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National Aerospace Laboratory NLR

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Issues of Various Two-Phase Heat Transfer Devices in Gravity Environments Ranging from Micro-Gravity to Super-Gravity

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This report is based on an invited tutorial and two technical papers, presented at the Fundamentals of Two-Phase Flow in Microgravity Session of the Thermophysics in Microgravity Conference, during the Space Technology & Applications International Forum STAIF-2001, in Albuquerque (NM), USA, from 11-15 February 2001, and on an invited keynote lecture presented at the 6th International Heat Pipe Symposium IHPS-2000, in Chiang Mai, Thailand, 5-9 November 2000.

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Summary

This publication, discussing different aspects of various heat transfer devices in gravity environments ranging from micro-gravity to super-gravity, gives an overview of past activities and current state of the art. It consists of three parts and an appendix:

- PART 1. Fundamentals of Gravity Level Dependent Two-Phase Flow and Heat Transfer - A Tutorial.
- PART 2. Thermal-Gravitational Modelling and Scaling of Two-Phase Heat Transport Systems from Micro-Gravity to Super-Gravity Levels.
- PART 3. Modelling and Scaling of Oscillating or Pulsating Heat Transfer Devices Subjected to Earth Gravity and to High Acceleration Levels.
- APPENDIX. Compilation of Slides.

The first three parts are the NLR contributions (an invited tutorial and two technical papers) to the Fundamentals of Two-Phase Flow in Microgravity Session of the Thermophysics in Microgravity Conference, held during the Space Technology & Applications International Forum STAIF-2001, in Albuquerque (NM), USA, from 11-15 February 2001. They are included, as separate papers, in the STAIF-2001 Conference Proceedings published by the American Physical Society (Edited by Mohamed. El-Genk). The appendix contains a compilation of the slides of an invited keynote lecture presented at the 6th International Heat Pipe Symposium IHPS-2000, in Chiang Mai, Thailand, 5-9 November 2000, and the slides of the above STAIF-2001 presentations.

Contents

(63 pages in total)

Part 1.

Fundamentals of Gravity Level Dependent Two-Phase Flow and Heat Transfer - A Tutorial

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Abstract. Multiphase flow, the simultaneous flow of the different phases (states of matter) gas, liquid and solid, strongly depends on the level and direction of gravitation, since these influence the spatial distribution of the phases, having different densities. Many investigations concern behaviour of liquid-solid flows (e.g. in mixing, crystal growing, or materials processing) or gas-solid flows (e.g. in cyclones or combustion equipment). But of major interest for aerospace applications are the more complicated liquid-vapour or liquid-gas flows, being 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 become extremely complicated, because of heat and mass exchange between the phases by evaporation, condensation, and flashing. Though very many publications (textbooks, conference proceedings, journal articles) concern two-phase flow and heat transfer, publications on the impact of reduced gravity are very scarce. This is the main driver for carrying out research in micro-gravity. Various heat and mass transfer issues of two-phase heat transport technology for space applications are discussed, focusing on the most complicated case of liquid-vapour flow with heat and mass exchange. Simpler cases, like adiabatic or isothermal liquid-vapour flow or liquid-gas flow, can be derived from this case, by setting various terms in the constitutive equations equal to zero. The discussions start with the background of the research, followed by a short description of two-phase flow and heat transfer phenomena. The impact of the gravity level will be assessed., including development supporting theoretical work: Thermal/gravitational scaling of two-phase flow and heat transport in two-phase thermal control loops, including gravity level dependent two-phase flow pattern mapping and condensation issues. Outcomes of theoretical work are compared with results of experiments, done on earth and in micro-gravity.

WHY LOOKING AT TWO-PHASE FLOW & HEAT TRANSFER IN MICROGRAVITY?

Thermal control systems for future large spacecraft have to transport large amounts of dissipated power (say hundreds of kW) over large distances (say 100 meters). Conventional single-phase systems (based on the heat capacity of the working fluid) are simple, well understood, easy to test, inexpensive and low risk. However, for proper thermal control with small temperature drops from equipment to radiator (to limit radiator size and mass), they require thick walled, large diameter lines and noisy, heavy, high power pumps, hence large solar arrays. Alternatives for 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 rejection into space. Such systems, relying on the heat of vaporisation, operate nearly isothermally. Consequently pumping power is reduced by orders of magnitude, thus minimising radiator and solar array sizes. Ammonia is the best working fluid. The stations can be in a pure series, pure parallel, or mixed configuration. A very important near-future two-phase heat transport system application is the two-phase thermal control system of the Russian segment of the International Space Station (Grigoriev, 1996; Cykhotsky, 1999; Ungar, 1996). 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 better working fluid for capillary-pumped two-phase loops also. Two systems can be distinguished (Figs. 1 and 2): The western-heritage Capillary Pumped Loop CPL and the Russian-heritage Loop Heat Pipe LHP (Maidanik, 1995). Active loop temperature set-point control can be done by controlling the temperature of the reservoir or of the compensation chamber, thus influencing 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).

Because of performance advantages and unique operational characteristics CPL's and LHP's are planned for several future spacecraft missions, not only low-orbit or geo-synchronous satellites, but also for missions to planets (Butler, 1999). Examples are the American Earth Observation Satellite EOS-AM, the European Atmospheric Lidar earth observation spacecraft ATLID, the French technology demonstration satellite STENTOR, the Russian spacecraft OBZOR, the Hubble Space Telescope retrofit mission, the Hughes 702 satellites, and other commercial geo-synchronous communication satellites.

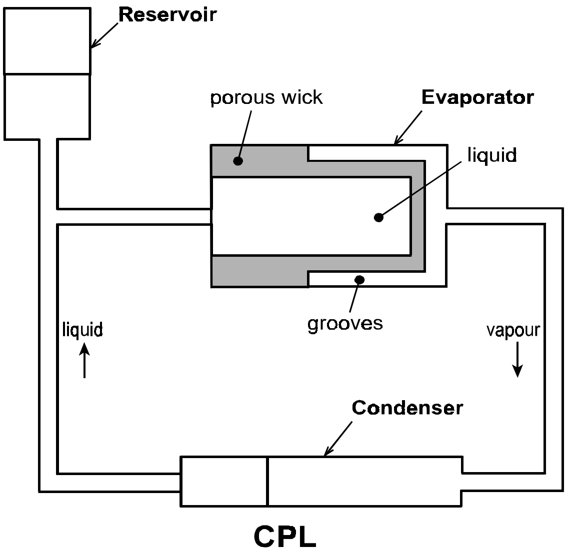

FIGURE. 1 Schematics of Capillary Pumped Loop

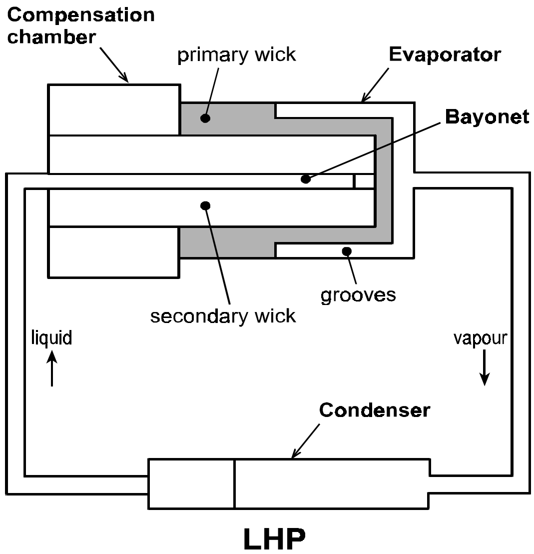

FIGURE. 2 Schematics of Loop Heat Pipe

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 has to be demonstrated in space. Therefore several in-orbit experiments were carried out. The most recent ones are: ESA's Two-Phase eXperiment TPX I (Delil, 1995, 1997), NASA's CApillary Pumped Loop experiments CAPL 1&2 (Butler, 1995), the Loop Heat Pipe Flight eXperiment LHPFX (Bienert, 1998), the all US Loop Heat Pipe with Ammonia ALPHA, the Cryogenic Capillary Pumped Loop CCLP (Hagood, 1998), and the Two-Phase Flow experiment TPF (Ottenstein, 1998). Others were planned and done on other flights: TPX II (Delil, 1997), CAPL 3 (Kim, 1997), STENTOR (Amadieu, 1997), Two-phase flow Extended Evaluation in Micrograviy TEEM (Miller-Hurlbert, 1997), and Granat (Orlov, 1997).

Figure 3 shows a photograph of the TPX I hardware after the successful flight as Get Away Special G557, aboard Space Shuttle STS-60, February 1994. The bottom part consists of the 1.8 kWhr battery, the middle part of Payload Measurement and Control Unit of this self-contained experiment. The top part is the two-phase loop attached to the radiator, being the GAS canister lid. 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 (e.g. Leontiev, 1997). Some experiments were done in drop towers (e.g. Wőlk, 1999) or during Microgravity Science Laboratory missions on the Space Shuttle (Allen, 1999, 1998). Many others were executed during low-gravity aircraft flights (Lebaigue, 1998; Hamme, 1997; Antar, 1996; Fore, 1996; Jayawardena, 1996; Reinarts, 1996, 1995; McQuilen, 1996; Bousman, 1993; Rite, 1994; Miller, 1993; Huckerby, 1992; Crowley, 1991).

