Nationaal Lucht- en Ruimtevaartlaboratorium
National Aerospace Laboratory NLR

NLR TP 96380

High efficiency low pressure drop two-phase condenser for space

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DOCUMENT CONTROL SHEET

	ORIGINATOR'S REF. **NLR TP 96380 U**		SECURITY CLASS. **Unclassified**

ORIGINATOR
National Aerospace Laboratory NLR, Amsterdam, The Netherlands

TITLE
High efficiency low pressure drop two-phase condenser for space.

PRESENTED AT
26th International Conference on Environmental Systems, Monterey, CA, USA, 8-11 July 1996.

AUTHORS	DATE	pp	ref
A.A.M. Delil, J.F. Heemskerk, O. Mastenbroek and W. Supper	**960113**	**17**	**3**

DESCRIPTORS

Capillary flow	**Performance tests**
Condensers (liquefiers)	**Pressure drop**
Equipment specifications	**Reduced gravity**
Heat transfer	**Spacecraft temperature**
Heat pipes	**Two phase flow**

ABSTRACT
This paper describes all analytical and experimental activities (of NLR, prime contractor, and the subcontractors Bradford Engineering and Daimler Benz Aerospace-Space Infrastructure) to develop a high efficiency low pressure drop condenser and to bring it up to pre-qualification level:

- **The inventory, dedicated to a literature review of potential concepts and means to improve condenser efficiency and to lower pressure drop, followed by a trade-off, supported by the necessary thermo-hydraulic analysis, yielding the definition of the design.**
- **The manufacture of the condenser defined and the execution of an extensive test programme, performed under conditions reflecting, as close as possible, realistic in-orbit conditions. Thermo-hydraulic modelling was done to predict test results prior to the execution of the tests.**

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Contents

(17 pages in total)

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INTRODUCTION

For the next generation of large satellites and platforms, currently widely used heat pipes might meet their capability limits. Capillary pumped two-phase systems are considered prime candidates to provide efficient thermal management systems for such applications. Typical heat loads range from a few hundred Watts to a few kW. Transport lengths are typically several metres.

Another promising potential application for capillary pumped two-phase loops is the thermal control of advanced larger instruments (e.g. laser-based earth observation instruments) or microgravity experiments. For such applications the typical heat loads are in the order of several hundred Watts to be transported over distances of up to several meters, with very stringent requirements for the temperature stability (0.1 to 1 K) and in most cases the demand to actively control the working temperature setpoint.

It is obvious that, in such future applications, condensers with high-efficient condensation heat transfer (under zero-g) and low-pressure drop will be required.

Two different condenser types can be distinguished:

- One for a direct condensing radiator, where the heat of condensation is directly radiated to space.
- One for a hybrid condenser radiator, containing a central heat pipe to transport the heat from the condenser to the distribution heat pipes on the radiator.

The disadvantage of a hybrid condenser with respect to the direct one is its more complex construction. Advantages of hybrid condensers are low vulnerability for micro-meteoroid puncture and inherent protection against condenser freezing.

Since direct condenser features are available from the work done by MMS-UK, BE and NLR within the ESA study on the ATLID laser head thermal control breadboard (Ref. 1), it has been decided to focus on the hybrid condenser.

REQUIREMENTS SPECIFICATIONS

The requirement specifications following from the statement of work, are listed next.

General Conditions and Requirements

The condenser design has to reflect awareness of realistic spacecraft constraints regarding size, mass and material selection. It has to be capable of meeting typical spacecraft shock, vibration and other environmental conditions. Whenever possible, ESA approved parts and materials are to be selected. The condenser has to comply with common spacecraft requirements regarding quality control and has to undergo comprehensive testing.

Lifetime and Mission Profile

As a guideline, the condenser is intended to be compliant with either one mission with three years storage, two years testing time, one launch and ten years active life on orbit (single launch mission) or up to five missions with a total of five years storage, five years testing time, five launches, five recoveries and a total of thirty months active life orbit (EURECA type mission) or up to ten missions with a total of ten years storage, five years testing time, ten launches, ten recoveries and a total of one year active life orbit (SPACELAB type mission).

