# FUNDAMENTAL DIFFERENCES BETWEEN LIQUID-VAPOUR AND LIQUID-GAS TWO-PHASE FLOW AND HEAT TRANSFER

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**Abstract**

Single-component two-phase Mechanically Pumped Loops, Vapour Pressure Driven Loops, Capillary Pumped Loops and Loop Heat Pipes are crucial for future aerospace-related thermal control applications in various g-environments. The latter range from micro-g, via reduced-g for Mars and Moon bases and 1-g during terrestrial testing to hyper-g in rotating spacecraft, during combat aircraft manoeuvres and in systems for outer planets. In several sections of single-component two-phase systems, the fluid is a mixture of the working and saturated vapour. Results of two-phase two-component flow and heat transfer research (pertaining to liquid-gas mixtures, e.g. water/air) are often applied to support two-phase single-component flow and heat transfer research. Different aerospace-related two-phase flow and heat transfer research issues are discussed (pressure drop and flow pattern issues, flashing, thermal-gravitational scaling) to assess the differences between single- and two-component flow and heat transfer. Though these look physically similar and close, they are essentially different. The usually not negligible differences illustrate that one must be very careful to use two-phase two-component data to develop single-component two-phase thermal control systems for aerospace applications.

**KEYWORDS**

Thermal control, Thermal management, Two-phase flow, Two-phase heat transfer, Thermal-gravitational modelling & scaling, Micro-g, Hyper-g, Reduced gravity, Similitude, Similarity considerations, Dimensionless numbers, Pressure drop, Gas-liquid mixtures, Liquid-vapour mixtures, Flow patterns, Aerospace applications.

## TWO-PHASE THERMAL CONTROL FOR SPACECRAFT APPLICATIONS

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.

The discussion starts with the background of the research, followed by a short general description of two-phase flow and heat transfer phenomena. The impact of the gravity level will be assessed. The discussions include development supporting theoretical work on thermal/gravitational scaling of two-phase flow and heat transfer in two-phase thermal control loops (including gravity level dependent two-phase flow pattern mapping and condensation), in-orbit technology demonstration experiments and also some current R&D. They constitute the elements to assess the fundamental differences between single-component (liquid-vapour) and two-component (liquid-gas) two-phase flow and heat transfer.

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 other 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. As compared to the parallel concept, the series concept

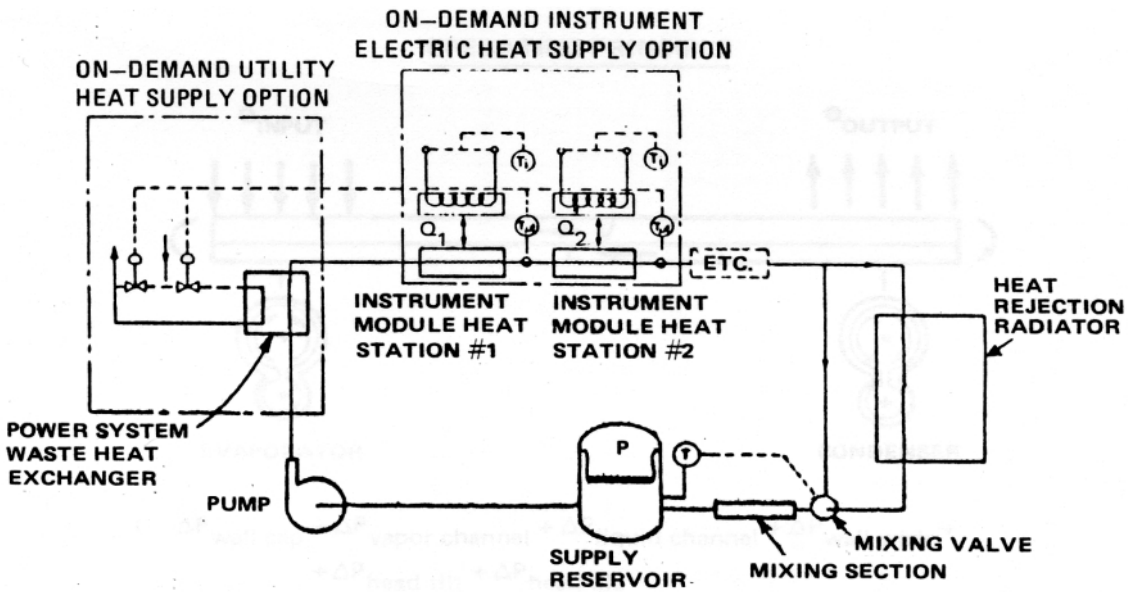


Figure 1. Schematic of a mechanically pumped two-phase thermal bus series configuration [1].

(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 the modular approach, in which 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

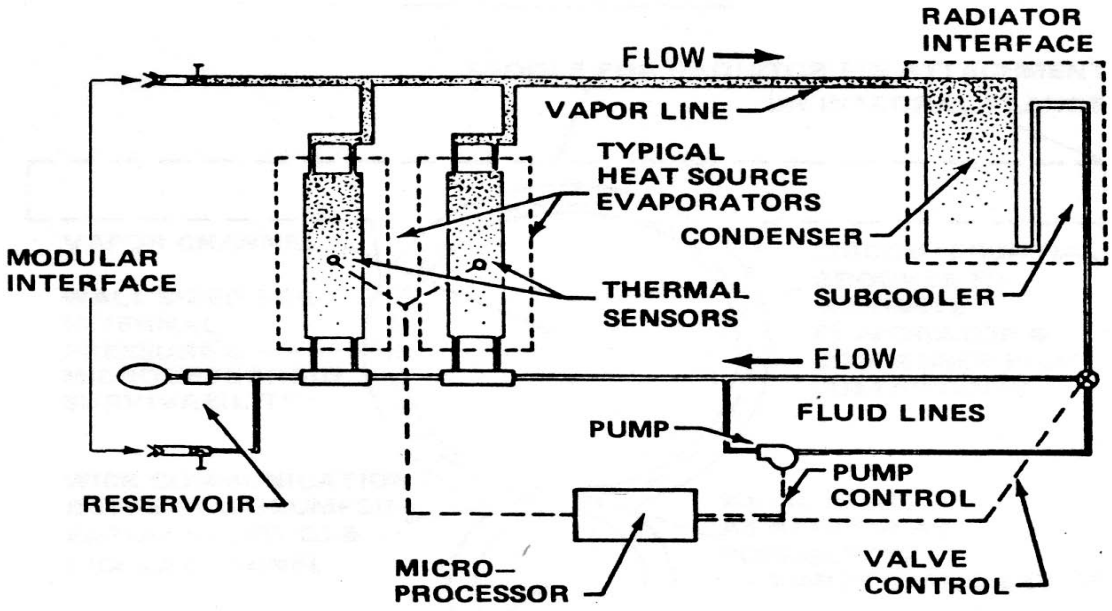


Figure 2. Schematic of mechanically pumped two-phase thermal bus, parallel configuration [4].

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.

Most important issues in developing two-phase thermal busses were formulated in the early 80'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.

Two important European future mechanically pumped two-phase heat transport applications are:

- 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 [13-18]. AMS-2, the Alpha Magnetic Spectrometer experiment [19] 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.

Concerning this Tracker Thermal Control System (TTCS) 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 $kW/m^2.K$ [20]. 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 [20]. Data from experiments with $CO_2$ in small diameter tubes confirm this [22]. 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 [23]. Ethane is an exception: This issue (called flashing) will be discussed later, since it one of the crucial differences between single- and two-component two-phase flow.
- A dedicated hybrid two-phase loop will guarantee the required isothermality and quality range.

Alternatives for mechanically pumped systems are capillary pumped systems, using surface tension driven pumping of capillary evaporators, to transport (like in a heat pipe) the condensate back from condenser to evaporator. Such capillary two-phase systems can be used in spacecraft not allowing vibrations induced by mechanical pumping. Ammonia is the best working fluid for capillary-pumped two-phase loops also. Two systems can be distinguished: the western-heritage Capillary Pumped Loop CPL [24] and the Russian-heritage Loop Heat Pipe LHP [25]. Active set point temperature control of any two-phase loop can be realised by control of the temperature of the reservoir or the compensation chamber. This influences their liquid contents, hence the amount of liquid in the rest of the loop and consequently the condenser flooding, hence the condenser area available for condensation. In this way the loop set point can be maintained independent of variations in heat load (power to be transported) or in heat sink (radiator temperature). CPLs and LHPs offer design flexibility, several performance advantages and unique operational characteristics [9], substantial heater power savings by their diode function, very tight temperature control (tenths of °C), and large area isothermalisation (tenths of $dm^2$).