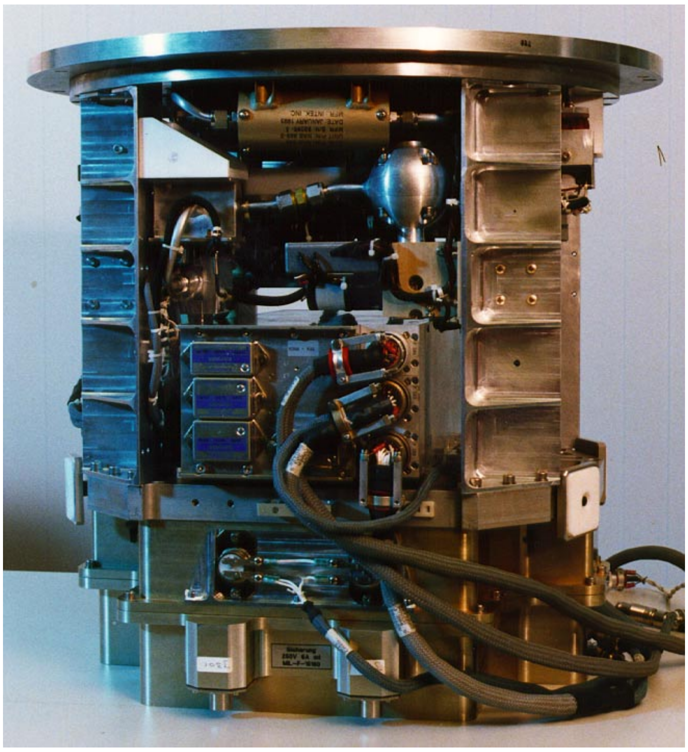

FIGURE 3. TPX I Hardware

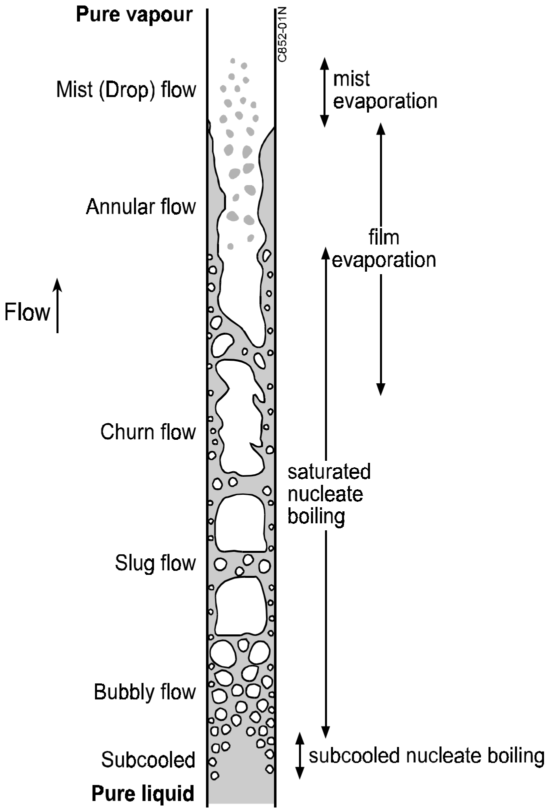

FIGURE. 4 Flow Patterns and Boiling Mechanisms for Up-Flow in a Vertical Line on Earth

WHAT IS TYPICAL IN TWO-PHASE FLOW AND 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 almost the same equations, as long as diffusion due to concentration gradients can be neglected. Results of calculations and experiments in one system can be used in the other, as long as they pertain to flow phenomena only, hence if 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. Consequently, complicated two-phase single-component systems cannot be properly understood by using modelling and experimental results of simpler two-phase two-component systems. 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 complicated than equations for single-phase flow. The complications are due to the fact that inertia, viscosity and buoyancy effects can be attributed to the liquid and to the vapour phase, and also due to surface tension effects. An extra, major complication is the spatial distribution of liquid and vapour: The flow pattern. Figure 3 schematically shows the various flow patterns occurring in a vertical tube evaporator: the entering pure liquid gradually changes to the exiting pure vapour flow, via the main (morphological) patterns for bubbly, slug (or plug), 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. The corresponding behaviour in a horizontal evaporator on earth is depicted in figure 5. Figure 6 gives the patterns in a horizontal condenser tube, for high and low liquid loading. These figures clearly illustrate the stratification induced by gravity, leading to non-symmetric flow patterns. The problem is that each flow pattern (regime) requires its own mathematical modelling. In addition, transitions from one pattern to another are to be modelled also. Within a regime, further refinement of the modelling can be based on additional criteria: The relative magnitudes of the various forces or the difference between laminar and turbulent flow.

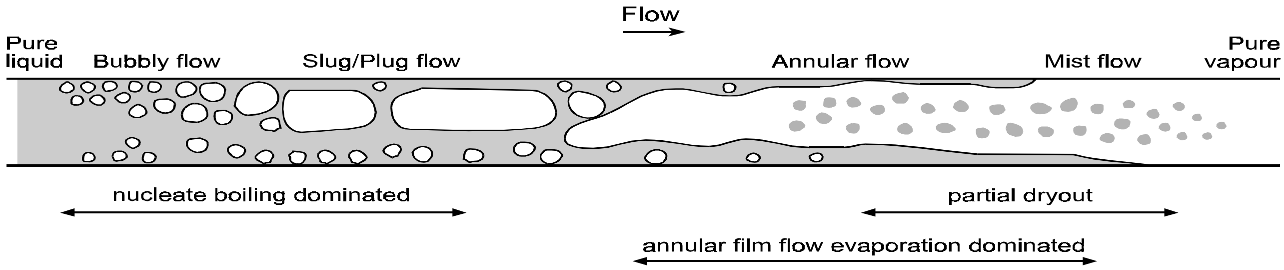

FIGURE 5. Horizontal Evaporator Line on Earth

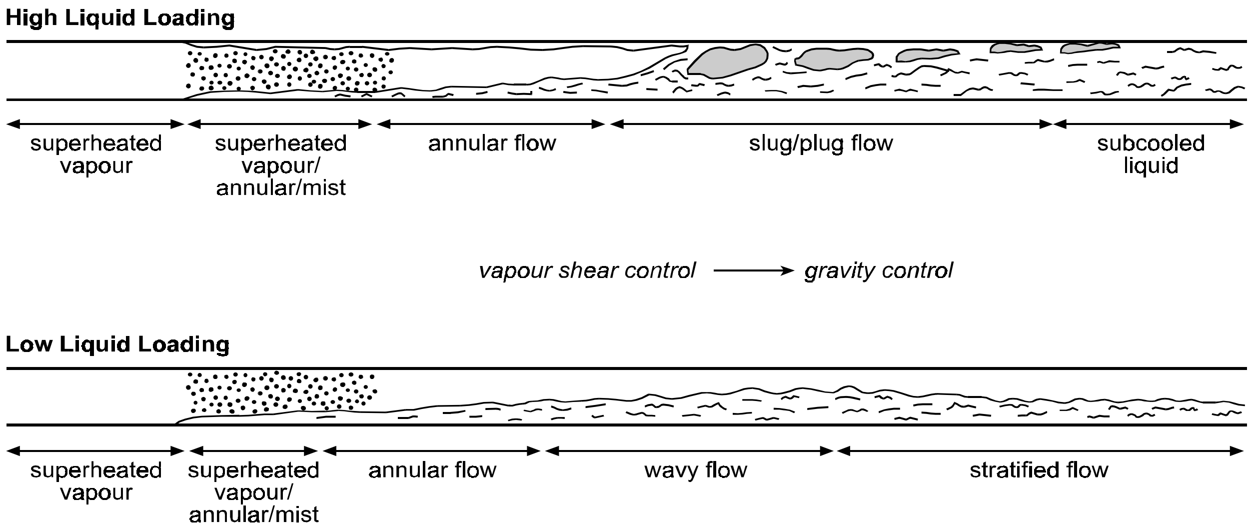

FIGURE 6. Horizontal Condenser Line on Earth

Various text books on two-phase flow and heat transfer (Wallis, 1969; Collier, 1980; Mayinger, 1982; Van Carey, 1992), derive and discuss in detail the constitutive (conservation) equations for the various (main) flow patterns, focusing on one-dimensional liquid-vapour (or gas) flow. Such one-dimensional models, especially those for homogeneous (bubbly and mist) flow, slug and annular vertical downward flow in lines of circular cross section, are relevant for the various aerospace-related two-phase issues (discussed here), as the non-terrestrial gravity levels in various space environments are circular symmetric also. By writing these equations in dimensionless form, 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 flow pattern maps, like the maps in the figures 7 and 8. An alternative way to derive these dimensionless numbers is by dimension analysis, a useful baseline for similitude in engineering approaches, discussed in special textbooks (e.g. Murphy, 1950). It is remarked that the discussions here will be based on dimension-analytical considerations, assuming lines having circular cross-section, the problem is circle-symmetric, one-dimensional. The homogeneous flow model is based on homogeneous mixture properties and on zero slip between the phases (equal velocities of both phases). The annular flow model, considering the two phases to move separately with different velocities, is valid in the adiabatic two-phase thermal control system lines, in almost the full condenser length, and also - in case of (swirl) tube evaporators - in evaporator lines.

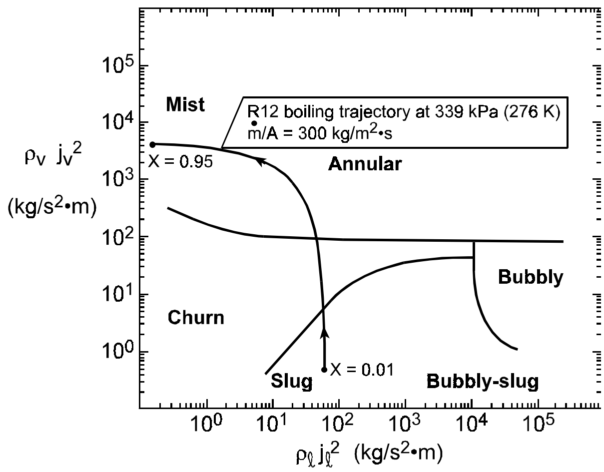

FIGURE 7. Flow pattern map for vertical flow

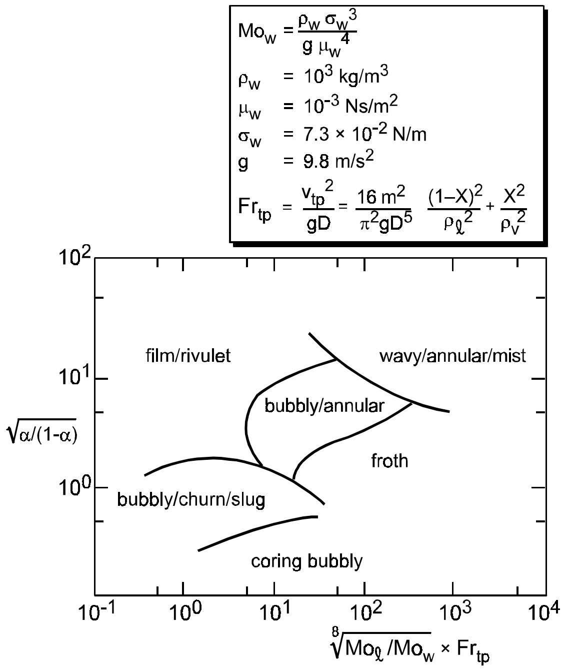

FIGURE 8. Flow Pattern Map for Vertical Down Flow

WHY THERMAL-GRAVITATIONAL MODELLING AND SCALING?