Working Fluid

The condenser has to meet all requirements induced by operation with common working fluids, in particular anhydrous ammonia, the fluid used in this study.

Performance Requirements

The performance requirements defined below concern an operating two-phase loop, i.e. when heat is transferred by circulation and phase change of the working fluid.

Temperature and pressure ranges

The condenser has to meet all performance requirements within the following ranges:

- Under operational conditions: -40 °C to +60 °C. The 60 °C fluid saturation pressure determines the maximum operating pressure (MOP)
- Under non-operational conditions: -60°C to +80 °C. The 80 °C fluid saturation pressure determines the maximum design pressure (MDP)

Heat load

The heat load to be rejected by the condenser depends on the application. The heat load envelope of applications envisaged ranges from 100 to 5000 W. The condenser shall be able to reject any of these loads by a modular approach using nominal 300-350 W condenser-radiator elements as baseline.

Condensation film coefficient

The condensation film coefficient based on the internal hydraulic diameter of an elementary condenser channel shall be at least 2000 W/m^2K. As a goal, a film coefficient of 5000 to 10000 W/m^2K shall be met.

Pressure drop

The pressure drop of the condenser depends on the selected working fluid. By appropriate design the pressure drop of the condenser shall be, as far as possible, independent of the mission specific design heat load. The pressure drop shall be minimised (aim: below 300 Pa for ammonia).

Control accuracy and response time

The design shall aim for a minimum fluid content, optimise the overall loop accuracy and minimise the response time.

Testing on earth

The dependence of the condenser performance upon level and direction of gravity shall be minimised as far as possible. Ideally, the performance shall be independent of the gravity field. An attempt shall be made to predict μ-g condenser performance figures, based on the outcomes of horizontal and vertical tests.

TRADE-OFF

Preliminary Concept Selection

The selection & design of a hybrid radiator condenser can be split into:

- The internal configuration of condensing flow passages.
- The thermal connection to the heat pipe.

Various internal condenser design solutions and thermal connections can be combined. To reduce the number of concepts to be evaluated, the heat pipe/condenser connection and the condenser flow passage design are separately traded. From these trades the optimum combination for a hybrid radiator will be derived.

Internally the condenser concepts differ by:

- Arrangement of the condenser channels: axial or helical grooves.
- Channel cross-section: square, trapezoidal, triangular.
- Arrangement of the wick structure: without wick, wick at condenser outlet, cylindrical wick separating the axial vapour and liquid grooves.

Many combinations of these three features are possible. Fortunately, some can be excluded (Table 1), as elucidated in reference 2.

Selection of the flow passage concept

Four flow passage concepts (Figs. 1, 2) remain for the detailed evaluation presented in table 2. The concept H-A obviously is the best condenser flow passage design, followed by concept H-D. But as the patent situation of the concept H-D is not clarified yet, the concept H-C is the back-up solution, instead H-D.

Selection of the thermal connection to the heat pipe

The thermal connection can be subdivided into dry or wet concepts. In wet connections the vapour channels are directly milled into the heat pipe wall (Fig. 3). The condenser is an integral part of the heat pipe. In contrast, the condenser of a dry connection is a separate assembly of the condenser. The figures 3 and 4, show some possible concepts. Concept IC-B consists of a cylindrical condenser, which is slipped over the heat pipe using a high conductivity epoxy, adhesive or grease. Another possibility is a Morse-cone connection with interface filler (concept IC-C). If the cone angle exceeds 15°, the connection is detachable (concept IC-D).

The trade-off for the thermal connection, using the relevant evaluation criteria, is shown in table 3. The wet connection (IC-A) is the best, followed by the cylindrically shaped dry connection (IC-B), the back-up solution.

Conclusion: The most promising condenser concept is the combination of wet connection IC-A and H-A passage.

