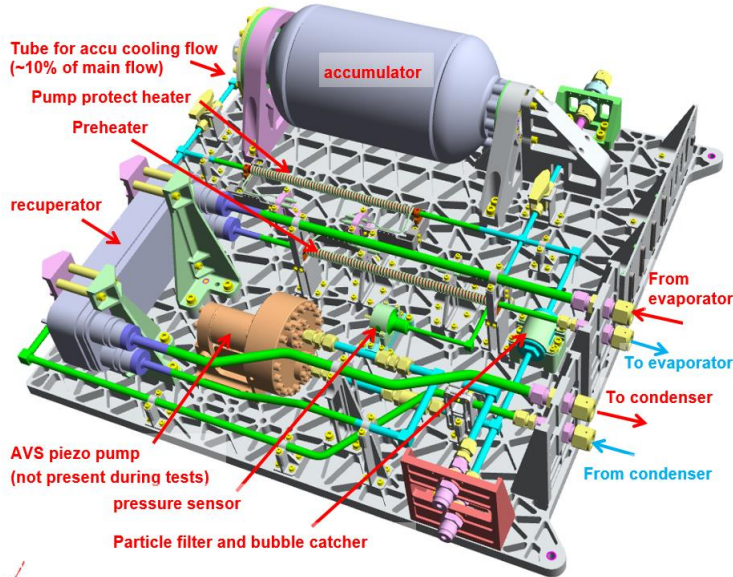


Figure 6. CAD drawing of the MPL equipment box

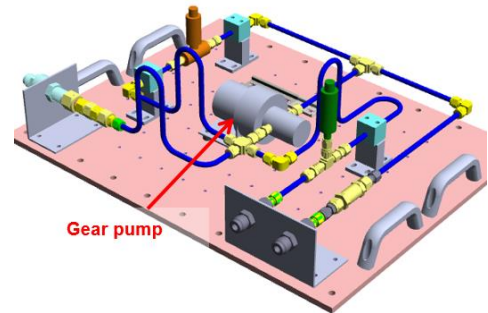


Figure 7. CAD drawing of the gear pump

B. Evaporator section

The IMPACTA system is intended to cool an active antenna. The Solid-State Power Amplifiers (SSPA) generate most of the waste heat in the antenna and have to be actively cooled. In a typical active antenna application, there are 98 SSPA units that generate a waste heat of 100 W each (9.8 kW in total). The SSPA units are arranged in rows. Each row of SSPA units is cooled by a separate evaporator branch. In a typical active antenna application, there will be 10 rows of SSPA units (and therefore 10 evaporator branches).

In the evaporator section of the IMPACTA demonstrator, there are 4 parallel evaporator branches that can be used to test different evaporators. Three of these branches are equipped with ‘NLR’ evaporators with a ‘conventional’ geometry (straight channels, no advanced surface structures) based on aluminum Multi Port Extrusions (MPE) profiles. These evaporators are described in more detail in the preliminary design paper¹. One branch is equipped with an evaporator designed by CEA. Each evaporator branch has 10 heat sources of 100 W each that represent the SSPA units.

An actual flight system will contain a primary cooling system and a redundant cooling system, so a flight system will have two MPL equipment boxes. In the IMPACTA project, only one MPL has been build. However, the evaporators for both the primary and redundant system are included. A 3-way valve is used to direct the fluid flow via the primary evaporator, or the redundant evaporator.

In addition to the 4 evaporator branches that are used to test different types of evaporators, there are also 6 branches with a single ‘NLR’ evaporator of 980 W. The total heat load of the 4 evaporator branches and the six 980 W branches is 9.8 kW. Figure 8 shows CAD drawings of the complete evaporator section while Figure 9 shows a CAD drawing of a single evaporator branch. The heater blocks are numbered 1 (at the inlet) to 10 (at the outlet). Heater blocks 1, 4, 7, and 10 are equipped with temperature sensors. These temperature sensors have a diameter of 0.5 mm and are inserted into holes (with a diameter of 0.6 mm) in the heater block and heat spreader. The locations of the temperature sensors in heater block 10 are indicated in Figure 10. The other heater blocks have temperature sensors at the same locations, except that there are no sensors at ‘A’ and no sensors in the bottom of the heat spreader.

The IMPACTA system is tested in three different orientations. In orientation C, the evaporator branches are in the same plane. In orientation A, the CEA branch is at the top, while NLR1a branch is at the bottom. In orientation B, the CEA branch is at the bottom and the NLR1a branch is at the top.

At the inlet of each branch is a flow restriction that consists of an orifice that has a diameter of 0.7 mm. This flow restriction results in a pressure difference of 0.3 bar. This pressure difference is larger than the pressure difference over the evaporators, and this ensures an equally distributed flow over all 10 branches.

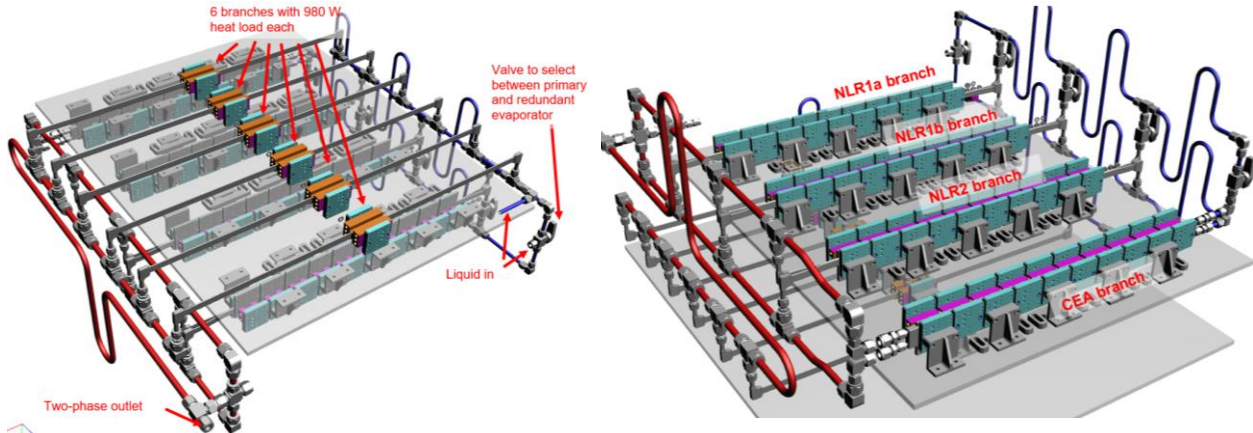


Figure 8. CAD drawings of the evaporator section with 10 branches (left: top view, right: bottom view)