Development supporting theoretical work like thermal-gravitational modelling and scaling of two-phase heat transport systems (Delil, 1989; 1991 a, b; 1998 a, b; 1999 a, b; 2000 a, b) is being done to get better understanding of the impact of gravitation level on two-phase flow and heat transfer phenomena, to provide means for comparison and generalisation of data, to develop tools to design space-oriented two-phase loops (components), based on terrestrial tests, to and reduce costs. Scaling of physical dimensions is of major interest in the process industry: large-scale industrial systems are studied using reduced scale laboratory systems. Scaling of the working fluid is of principal interest in the power industry: large industrial systems, characterised by high heat fluxes, temperatures, and pressures, are translated in full size systems operating at more attractive lower temperature, heat flux and pressure.

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 using results of experiments with scale models on earth. Scaling spacecraft systems can 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. Also in-orbit experiments are defined 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 for terrestrial scaling of orbiting spacecraft, to 0.16 g on Moon and 0.4 g on Mars s, and to super-g on larger planets.

Even in single-phase systems scaling is anything but simple, since flow and heat transfer are equivalent in model and prototype only if the corresponding velocity, temperature and pressure fields are identical. Dimensionless numbers can be derived from conservation equations (mass, momentum, energy) or from similarity considerations, based on dimension analysis. Identity of velocity, temperature and pressure fields is obtained if all dimensionless numbers are identical in model and prototype. Scaling two-phase systems is far more complicated because also the spatial density distribution (void fraction, flow pattern) is to be considered, and also because of a proportion problem at high power densities, typical for two-phase flow boiling heat transfer.

DO SIMILARITY CONSIDERATIONS & DIMENSION ANALYSES YIELD RESULTS?

Similarity considerations (Delil, 1991 a, b,c) led to the identification of 18 dimensionless numbers (so-called π-numbers) relevant for thermal gravitational scaling of mechanically and capillary pumped two-phase loops. This set of 18 π-numbers, called "the most complete set for two-phase flow" (Isaacci, 1995), is listed in the table below.

Relevance of π-numbers for thermal Gravitational scaling of two-phase loops	Liquid Parts		Evaporators Swirl & Capillary	Non-liquid Lines Vapour/2-Phase	Condensers
	Adiabatic	Heating/Cooling			
π_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	•	•	•	/•	•
π_6 = S = slipfactor = v_v/v_l			•	•	•
π_7 = density ratio = ρ_v/ρ_l			•	•	•
π_8 = viscosity ratio = μ_v/μ_l			•	•	•
$\pi_9 = We_l = (\rho v^2 D/\sigma)_l$ = inertia/surface tension			•	/•	•
$\pi_{10} = Pr_l = (\mu Cp/\lambda)_l$		•	•		•
$\pi_{11} = Nu_l = (hD/\lambda)_l$ = convective/conductive		•	•		•
$\pi_{12} = \lambda_v/\lambda_l$ = thermal conductivity ratio			•		•
$\pi_{13} = Cp_v/Cp_l$ = specific heat ratio			•		•
$\pi_{14} = \Delta H/h_{lv}$ = Bo =enthalpy nr. = X = quality		•	•	•	•
$\pi_{15} = Mo_l = (\rho_l \sigma^3/\mu_l^4\ g)$ = capillarity/buoyancy			•	/•	•
$\pi_{16} = Ma = v/(\partial p/\partial \rho)^{1/2}{}_s$			•	•	•
$\pi_{17} = (h/\lambda_l)(\mu_l^2 g)^{1/3}$			•		•
$\pi_{18} = L^3 \rho_l^2\ g\ h_{lv}/\lambda_l \mu_l (T-T_o)$			•		•

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 evident that perfect scaling is not possible for two-phase flow and heat transfer: the phenomena are too complex, the number of important parameters or π-numbers is too large. But also distorted scaling can give useful results (Murphy, 1950), if the relative magnitude of the effects is carefully estimated. Effects identified as minor important, make the need for identity of some π-numbers superfluous for the problem, e.g. the Mach number is not important for pure liquid flow, the Froude number is unimportant for pure vapour flow. A limit of dimension analysis is the fact that the proportionality factor between the various π-numbers is not always known. Such a factor might be derived, by depicting data in graphs, showing relations between adequately chosen π-number groupings.

Important conclusions drawn from detailed considerations given before (Delil, 1989; 1991 a, b; 1998 a, b; 1999 a, b; 2000 a, b) are that condensers and, in mechanically pumped systems, also two-phase lines, are crucial in scaling with respect to gravity. They determine the conditions for evaporators and single-phase sections. In adiabatic two-phase lines (in mechanically pumped systems) under low-gravity conditions, only shear forces are expected to cause separation of phases in a high-quality mixture. This leads to annular flow (a fast moving vapour in the core and a, by frictional drag induced, slowly moving liquid annulus at the inner line wall) for the lower flow rates. For increasing power, hence flow rate, the slip factor will increase introducing waves on the liquid-vapour interface and entraining of liquid droplets in the vapour: wavy-annular-mist flow. A similar flow pattern can be predicted for vertical downward flow on earth, as it easily can be derived from a flow pattern map for downward two-phase flow (Fig. 8). In this map (Oshinowo, 1974), water properties at 293 K (Fig. 8) must be used to set the scale of the abscissa. Comparing low-g and vertical downward terrestrial flow one has to correct the latter for the reduction of the slip factor by gravity assisting the liquid layer flowing down. Anyhow, vertical down flow is the preferred two-phase line orientation in a terrestrial model because of the axial-symmetric flow pattern. A similar conclusion can be drawn for straight tube condensers, where flow will change from wavy, annular, mist to liquid flow, passing several flow patterns, depending on the condensation path.

Consequences of scaling can be derived from the figures 9 and 10, depicting the temperature dependence of the groups $g.Mo_l = \rho_l.\sigma^3/\mu_l^4$ and $(\sigma/\rho_l)^{½} = D.g^{½}/(We/Fr)^{½}$.

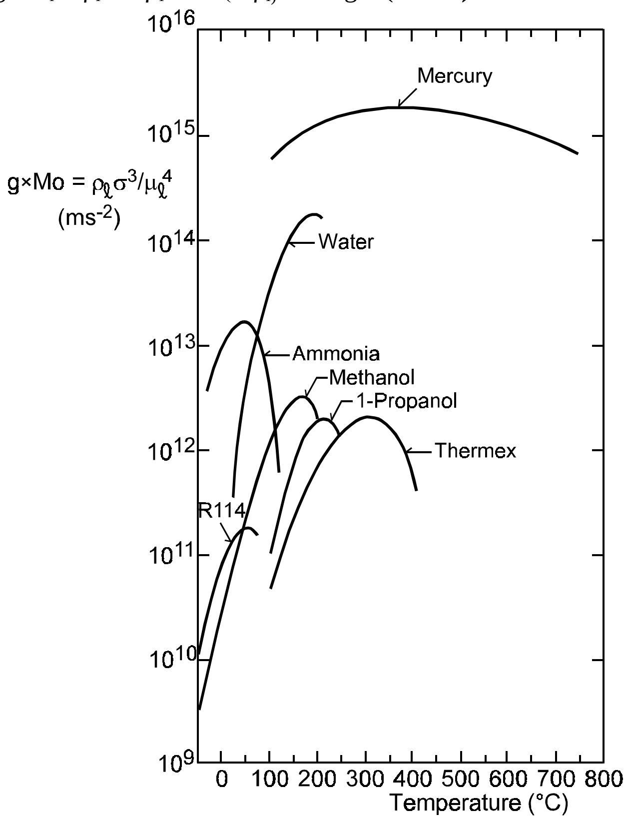

FIGURE 9. $\rho_l.\sigma^3/\mu_l^4$ Versus Temperature for Six Fluids

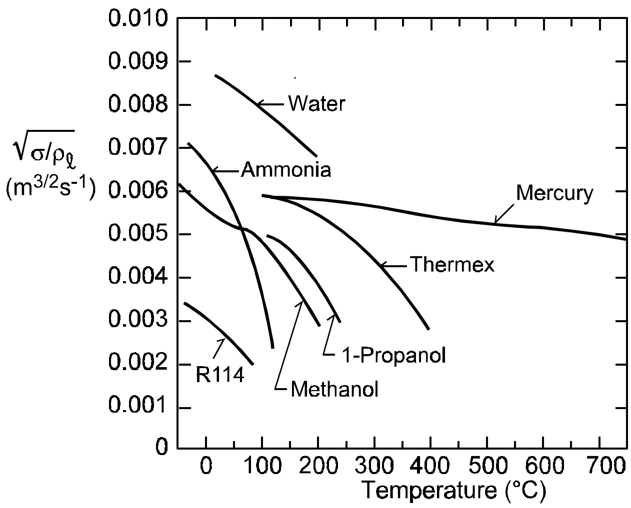

FIGURE 10. $(\sigma/\rho_l)^{½} = Dg^{½}/(We/Fr)^{½}$ Versus Temperature

Scaling at the same gravity level means a fixed $gMo = \rho_l\sigma^3/\mu_l^4$-value for prototype and model. Figure 9 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 scales follow from reading the with these temperatures corresponding (σ/ρ_l)-values in figure 10, and inserting identity in $g/(We/Fr)^{½}$, the geometric ratios 2.5 : 4.5 : 8.4 : 4.2 : 3.0 : 5.0 : 3.6. Figure 9 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 10 that the geometric scaling ratio between high-pressure prototype and low-pressure model (both characterised by $\rho_l\,\sigma^3/\mu^4_l = 2.10^{12}$ m/s^2) is about 0.4. Figure 9 shows also that scaling with respect to gravity is restricted to maximal two decades, if the fluid in prototype and model is the same (water or methanol). The figures 9 and 10 illustrate also that "fluid to fluid"-scaling offers many possibilities, hence is far more interesting. A very attractive scaling 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 (cos υ) of non-horizontal lines in the terrestrial model may lead to almost perfect scaling. Further details are given in Delil, 1989; 1991 a, b; 1998 a, b; 1999 a, b; 2000 a, b.