Preliminairy Design Considerations

The agreed option for a hybrid radiator condenser is a 250 W to 300 W unit. This unit is coupled to a central heat pipe which transfers the received heat to secondary heat pipes spreading the heat over the radiator panel.

The radiated power for Low Earth Orbit (LEO) and Geostationary Earth Orbit (GEO) can be calculated under the following assumptions:

- LEO is characterised by a typical sink temperature of about 190 K (ATLID 800 km high polar orbit).
- GEO is characterised by no solar or earth inputs.
- Radiator absorptivity α = 0.06, emissivity ε = 0.80.

The radiator surface and the heat pipe dimensions are estimated under the following assumptions:

- Object to be cooled:

Electronics with target temperature	298 K

- Temperature differences:

Payload to condensing vapour	15.0 K
Vapour to material condenser	2.5 K
Material condenser to main HP vapour	2.5 K
Main HP vapour to vapour of secondary HP	3.0 K

This means a secondary heat pipe temperature, radiator root temperature, of about 275 K. Consideration of radiator fin efficiency and the properties of appropriate secondary heat pipes lead to the need of six secondary heat pipes.

For the above estimated radiator root temperature and six secondary heat pipes the radiated power for LEO (1 m^2) is 175 W and for GEO 230 W. To obtain 300W (GEO) and 200 W (LEO), the radiator area should be extended by some 30% (GEO) (1.3 m^2) or 14% (LEO) (1.14 m^2), since area and radiated power are linearly related.

The main heat pipe shall transport 300 W. A condenser length of say 0.4 m and a radiator of roughly 1 m wide, yields an effective length of (0.4 + 1)/2 = 0.7 m, meaning a heat pipe transport capability of 210 Wm. The performance of the heat pipe for this task should be twice this value to guarantee a wide working range. Reason for doubling is that the typical performance curve of a heat pipe resembles a parabola and the performance quoted in data sheets pertains to the parabola top.

Various heat pipes meet the 400 Wm requirement: The ERNO AGC 13-P1 & AGC 11.5-P1, similar Dornier types, and SABCA AG200-1 (770 Wm) & AG250-1 (1210 Wm).

As a large diameter heat pipe is considered best (lowest radial heat flux) and finned heat pipes have to be machined to obtain a cylindrical joint design, the cylindrical 25 mm diameter SABCA pipe seems the best choice. Therefore values used in the sequel are: 25 mm OD, 2 mm aluminium wall thickness, 15 mm vapour core diameter. The heat pipe film coefficient is around 3000 W/m^2K, which implies a temperature difference of 4.5 K between the condenser and the vapour in the heat pipe (for a 300 W, 300 mm long, 25 mm diameter configuration).

The thermally most effective construction is the use of the heat pipe outside as condensing surface. This means that the main heat pipe is an essential condenser part and the vapour supplying two-phase system has to be opened if the main heat pipe is removed.

A way to keep the two-phase system closed and apart from the main heat pipe is to attach a special condenser piece around the main heat pipe evaporator. Penalty for this construction is a temperature drop over joint and condensing piece wall. A shrink joint has been chosen for this discussion, but a soldered joint or a thermal grease filled joint are equally possible. The latter is easily dismountable, but it has a higher temperature drop.

An estimate for a shrink joint heat transfer coefficient is 10000 W/m^2K, implying a temperature drop of 1.25 K for a 0.3 m long condenser, 0.95 K for a 0.4 m long condenser.

The extra temperature drops have to be compensated for by enlarging the condenser area.

<u>Channelled and annular condenser concepts</u>

The design supporting discussions and calculations must lead to a condenser, characterised by:

- A hybrid condenser unit with a condensation capability of about 200 W for LEO and about 300 W for GEO.
- A radiator area of 1.3 m^2 for GEO, for LEO 1.14 m^2. The heat is distributed by six secondary heat pipes and one main heat pipe, 25 mm OD, 400 Wm.