Since two-phase flow and heat transfer is essentially different in earth gravity, lunar gravity, Mars gravity and micro-gravity, the two-phase heat transport system technology is to be demonstrated in space. Therefore several in-orbit experiments were carried out. Examples are: ESA's Two-Phase eXperiment TPX I [7], NASA's CApillary Pumped Loop experiments CAPL 1&2&3 [26-28], the Loop Heat Pipe Flight eXperiment LHPFX [8], the Cryogenic Capillary Pumped Loop CCLP [29], the Two-Phase Flow experiment TPF [30], and COM2PLEX [30]. Other experiments are planned for future flights: A re-flight of TPF with a miniature LHP [32], as piggy bag.

Because of their merits, CPLs are already operating in space. On the Hubble Space Telescope a CPL/cryo-cooler is used to adjust a sensor temperature for maximum sensitivity, a CPL removes waste cryo-cooler heat to cool the Near IR Camera Multi-Object Spectrometer. Three CPL's serve the SWIR, TIR and MOPPIT instruments on TERRA, the Earth Observation Satellite EOS-AM. LHP's are on or planned for future spacecraft missions, not only for low-orbit satellites or geo-synchronous, but also for missions to planets [25]. Examples are the US spacecraft [9] COMET, GLAS (the LHP GLAS Laser Thermal Control System), SWIFT (two LHP's, one at either side of the Gamma Ray Detector Array Plate of the Burst Array Telescope) and AURA (better packaging of NASA/JPL's TES instrument by five LHP's). There are also the Hughes 702 and other commercial geo-synchronous communication satellites (where LHPs permit deployable radiators to accommodate high power spacecraft in smaller faring), the European earth observation spacecraft ATLID [33] and the Russian spacecraft OBZOR.

Emerging two-phase technologies have to meet severe future requirements [9] for:

- The dimensional stability of very large deployable structures, e.g. a two-phase isothermaliser application for 10-km resolution soil moisture measurements from space.
- The need for cold telescope optics at IR and longer wavelengths, applying Cryogenic CPL's/ LHP's, with a transport length of say 1 metre, to reduce the need for cryo-coolers in cryogenic optical telescope assemblies. Typical figures are 1 to 5 W at 30 to 38 K, using neon [34-36], 0.1 to say 3 W at 18 to 25 K (using hydrogen), and tenths of mW at 3 K (using helium).
- Increasingly integrated designs, such as the Next Generation Space Telescope and the Fourier-Kelvin Stellar Interferometer.
- Tighter temperature control, higher heat fluxes (e.g. from lasers) and diode function.
- Fleets of micro- and nano-spacecraft: Miniaturised CPL's/LHP's [32, 36-38].

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.

## TWO-PHASE FLOW & HEAT TRANSFER

Two-phase flow is the simplest case of multiphase flow, the latter being the simultaneous flow of different phases (states of matter): gas, liquid and solid. The nature of two-phase flow in spacecraft thermal control systems is single-component, meaning that the vapour and the liquid phase are of the same chemical substance. If the phases consist of different chemical substances, e.g. in air-water flow, the flow is called two-phase two-component flow. Flow-related (hydraulic) two-phase, single-component and two-component flows are described by the same mathematical model equations. Therefore results of calculations and experiments in one system can be used in the other, as long as they pertain to flow phenomena only, consequently there is no heat transfer.

Heat transfer in a two-phase two-component system has a relatively simple impact on the system behaviour: only the physical (material) properties of the phases are temperature dependent. Two-phase single-component systems are far more complicated, because the heat transfer and the temperature cause (in addition to changes of the physical properties of the phases) mass exchanges between the phases, by evaporation, flashing and condensation. Hence complicated two-phase single-component systems can not be properly understood by using modelling and experimental results of simpler two-phase two-component systems, as it will be discussed later in this tutorial. Two-phase single-component systems, like the liquid-vapour systems in spacecraft thermal control loops, require their own, very complicated mathematical modelling and dedicated two-phase single-component experiments. Though liquid-vapour flows obey all basic fluid mechanics laws, their constitutive equations are more numerous and more complicated than the equations for single-phase flows. The complications are due to the impact of surface tension and the fact that inertia, viscosity and buoyancy effects can be attributed both to the liquid phase and vapour phase.

## FLOW PATTERN ISSUES

An extra, major, complication is the spatial distribution of liquid and vapour, the so-called flow pattern. The figures 3a and 3b schematically show the various flow patterns and boiling mechanisms for up-flow in a radially heated vertical tube evaporator and for flow in a horizontal evaporator. The entering pure liquid gradually changes to the exiting pure vapour flow, via the main (morphological) patterns for bubbly, slug, annular and mist (or drop) flow. The hybrid flow patterns, bubbly-slug, slug-annular (churn), and annular-wavy-mist, can be considered as transitions between main patterns.
Figure 4 shows the flow patterns for horizontal condensing flow in a gravity field. It is clear that each flow pattern (regime) requires its own mathematical modelling. Also transitions from one pattern to another are to be modelled. Within each regime, further modelling refinement can be based on extra criteria: Relative magnitudes of the various forces or the difference between laminar and turbulent flow.

Various textbooks on two-phase flow and heat transfer [20, 39] discuss the constitutive equations for the various flow patterns, focusing on one-dimensional liquid-vapour (or gas) flow. Such 1-D

models, especially for homogeneous (bubbly and mist) flow, and slug and annular vertical downward flow in lines of circular cross section, are relevant for aerospace, as non-terrestrial gravity levels in space environments often are circular-symmetric. From the dimensionless equations, one can identify dimensionless numbers (groups of fluid properties and dimensions), that determine two-phase flow and heat transfer. Such numbers are very useful for similarity considerations in thermal-gravitational scaling exercises and for the creation of various flow pattern maps, like the maps in the figures 5 to 7.

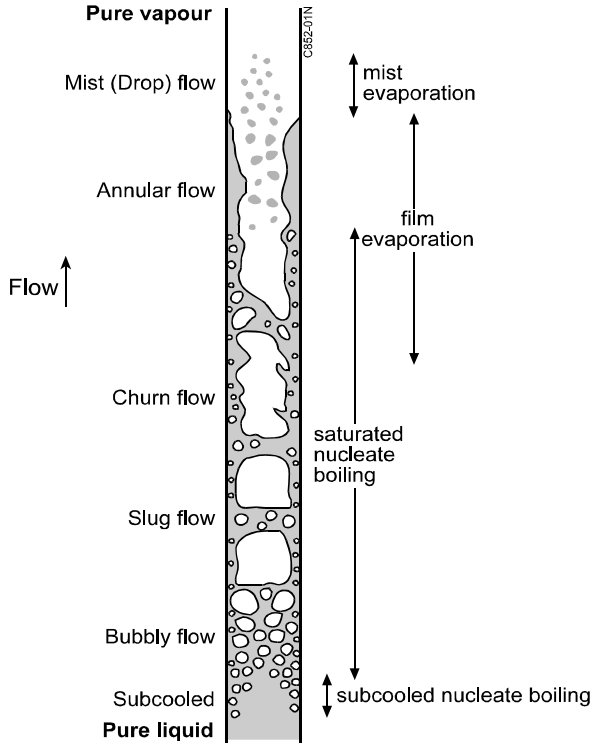


Figure 3a. Flow patterns/boiling mechanisms, up-flow in vertical tube in 1-g.