ARE MODELLING OUTCOMES VERIFIED BY EXPERIMENTAL RESULTS?

An important quantity (to be measured during two-phase flow experiments) is the pressure drop in adiabatic sections and in condensers: sections considered crucial for two-phase system modelling and scaling. The equations for annular flow pressure drops in straight tube condensers and adiabatic lines are extensively discussed (Soliman, 1968; Delil, 1992 a, b; 1998 a, b; 1999 a, b; 2000 a, b)). The total local (z-dependent) pressure gradient for annular flow is the sum of friction, momentum and gravity gradients. These constituents, calculated for ammonia at 25 °C (298K) and -25 °C (248K) prove that at low temperature the gravity constituent is overruled by the other contributions (Fig. 11). This confirms the earlier statement that low-gravity behaviour can be investigated by terrestrial tests at low temperature. Figure 12 shows curves calculated (Delil, 1991), assuming a constant 10^{-2}-g acting co-current with the flow, counter-current and perpendicular to the flow. As hydraulic changes in thermal systems are relatively slow, each measured value represents a mean of many measurements (Chen, 1991) at an average g of the order 10^{-2}-g. These measured data lie within the boundaries of the calculated curves.

Modelling and calculations were extended from adiabatic to condensing flow in a straight duct (Delil, 1992 a, b) to investigate the impact of gravity level on the duct length required to achieve complete condensation. This impact, reported to lead to duct lengths being more than one order of magnitude larger for zero gravity, as compared to horizontal orientation in earth gravity (Da Riva, 1991), was assessed for various mass flow rates, duct diameters and thermal (loading) conditions, for ammonia and R114. A summary of results of calculations for ammonia is presented next. To compare the results of calculations with data from literature, the condenser defined by Da Riva was chosen as the baseline (power 1 kW, line diameter 16.1 mm, ammonia temperature 300 K and temperature drop to sink 10 K).

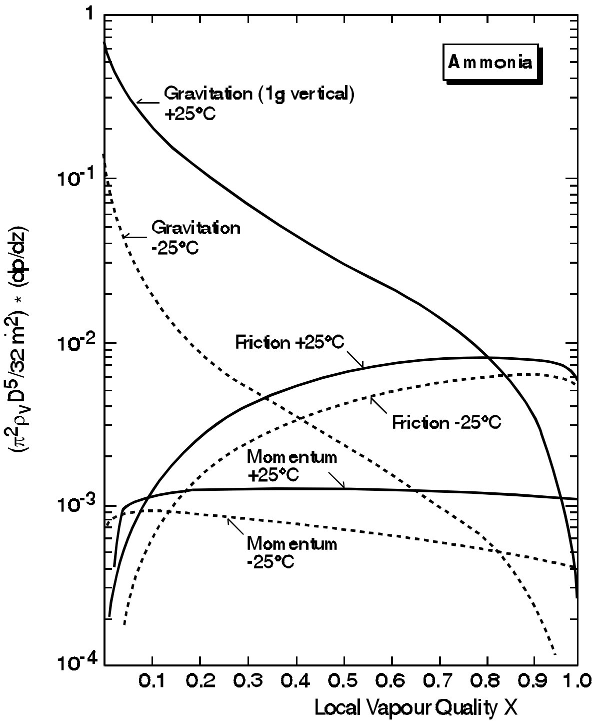

FIGURE. 11. Friction, Momentum and Gravity Contributions as a Function of the Vapour Quality

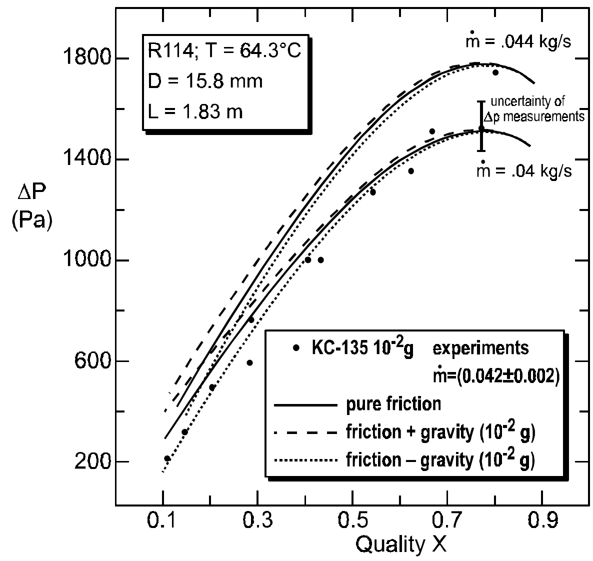

FIGURE 12. Measured and Predicted Adiabatic Pressure Drops

Figure 13 shows the vapour quality X along the condensation path (as a function of non-dimensional length z/D) for all gravity levels mentioned, including the curves for zero-g and horizontal condensation on earth, found in literature (Da Riva, 1991). From this figure it can be concluded that: the length required for full condensation strongly increases with decreasing gravity. Zero-gravity condensation length is roughly 10 times the terrestrial condensation length. Da Riva's data can be considered as extremes. To assess the impact of saturation temperature on condensation, similar curves were calculated for two other temperatures, 243 K and 333 K, and the parameter values given above (1992 a, b). 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 and low-g outcomes decrease with decreasing temperature. It confirms the statement that gravity impact 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 (Delil, 1992) that 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). Also are, under earth gravity conditions, power and full condensation length strongly interrelated: from L_c = 554 D at 25 kW to only 19 D at 500 W. The gravity dependence of the full condensation length decreases with increasing power, until the differences vanish at roughly 1 MW condenser choking conditions. Calculation of the vapour quality along the duct for three gravity levels (0, Earth and 2-g) and three duct diameters (8.05, 16.1 and 24.15 mm) at 300 K, yielded the ratio of the absolute duct lengths L_c(m) needed for full condensation under zero-g and one-g respectively (Delil, 1992 a, b). It is concluded that 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 TPX I flight data (Delil, 1995).

WHAT ARE THE FLOW PATTERN ISSUES?

As the model developed is valid for annular flow, 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 other words, it is to investigate whether the pure annular flow assumption, leads towards slightly or substantially overestimated full condensation lengths. In addition, flow pattern transitions occur at quality values, which strongly depend on temperature and line diameter.

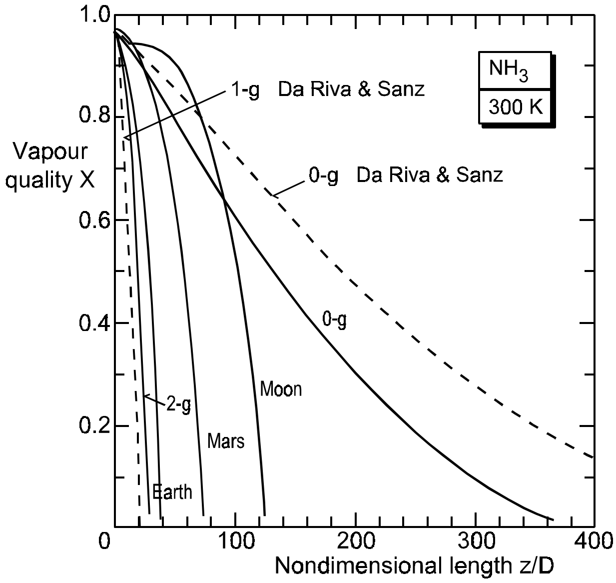

FIGURE 13. Vapour Quality Along the 16.1 mm Duct for Ammonia at 300 K, 1 kW, for All Gravity levels

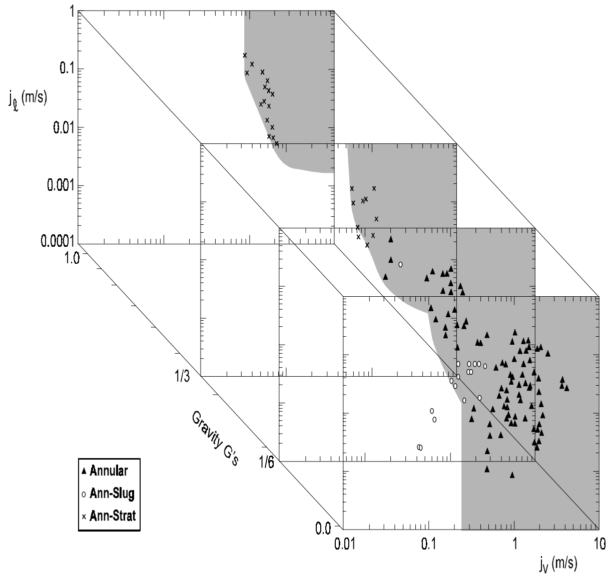

FIGURE 14. Annular Flow: Gravity Dependent Three-Dimensional Flow Pattern Map

Accurate knowledge of gravity level dependent flow regimes is crucial to model and design two-phase heat transport systems, as flow patterns directly affect thermal-hydraulic characteristics of two-phase flow and heat transfer. Therefore flow pattern maps are to be created, preferably in the non-dimensional format of figure 8.The three-dimensional flow pattern maps, shown in the figures 14 and 15, were created by using many K135 aircraft flight data obtained with a R12, 10.5 mm line diameter experiment (Hamme, 1997). Data were obtained at various g-levels, realised during flights. They clearly show the gravity level dependency of the shifts in transitions from annular flow to slug or stratified flow, and from slug/plug flow to annular and stratified flow.