Calculations for a channelled condenser consisting of 15 channels (4.0 x 2.5 x 300 mm^3) yield a temperature drop from condensing vapour to main heat pipe vapour of about 6 °C and a very low maximum pressure drop of 10 Pa over 30 cm length.

For reliable testing of the hybrid condenser the heat pipes must be horizontally positioned: hence also the condenser must perform well in horizontal orientation. It is preferred that the condenser gives a clear correlation between vertical and horizontal performances. To verify calculations and to assess the sensitivity for orientation with respect to gravity, a channelled condenser sample had to be defined.

In the channelled condenser, the vapour shear must dominate over gravity to yield a clear correlation between the vertical and horizontal performances. Considerations on vapour shear led to a second condenser option: the annular concept, in which capillary forces assist vapour shear to overrule gravity inside the condenser.

If the gap size decreases the capillary forces will become more dominant. However the pressure drop increases with the cube of the reciprocal gap distance, which means for a 0.3 m long, 0.15 mm gap condenser at 300 W and 5 °C, a pressure drop of 940 Pa for pure vapour, 110 Pa for pure liquid. This means a maximum pressure drop above 1000 Pa, being much higher than the pressure drop of the channelled condenser. As a compromise, an annular type concept with a gap size 0.3 mm was chosen to be tested.

The test samples have been tested to determine:

- The thermal and hydraulical performances.
- The performance differences in vertical and horizontal orientation (with or without vapour stops), to assess terrestrial testability constraints.

The power was varied between 100 and 280 W. Tests have been performed with and without vapour stops at the outlet, the heat pipe simulated by forced liquid convection cooling. A glass tube allowed observation of the liquid/vapour front.

The annular concept sample consisted of two concentric tubes, one glass and one aluminium. The condensation takes place in the 0.3 mm gap between the glass and aluminium tube.

The channelled sample consisted of a glass tube and a channelled aluminium tube. The channelled tube included fins, yielding 11 condensation channels of 4.0 x 2.5 x 340 mm^3. The tube was also cooled by forced liquid convection.

<u>Test setup and evaluation of test results</u>

The NLR ammonia test rig (Fig. 5) contains a pump circulating the working fluid through a mass flowmeter, evaporator and condenser. A control reservoir in the loop allowed to set the saturation temperature to a constant value. To ensure only saturated vapour at the condenser inlet, the mass flow was controlled using the evaporator input power and the saturation temperature. By using flexible tubes it was possible to study the condenser behaviour in horizontal and vertical orientation. The pressure drop over the condenser was measured, using a differential pressure sensor. Reliable pressure drop measurements turned out to be possible only for horizontal orientation. In vertical orientation the liquid head deteriorates the measurement. The tested power range was 100 W up to 280 W. The temperature of the incoming vapour, of the water inlet and outlet and the water massflow were measured. As the loop temperature was about 2 °C above ambient temperature and the majority of the loop parts were insulated, thermal losses could be neglected.

Results of the tests are:

- The annular sample performed well in all orientations, in the power range 100 to 280 W, with a condenser temperature drop of 3 to 5 K (between vapour at condenser inlet and heat pipe vapour temperature) and a pressure drop of about 500 to 850 Pa.
- The channelled sample performed well, if equipped with a vapour stop, in the power range 100 to 280 W. The temperature drop was 2 to 6 K and the pressure drop less than 100 Pa. For horizontal orientation, its performance increased by about 20% due to gravity induced stratified flow: the upper channels filled with vapour only, the lower filled with liquid only.
- The tests confirmed that, for both hybrid condenser designs, the thermal resistance from the vapour inlet to condenser surface is relatively low as compared to the other radial resistances (joint, heat pipe evaporator).

FINAL CONDENSER DESIGN

For the final trade-off, the two concepts, annular and channelled, were to be compared with respect to the performances requirements and the trade-off criteria (Table 4). Dimensions are: 25 mm ID, 340 mm length, 0.3 mm gap (annular sample), 15 channels 4.0 mm wide, 2.5 mm high (channelled sample).