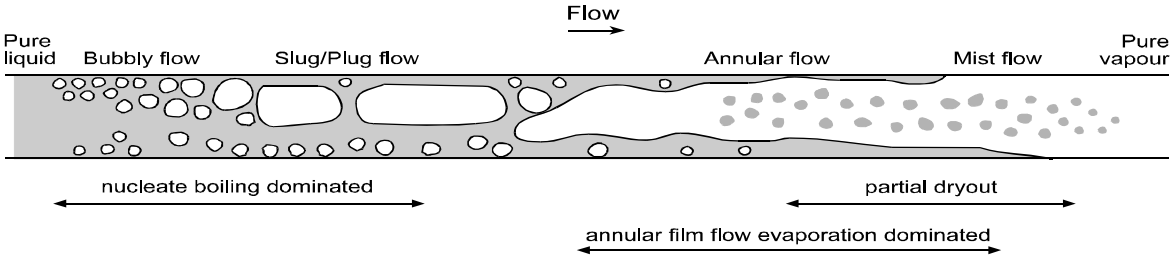


Figure 3b. Flow patterns for horizontal evaporation in a gravity field.

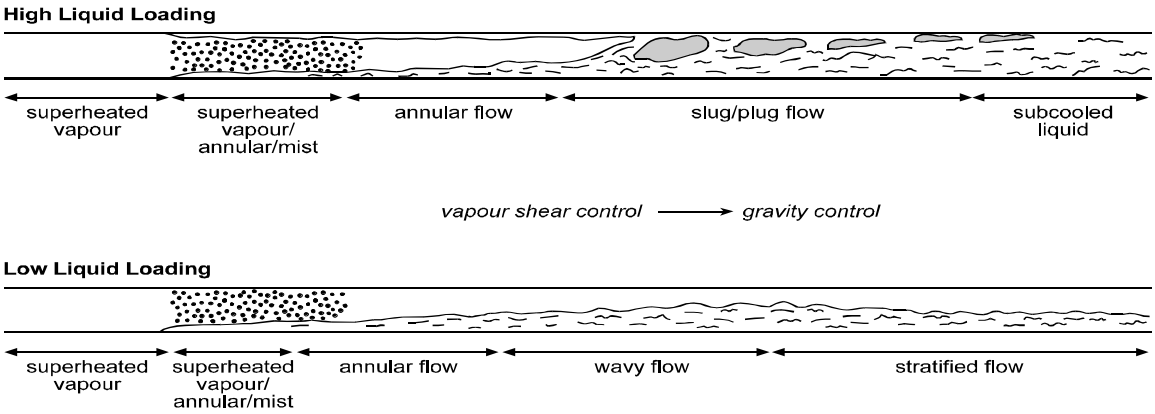


Figure. 4. Flow patterns for horizontal condensation on earth.

Alternatively one derives dimensionless numbers by dimension analysis or similitude in engineering, discussed in specialised textbooks [40]. Figure 5 [20] depicts a boiling trajectory crossing the different flow regimes. Figure 6 [41] shows a map in a normalised form. By combining cross-

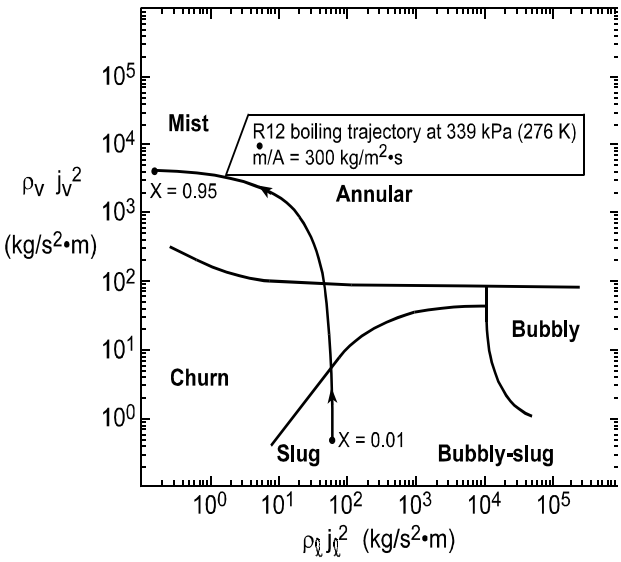


Figure 5. Vertical flow pattern map.

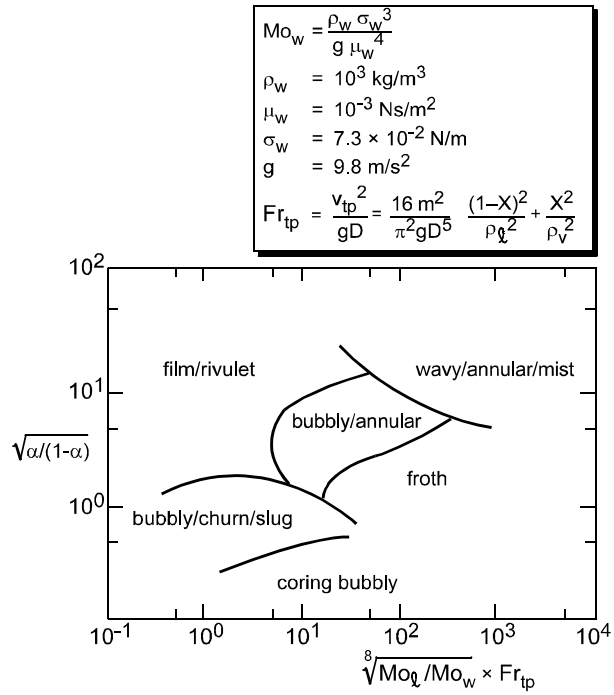


Figure 6. Vertical co-current flow pattern map.

sections of figures like 7 [42] one can create flow pattern maps for a chosen g-level. A comparison of the latter two maps and maps produced during the Cyrene experiment [43], and TPX I [7] indicate that they partly contradict each other. A comparison between the figures suggests that the transition to annular flow occurred in these three systems more or less at the same $j_v$-value 0.2-0.25 m/s, but at different $j_l$-values. This can be due to either by different working fluids used (R12/ammonia/ammonia) or the different inner line diameter (10.5 mm/4.7 mm/4.93 mm). More data are to be gathered to draw final conclusions on this.

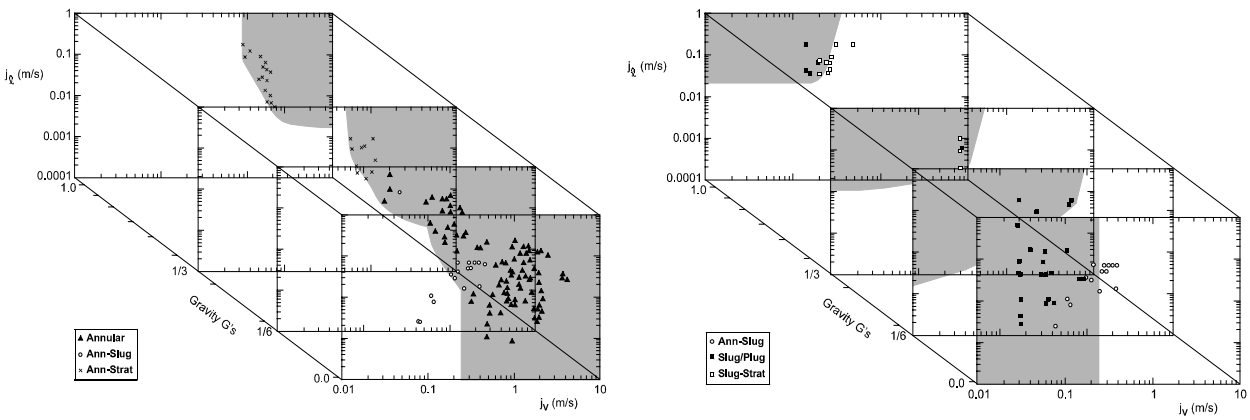


Figure 7. Gravity dependent 3-D maps: Annular flow (left), slug/plug flow (right).

## THERMAL-GRAVITATIONAL MODELLING & SCALING ISSUES

Development supporting, theoretical research activities, like thermal-gravitational modelling and scaling of two-phase heat transport systems [44-47] are being done not only to reduce costs but also:

- To better understand the impact of gravitation level on two-phase flow and heat transfer phenomena, provide means for comparison and generalisation of data.
- To develop tools to design space-oriented two-phase loops (components), based on terrestrial tests.