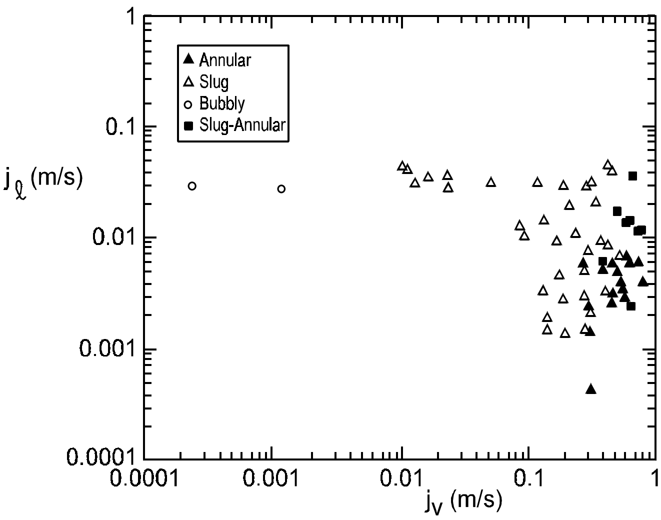

FIGURE 17. Cyrène Flow Pattern Map

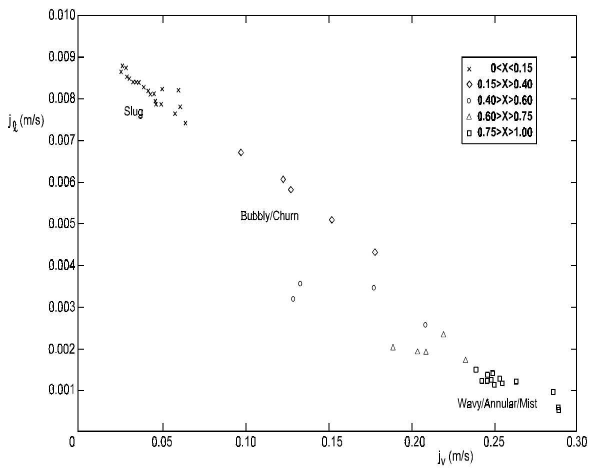

FIGURE 18 Flow Patterns Derived from TPX I Vapour Quality Sensor Data

Figure 16 summarises the 0-g data. It is a cross-section at 10^{-2}-g of figures 14 and 15. Figure 17 shows data of low-g aircraft experiments with Cyrène, a 4.7 mm line diameter ammonia system a (Lebaigue, 1998). Figure 18 shows the 0-g map, derived from TPX I (ammonia, 4.93 mm line) flight data obtained during the STS-60 flight (Delil, 1995). The maps partly contradict each other. Comparison between data suggests that the transition to annular flow occurs in the three systems more or less at the same j_v-value 0.2-0.25 m/s, but at different j_l-values. This may be due to different line diameters (10.5 mm/4.7 mm/4.93 mm) or to different fluids (R12/ammonia/ammonia. More data are needed to draw a final conclusion on the actual cause.

In conclusion it can be said that the above illustrates that a lot of work has to be done before adequate flow pattern (regime) maps will be produced and will become mature. Such maps preferably have to be in the normalised format of figure 8 or in the very good alternative (three-dimensional) j_v - j_l - g format, given in the figures 14 to 18. They can then be used to determine in an iterative way, via the flow pattern dependent constitutive equations for two-phase flow and heat transfer, the actual trajectories of condensing or evaporating (boiling) flow. The latter will finally lead to an accurate determination of the pressure drops in the various sections and of the heat transfer in the evaporator or condenser sections of a two-phase heat transport system.

NOMENCLATURE

A	area (m^2)
Boil	boiling number = $\Delta H/h_{lv}$ = Boil (-)
Cp	specific heat at constant pressure (J/kg.K)
D	diameter (m)
Eu	Euler number = $\Delta p/\rho v^2$ (-)
Fr	Froude number = v^2/gD (-)
g	gravitational acceleration (m/s^2)
H	enthalpy(J/kg)
h	heat transfer coefficient ($W/m^2.K$)
h_{lv}	latent heat of vaporisation (J/kg)
j	superficial velocity (m/s)
k	thermal conductivity (W/m.K)
L	length (m)
Ma	Mach number = $v/(\partial p/\partial \rho)_s^{1/2}$ (-)
Mo	Morton number = $\rho_l \sigma^3/\mu_l^4$ g (-)
$\dot{m}$	mass flow rate (kg/s)
Nu	Nusselt number = hD/λ (-)
p	pressure (Pa = N/m^2)
Pr	Prandtl number = $\mu Cp/\lambda$ (-)
Q	power (W)
Re	Reynolds number = $\rho vD/\mu$ (-)
S	slip factor (-)
T	temperature (K = 273 + °C)
v	velocity (m/s)
We	Weber number = $\rho v^2 D/\sigma$ (-)
X	vapour quality = vapour mass fraction (-)
z	axial or vertical co-ordinate (m)
α	vapour/void fraction (volumetric) (-)
Δ	difference, drop (-)
λ	thermal conductivity (W/m.K)
μ	viscosity ($N.s/m^2$)
ν	angle (with respect to gravity) (rad)
π	dimensionless number (-)
ρ	density (kg/m^3)
σ	surface tension (N/m)

Subscripts

a	acceleration, adiabatic, axial
c	condenser, cold
e	evaporator
f	friction
g	gravitation
l	liquid
m	momentum, model
o	reference condition, outer
p	pore, prototype
t	total
tp	two-phase
v	vapour

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Part 2.

Thermal-Gravitational Modelling and Scaling of Two-Phase Heat Transport Systems from Micro-Gravity to Super-Gravity Levels

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Abstract. Earlier publications extensively describe NLR research on thermal-gravitational modelling and scaling of two-phase heat transport systems for spacecraft applications. These publications on mechanically and capillary pumped two-phase loops discuss pure geometric scaling, pure fluid to fluid scaling, and combined (hybrid) scaling of a prototype system by a model at the same gravity level, and of a prototype in micro-gravity environment by a scale-model on earth. More recent publications include the scaling aspects of prototype two-phase loops for Moon or Mars applications by scale-models on earth. Recent work, discussed here, concerns extension of thermal-gravitational scaling to super-g acceleration levels. This turned out to be necessary, since a very promising super-g application for (two-phase) heat transport systems will be cooling of high-power electronics in spinning satellites and in military combat aircraft. In such aircraft, the electronics can be exposed during manoeuvres to transient accelerations up to 120 m/s^2. The discussions focus on “conventional” (capillary) pumped two-phase loops. It is considered as introduction to the accompanying article, which focuses on pulsating/ oscillating devices.

BACKGROUND

Summarizing earlier discussions (Delil, 1989; 1991 a to c; 2000), it is repeated that there are many examples of scaling of two-phase flow and heat transfer in the power and the process industries. Scaling physical dimensions is the principal interest in the process industry: Large-scale industrial systems are studied by reduced-scale laboratory model systems. Scaling the working fluid is of principal interest in the power industry: Large-scale industrial systems (high heat fluxes, temperatures, and pressures) are translated in full size systems, operating at lower temperature, heat flux and pressures (e.g. high pressure water-steam systems scaled by low pressure refrigerant systems of identical sizes). Thermal/ gravitational modeling and scaling of two-phase heat transport systems was done at NLR to get a better understanding of two-phase flow and heat transfer phenomena, to provide means for comparison and generalization of data, and to develop a useful tool to design two-phase systems and components, to save money, to reduce costs.

Main goal of scaling space-oriented two-phase heat transport systems is developing reliable spacecraft systems, whose reduced-g performance can be correctly predicted by results of terrestrial experiments with scale models. Scaling of spacecraft systems can be useful also for in-orbit technology demonstration (as the performance of spacecraft heat transport systems prototypes can be predicted using the outcomes of in-orbit experiments on models with reduced geometry or a different working fluid) and for defining in-orbit experiments to isolate a particular phenomenon to be looked at (e.g. excluding gravity-induced disturbing buoyancy effects on alloy melting, diffusion and crystal growth) for a better understanding of the physics. The magnitude of the gravitational scaling varies with the objectives: From 1 g to 10^{-6} g for terrestrial scaling of orbiting spacecraft, from 1 g to 0.16 g for Moon base and 0.4 g for Mars base systems, from 10^{-2} g or 10^{-6} g to 1 g for isolating gravity-induced disturbances on physical phenomena investigated, from a low-g level to another or the same low-g level in low-g aircraft or sounding rockets. NLR developments originally pertained to the scaling of mechanically and capillary pumped two-phase loops for use in micro-gravity. The activities were based on dimension analytical similarity considerations. They were extended later to applications for Moon and Mars bases (Delil, 1998 a, b; 1999 a to c).

APPROACH: DIMENSION ANALYSIS & SIMILARITY CONSIDERATIONS

Similarity considerations (for details see the aforementioned references) led to the identification of 18 dimensionless groups (so-called π-numbers), considered crucial for the thermal-gravitational scaling of two-phase loops. These 18 π-numbers are called "the most complete list of dimensionless numbers for two-phase flow and heat transfer" (Issacci, 1995), are shown in the first column of the table below. It is marked by • in the other columns whether a typical π-number is relevant in a particular section of a two-phase heat transport system: The liquid lines (with and without heat exchange), capillary or swirl evaporators, vapour or two-phase lines, and condensers. 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 evident that this is not possible for two-phase flow and heat transfer, as the phenomena are too complex, the number of important parameters or π-numbers is too large. Fortunately also imperfect (distorted) scaling can give useful results (Murphy, 1950). But, therefore a careful estimation of the relative magnitudes of the effects is required. Unimportant effects make the identity requirement for some π-numbers superfluous for the particular problem.