From the sample test results it can be concluded that:

- An annular concept has problems concerning the pressure drop (the gap size must be small to give capillary forces of sufficient magnitude). 1000 Pa has been measured, being far above the required value.
- A channelled concept, equipped with a vapour stop in each channel, works well in horizontal and vertical orientation. For a condensed power of 275 W, the average condensation length for horizontal orientation was 18 cm, for vertical orientation 26 cm. The horizontal performance was better, due to flow stratification. The pressure drop in the channelled concept, equipped with vapour stops, was about 100 Pa, meeting the requirement.
- Vertical testing, yielding information of crucial importance for μ-g performance predictions, will be possible only by simulating the central heat pipe.

The annular concept has the highest efficiency and lowest thermal resistance, because of direct condensation at the outer surface. But it has the highest pressure drop, induced by the small capillary gap to guide the condensate. The channelled concept has the lower efficiency induced by the thermal interface resistance between heat pipe and condenser. But it has the lowest pressure drop.

The above suggests that the requirements high efficiency and low pressure drop are hard to combine, meaning that the application determines the choice to be:

- Annular, if thermal efficiency is the driver and pressure drop of minor importance (as in a mechanically pumped two-phase loop).
- Channelled, if the pressure drop is critical, e.g. in capillary pumped systems.

An improved direct condensation channelled concept will be more promising. In such a concept the efficiency will be high by considerable reduction of the interface resistance. This concept uses extruded heat pipes having also an external profile with channels. But the development of such heat pipes is out of the scope of the study. Only when an application requires such a dedicated heat pipe, such an extruded option has to be developed.

An alternative direct condensing channelled option, using standard circular heat pipes, is the condenser concept recommended to proceed with. This concept is simple: two concentric circular tubes, the inner can be a heat pipe or a convectively cooled tube. The spacing is realised by aluminium wires, creating channels. The concept is flexible with respect to the dimensions of the internal tube (heat pipe OD). Thermo-hydraulic requirements determine the gap, hence the wire diameter and number of wires.

Calculations based on a 25 mm OD inner tube/heat pipe yielded a pressure drop close to the 10 Pa of the earlier discussed channelled concept and a direct condensation area close to the earlier discussed annular concept. As the capillary action of the four sharp corners of each channel is far better than in the rectangular channelled concept, the thermal hydraulic behaviour, hence also the horizontal testability will be far better. Thermal hydraulic behaviour can be further improved by a swirling action by routing the wires helically instead of straight around the inner tube.

For this condenser, vapour stops delivering 200 Pa liquid pressure drop, are considered appropriate for:

- Preventing vapour bubbles to leave the condenser exit.
- An approximately equal mass flow distribution between the channels and an improved horizontal testability.

A reliable, well defined way to produce vapour stops is to drill holes in an end cap covering the exits of the channels. Three 0.25 mm diameter holes per channel correspond to 200 Pa liquid drag (at capillary pressure $2\ \sigma/r \approx 400$ Pa).

The above design is straightforward with respect to manufacture. Special attention has to be given to the interface(s) between inner tube and the condenser entrance/exit, which are to be high pressure ammonia leak tight. The condenser had to be tested using a convectively cooled inner tube, allowing simple non-welded interfaces and the possibility of vertical testing.

The drawing (Fig. 6) shows three concentric tubes. At the tube in the middle the vapour condenses. The vapour inlet is a tube with an ID of 4 mm, the vapour is distributed by a cone (part 2). The vapour flows then through a annulus with an ID of 25 mm and an OD of 28 mm. The inner surface simulates a heat pipe, it is liquid cooled. In this way fast exchange of parts and adaptation of parameters was possible during the test campaign, while the realistic simulation of heat transfer coefficients relevant to an inner heat pipe configuration remained possible. The cooling liquid flow trough the innermost tube (part 6) and than flows back through the water outlet (part 9). The vapour stops are located at the end of the condenser (part 8) and consists of 6 x 3 holes with a diameter of 0.25 mm. Part 10 is the outlet for the condensed liquid. Six wires with a diameter of 1.5 mm subdivide the annulus into six parts, the wires are coiled around the tube in the middle, which results into helical condensation channels.