The main goal of the scaling of space-related two-phase heat transport systems is to develop reliable spacecraft systems, whose reduced gravity performance can be predicted via results of experiments with scale models on earth. Scaling spacecraft systems proved to be useful also:

- For in-orbit technology demonstration, e.g. the performance of spacecraft heat transport systems can be predicted based on the outcomes of in-orbit experiments on model systems with reduced geometry or different working fluid.
- To define in-orbit experiments to isolate phenomena to be investigated, (e.g. excluding gravity-induced disturbing buoyancy effects on alloy melting, diffusion and crystal growth), for a better understanding of the phenomena. The magnitude of the gravitational scaling varies with the objectives from 1 g to $10^{-6}$ g, to reduced g (0.16 g for Moon base, 0.4 g for Mars base systems), and to super-g values, pertaining to larger planets or rotating spacecraft.

Similarity considerations [44-47] identified 18 dimensionless numbers (π-numbers) relevant for thermal gravitational scaling of mechanically/capillary pumped two-phase loops. The 18 π-numbers are listed in the first column of Table 1. There is perfect similitude between model and prototype if all dimensionless numbers are identical in prototype and model: Only then scaling is perfect. It is clear that perfect scaling is impossible for two-phase flow and heat transfer, as the phenomena are too complex, the number of crucial parameters/π-numbers is too large. Fortunately imperfect (distorted) scaling can give useful results [40]. Therefore a careful estimation of the relative magnitudes of the different effects is required. Effects considered unimportant for the identity requirement set some π-numbers superfluous for the particular problem.

Table 1. Relevance of π-numbers for thermal-gravitational scaling of two-phase loops.

| | Liquid Parts | | Evaporator Swirl & Capillary | Vapour/ and Two-Phase Lines | Condenser |
|---|---|---|---|---|---|
| | Adiabatic | Heating/Cooling | | | |
| $\pi_1 = D/L$ = geometry | • | • | • | • | • |
| $\pi_2 = Re_l = (\rho v D/\mu)_l$ , inertia/viscous | • | • | • | • | • |
| $\pi_3 = Fr_l = (v^2/gD)_l$ , inertia/gravity | • | • | • | /• | • |
| $\pi_4 = Eu_l = (\Delta p/\rho v^2)_l$ , pressure head/inertia | • | • | • | • | • |
| $\pi_5 = \cos \nu$ = orientation with respect to g | • | • | • | /• | • |
| $\pi_6 = S$ = slip factor = $v_v/v_l$ | | | • | • | • |
| $\pi_7$ = density ratio = $\rho_v/\rho_l$ | | | • | • | • |
| $\pi_8$ = viscosity ratio = $\mu_v/\mu_l$ | | | • | • | • |
| $\pi_9 = We_l = (\rho v^2 D/\sigma)_l$ , inertia/surface tension | | | • | /• | • |
| $\pi_{10} = Pr_l = (\mu Cp/k)_l$ | | • | • | | • |
| $\pi_{11} = Nu_l = (hD/k)_l$ , convective/conductive | | • | • | | • |
| $\pi_{12} = k_v/k_l$ = thermal conductivity ratio | | | • | | • |
| $\pi_{13} = Cp_v/Cp_l$ = specific heat ratio | | | • | | • |
| $\pi_{14} = \Delta H/h_{lv}$ = enthalpy number = X = quality | | • | • | • | • |
| $\pi_{15} = Mo_l = (\rho_l \sigma^3/\mu_l^4\, g)$, capillarity/buoyancy | | | • | /• | • |
| $\pi_{16} = Ma = v/(\partial p/\partial \rho)_s^{1/2}$ | | | • | • | • |
| $\pi_{17} = (h/k_l)(\mu_l^2 g)^{1/3}$ | | | • | | • |
| $\pi_{18} = L^3 \rho_l^2\, g\, h_{lv}/k_l \mu_l (T-T_o)$ | | | • | | • |

A first step in a practical approach to scale two-phase heat transport systems is identification of important phenomena, to obtain π-numbers for which identity in prototype and model must be required to realise perfect scaling according to the Buckingham pi theorem (crucial in similarity considerations). Distortion will be permitted for π-numbers pertaining to less important phenomena. Important phenomena and the relevant π-numbers will be different in different parts of a system. The relevance of the π-numbers in the various loop sections is indicated by • in the table (π-numbers for thermal gravitational scaling of two-phase loops), given earlier in this section. The best scaling approach is to choose combinations of π-numbers that suit the problem under investigation.

Referring to detailed discussions [44-47] it is remarked that, considering only the identity of Mo and of We/Fr for prototype and scale model, important conclusions can be drawn from the figures 8 and 9, showing the temperature dependence of $g.Mo_l = \rho_l.\sigma^3/\mu_l^4$ and of $(\sigma/\rho_l)^{½} = D.g^{½}/(We/Fr)^{½}$ :

- First, scaling at the same gravity level means a fixed $gMo = \rho_l\sigma^3/\mu_l^4$-value for prototype and model. Figure 8 shows that the value $\rho_l\sigma^3/\mu_l^4 = 2*10^{12}$ m/s$^2$ can be realised by 115°C ammonia, 115°C methanol, 35°C water, 180°C propanol, 235°C propanol, 250°C thermex and 350°C thermex. The length scale ratios follow from reading with these temperatures corresponding $(\sigma/\rho_l)$-values in figure 9, being 2.5 / 4.5 / 8.4 / 4.2 / 3.0 / 5.0 / 3.6.

- Second, figure 8 also shows that scaling a high-pressure (say 110 °C) ammonia system can be done by a low-pressure (say -50 °C) ammonia system, which might be attractive for safety reasons or will to reduce the impact of earth gravity in vertical two-phase sections. It follows from figure 9 that the geometric scaling ratio between high-pressure prototype and low-pressure model (both characterised by $\rho_l \sigma^3/\mu_l^4 = 2.10^{12}$ m/s$^2$) is about 0.4.
- Third, figure 8 shows also that scaling with respect to gravity is restricted to maximal two decades, if the fluid in prototype and model is the same.
  Fourth, the figures 8 and 9 illustrate also that "fluid to fluid" scaling is far more interesting. A very attractive possibility is the scaling of a two-phase prototype for a Mars or a Moon base, by a terrestrial model with the same or a scaled working fluid. As the ratio of gravity levels between prototype and model is not far from 1 (Mars 0.4, Moon 0.16), the sizes of the model have to be only slightly larger than the geometric sizes of the prototype. Adjustment of the inclinations υ of non-horizontal lines in the terrestrial model leads to almost perfect scaling.
- Fifth, also the consequences for the scaling of a super-gravity prototype system by a 1-g model straightforwardly follow from the figures 8 and 9. This is illustrated by the following example [48]. For an ammonia prototype system P, intended for operation around 320 K in a 10-g environment, figure 8 reads a value (10g x.$Mo_P$) of about 1.5 $10^{13}$, since g is about 10 m/s$^2$. As proper scaling requires that the Morton number in prototype and model are to be identical, the ordinate for the 1-g model becomes 10x1.5 $10^{11}$ = 1.5 $10^{12}$. The latter value corresponds to acetone at say 310 K. As for proper scaling Eö = (We/Fr) in prototype and model are to be the same, one obtains the relation $(D_m/D_p)^2 = (g_P/g_m)\,(\sigma/\rho_l)_P/(\sigma/\rho_l)_m$. Figure 9 yields the geometric scaling factor by inserting the g-ratio (10) and the ordinate values corresponding to ammonia at 320 K (0.0055) and acetone at 35°C (0.0053). The result is a geometric scaling factor $D_M/D_P$ around 3.2, maybe too large for novel pulsating/oscillating devices (as these have to fulfil an additional capillary criterion, as it was elucidated earlier in this report), but not unrealistic or impossible for two-phase loops. Similar considerations for water (at 310 K) as the model fluid yield a $D_M/D_P$ somewhat less than 2, which is ideal for scaling two-phase systems, both loops and pulsating/oscillating ones.
- As it will be discussed later on: Many things will be different in "g-assist & anti-g" conditions.