Relevance of π-numbers for thermal Gravitational scaling of two-phase loops	Liquid Parts		Evaporators Swirl & Capillary	Non-liquid Lines Vapour/2-Phase	Condensers
	Adiabatic	Heating/Cooling			
π_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	•	•	•	/•	•
π_6 = S = slipfactor = v_v/v_l			•	•	•
π_7 = density ratio = ρ_v/ρ_l			•	•	•
π_8 = viscosity ratio = μ_v/μ_l			•	•	•
$\pi_9 = We_l = (\rho v^2 D/\sigma)_l$ = inertia/surface tension			•	/•	•
$\pi_{10} = Pr_l = (\mu Cp/\lambda)_l$		•	•		•
$\pi_{11} = Nu_l = (hD/\lambda)_l$ = convective/conductive		•	•		•
$\pi_{12} = \lambda_v/\lambda_l$ = thermal conductivity ratio			•		•
$\pi_{13} = Cp_v/Cp_l$ = specific heat ratio			•		•
$\pi_{14} = \Delta H/h_{lv}$ = Bo =enthalpy nr. = X = quality		•	•	•	•
$\pi_{15} = Mo_l = (\rho_l \sigma^3/\mu_l^4\, g)$ = capillarity/buoyancy			•	/•	•
π_{16}=Ma = $v/(\partial p/\partial \rho)^{1/2}{}_s$			•	•	•
$\pi_{17} = (h/\lambda_l)(\mu_l^2 g)^{1/3}$			•		•
$\pi_{18} = L^3 \rho_l^2\, g\, h_{lv}/\lambda_l \mu_l (T-T_o)$			•		•

Sometimes it is more convenient to replace quality X by the volumetric vapour fraction (void fraction) α via

$$(1 - \alpha)/\alpha = S\, (\rho_v / \rho_l)\, X/(1 - X)\,. \qquad (1)$$

The set of 18 π-numbers is rather arbitrarily chosen: Several numbers contain only liquid properties. They can be transferred into vapour properties containing numbers, using π_6 to π_8. π_1 can be used to interchange a characteristic length (duct length, bend radius) and some characteristic diameter (duct diameter, hydraulic diameter, surface roughness, bubble size). The best scaling approach is to choose π-number combinations suiting the problem under investigation. Examples are the Morton number Mo_l (used for scaling two-phase flow with respect to gravity, as it contains only liquid properties, surface tension and gravity), the Mach number Ma (a crucial quantity, if the compressibility is important, because choking depends on the vapour quality of the two-phase mixture), and the enthalpy or boiling number Boil (Q is the power fed to the boiling liquid):

$$\pi_{15} = Mo_l = Re_l^4 Fr_l / We^3 = \rho_l \sigma^3 / \mu_l^4 g \tag{2}$$

$$\pi_{16} = Ma = v/(\partial p/\partial \rho)^{1/2} \tag{3}$$

$$\pi_{14} = Boil = X = \Delta H(z)/ h_{lv} = Q/ \dot{m} h_{lv} = Q/\dot{m}\, Cp_l\, \Delta T = \Delta H(z)/h_{lv} = \Delta H_{in}/h_{lv} + \pi Dzq/\dot{m}h_{lv}. \tag{4}$$

The right hand side is the dimensionless enthalpy at any z in a line heated from outside (q is heat flux) for sub-cooled liquid (ΔT is temperature drop). This implies that, if the dimensionless entrance enthalpies are equal for different fluids flowing in a similar geometry, boiling number identity means equal non-dimensional enthalpies at similar axial locations. Thermodynamic equilibrium means equal quality at similar locations, equal sub-cooling & boiling lengths. Other examples are the condensation number π_{17} (in which h is the local heat transfer coefficient) and the vertical wall condensation number π_{18} (with T_o as the local sink, T as the local saturation temperature):

$$\pi_{17} = (h/\lambda_l)\, (\mu_l^2/g\, \rho_l^{\,2})^{1/3}, \tag{5}$$

$$\pi_{18} = L^3 \rho_l^2\, g\, h_{lv}/\mu_l\, \lambda_l (T - T_o)\, . \tag{6}$$

A first step, in a practical approach to scale two-phase heat transport systems, is identification of the important phenomena, to obtain π-numbers for which identity in prototype and model must be required to realise perfect scaling according to the so-called Buckingham Pi Theorem, being crucial in similarity considerations (Murphy, 1956). Distortion will be permitted for π-numbers pertaining to less important phenomena. Important phenomena and the relevant π-numbers will differ in different parts of a system. As said before, 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).

For refrigerants, forced convection heat transfer overrules conduction completely. Hence π_{10}, π_{11} and π_{12}, are not critical in gravitational scaling. π_{16} can be neglected if the system maximum power level and line diameters correspond with flow velocities far below the sonic velocity in all system parts. On π_3/π_5, it can be remarked that inertia overrules buoyancy not only in pure vapour flow or in a low gravity environment, but also for horizontal liquid sections on earth ($\nu \rightarrow \pi/2$). This implies that there is π-number identity for these sections in the low-g prototype and the terrestrial model, for a horizontal arrangement of these sections. Also it is remarked that, in the porous (liquid) part of a capillary-pumped evaporator, surface tension (σ/D_p) is dominant over inertia ($\pi_9 \rightarrow 0$): evaporator exit quality approaches 1 (pure vapour). This means that gravity is less important for the vapour part of the evaporator and the vapour line to the condenser.

A first conclusion can be drawn now: Condensers and, in mechanically pumped systems, also two-phase lines, are crucial in scaling with respect to gravity. They set conditions for evaporators and single-phase sections. A second conclusion is: In adiabatic two-phase lines of mechanically pumped systems in low-gravity, only shear forces will cause separation of phases in a high-quality mixture, leading to annular flow (fast moving vapour in the core and a frictional drag induced, slowly moving liquid annulus at the inner line wall) for the lower flow rates. For increasing power, hence flow rate, the slip factor will increase introducing waves on the liquid-vapour interface and entraining of liquid droplets in the vapour: wavy-annular-mist flow. A similar flow pattern can be predicted for vertical downward flow on earth.

Comparing low-g and vertical downward terrestrial flow one has to account in the latter for the reduction of the slip factor by the gravity forces assisting the liquid layer flowing down (draining effect). Anyhow, vertical down flow is the preferred two-phase line orientation in the terrestrial model, because of its axial-symmetric flow pattern. A similar conclusion can be drawn for the straight tube condenser. In condensers the flow will change from wavy annular mist to pure liquid flow, passing several flow patterns, depending on the path of the condensation.

Scaling consequences are derived from figures 1 and 2, showing the temperature dependence of $g.Mo_l = \rho_l.\sigma^3/\mu_l^4$ and $(\sigma/\rho_l)^{1/2} = D.g^{1/2}/(We/Fr)^{1/2} = D.g^{1/2}/(Eö)^{1/2} = D.g^{1/2}/2(Bo)^{1/2}$. Eötvös number Eö and Bond number Bo are defined by:

$$Eö = 4\,Bo = g\,D^2\,(\rho_l - \rho_v)/\,\sigma \approx g\,D^2\,\rho_l. \tag{8}$$

Conclusions on scaling possibilities were drawn from these two figures, in the earlier NLR publications. They pertain to scaling at the same gravity level or to scaling of a prototype in a lower than 1-g environment, and a model on earth, preferably operating in the vertical down-flow (gravity assist) mode. Shortly summarizing extensive discussions in these publications, it can be said that scaling two-phase heat transport systems is very complicated. Only distorted scaling offers some possibilities, if not the entire loop but only sections are involved. Scaling with respect to gravity is hardly discussed in literature. Very few possibilities can be identified, and for restricted conditions only. Scaling at the same gravity covers only a limited range.

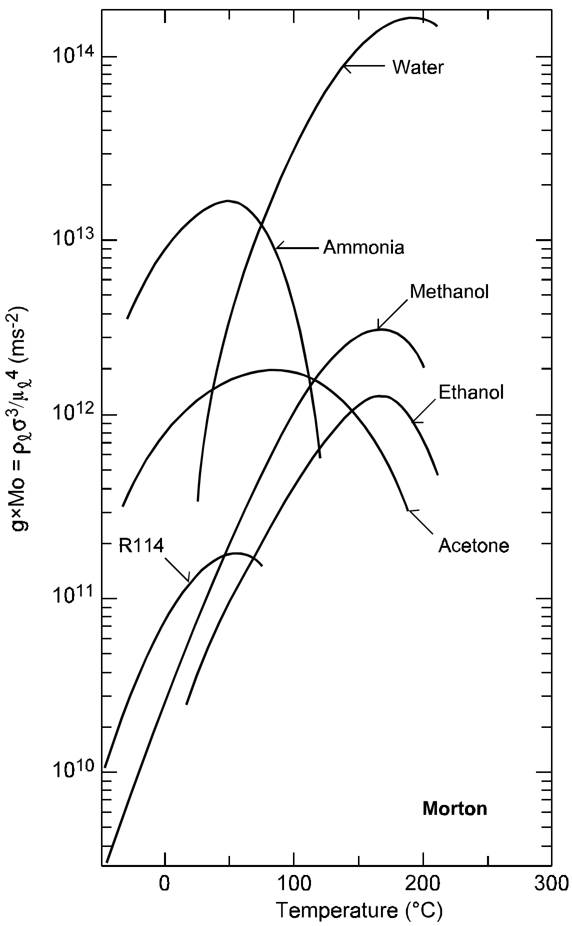

FIGURE 1. $\rho_l.\sigma^3/\mu_l^4$ Versus Temperature for Six Fluids

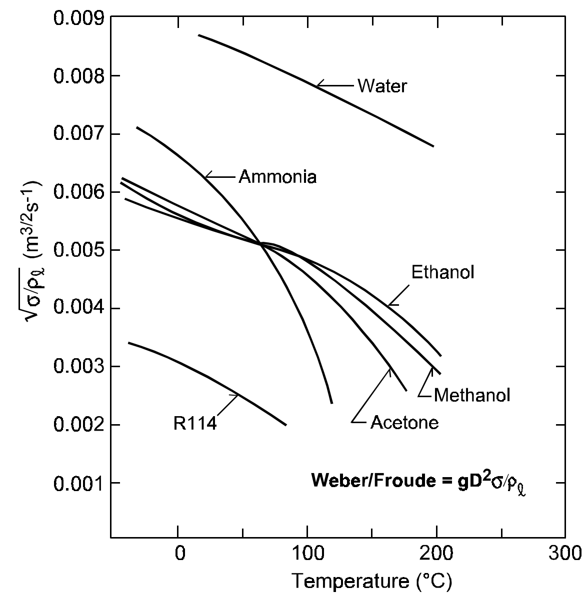

FIGURE 2. $(\sigma/\rho_l)^{1/2} = D.g^{1/2}/(We/Fr)^{1/2}$ Versus Temperature

.

Scaling high-pressure systems, e.g. ammonia at 110 °C (383 K), by low-pressure system (parts), e.g. -50 °C (223 K) ammonia, may be attractive for safety reasons or to reduce the impact of earth gravity in vertical two-phase sections. It follows from the figures, that the length scale ratio between high-pressure prototype and low-pressure model (characterized by $\rho_l\,\sigma^3/\mu^4_l = 2.10^{12}$ m/s^2) is $L_p/L_m = [(\sigma/\rho_l)_p/\sigma/\rho_l)_m]^{1/2} \approx 0.4$. Anticipating later discussions, it is remarked that in low temperature vertical down flow on earth the impact of gravity is minimised, hence it is suitable to simulate conditions below 1-g. Gravitational scaling is restricted to say two decades, if prototype and model fluid are the same.