TEST PROGRAMME AND PREDICTIONS

The condenser has been tested for a vapour inlet temperature of 40°C, 20°C and -10 °C, in horizontal and vertical position, for a power range from 100 to 300 W. For each vapour temperature and orientation two graphs have been produced, i.e.:

- The condensation length as a function of the condensed power, for a temperature difference between vapour and cooling liquid needed to keep the vapour/liquid front just in front of the condenser exit (maximum power).
- The minimum temperature difference between vapour and cooling liquid temperature as a function of the condensed power, to keep the vapour/liquid front in the condenser.

The condenser has been tested with the NH_3-test rig. The loop heat losses were minimised by the use of guard heaters/coolers. Temperature controlled dry nitrogen has been used around the test rig to prevent condensation of water from the humid air and to minimise heat losses. The condenser instrumentation included:

- 32 thermocouples on the outside along the stream: 8 in line on top, 8 in line at left side, 8 in line at right side, and 8 in line at the bottom.
- 4 thermocouples on the inside to measure the condenser innerwall temperature .

A thermostat bath has been used for cooling. All temperatures, differential pressure, absolute pressure and heater power were recorded at least every minute.

Heat Pipe Simulation Aspects

The condenser was cooled by convection. To derive the performance of the condenser integrated with a heat pipe from the convective cooling data, the measured temperature difference must be corrected, as the heat transfer coefficient of the convective cooling differs from the heat transfer coefficient of an actual heat pipe. Fig. 7 shows the heat transfer coefficient as a function of the distance from the inlet (for the tested temperatures) calculated using CFDS-FLOW-3D. For 40 °C and 20 °C the flow is turbulent, therefore a K-ε model has been used. For -20°C the flow is laminar (due to the high viscosity of a 50/50 water ethanol mixture to be used at lower temperatures). The fluid properties are temperature dependent, especially the viscosity which increases by a factor 20 (10^{-3} Pa s for 20 °C and 20 10^{-3} Pa s for -20 °C, the temperature of the coolant during tests at -10°C). This means a temperature dependent heat transfer coefficient, for which a correction must be made to derive T_{vap} - $T_{heat\ pipe}$ (vapour core) from T_{vap} - $T_{coolant}$.

The maximum value of the correction term for a power of 300 W, for a 25 mm OD/19 mm ID heat pipe with an assumed evaporator heat transfer coefficient of 3000 W/m²K, turned out to be:

- 2.3K at 40°C setpoint (7500 W/m²K convective cooling coefficient).
- 0.7K at 20°C (4500 W/m²K cooling coefficient).
- -4.3K at -10°C (2000 W/m²K cooling coefficient).

The above implies that only at 20 °C the convective cooling properly simulates the heat pipe evaporator. For the other vapour temperatures, corrections have been made.

TEST RESULTS, EVALUATION, CONCLUSION

Results

The figures 8 and 9 depict measured results: T_{vap} - $T_{coolant}$ versus the condensed power (ΔT curves) and the condensation length versus the condensed power (L_c curves). The ΔT curves were measured for liquid front positions just before the vapour stops (totally effective condenser). The L_c curves were measured for a difference T_{vap} - $T_{coolant}$ corresponding to the measured difference for a power of 275 W and a liquid front position just before the vapour stops. Pressure drops (measurable only in the horizontal setup) are also shown.