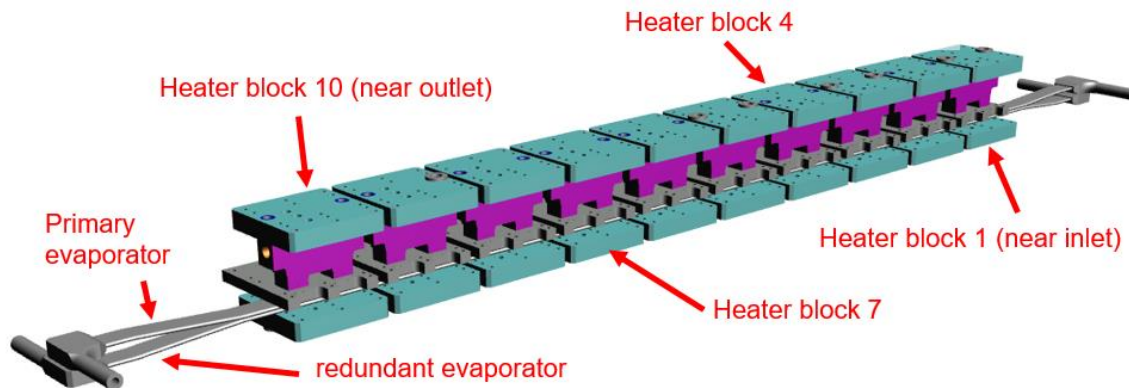


Figure 9. CAD drawing of single (NLR) evaporator branch

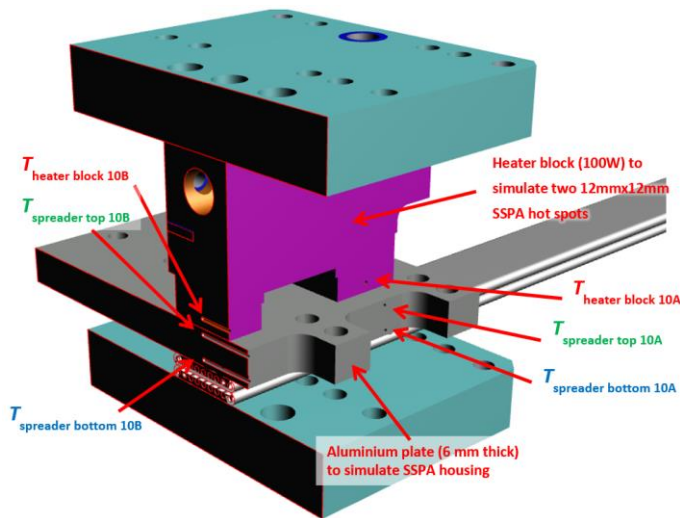


Figure 10. CAD drawing with cross section of heater block 10 with the location of the sensors

C. Condenser section

The condenser consists of an Alfa Laval AlfaNova XP27-10 plate heat exchanger. This condenser is cooled by building cooling water. The temperature of the building cooling water is approximately 12°C. Downstream of the condenser is a Rheonik RHM04 flowmeter that measures the ammonia flow out of the condenser. Figure 11 shows a CAD drawing of the condenser heat exchanger and the flowmeter.

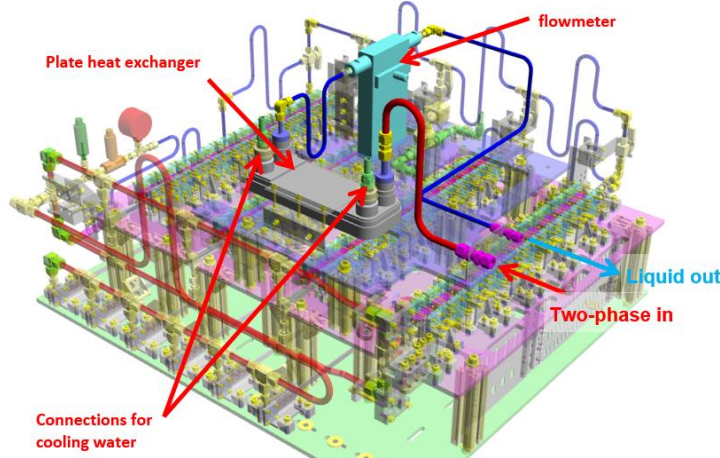


Figure 11. CAD drawing of the condenser heat exchanger and the flowmeter

D. Transport tubes

The length of the tubing has a large influence on the transient behavior of the system and the pressure difference over the pump, and it is therefore important that a representative length of tubing is included in the IMPACTA demonstrator. In the IMPACTA demonstrator, 6 meter of tubing is used between the MPL equipment box and the evaporator section and 8 meter is used between the MPL equipment box and the condenser section. Figure 12 shows the CAD of the tubing section. The liquid tubes have an outer diameter of 8 mm and the two-phase tubes have an outer diameter of 1/2 inch.

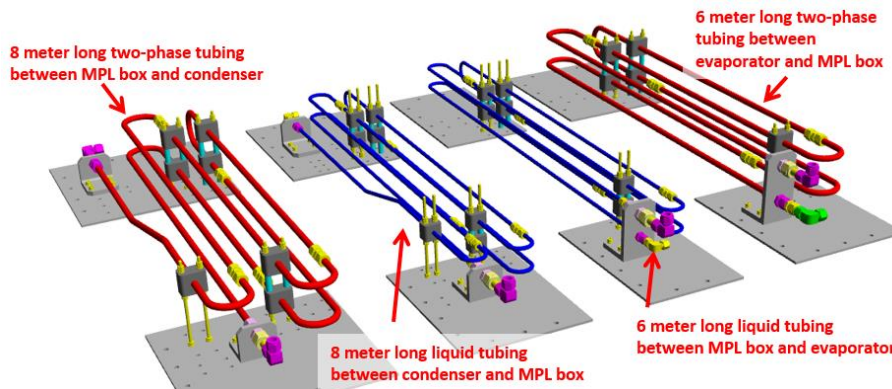


Figure 12. CAD drawing of the tubing between the MPL equipment box and the evaporator and condenser sections

III. Test results with complete system

After a successful helium leak test and proof pressure test, the system was filled with ammonia and tests have been carried out in 3 orientations. Most tests are performed in orientation C (horizontal accumulator and evaporators) since this orientation is most representative for micro-gravity. The instantaneous full evaporator power test has also been carried out in orientation A (vertical accumulator) and orientation B ('upside down' vertical) for both the primary and redundant evaporators. Figure 13 to Figure 16 show cross-sections of the accumulator in these 3 orientations. Under

gravity conditions, the liquid ammonia is at the bottom of the accumulator, which is indicated by blue in the CAD drawings. Under micro-gravity conditions, the liquid will be located around the porous tube, see Figure 14 for an example of a liquid distribution simulated with OpenFOAM⁶.

The saturation temperature in the MPL is controlled by the accumulator. For this reason, the accumulator is equipped with 4 primary foil heaters (indicated with green in figures) and 4 redundant foil heaters (indicated with yellow) to keep the accumulator at the desired temperature (approximately 80°C) during operations. The four primary heaters have a total combined heating capacity of 212 W. The redundant heaters also have a total heating capacity of 212 W. Besides these flight heaters, the accumulator is also equipped with 2 additional ground test heaters, which can be used to quickly increase the accumulator temperature during terrestrial testing in orientation A (these ground heaters are not intended to be used during flight). These heaters have a heating capacity of 212 W each. Note that the heaters are partly located above the liquid level. For this reason, the inner wall of the accumulator is lined with a 3 mm thick porous material, which transports the liquid to the heaters by capillary forces.