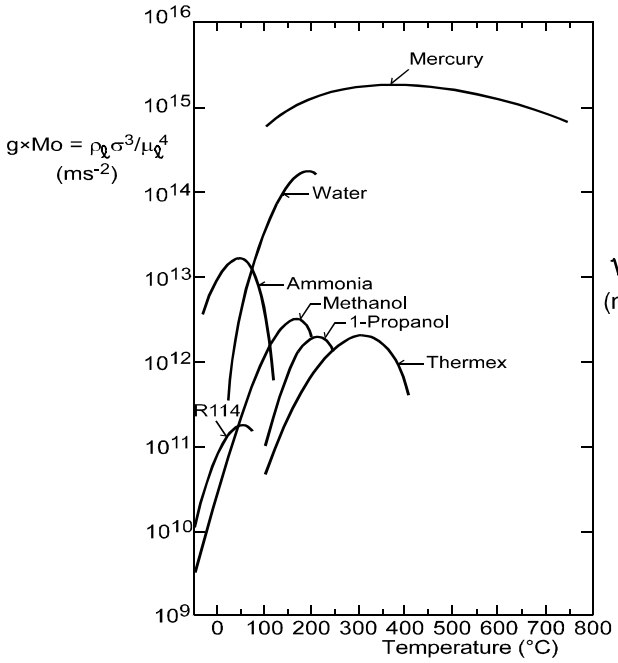


Figure 8. $\rho_l.\sigma^3/\mu_l^4$ versus temperature for various fluids.

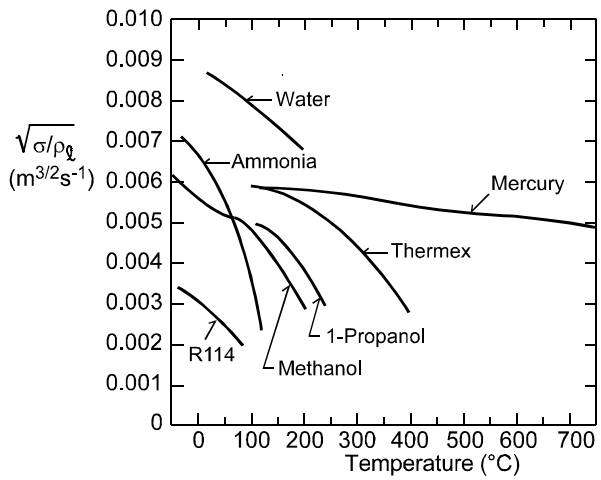


Figure 9. $(\sigma/\rho_l)^{1/2} = D.g^{1/2}/(We/Fr)^{1/2}$ versus temperature for various fluids.

Finally it is remarked that though it took almost a decade for the thermal-gravitational scaling approach described to be accepted and followed by other aerospace researchers [49]. One of the drivers for this is it usefulness for interpretation of experimental data recorded during low-g aircraft flights The latter data concern not only $10^{-2}$-g flight conditions, but also data obtained during flights with trajectories delivering extended periods (of the order of 10 seconds) lunar or Martian gravity.

## PRESSURE DROP ISSUES

An important quantity (to be measured during two-phase flow experiments) is the pressure drop in adiabatic sections and in condensers: sections, being considered crucial for two-phase system modelling and scaling. The equations for annular flow pressure gradients in straight tube condensers and adiabatic lines, extensively discussed in literature [23, 44-48] are based on an elaborate journal article [50],). The total local (local position z-dependent) pressure gradient for annular flow is the sum of friction, momentum and gravity gradients. The equations to be solved are:

$$(dp/dz)_t = (dp/dz)_f + (dp/dz)_m + (dp/dz)_g \tag{1}$$

$$(dp/dz)_f = -(32m^2/\pi^2\rho_v D^5)(0.045/Re_v^{0.2})[X^{1.8}+5.7(\mu_l/\mu_v)^{0.0523}(1-X)^{0.47}X^{1.33}(\rho_v/\rho_l)^{0.26} + 8.1(\mu_l/\mu_v)^{0.105}(1-X)^{0.94}X^{0.86}(\rho_v/\rho_l)^{0.52}] \tag{1a}$$

(X is local quality X(z), Reynolds number $Re_v = 4\dot{m}/\pi D\mu_v$, $\beta$=2 (laminar), 1.25 (turbulent liquid flow).

$$(dp/dz)_m = -(32\dot{m}^2/\pi^2\rho_v D^5)(D/2).(dX/dz)[2(1-X)(\rho_v/\rho_l)^{2/3} + 2(2X-3+1/X)(\rho_v/\rho_l)^{4/3} + +(2X-1-\beta X)(\rho_v/\rho_l)^{1/3} + (2\beta-\beta X-\beta/X)(\rho_v/\rho_l)^{5/3} + 2(1-X-\beta+\beta X)(\rho_v/\rho_l)] \tag{1b}$$

$$(dp/dz)_g = (32\dot{m}^2/\pi^2\rho_v D^5)\{1-[1+(\rho_v/\rho_l)^{2/3}(1-X)/X]^{-1}\}[\pi^2 D^5 g\cos\nu(\rho_l-\rho_v)\rho_v/32\dot{m}^2] \tag{1c}$$

$$(1-\alpha)/\alpha = S(\rho_v/\rho_l)X/(1-X) \tag{2}$$

$$S = (\rho_l/\rho_v)^{1/3} \tag{3}$$

$$\dot{m}\, h_{lv}(dX/dz) = -h\pi D[T(z)-T_s] \tag{4}$$

$$h = 0.018(\lambda_l\rho_l^{1/2}/\mu_l)Pr_l^{0.65}|-(dp/dz)_t|^{1/2}D^{1/2} + R(4\lambda_l/D)\ln[1+(\rho_v/\rho_l)^{2/3}(1-X)/X] \quad 0<R<1. \tag{5}$$

$$\Delta p_t = \int_0^{L_c}(dp/dz)_t.dz\,. \tag{6}$$

$$F(dX/dz, X) = 0. \tag{7}$$

Results of calculations for ammonia are shown in the figures 10 and 11. Figure 11 shows that at 25 °C the gravity constituent overrules the sum of the two other constituents at vapour qualities below 0.8. At -25 °C this overruling holds for vapour qualities below say 0.4. This confirms the scaling statement, that room temperature low-gravity behaviour can be simulated by terrestrial tests at far lower temperatures. Figure 10 depicts the results of calculations of the vapour quality along the duct for three gravity levels (0, Earth and 2-g) and three duct diameters (8.05, 16.1, 24.15 mm) at 300 K, gave the ratio of duct lengths $L_c$(m) needed for condensation under zero-g and one-g respectively [23, 46]. The ratio between full condensation lengths in zero-g and on Earth ranges from roughly 1.5 for the 8.05 mm duct, via 11 for the 16.1 mm duct, up to more than 30 for the 24.15 mm duct. In other words, small line diameter systems are less sensitive to differences in gravity level as compared to larger diameter systems. This is confirmed by flight data of TPX I [7]. However, it must be remarked that, since the model developed is valid for annular flow, it is worthwhile to investigate the impact of other flow patterns inside an evaporator or a condenser duct: Mist flow at high quality, slug and bubbly flow at low quality and wavy-annular-mist in between. It is to be investigated if an annular flow assumption leads towards slightly or substantially overestimated full condensation lengths. To assess the impact of the saturation temperature on condensation, similar curves were calculated for two other temperatures, 243 K and 333 K [23, 46]. The calculations show that the full condensation length increases with the temperature for zero-g conditions, but decreases with temperature for the other gravity levels. This implies that the differences between earth gravity, partial-g and low-g outcomes decrease with decreasing temperature. It confirms that the impact of the gravity level is reduced in low temperature vertical downward flow.

Calculations of the vapour quality distribution along the 16.1 mm reference duct for condensing ammonia (at 300 K) under Earth gravity and 0-g conditions, for power levels ranging from 0.5 kW up to 25 kW, yielded [23, 46] that:

- Under earth gravity conditions, power and full condensation length are strongly interrelated: from $L_c$ = 554 D at 25 kW to only 19 D at 500 W.

- A factor 50 in power, 25 kW down to 500 W, corresponds in a zero gravity environment to a relatively minor reduction in full condensation length, i.e. from 600 D to 400 D (9.5 to 6.5 m).
- The gravity dependence of the full condensation length decreases with increasing power, until differences vanish at 1 MW condenser choking.