From the graphs it can be concluded that:

- T_{vap} - $T_{coolant}$ and the condensation length L_c are directly proportional to the condensed power.
- T_{vap} - $T_{coolant}$ depends on the coolant temperature induced different heat transfer coefficients. Figure 10 yields the corrected (T_{vap} - $T_{heat\ pipe}$) curves. The latter figure shows that the performance is almost independent of the inlet vapour temperature. For 300 W condensed power, the drop between vapour inlet temperature and the heat pipe vapour core temperature is between 5 and 7.5 °C, being slightly above estimated value 5 °C.
- The condenser performance in horizontal and vertical orientation is equal. This is confirmed by infrared pictures taken (Refs. 2, 3).
- Pressure drop is independent of the vapour temperature.

The following has been observed during testing:

When some vapour is forced to reach a vapour stop (while testing, at fixed loop set point, by decreasing the temperature drop between vapour and coolant) the pressure drop increases rapidly up to values higher than 2000 Pa. This is because of blockage of condenser holes, forcing the liquid to pass through a smaller area at the condenser exit.

Comparison with Predictions and Evaluation

The predicted temperature differences T_{vap} - $T_{coolant}$, turned out to be less than 0.3 K above the measured values. This suggests that the heat transfer coefficient value used for the predictions is too low (though it was obtained by correcting the annular flow heat transfer algorithm by heat transfer enchancement induced by swirl, direct condensation in the entrance region, slug flow in the exit region and and the very efficient drainage in the corners between the separating wires and annular surfaces).

The measured pressure drop (1000 Pa at 275 W) was far above the 300 Pa predicted. This discrepancy could be attributed to a large pressure drop generated in the very narrow gap at the condenser entrance cone, which was not taken into account for the predictions. Therefore, after increasing the cone gap from 0.3 mm to 2 mm the condenser was re-tested at room temperature (at power values up to even 4-50 W).

Figure 11 depicts the measured power dependence of the difference between vapour temperature and coolant temperature for a fully active condenser (the vapour liquid front being close to the exit, having 6*3 vapour stop holes) and also the corresponding total pressure drops during this horizontal testing. The figure clearly shows low pressure drop performance: below 400 Pa at a power of 300 W.

Increasing the number of vapour stops will not only reduce the risk for hole blockage by particles, but will also reduce the contribution of the liquid pressure drop across these holes. In other words, an increase of the number of holes will lead to a further condenser pressure drop reduction.

Figure 12 can be used to estimate the magnitude of this reduction. The figure shows the calculated liquid pressure drop for the condenser tested (6*3 = 18 holes) confirmed by pressure drops measured for pure liquid flow rates, corresponding to power values up to 450 W. The calculated "6*6 holes and 6*9 holes" curves show the appreciable liquid pressure drop reduction that can be achieved by increasing the number of holes per channel from 3 to 6 and 9 respectively.

Figure 13 depicts the resulting fluid dynamic condenser pressure drops predicted for the different number of holes. It illustrates that the target low pressure drop value for a fully operational condenser (below 300 Pa, at 300 W) can be easily reached. Finally it is remarked that the calculated fluid dynamic pressure drops have been confirmed by measurement derived values at higher power values only (say above 200 W).

<u>Conclusion</u>

Recalling the trade-off criteria and the requirements (Table 5), it can be concluded that the condenser, with a cone gap of 2 mm, an annular gap width of 1.5 mm and four or five vapour stop holes in each of the six channels, will meet all requirements for the High Efficiency Low Pressure Drop Condenser envisaged.

ACKNOWLEDGEMENT

We express our appreciation for the efforts and support of our colleagues in the project:

- R. Müller, W.D. Ebeling and K. Beckmann (Daimler Benz Aerospace-Space Infrastructure): Requirements Specifications, Trade-off and Test Plan.
- R. Voeten, P. van Put and R. Jacobs (Bradford Engineering): Production of Test Samples & Condenser Hardware, Structural Calculations.
- A. Pauw (NLR Space Division): Testing.