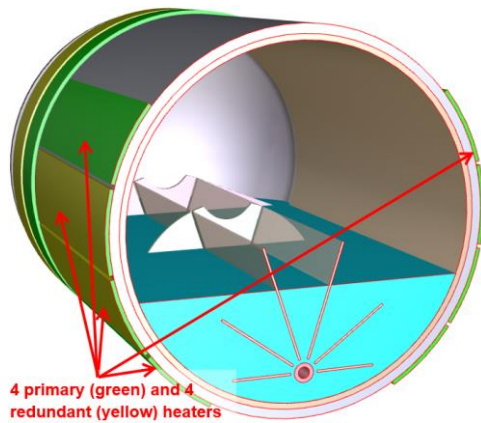


Figure 13. Cross section drawing of the accumulator in orientation C

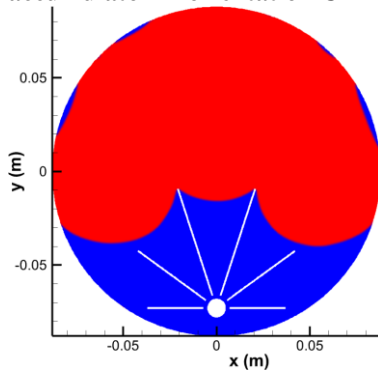


Figure 14. Simulated liquid distribution with void fraction of 0.7 in zero gravity

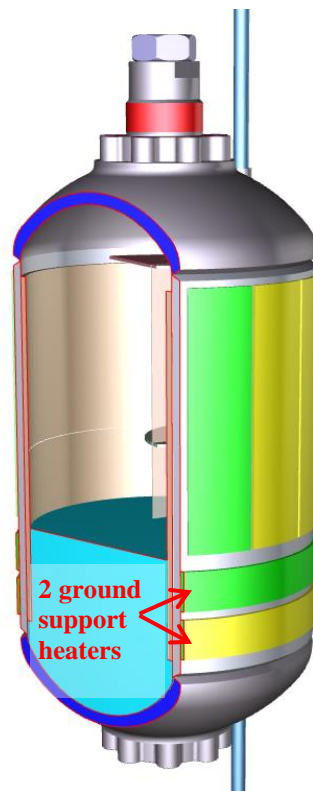


Figure 15. Cross section drawing of the accumulator in orientation A (vertical)

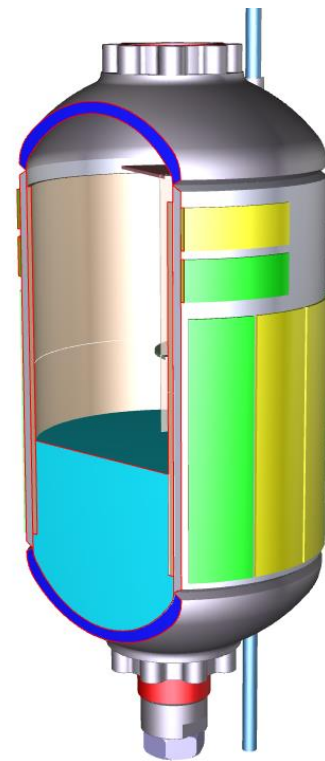


Figure 16. Cross section drawing of the accumulator in orientation B ('upside down')

A. Orientation C, heat load turned on or off instantaneously

The IMPACTA system has to be able to handle variations in the heat load. The most extreme heat load variation is when all evaporator heaters (9.8 kW in total) are turned on or off instantaneously, and this is tested with the following test sequence:

- The pump is started one minute after the start of the measurement.
- 5 minutes after the start of the pump, the evaporator heat load is turned on for all branches (9.8 kW heat load).
- After 4 hours, the heat load is turned off.
- After 20 minutes, the pump is turned off, and after 5 more minutes the measurement is ended.

Figure 17 and Figure 18 show the massflow and applied heater power on the evaporator, while Figure 19 and Figure 20 show the temperature in the heater blocks of the 6 branches, and the fluid temperature (measured on the

outside of the tubing) at several locations (see Figure 1 for the location of the sensors). The saturation temperature in these figures is calculated from the pressure at the evaporator outlet. As soon as the evaporator heater power is turned on, the fluid temperature in the evaporator quickly rises until it reaches the saturation temperature after which it starts to evaporate and the fluid temperature no longer rises. As a result of the heat input of 9.8 kW, part of the liquid flow (approximately 60%) is evaporated. This vapour displaces the liquid that is present in the 15 meters of tubing between the evaporator and the condenser, and this displaced liquid flows into the accumulator. This has two effects on the saturation temperature in the system:

1. The liquid that flows into the accumulator compresses the vapour inside the vessel. As a result, the saturation temperature increases
2. The liquid that flows into the accumulator has a low temperature. This cold liquid cools (and condenses) the vapour in the vessel. As a result, the saturation temperature decreases

Both effects can be seen in Figure 19; the saturation temperature first rises from the initial value of 74°C to 82°C because of the compression of the vapour, and then becomes lower (79°C) because of the cold liquid inflow. The saturation temperature in the accumulator is controlled by the accumulator heater. When the temperature is lower than the setpoint temperature of 80°C, the heater power is increased and when it is higher than 80°C, the power is decreased, see Figure 18. In steady state, 140 W of accumulator heater power is required to keep the saturation temperature at 80°C. In a vacuum, the heat leak to the environment will be smaller, which will result in a smaller required accumulator heater power.

When the evaporator power is suddenly decreased, relatively warm liquid flows out of the accumulator. As a result, the temperature of the liquid that goes to the pump is temporarily increased (see red line Figure 20), and there is a drop in the liquid flow out of the condenser (see Figure 17). At the same time, the saturation temperature of the accumulator decreases, because the liquid flow out of the accumulator expands the remaining vapour in the accumulator, which results in a reduction in the saturation pressure (and thereby saturation temperature).

The liquid at the pump outlet is relatively cold (~25°C). The recuperator warms this cold liquid to 75°C before it enters the evaporator section.

Figure 21 to Figure 23 show the measured temperatures in the NLR evaporator branches, see Figure 10 for the location of the temperature sensors. Evaporators NLR1a and NLR1b are the same and should give similar results. These evaporators are described in the paper¹ about the preliminary design of the system, which also contains test results of evaporator samples. Evaporator branch NLR2 is similar to NLR1a and NLR1b, except that the heater blocks are mounted differently, which results in slightly higher temperatures.