Scaling the full size low-gravity ($< 10^{-2}$ g) mechanically-pumped R114 ESA mechanically pumped Two-Phase Heat Transport System, can be adequately done by NLR's ammonia test rig, since the 10^{-2} - 10^{-3} g R114 prototype and the terrestrial ammonia model have approximately identical Morton numbers. This fluid to fluid scaling yields a length ratio of

4.5 to 6.5, meeting the ratio of actual diameters: 21 mm and 4.9 mm respectively. NLR's mechanically pumped two-phase ammonia test rig offers also some opportunities to scale the TPX ammonia loops.

An 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 only 0.4 for Mars, 0.16 for the Moon, the sizes of the model have to be slightly larger than the geometric sizes of the prototype. Adjustment of inclinations ($\cos \nu$) of non-horizontal lines in the terrestrial model may lead to almost perfect scaling.

PRESSURE DROP & HEAT TRANSFER EQUATIONS, CONDENSATION LENGHTS

An important quantity, the pressure drop in the different two-phase sections, is considered crucial for two-phase loop modelling and scaling. The next considerations therefore will concentrate on pressure drops in condensing and adiabatic flow and will restrict the discussion to straight tubes. The total local pressure gradient for annular flow $(dp(z)/dz)_t$ is the sum of three constituents: friction, momentum and gravity. The equations for these constituents are given below (deleting the z-dependence to shorten the notation). Fluid properties are assumed independent of z, since they depend only on the mixture temperature, which usually is almost constant in adiabatic and condensing sections. For details it is referred to elaborate articles on the subject (Soliman, 1968; Delil, 1991).

$$(dp/dz)_f = -(32m^2/\pi^2\rho_v D^5)(0.045/Re_v^{0.2})[X^{1.8}+5.7(\mu_l/\mu_v)^{0.0523}(1-X)^{0.47}X^{1.33}(\rho_v/\rho_l)^{0.261}+8.1(\mu_l/\mu_v)^{0.105}(1-X)^{0.94}X^{0.86}(\rho_v/\rho_l)^{0.52}]$$

(X is local quality X(z), Reynolds number $Re_v = 4\dot{m}/\pi D\mu_v$, β=2 (laminar), 1.25 (turbulent liquid flow). (9)

$$(dp/dz)_m = -(32\dot{m}^2/\pi^2\rho_v D^5)\,(D/2).(dX/dz)\,[2(1-X)(\rho_v/\rho_l)^{2/3} + 2(2X-3+1/X)(\rho_v/\rho_l)^{4/3} + (2X-1-\beta X)(\rho_v/\rho_l)^{1/3} + (2\beta-\beta X-\beta/X)(\rho_v/\rho_l)^{5/3} + 2(1-X-\beta+\beta X)(\rho_v/\rho_l)]. \quad (10)$$

$$(dp/dz)_g = (32\dot{m}^2/\pi^2\rho_v D^5)\{1-[1+(\rho_v/\rho_l)^{2/3}\,(1-X)/X]^{-1}\}[\pi^2 D^5 g \cos\nu(\rho_l-\rho_v)\rho_v/32\dot{m}^2]. \quad (11)$$

$g \to 0$ for micro-gravity conditions and $g \cos\nu$ equals 9.8 m/s^2 for vertical down flow on Earth, 3.74 m/s^2 for vertical down flow on Mars and 1.62 m/s^2 on the Moon. The slip factor S, used in the derivations (for simplicity reasons and since it allows comparison with results of calculations found in literature), was specified according to the principle of minimum entropy production (Zivi, 1964) for ideal annular flow, as

$$S = (\rho_l/\rho_v)^{1/3}. \quad (12)$$

To solve (9) to (11) an extra relation is necessary, defining the z-dependence of X. The missing equation

$$\dot{m}\, h_{lv}(dX/dz) = -\, h\pi D[T(z)-T_s]\,, \quad (13)$$

relates the local vapour quality and heat transfer. h is the local heat transfer coefficient h(z), for which one can write

$$h = 0.018(\lambda_l\rho_l^{1/2}/\mu_l)Pr_l^{0.65}|-(dp/dz)_t|^{1/2}\,D^{1/2}, \quad (14)$$

assuming that the major thermal resistance is in a laminar sub-layer of the turbulent condensate film (Soliman, 1968).

As already mentioned the two-phase flow path is almost isothermal, which implies constant temperature drop $T(z) - T_s$ (for a constant sink temperature T_s), constant fluid properties and a constant Prandtl number, defined in Table 1.
The total condensation pressure drop is

$$\Delta p_t = \int_0^{L_c} (dp/dz)_t dz\,. \quad (15)$$

The equations (1, 9 to 15) can be combined. This yields an implicit non-linear differential equation in the variable X(z), which can be rewritten into a solvable standard form for differential/algebraic equations

$$F(dX/dz, X) = 0. \quad (16)$$

Figure 3 compares the pressure gradient constituents for ammonia vertical down-flow at 25˚C (298 K) and -25˚C (248 K). The curves prove that the impact of gravity decreases with decreasing temperature. This confirms the earlier statement that low-gravity behaviour can be simulated the best by terrestrial down-flow tests at low temperature.

Modelling and calculations were extended from adiabatic to condensing flow in a straight duct (Delil, 1992 a, b), in order to investigate the impact of gravity level on the duct length required to achieve complete condensation. This impact, reported to lead to duct lengths being more than one order of magnitude larger for zero gravity, as compared to horizontal orientation on earth (Da Riva 1991), was assessed for various mass flow rates, duct diameters and thermal (loading) conditions, for ammonia and R114. A summary of results of calculations for ammonia is presented next. To compare the results of calculations with data from literature, Da Riva's (1991) condenser was chosen as the baseline. Main characteristics are: power 1 kW, line diameter 16.1 mm, temperature 300 K and temperature drop to sink 10 K. Gravity levels considered are: zero gravity g=0, Earth gravity (1-g) g=9.8 m/s^2, Mars gravity g=3.74 m/s^2, Moon gravity g=1.62 m/s^2, and 2-g super-gravity 19.6 m/s^2. Illustrative results of calculations are discussed next t.

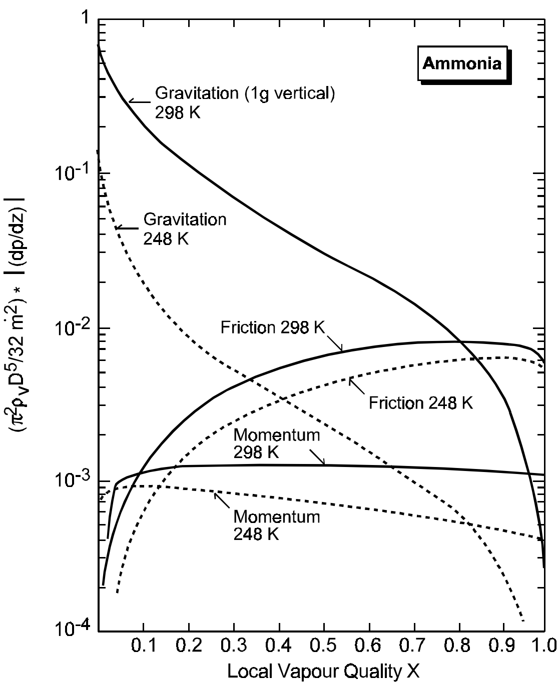

FIGURE 3. Pressure Gradient Constituents Versus Vapour Quality, for Ammonia at 298 K (25˚C) and 248 K (-25˚C)

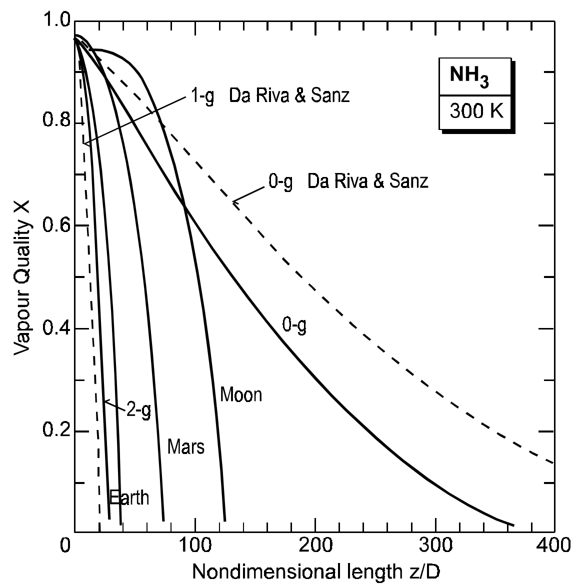

FIGURE 4. Vapour Quality Along the Reference Duct

Figure 4 depicts the vapour quality X along the condensation path (as a function of non-dimensional length z/D) for all gravity levels mentioned, including Da Riva's curves for zero-g and horizontal condensation on earth. From this figure it can be concluded that: the length required for full condensation strongly increases with decreasing gravity. Zero-gravity condensation length is roughly 10 times the terrestrial condensation length. Da Riva's outcomes are clearly extremes. To assess the impact of the saturation temperature on condensation, similar curves were calculated for 243 K and 333 K, and the above parameter values. They indicate that the full condensation length increases with temperature for zero-g conditions, but decreases with temperature for other gravity levels. This implies that differences between earth gravity and low-g decrease with decreasing temperature. It confirms the remark that gravity impact is smaller at lower temperatures. 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 (Delil, 1992a,b) that 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). It turned out that in earth gravity, power and full condensation length are strongly interrelated: from L_c = 554 D at 25 kW to 19 D at 500 W. The gravity dependence of the full condensation length decreases with increasing power, until the differences vanish at roughly 1 MW condenser choking conditions. This value is an upper limit, calculated for ideal annular flow.