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Table 1: Flow passages for evaluation

Name	Grooves		Grooves		Wick	
	Axial	Helical	Square	Triangular	Outlet	Cylindrical
H-A	X		X		X	
H-B	X		X			X
H-C	X			X	X	
H-D		X	X		X	

Table 2: Flow passage trade-off

Trade-off approach: weight factors assigned to the criteria (as listed) are multiplied with marks between 1 for the worst concept and 10 for the best.

Evaluation criterion	Weight Factor	H-A	H-B	H-C	H-D
		Weight x Marks			
Thermal coupling	16	16	160	32	144
Low pressure drop	15	150	45	105	15
Compactness	10	100	10	100	100
Micrometeroid/SpaceDebris Vulnerability	7	70	7	70	70
Reproducible condenser efficiency	8	-	-	-	-
Prevention vapour migration/ NCG trapping	5	5	50	5	5
Dismountability	7	-	-	-	-
Scalability	5	-	-	-	-
1-g	8	-	-	-	-
1-g/μ-g Correlation	5	50	5	50	35
Design Complexity	6	60	6	60	42
Heritage/ Development risk	8	80	56	8	56
	100	531	339	430	467

Table 3: Thermal connection trade-off

Trade-off approach: weight factors assigned to the criteria (as listed) are multiplied with marks between 1 for the worst concept and 10 for the best.

Evaluation criterion	Weight Factor	IC-A	IC-B	IC-C	IC-D
		Weight x Marks			
Thermal coupling	16	160	16	48	16
Compactness	10	100	80	30	10
Reproducible Condenser efficiency*	8	80	8	56	40
Dismountability	7	7	21	7	70
Design complexity	6	60	60	6	6
Heritage/ Development risk	8	80	40	8	8
	55	487	225	155	150

* Interface conductance depends on temperature of and temperature difference across interface.

Table 4: Comparison of sample performances and requirements/trade-off criteria

Criteria	Annular concept		Channelled concept	
Thermal coupling	+++	highest possible	++	depends on joint, e.g. shrink, grease
Low pressure drop < 300 Pa	+ -	approx. 1000 Pa *) approx. 1300 Pa	+++ ++	< 30 Pa *) approx. 100 Pa
Compactness	+++	very compact	++	
Reproducible condenser efficiency	++	film coefficient > 5000 W/m^2 K	++	film coefficient > 5000 W/m^2 K
Prevention of vapour leaving outlet (horizontal)	+ +++	for low powers *)	- +++	stratified flow *)
Dismountability/ easy to integrate	+	impossible without emptying loop	+++ -	grease filled soldered/ shrink
Scalability	++		++	
1-g Testing vertical horizontal	 ++ ++	 if gap < 0.3 mm	 ++ ++	 *)
1-g/µ-g Correlation vertical horizontal	 ++ ++	 capillary/shear dominant	 ++ +	 Annular flow *) Stratified flow
Design complexity	++		+	thermal joint
Heritage/ development risk	+	compromise, gap size - pressure drop	++	low risk

+++ best ++ good + moderate - worst *) Sample equipped with vapour stop

Table 5: Condenser performance, requirements and criteria

Criterion/Requirement	Satisfied	Remarks
Thermal coupling	+++	Highest possible due to direct condensation on the heat pipe
Low pressure drop < 300 Pa at 300 W	++	For cone gap 2mm and 4 or 5 vapour stop holes in each of the 6 channels
Compactness	++	
Reproducible condenser efficiency > 5000 W/m^2 K	+++	High efficiency due to helical channels, heat transfer above 10^4 W/m^2 K
Prevention of vapour leaving outlet (horizontal)	++	Vapour stops perform well, also in horizontal operation
Dismountability/ Easy to integrate	+	Ammonia circuit must be empty
Scalability	++	Reliable condenser thermal model guarantees proper scalability
1-g Testing vertical horizontal	+++	No significant difference in performance for horizontal and vertical orientation, as shear forces dominate condensation process
1-g/µ-g Correlation vertical horizontal	+++	Shear dominates condensation process
Design complexity	+	Connection to heat pipe is critical

+++ best ++ good + moderate - worst