The maximum steady-state temperature in the heater blocks is 94.6°C for branch NLR1a and 95.6°C for branch NLR1b. The measured temperatures in heater blocks 1,4,7, and 10 are very similar, the difference is less than 2°C, i.e. the temperature uniformity in the evaporator branches is better than 2°C. The maximum temperatures in the heater blocks are very similar to the maximum temperature of 96°C that was measured in the evaporator samples¹. Before the liquid in the NLR evaporator branches starts to boil, the liquid is slightly superheated (i.e. the liquid temperature is above the boiling temperature) which results in a small temperature overshoot, see Figure 22. This overshoot was not observed in the evaporator sample tests¹. A possible reason for the temperature overshoot in NLR branches could be that the evaporators for the IMPACTA demonstrator were more thoroughly cleaned before installation, which may have removed some nucleation sites to initiate the onset of boiling. Another possible explanation is that during the evaporator sample test, there might have been a small amount of nitrogen dissolved in the ammonia. This dissolved nitrogen is released from the liquid just before the boiling temperature is reached (the solubility of nitrogen in ammonia approaches zero close to the saturation temperature) and these small nitrogen gas bubbles enhance the onset of boiling. Although the temperature overshoot is not so large, it is recommended to investigate the onset of boiling in a future project. Note that the temperature difference between the fluid and the heater block can be made significantly smaller by reducing the thickness of the heat spreader. For example, the temperatures in the 6 branches are lower (despite the much higher heat flux on the evaporator) because these branches do not have a heat spreader.

Figure 24 shows the measured pressure difference over the pump and evaporator section. There is no large dip in the pressure difference over the evaporator when the evaporator heat load is suddenly decreased. This shows that there is no pump cavitation when the heat load is suddenly reduced.

The measured massflow is very noisy in orientation C, see Figure 17. In orientation A and B, the massflow is far less noisy, see Figure 39. A possible explanation is that the noisy signal is caused by small vapour bubbles that enter the flow meter. The tube between the condenser and flowmeter is short, and vapour bubbles might not be fully condensed by the surrounding cold liquid before they reach the flow meter. The connection manifolds of the condenser have a large diameter (30 mm, which is much larger than the 10 mm internal diameter of the two-phase tubing). Due to this large diameter, the fluid velocity in the manifold is very low, which results in stratified flow. As a result, the

liquid tends to flow through the bottom part of the heat exchanger and the vapour through the top part. This makes the condenser less effective (because vapour cannot condense in the bottom part of the condenser) and this might result in small vapour bubbles that can escape the condenser. In orientation A and B, the vapour flow is evenly distributed over the plates of the condenser and this noisy massflow is not observed. The measured averaged massflow is the same in orientation A, B, and C, so the noise does not seem to have an influence on the measured average massflow.