But it is repeated once more that the model developed is for annular flow, hence it is worthwhile to investigate the impact of other flow patterns inside the condenser duct: Mist flow at high quality, slug and bubbly flow at low quality and wavy-annular-mist in between.

In summary it is remarked that the curves for a particular non-1g condition (say n-g) can be simply obtained by shifting the curves of the gravity constituents upwards (if n > 1) or downwards (if n < 1). For instance, the curves for 10 g are obtained by shifting the gravity curves a decade upwards, which means that gravity is overwhelming the other constituents up to a quality of say 0.97 (where the flow pattern is homogeneous). When the direction of gravity is reversed, the situation is fully different. Gravity will act against the other constituents. This means fall back of the liquid, which initially leads to a steep increase of the saturation temperature to deliver the vapour pressure needed to maintain transport. A complicating factor is the fact that if the pressure drops substantially increase the equations, derived for nearly isothermal conditions (hence constant fluid properties), no longer hold and have to be replaced by far more complicated ones. The latter is certainly valid for pulsating/oscillating devices, as these essentially need to operate relatively large temperature differences between evaporator and condenser.

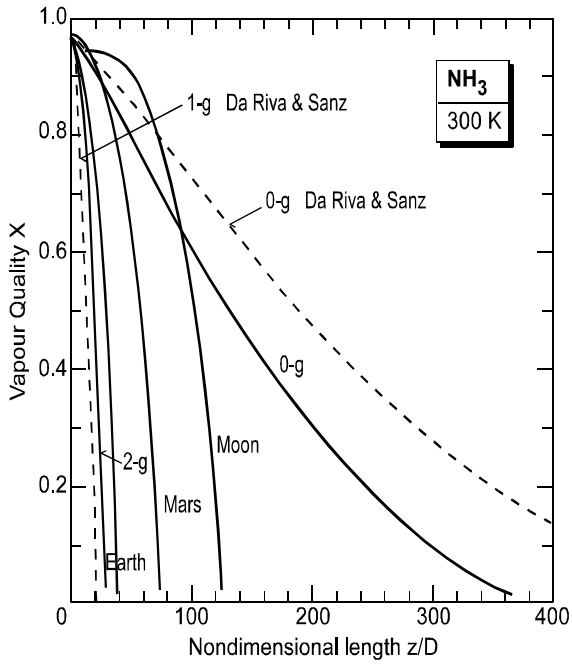


Figure 10. Vapour quality along a reference duct.

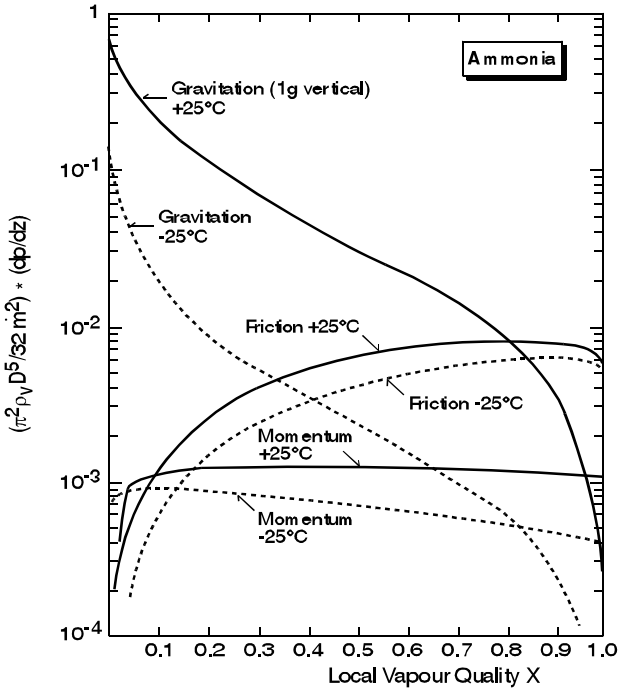


Figure 11. Pressure gradients at –25 and +25 °C.

## FLASHING ISSUES

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

- In case of 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 quality increase. Ethane is an exception, as illustrated by its Mollier chart (Fig. 12). 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 through a line or valve, by writing according to the 1st Law of Thermodynamics for a steady state process[51]:

$$Q = m\,(H_e - H_i) + M + m\,(\Delta E_k + \Delta E_p) \tag{8}$$

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 fluid and surroundings.

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

$$Q/T + S_{gen} = m\,(s_e - s_i) \tag{9}$$

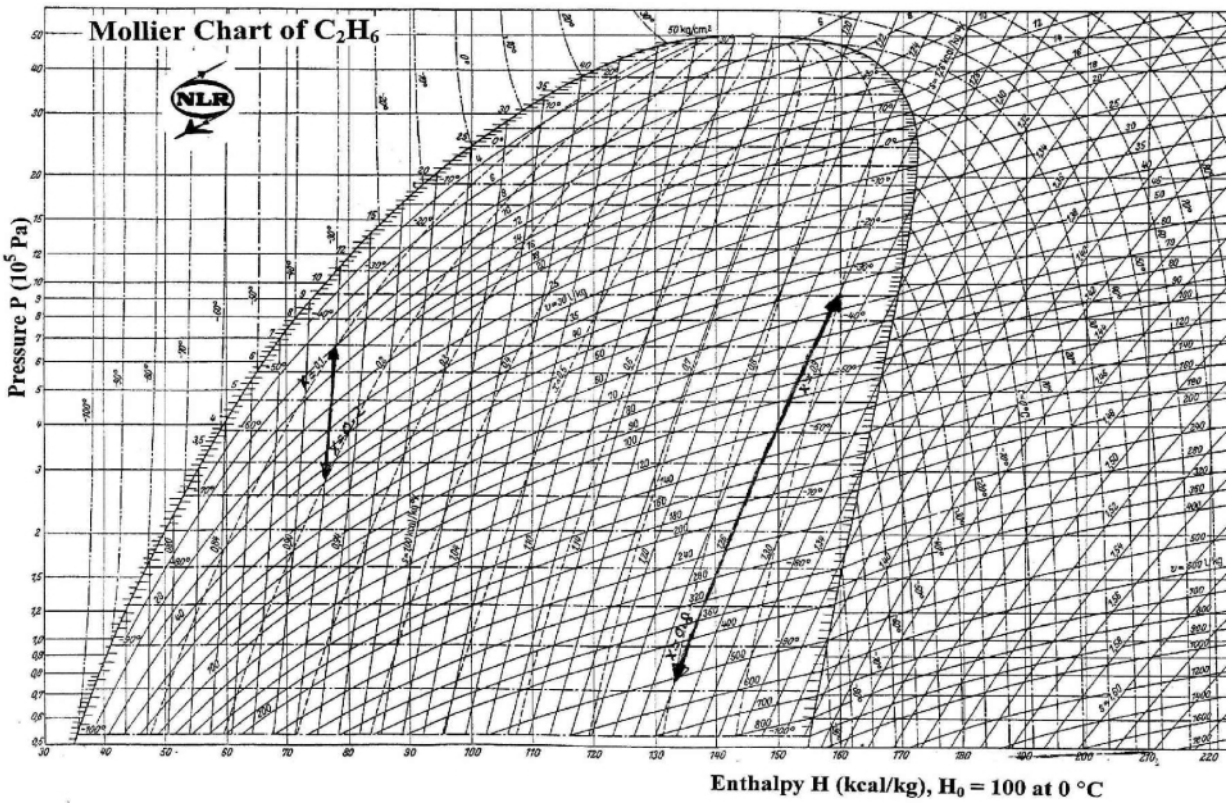


Figure 12. Mollier chart of ethane (1 kcal = 4.17 kJ).

Hence for an adiabatic (Q = 0) real (hence irreversible) process the fluid entropy increases from inlet to outlet. 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.

## CONCLUSIONS ON SINGLE- VERSUS TWO-COMPONENT FLOW

The background of the developments is described. Several critical issues are discussed in detail. The main differences between single-component (liquid and its saturated vapour) and two-component (liquid and gas) flow and heat transfer can be summarised from the preceding text by:

- The heat transfer process in two-component systems is based on caloric heat only, the mechanisms are restricted to conduction and convection. Heat transfer in single-component systems is far more efficient, as the transport is not only by caloric heat but also by the larger contribution of latent heat (evaporation or condensation). The rudimentary sets of constitutive equations and relevant dimensionless numbers of liquid-gas systems do not represent at all single-component two-phase

(liquid-vapour) flow and heat transfer. Consequently liquid-gas models and data are hardly useful foe thermal modelling and thermal-gravitational scaling of (space-related) two-phase thermal control systems.