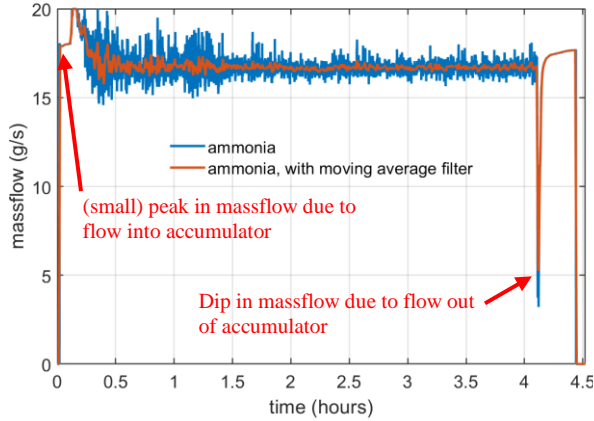


Figure 17. Measured massflow after the condenser

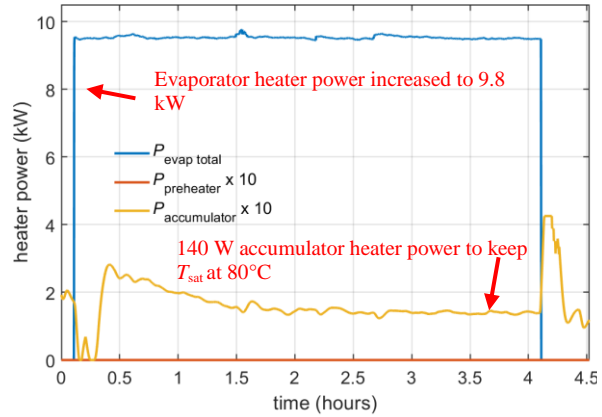


Figure 18. Applied heater power

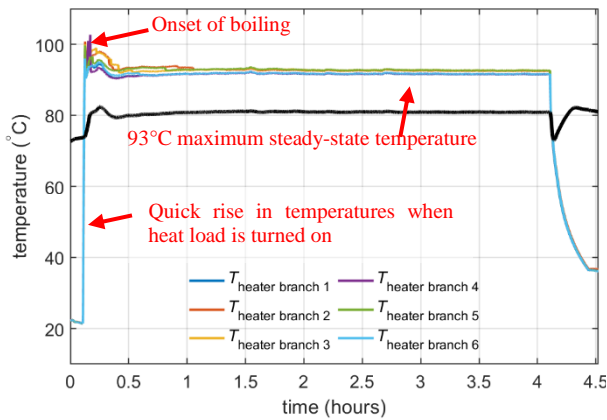


Figure 19. Measured temperatures in the 6 heater blocks

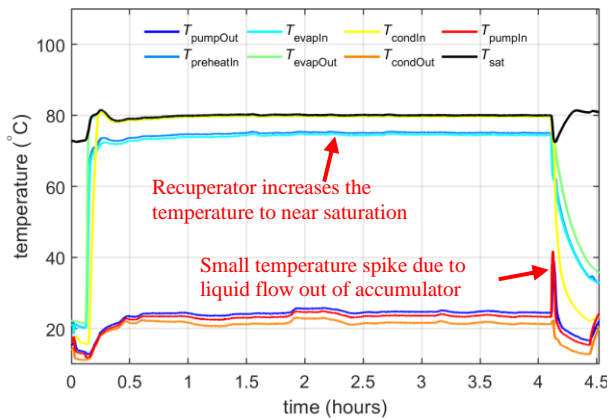


Figure 20. Measured temperatures at various locations

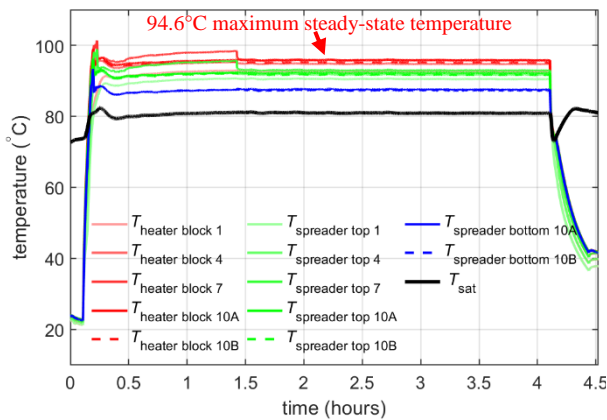


Figure 21. Measured temperatures in evaporator branch NLR1a

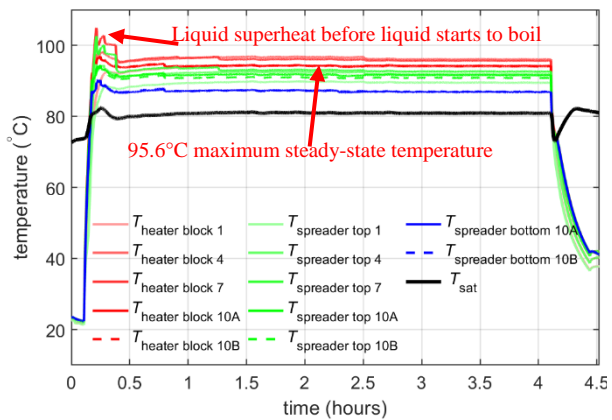


Figure 22. Measured temperatures in evaporator branch NLR1b

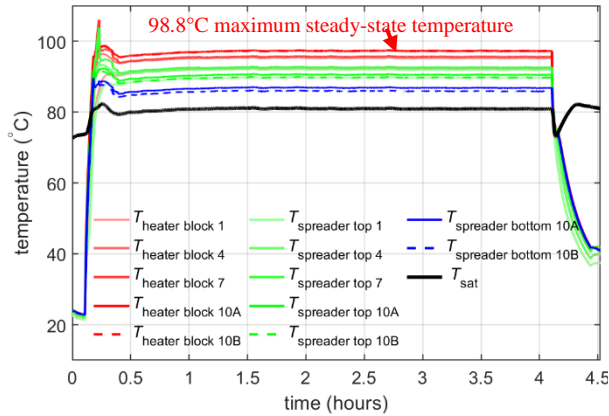


Figure 23. Measured temperatures in evaporator branch NLR2

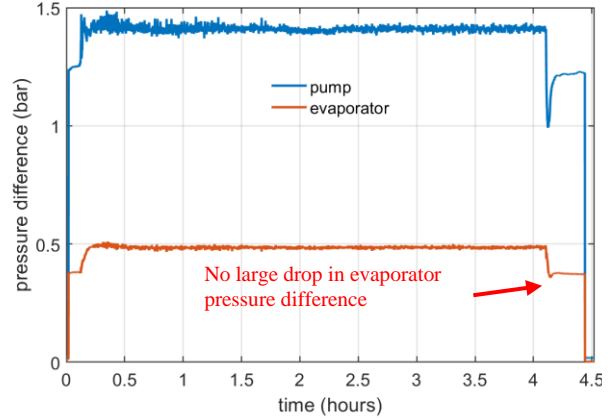


Figure 24. Measured pressure difference over the pump and evaporator

B. Orientation C, imbalance in the heat load

In the previous test, the evaporator heat load was applied to all 10 parallel branches. However, in an actual application, the heat load can be unevenly distributed over the different branches. As a result of this imbalance, the vapour mass fraction can differ between the branches. The frictional pressure drop strongly increases with increasing vapour mass fraction. When the heat load on one evaporator branch is higher than on another evaporator branch, more vapour is generated in that branch. This will result in an increase of the pressure difference over this branch. Since there has to be an equal pressure difference over each branch, the increase in vapour fraction will result in a smaller massflow through this branch if no precautions are taken. For this reason, a flow restriction that dominates the pressure difference over each branch is placed at the inlet of each branch. This flow restriction consists of an orifice that has a diameter of 0.7 mm which results in a pressure difference of 0.3 bar over this flow restriction. In order to test if the evaporator section can handle an imbalance in the heat load between the branches, a ‘worst case’ test was carried out in which half of the branches receive full power and the other branches receive no power:

- The pump is started one minute after the start of the measurement.
- 5 minutes after the start of the pump, the evaporator heat load is turned on for branches 1,2,3, NLR1a and NLR1b (4.9 kW heat load in total). No heat load is applied on the other 5 branches.
- After 3 hours, the heat load on the active branches is reduced to zero, and the other 5 branches (branch 4,5,6, NLR2 and CEA) are turned on (4.9 kW heat load in total).
- After 2 hours, the heat load on the active branches is reduced to zero, and the evaporator heat load is turned on again for branches 1,2,3, NLR1a and NLR1b (4.9 kW heat load in total).
- After 2 hours, the heat load for all branches is turned off.
- After 20 minutes, the pump is turned off, and after 5 more minutes the measurement is ended.

Figure 25 to Figure 29 show the test results. Even with this extreme imbalance in the heat load, none of the evaporator branches shows any sign of dry-out. This means that the system can handle an imbalance in the heat load.

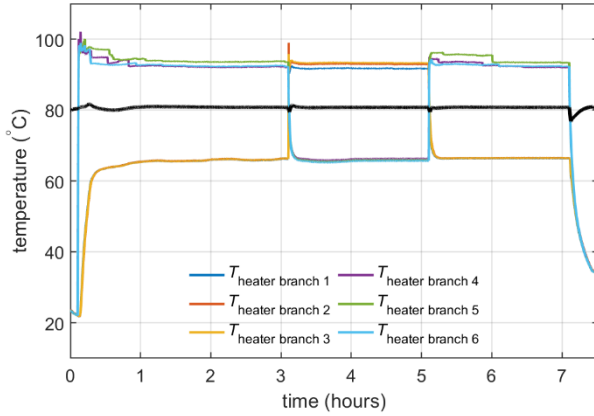


Figure 25. Measured temperatures in the 6 heater blocks

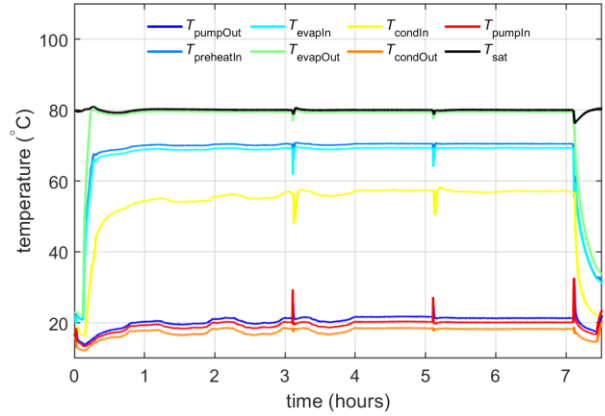


Figure 26. Measured temperatures at various locations

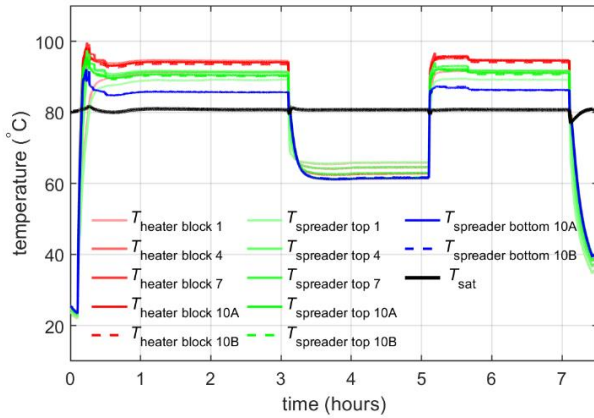


Figure 27. Measured temperatures in evaporator branch NLR1a

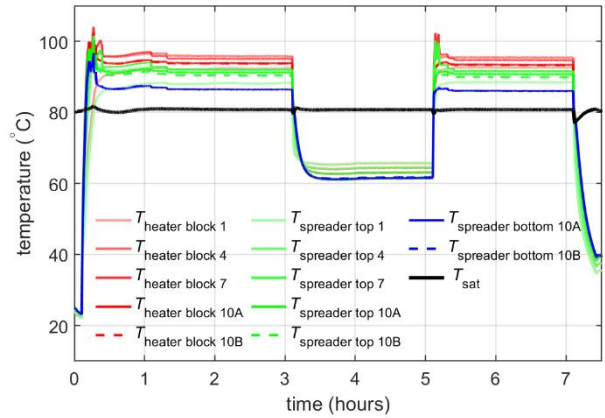


Figure 28. Measured temperatures in evaporator branch NLR1b

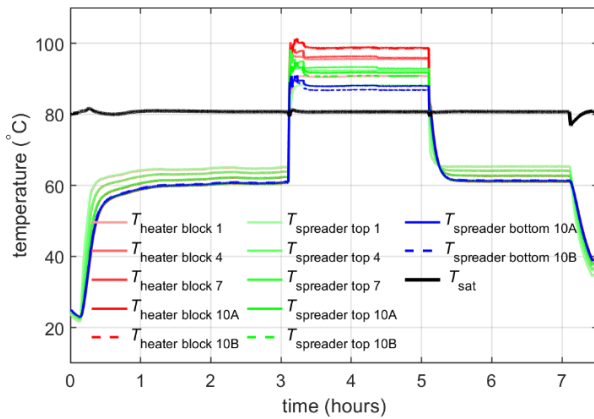


Figure 29. Measured temperatures in evaporator branch NLR2

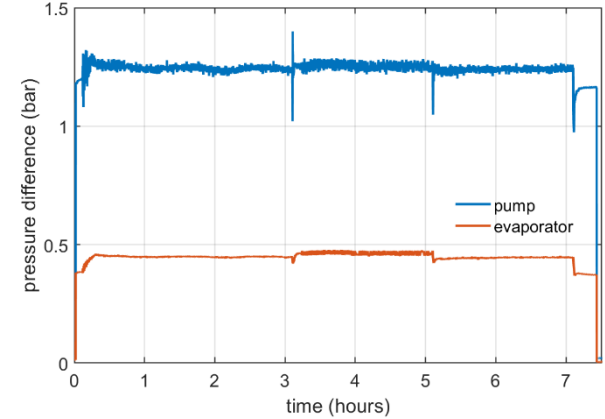


Figure 30. Measured pressure difference over the pump and evaporator

C. Orientation C, variable heat sink temperature over a 24 hours cycle

In a typical active antenna application, the heat sink temperature for the radiator varies as a sinusoid between -40°C and -170°C over a 24 hours period. As a result of the varying radiator heat sink temperature, the ammonia liquid

temperature at the condenser outlet varies approximately between approximately 64°C and 17°C with an evaporator heat load of 9.8 kW. For this reason, a test was carried out in which the building cooling water flow rate was varied such that the condenser ammonia outlet temperature varied approximately between these values (it is not possible to vary the temperature of the building cooling water), see Figure 31. The ammonia pump speed was set to a constant value and the massflow varies between 14 and 16.5 g/s (see Figure 32), because the ammonia density varies with temperature. Figure 34 to Figure 37 show the temperatures in the evaporator branches. These figures show that the temperatures in the evaporator branches are not influenced by the heat sink temperature.