- If a substantial heat rate is added to liquid-gas flow the mixture temperature will increase in the down-flow direction. In case of heat withdrawal the mixture temperature will decrease. This implies that the various properties of both components will (substantially) change along the flow path. In case of single-component flow the temperature will always (in most cases only slightly) decrease in the down-flow direction, since a small pressure drop needed for the flowing corresponds to a temperature drop of the saturated mixture. This means almost isothermal flow, hence constant fluid properties, except for very long, small diameter lines for which the pressure decreases can be large.
- The effect of flashing, being sometimes certainly not negligible in single-component systems, is completely absent in gas-liquid systems. Shortly said this means that in steady-state adiabatic liquid-gas flow the quality remains constant along the flow path, even in long, small diameter lines with bends, restrictions, etc. This is not true for the corresponding liquid-vapour flow case, for which the vapour quality will increase along the flow path.
- Flow pattern maps, created from two-component experiments, may in a few cases be used for single-component system design. But in most cases these maps are created for properties of the two phases, which considerably differ from the actual single-component case. In other words, one shall be careful to use such maps.

**NOMENCLATURE**

| | |
|---|---|
| A | Area ($m^2$) |
| Bo | boiling number (-) |
| Cp | specific heat at constant pressure (J/kg.K) |
| D | diameter (m) |
| E | energy per mass unit (J/K) |
| Eu | Euler number = $(\Delta p/\rho v^2)$ (-) |
| Fr | Froude number = $v^2/gD$ (-) |
| g | gravitational acceleration ($m/s^2$) |
| H | enthalpy per mass unit (J/kg) |
| h | heat transfer coefficient ($W/m^2.K$) |
| $h_{lv}$ | latent heat of vaporisation (J/kg) |
| j | superficial velocity (m/s) |
| L | length (m) |
| M | mechanical power (W) |
| Ma | Mach number = $v/(\partial p/\partial \rho)_s^{1/2}$ (-) |
| m | mass flow rate (kg/s) |
| Mo | Morton number = $\rho_l \sigma^3/\mu_l^4$ g (-) |
| Nu | Nusselt number = hD/k (-) |
| p | pressure (Pa = $N/m^2$) |
| Pr | Prandtl number = $\mu Cp/k$ (-) |
| Q | (thermal) power (W) |
| Re | Reynolds number = $\rho vD/\mu$ (-) |
| S | slip factor = $v_v/v_l$ (-) or entropy (J/K) |
| s | fluid entropy per mass unit (J/kg.K) |
| T | temperature (K) or (°C = K – 273.15) |
| v | velocity (m/s) |
| We | Weber number = $\rho_l.v^2D/\sigma$ (-) |
| X | vapour quality ($m_v/m_{tot}$) (-) |
| z | axial or vertical co-ordinate (m) |
| $\alpha$ | vapour fraction (volumetric) (-) |
| $\Delta$ | difference, drop (-) |
| $\mu$ | viscosity ($N.s/m^2$) |
| $\sigma$ | surface tension (N/m) |
| $\lambda$ | thermal conductivity (W/m.K) |
| $\pi_{1, etc}$ | dimensionless number (-) |
| $\rho$ | density ($kg/m^3$) |
| $\nu$ | angle (with respect to gravity) (rad)) |

Subscripts: c = condenser, e = exit, i = inlet, k = kinetic, l = liquid, g = gravity, gen = generation rate, m = model, p = potential/prototype, s = constant entropy, v = vapour, t = total tp = two-phase, w = water

**References**


1. Almgren, D.W., et al., The Use of a Thermal Utility with Space Platform-Mounted Instruments, AIAA 81-1114, AIAA 16th Thermophysics Conference, Palo Alto, USA, 1981.
2. Stalmach, D.D., Oren, J.A., 1982, Systems Evaluation of Thermal Bus Concepts, NASA CR-167774.
3. Delil, A.A.M., Some considerations concerning two-phase flow thermal bus systems for spacecraft, NLR-RL-84-028, 1984.
4. Oren, J.A., Study of Thermal Management for Space Platform Applications, NASA CR-165307, 1981.
5. Haslett, B., Space Station Technology 1983, Thermal Control, NASA CP-2293, 1983, pp. 165-185.
6 Delil, A.A.M., Aerospace-Related Fluid Physics, Heat Transfer, and Thermal Control Research at the NLR Space Division," NLR-TP-2001-300, *Keynote, Proc. HEFAT 2002, 1st Int. Conf. on Heat Transfer, Fluid Mechanics and Thermodynamics*, Kruger Park, South Africa, 2002, pp. 75-86.
7. Delil, A.A.M., et al., TPX for In-Orbit Demonstration of Two-Phase Heat Transport Technology - Evaluation of Flight & Postflight Experiment Results, NLR-TP-95192, SAE 95150, 25th Int. Conf.

on Environmental Systems, San Diego, USA, 1995.
8. Bienert, W.B., Baker, C.L., Ducao, A.S, Loop Heat Pipe Flight Experiment, SAE 981580, 28th Int. Conf. on Environmental Systems, Danvers, USA, 1998.
9. Swanson, T., Thermal Control Technology Development at Goddard, *Proc. Int. Two-Phase Thermal Control Technology Workshop*, Mitchellville, USA, 2002.
10. Grigoriev, Y.I., et al., Two-Phase Thermal Control System of International Space Station Alpha Russian Segment, *Proc. Int. Workshop Non- Compression Refrigeration & Cooling*, Odessa, Ukraine, 1999, pp. 73-83.
11. Cykhotsky, V.M., *et al*., Development and Analyses of Control Methods of the International Space Station Alpha Russian Segment Thermal Control System Parameters, *AIP Conference. Proc. 458, Space Technology & Applications Int. Forum*, Albuquerque, USA, 1999, pp. 848-853.
12. Leontiev, A.I., et al., Thermophysics of Two-Phase Flows in Microgravity: Russian-American Research Project, *AIP Conf. Proc. 387, Space Technology & Applications Int. Forum*, Albuquerque, USA, 1997, pp. 541-546.
13. Delil, A.A.M., Woering, A.A., Verlaat, B., AMS-2 Tracker Thermal Control System: Design and Thermal Modelling of the Mechanically Pumped Two-Phase $CO_2$ Loop, NLR-TP-2003-001, AIAA-2003-0345, 41st AIAA Aerospace Sciences Meeting & Exhibition, Reno, USA, 2003.
14. Delil, A.A.M., NLR's Novel Two-Phase Developments for the International Space Station ISS: - AMS-2, a Mechanically Pumped Two-Phase $CO_2$ Cooling Loop for the Alpha Magnetic Spectrometer Tracker Experiment – CIMEX-3, Versatile Two-Phase Loop for the Fluid Science Laboratory, NLR-TP-2002-184, *Proc. 12th Int. Heat Pipe Conf*., Moscow, Russia, 2002, pp. 241-252.
15. Delil, A.A.M., Woering, A.A., Verlaat, B., Development of a Mechanically Pumped Two-Phase $CO_2$ Cooling Loop for the AMS-2 Tracker Experiment, NLR-TP-2002-271, SAE 2002-01-2465, *Proc. 32nd Int. Conf. on Environmental Systems*, San Antonio, USA, 2002.
16. Delil, A.A.M., Woering, A.A., Verlaat, B., Development of a Mechanically Pumped Two-Phase $CO_2$ Cooling Loop for the AMS-2 Tracker Experiment, *Proc. Int. Two-Phase Thermal Control Technology Workshop*, Mitchellville, USA, 2002.
17. Delil, A.A.M., Development of a Mechanically Pumped Two-Phase $CO_2$ Cooling Loop for the AMS-2 Tracker Thermal Control System, *Proc. 13th Annual Spacecraft Thermal Control Technology Workshop*, El Segundo, USA, 2002.
18. Woering, A.A. et al., Thermal Modelling Issues Concerning the Mechanically Pumped Two-Phase $CO_2$ Cooling for the AMS-2 Tracker, NLR-TP-2002-272, SAE 2002-01-2466, *Proc. 32nd Int. Conf. on Environmental Systems*, San Antonio, USA, 2002.
19. Viertel, G.M., Capell, M., The Alpha Magnetic Spectrometer, *Nuclear Instruments & Methods in Physics Research*, Vol. A419, 1998, pp. 295-299.
20. Carey, P. V., *Liquid-Vapor Phase-Change Phenomena*, Hemisphere, Washington DC, USA, 1992.
21. Kandlikar, S.G., A general correlation for saturated two-phase flow boiling heat transfer inside horizontal and vertical tubes, *J. Heat Transfer*, Vol. 112, 1989, pp. 219-228.
22. Pettersen, J., Rieberer, R., Munkejord, S.T., Heat Transfer and Pressure Drop for Flow of Supercritical and Subcritical $CO_2$ in Microchannel Tubes, Norwegian Univ. of Science and Technology/SINTEF Energy Research TR A5127, 2000.
23. Delil, A.A.M., Gravity dependence of pressure drop and heat transfer in straight two-phase heat transport system condenser ducts, NLR TP 92167 U, SAE 921168, 22nd International Conference on Environmental Systems, Seattle, USA, 1992, *SAE Trans., J. Aerospace*, Vol.101, 1992, pp. 512-522.
24. Stenger, F.J., Experimental Study of Water-Filled Capillary Pumped Heat Transfer Loops, NASA TMX-1310, 1966.
25. Maidanik, Yu., Solodovnik, N., Fershtater, Y., Investigation of dynamic and stationary characteristics of loop heat pipe, *Proc. 9th Int. Heat Pipe Conf.*, Albuquerque, USA, 1995, pp. 1000-1006.
26. Butler, D., Overview of CPL & LHP Applications on NASA Missions, *AIP Conference Proc. 458, Space Technology & Applications Int. Forum*, Albuquerque, USA, USA, 1999, pp. 792-799.
27. Butler, D., Ottenstein, L., Ku, J., Flight Testing of the Capillary Pumped Loop Flight Experiments, SAE 951566, 25th Int. Conf. on Environmental Systems, San Diego, USA, 1995.
28. Ottenstein, L., Nienberg, J., Flight Testing of the Two-Phase Flow Flight Experiment, SAE 981816, 28th Int. Conf. on Environmental Systems, Danvers, USA, 1998.
29. Hagood, R., CCPL Flight Experiment: Concepts through Integration, SAE 981694, 28th Int. Conf. on Environmental Systems, Danvers, USA, 1998.