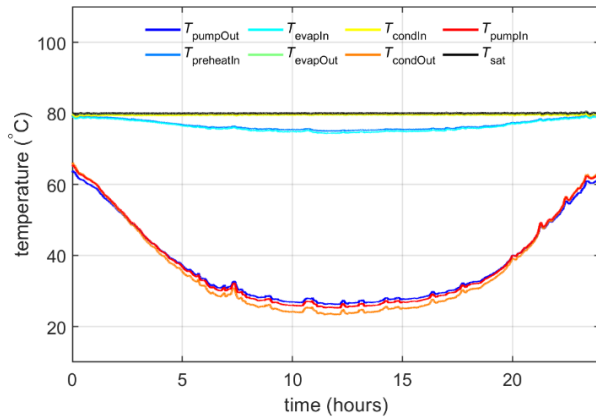


Figure 31. Temperatures at various locations

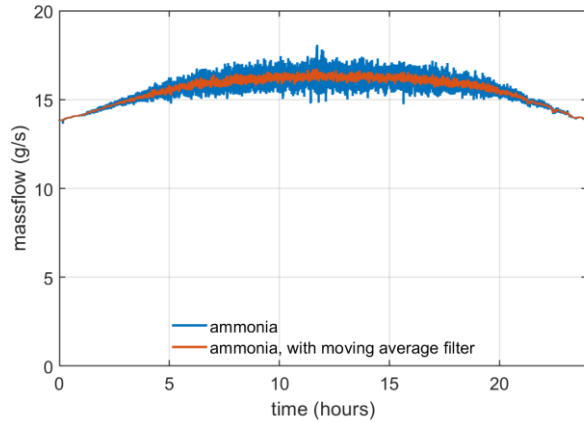


Figure 32. Measured massflow after the condenser

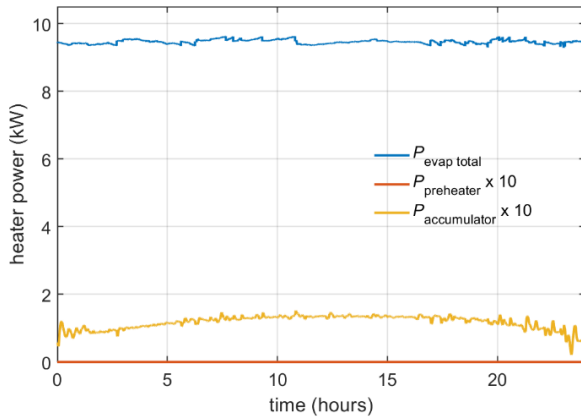


Figure 33. Applied heater power

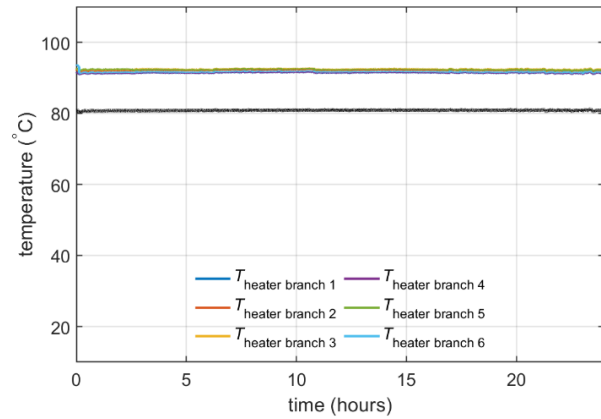


Figure 34. Temperatures in the 6 heater blocks

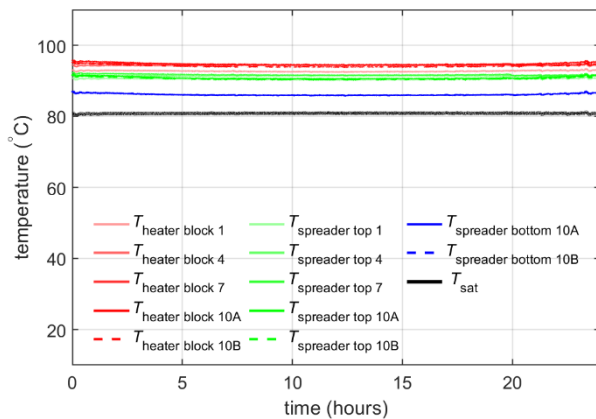


Figure 35. Temperatures in branch NLR1a

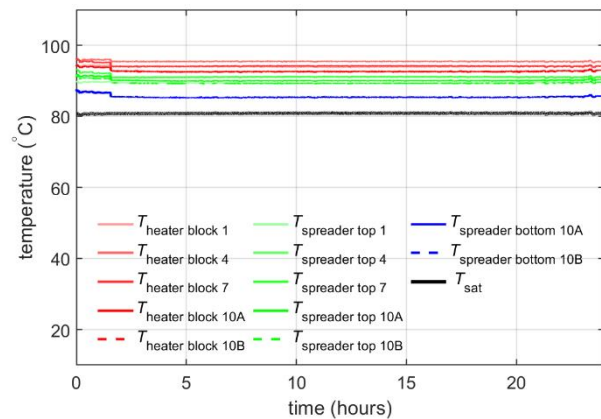


Figure 36. Temperatures in branch NLR1b

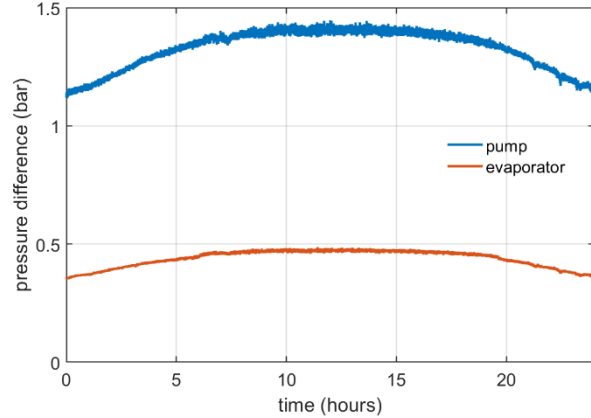
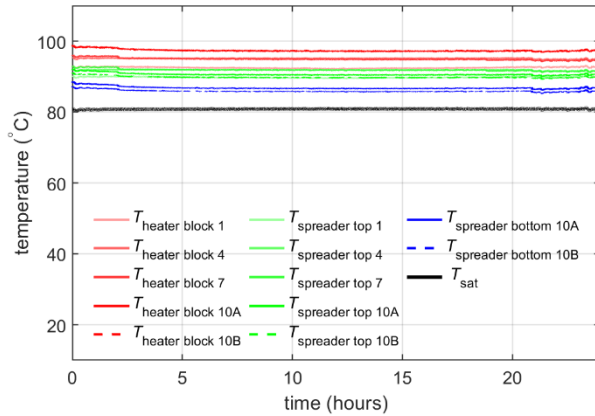


Figure 37. Temperatures in evaporator branch NLR2 **Figure 38. Measured pressure difference**

D. Orientation A, heat load turned on or off instantaneously

Figure 39 to Figure 46 show the measurement data of a test with the same parameters as described in section III.A, except that the system is now in orientation A (vertical accumulator and evaporator section). Another difference is that the ground support heaters of the accumulator were set to constant power of 100 W. The results are very similar as for orientation C (horizontal), which indicates that the system works also well in this orientation. However, there are some differences between the measurements:

- The flow meter data is far less noisy than for orientation C, see Figure 39.
- The temperature at the evaporator inlet is much closer to the saturation temperature, see Figure 42. This is because the recuperator works more effectively in orientation A and B than in orientation C. The reason for this is that the manifolds of the recuperator have a large diameter (30 mm, which is much larger than the 10 mm internal diameter of the two-phase tubing). Due to this large diameter, the fluid velocity in the manifold is very low, which results in stratified flow. As a result, more liquid flows through the bottom channels of the recuperator than through the top channels in orientation C, and this makes the recuperator less efficient. In micro-gravity, there will be no stratified flow in the manifold and it is expected that the recuperator works as efficient as in orientation A and B.
- The temperatures in the six evaporator branches (see Figure 41) and in branches NLR1a, NLR1b and NLR2 (see Figure 43 to Figure 45) are nearly identical as in horizontal orientation.
- The spike in the pump inlet temperature is much higher than in orientation C, see Figure 20. Pump cavitation can occur if the pump inlet temperature becomes too close to the saturation temperature. The reason why the spike in orientation A is larger than in orientation C is because in orientation C, the tube in the accumulator is surrounded by relative cold liquid, and this reduces the spike in the pump inlet temperature.
- In steady state, 248 W of accumulator heater power is required to keep the accumulator at 80°C, see Figure 18, which is much higher than in orientation C. The reason for this is that in orientation C, the tube through the accumulator is surrounded with relatively cold liquid, while in orientation A and B, part of the tube is in direct contact with the relative hot (80°C) vapour, which results in high condensation heat transfer.
- The increase in saturation temperature when the evaporator heat load is suddenly increased is much smaller (it only increases from 80°C to 81.4°C). The reason is that in vertical orientation, the heat leak from the accumulator to the cold liquid that flows through it is larger, which results in a smaller increase in the saturation temperature.

The measurement described above were also carried out with the three-way valve turned such that the flow is directed through the redundant evaporators instead of the primary evaporators. All measurement data is practically the same as for the test with the primary evaporator, except that for branches NLR1a, NLR1b and NLR2 the maximum temperature is approximately 2°C higher than for the measurements with the primary evaporator because the heat has to travel through the primary evaporator before it reaches the redundant evaporator, see Figure 9.

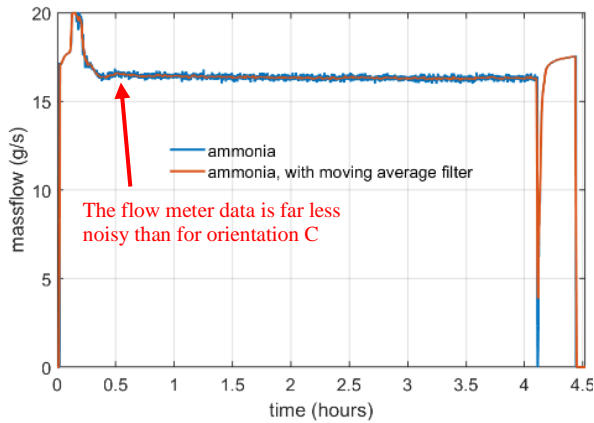


Figure 39. Measured massflow after the condenser

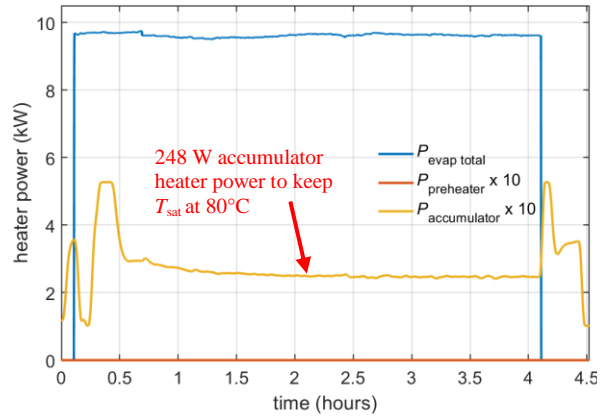


Figure 40. Applied heater power

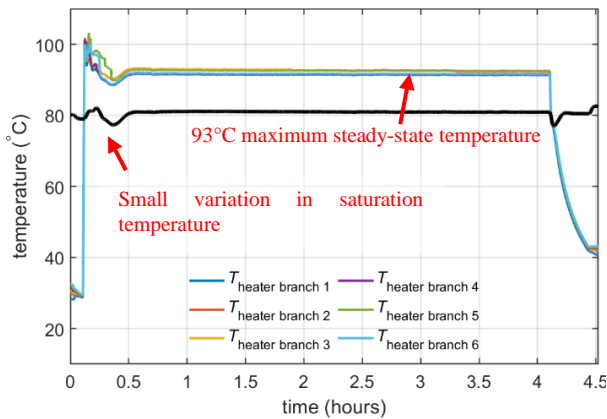


Figure 41. Measured temperatures in the 6 heater blocks

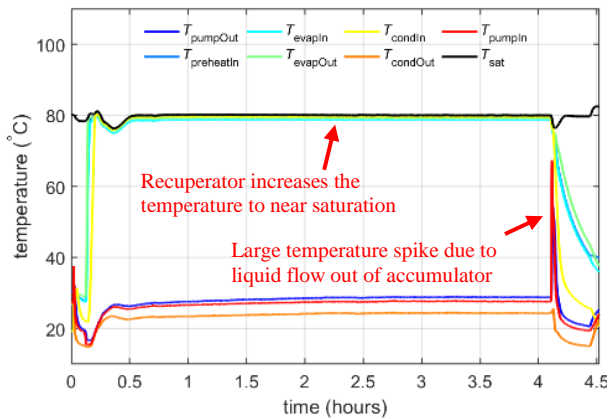


Figure 42. Measured temperatures at various locations

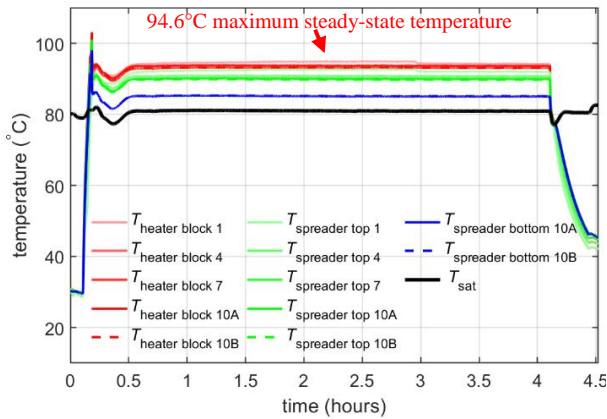


Figure 43. Measured temperatures in evaporator branch NLR1a

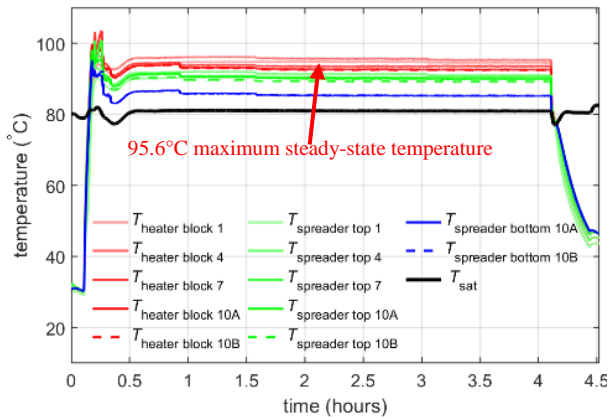


Figure 44. Measured temperatures in evaporator branch NLR1b

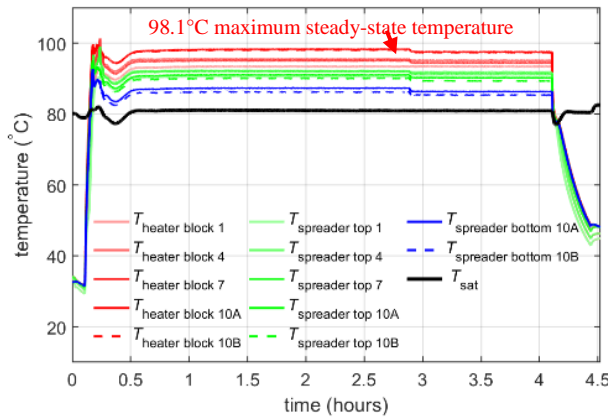


Figure 45. Measured temperatures in evaporator branch NLR2

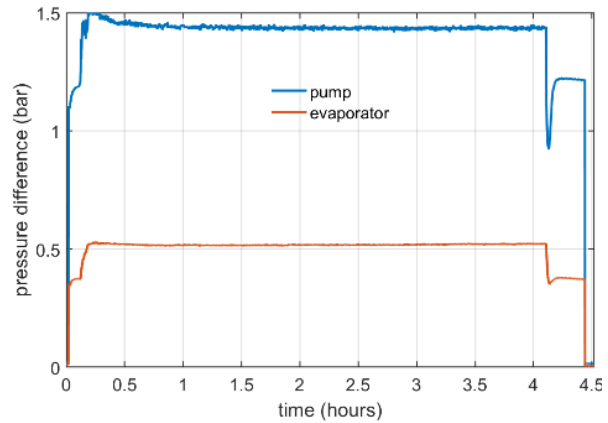


Figure 46. Measured pressure difference over the pump and evaporator

E. Orientation B, heat load turned on or off instantaneously

Test were carried out with the same parameters as described in the previous section, except with the system in orientation B (vertical ‘upside-down’ accumulator). The results are very similar to orientation A (vertical), which indicates that the system performs well in either orientation, and that gravity only has minor effects on the system.

IV. Conclusion

The IMPACTA demonstrator has been tested in different orientations. The system is able to cool a total heat load of 9.8 kW divided over 10 parallel branches and the test results correspond well with the analyses¹. The spatial temperature uniformity over the heat sources is better than 2°C. In an active antenna application, the heat load can be unevenly distributed over the different branches. Tests showed that even in the extreme case when half of the branches are turned off and the other half are set to full power, no sign of dry-out or too high temperatures is observed. The system is able to operate in 3 different orientations and the test results are similar for all orientations, indicating that the system is not sensitive to gravity effects. Using the redundant evaporators instead of the primary evaporators results in a 2°C higher temperature, which is well within the requirement.

Acknowledgments

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- ⁵Roldan, N., Castro, C., “Development of a Two-Phase Mechanically Pumped Loop for Active Phased Array Antennas”, European Space Thermal Engineering Workshop, 2022.
- ⁶OpenFOAM (version 5) with InterFOAM solver, URL: <https://openfoam.org/>.