30. Ottenstein, L., Flight Testing of the Capillary Pumped Loop 3 Experiment, *Proc. Int. Two-Phase Thermal Control Technology Workshop*, Mitchellville, USA. 2002.
31. Bodendieck, F., *et al*., Emerging Loop Heat Pipe Applications for Small-Sat, Mars Mission and ISS, SAE-2002-01-2464, *Proc. 32nd Int. Conf. on Environmental Systems*, San Antonio, USA, 2002.
32. Delil, A.A.M., Maidanik, Yu.F, Gerhart, C., Development of Different Novel Loop Heat Pipes within the ISTC-1360 Project, NLR-TP-2003-213, SAE-2003-01-2383, *Proc. 33rd Int. Conf. on Environmental Systems*, Vancouver, Canada, 2003.
33. Dunbar, N.W., ATLID Laser Head Thermal Control – Design and Development of a Two-Phase Heat Transport System for Practical application," SAE 961561, 26th Int. Conf. on Environmental Systems, Monterey, USA, 1996.
34. Kobel, M., Ku , J., Comprehensive Testing of a Neon Cryogenic Capillary Pumped Loop, *Proc. Int. Two-Phase Thermal Control Technology Workshop*, Mitchellville, USA, 2002.
35. Bugby, D., et al., Across Gimbal and Miniaturized Cryogenic Loop Heat Pipes, *Proc. Int. Two-Phase Thermal Control Technology Workshop*, Mitchellville, USA, 2002.
36. Hoang, T., O' Connell, T., Advanced Loop Heat Pipes for Spacecraft Thermal Control, *Proc. Int. Two-Phase Thermal Control Technology Workshop*, Mitchellville, USA, 2002.
37. Delil, A.A.M., Baturkin, V., Miniature loop heat pipe with a flat evaporator- Thermal modelling & Experimental results, NLR-TP-2002-273: Modelling of a Miniature Loop Heat Pipe with a Flat Evaporator, SAE-2002-01-2506, *Proc. 32nd Int. Conf. on Environmental Systems*, San Antonio, USA, 2002, and Experimental Results of Heat Transfer Phenomena in a Miniature Loop Heat Pipe with a Flat Evaporator, *Proc. 12th Int. Heat Pipe Conf.*, Moscow, Russia, 2002, pp. 134-143.
38. Maidanik, Yu.F., Pastukhov, V.G., Cheryshova, M.A., Delil, A.A.M., Developments and Test Results of Multi-Evaporator Condenser Loop Heat Pipe, , *AIP Conference. Proc. 654, Space Technology & Applications Int. Forum*, Albuquerque, USA, 2003, pp. 42-48.
39. Wallis, G.B., *One-dimensional Two-phase Flow*, McGraw-Hill Book Company, New York, 1969.
40. Murphy, G., *Similitude in Engineering*, Ronald Press, New York, USA, 1950.
41. Oshinowo, T., Charles, M.E., Vertical Two-Phase Flow, Flow Pattern Correlations, *Can. J. Chem. Engng*., Vol. 52, 1974, pp. 25-35.
42. Hamme, T.A., Best, F.R., Gravity Dependent Flow Regime Mapping, *AIP Conference Proc. 387, Space Technology & Applications Int. Forum*, Albuquerque, USA, 1997, pp. 635-640.
43. Lebaigue, O., Bouzou, N., Colin, C., Cyrène: An Experimental Two-Phase Ammonia Fluid Loop in Microgravity. Results of a Parabolic Flight Campaign," SAE 981689, 28th Int. Conf. on Environmental Systems, Danvers, USA, 1998.
44. Delil, A.A.M., Thermal Gravitational Modelling and Scaling of Two-Phase Heat Transport Systems: Similarity Considerations and Useful Equations, Predictions Versus Experimental Results, NLR-TP-91477, ESA SP-353, *Proc. 1st European Symp. Fluids in Space*, Ajaccio, France, 1991, pp. 579-599.
45. Delil, A.A.M., Two-Phase Heat Transport Systems for Space: Thermal Gravitational Modelling & Scaling, Similarity Considerations, Equations, Predictions, Experimental Data and Flow Pattern Mapping, NLR TP 98268, SAE 981692, *Proc. 28th Int. Conf. on Environmental Systems*, Danvers, USA, 1998.
46. Delil, A.A.M., Gravity dependent condensation pressure drop and heat transfer in ammonia two-phase heat transport systems, NLR TP 92121 U, AIAA 92-4057, National Heat Transfer Conference, San Diego, CA, USA, 1992.
47. Delil, A.A.M., Two-phase flow and heat transfer in various gravity environments, *Proc. 4th Int. Heat Pipe Symp.*, Tsukuba, Japan, 1994, pp. 223-234, and Survey of Heat Transfer Research Meeting of the Assembly of International Heat Transfer Conferences, ENEA C.R.E. report, Casaccia, Italy, 1992.
48. Delil, A.A.M., Pulsating & Oscillating Heat Transfer Devices in Acceleration Environments Ranging from Micro-Gravity to Super-Gravity, NLR-TP-2001-001, SAE 2001-01-2240, *Proc. 31st Int. Conf. on Environmental Systems*, Orlando, USA, 2001.
49. Miller Hurlbert, K., *Flow Dynamics for Two-Phase Flows in Partial Gravities*, PhD Thesis, University of Houston, USA, 2000.
50. Soliman, M., Schuster, J.R., Berenson, P.J., A General Heat Transfer Correlation for Annular Flow Condensation, *Trans. ASME, J. Heat Transfer*, Vol. 90, 1968, pp. 267-276.
51. Bejan, A., *Advanced Engineering Thermodynamics*, John Wiley & Sons, New York, 1988.