Choking may occur at considerably lower power values in the case of actual annular-wavy-mist flow, but the value exceeds anyhow the choking limit for homogeneous flow, roughly 170 kW. Calculation of the vapour quality along the duct for three gravity levels (0, Earth and 2-g) and three duct diameters (8.05, 16.1 and 24.15 mm) at 300 K, yielded the ratio of the duct lengths L_c(m) needed for full condensation under zero-g and one-g respectively. It was concluded that the ratio between full condensation lengths in zero-g and one-g ranges from roughly 1.5 for the 8.05 mm duct, via 11 for the 16.1 mm duct, up to over 30 for the 24.15 mm duct. In other words, small line diameter systems are less sensitive for differences in gravity levels as compared to larger diameter systems. This has been confirmed by TPX I flight data (Delil, 1995). As the model developed is mainly valid for annular flow, it is worthwhile to look at the impact of other flow patterns in the condenser duct (high quality mist flow, low quality slug and bubbly flow and wavy-annular-mist in between). Hence it is to be investigated whether the pure annular flow assumption, leads towards slightly or substantially overestimated full condensation lengths. Complications are that the flow pattern boundaries are not accurately known and that transitions strongly depend on temperature and line diameter. Good knowledge of the gravity level dependent two-phase flow regimes is crucial for modelling and designing two-phase systems for space, as flow patterns directly affect thermal-hydraulic characteristics of two-phase flow and heat transfer. Therefore flow pattern maps are to be created, preferably in a non-dimensional format (Oshinowo, 1954) or 3-dimensional format (Hamme, 1997). Such maps can be used to determine iteratively, via the flow pattern dependent constitutive equations for two-phase flow and heat transfer, the actual trajectories of condensing or evaporating flow. That will finally lead to an accurate determination of pressure drops in the various sections and of the heat transfer in the evaporator or condenser sections of a two-phase heat transport system. In summary: The information presented confirms the results of other models. When designing condensers for space applications, one should carefully use and interpret data obtained from terrestrial condenser tests, even when the latter pertain to vertical downward flow situations (characterised by the same flow pattern). The equations given are useful for a better understanding of the problems to be expected: i.e. the needed condenser lengths in space. Equations and calculated results suggest that hybrid scaling, combining geometric and fluid-to-fluid scaling, can support the design of space-oriented two-phase heat transport systems. The local heat transfer equation (14) has a wrong lower limit $h \rightarrow 0$ for $(dp/dz)_t \rightarrow 0$. It disappears by incorporating conduction via the liquid layer. Preliminary calculations indicate that this will lead to somewhat shorter full condensation lengths, in all gravity conditions. This implies only quantitative changes: Consequently the conclusions presented before remain valid.

Finally it is stressed, that the design of ammonia two-phase heat transport system prototypes for Mars and Moon base applications, looks very promising. This is because flow pattern maps for pplanetary gravity levels can be easily obtained from measurements in models on earth, with the same or with another working fluid. These models will have (almost) identical geometry and (if necessary) equal lines. They also might have a dedicated inclination with respect to the Earth gravity vector (i.e. cos ν = 0.4 in the case of Mars, 0.16 in the case of the Moon). These maps can also include data obtained from experiments with ammonia test loops, carried out during low-g aircraft trajectories.

1-G SCALING OF PROTOTYPES FOR SUPER-G

NLR's modelling and scaling (Delil, 2000 and experimental activities of the last two years (Es, 2000) are extensions to two-phase loops and oscillating heat transport devices for use in (transient) super-gravity environments (up to 12 g), encountered in spinning spacecraft during military combat aircraft manoeuvres. Similar activities are reported to be done elsewhere (Kiseev, 1999; Romestant, 1999, Ku, 2000). Though the extensions led to this paper, the paper started with preceding considerations, dealing also with gravity-assist performance (vertical down-flow in some gravity field), as they remain valid and useable for two-phase loops, including pulsating two-phase loops in super-gravity-assist conditions. A quantitative example is the simple extrapolation of the gravity dependent condensation curves (Fig. 4) to 12-g, which yields a 10-g gravity-assist full condensation length of the order of 10 D. Though many things will be different in the anti-gravity mode, general thermal-gravitational scaling rules are valid for gravity-assist and anti-gravity conditions.

The consequences for scaling a super-gravity prototype system by a 1-g model straightforwardly follow from the above two figures. This is illustrated by the following example. For an ammonia prototype system P, intended for operation around 320 KC in a 10-g environment, figure 1 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 2 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 fulfill an additional capillary criterion, as it will be elucidated later in this article), 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, ideal for scaling two-phase systems, both loops and pulsating/oscillating ones.

ANTI-SUPER-GRAVITY ISSUES AND CONCLUDING REMARKS

The impact of anti or against gravity or super-gravity can be illustrated by recalling figure 3, showing the absolute values of the three pressure drop constituents. The results of pressure drop calculations, presented in the preceding sections, pertained to gravity-assist condensation This implies that the calculations were done for cosν = +1, meaning (as it is gravity-assist) that the contribution of gravity had the opposite sign of the constituents of friction and momentum. But, when considering anti-(super)-gravity modes, some important issues are to be noted.

First, any pump has to deliver a pumping pressure that is at least the sum of all pressure losses in the system. In an anti-10g mode, this is mainly the gravitational pressure drop. Figure 3 illustrates that the contributions of friction and momentum become negligible, as compared to the 10-g contribution, which can by obtained by simply shifting the 1-g curve one decade upwards. The 10-g curve clearly overwhelms the curves of the other constituents, for almost all vapour qualities except for qualities over say 0.9, where the flow pattern probably will be homogeneous, instead annular. In case of a mechanical pump, the huge (super-gravity) pressure losses can be overcome by the use of a correctly designed displacement pump. Existing capillary pumped systems, however, will fail completely, since existing capillary structures can operate a two-phase loop against gravity, only if the maximum pumping height to be delivered is less than say 6 metres. The 6 m gravitational pressure drop is at least a factor 20 smaller than the 10-g super-gravity pressure drops. Current high-performance capillary pumps use reasonably permeable wicks with a pore size of 0.5 μm to realise the 6 metres pumping height. For a 20 times larger pumping height, the pore size shall be about 25 nm. But a wick of such small pores will not have the required permeability of a capable capillary pump.

Second, pressure and temperature drops are only minor in the gravity-assist condensation curves presented. It means that the condensation is almost isothermal, hence the fluid properties can be assumed constant. This will not be the case for anti-super-gravity conditions, meaning that for such conditions calculations will become very complicated. As said, gravity influences pressure drop and the corresponding temperature drop across the heat transport system. This is confirmed by experimental data (Kiseev, 1999): Compared to 0-g pressure and temperature drops, the drops decrease with increasing gravity-assist, and strongly increase with increasing anti-gravity. It is therefore clear that for pulsating loops (and other pulsating heat transfer devices) working fluids with a large dp/dT in the operating range are to be used, since the pressure drop is the system driver, as in detail discussed by Delil (2000 a, b; 2001).

Third, for super-g conditions there is no information at all on flow pattern maps and the boundaries between the different flow regimes. Therefore and as it is expected that the various items will substantially differ from the (hardly available) existing 1-g ones, the creation of flow pattern maps for super-gravity environment has been started at NLR.

NOMENCLATURE

A	area (m^2)	Eu	Euler number = $\Delta p/\rho v^2$ (-)
Bo	Bond number = $g\,D^2\,\rho_l/4\sigma$ (-)	Fr	Froude number = v^2/gD (-)
Boil	boiling number = $\Delta H/h_{lv}$ = Boil (-)	g	gravitational acceleration (m/s^2)
C	conductance (W/K)	H	enthalpy (J/kg)
Cp	specific heat at constant pressure (J/kg.K)	h	heat transfer coefficient (W/m^2.K)
D	diameter (m)	h_{lv}	latent heat of vaporisation (J/kg)
Eö	Eötvös number = $g\,D^2\,\rho_l/\sigma$ (-)	k	thermal conductivity (W/m.K)

L length (m)
Ma Mach number = $v/(\partial p/\partial \rho)_s^{1/2}$ (-)
Mo Morton number = $\rho_l \sigma^3/\mu_l^4$ g (-)
$\dot{m}$ mass flow rate (kg/s)
Nu Nusselt number = hD/λ (-)
p pressure (Pa = N/m^2)
Pr Prandtl number = $\mu Cp/\lambda$ (-)
Q power (W)
Q heat flux (W/m^2)
Re Reynolds number = $\rho vD/\mu$ (-)
T temperature (K = 273 + °C)
S slip factor (-)
v velocity (m/s)
We Weber number = $\rho v^2 D/\sigma$ (-)
X vapour quality = vapour mass fraction (-)
z axial or vertical co-ordinate (m)

α vapour/void fraction (volumetric) (-)
Δ difference, drop (-)
σ surface tension (N/m)
λ thermal conductivity (W/m.K)
μ viscosity (N.s/m^2)
ν angle (with respect to gravity) (rad)
π dimensionless number (-)
ρ density (kg/m^3)

Subscripts

a acceleration, adiabatic, axial
c condenser, cold
e evaporator
f friction
g gravitation
l liquid
m momentum, model
o reference condition, outer
p pore, prototype
t total
tp two-phase
v vapour

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Part 3.

Modelling and Scaling of Oscillating or Pulsating Heat Transfer Devices Subjected to Earth Gravity and to High Acceleration Levels

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Abstract. The discussions, presented in this article, suppose that the reader is familiar with the contents of the accompanying article "Thermal-Gravitational Modelling and Scaling of Two-Phase Heat Transport Systems from Micro-Gravity to Super-Gravity Levels". The latter article describes the history of this particular research at NLR, the approach (based on dimension analysis and similarity considerations), the derivation of constitutive equations for (annular) two-phase flow and heat transfer, the identification of thermal-gravitational scaling possibilities, condensation length issues, and the impact of the magnitude of super-gravity and its direction relative to the flow direction. But the discussions are restricted to "classical" two-phase loops. The most recent part of the research is discussed in this follow-up article. It concerns the extension of the research to the modelling, scaling and testing of the steady and transient performance of various types of oscillating or pulsating single-phase and two-phase heat transfer devices. This extension was opportune, as it turned out to be essential to properly support the research and development of such oscillating or pulsating heat transfer devices. For these devices several very promising applications have been identified, not only to cool commercial electronics, but also for cooling high-power electronics in spinning satellites and in military combat aircraft. In such applications, the electronics can be exposed to steady and transient accelerations up to levels around 120 m/s^2.

INTRODUCTION

Overview articles (Delil, 2000 a to c; 2001) discuss NLR publications of the last decade on thermal-gravitational modelling & scaling of two-phase heat transport systems for spacecraft applications (Delil, 1989; 1991 a to c; 1998 a, b; 1999 a to c). The initial research focused on mechanically and capillary pumped two-phase loops for use in micro-gravity. The activities dealt with pure geometric, pure fluid to fluid, or hybrid (combined) scaling of a prototype system by a model at the same gravity level, and of a prototype in micro-g by a model on earth. That scaling was based on dimension analysis and similarity considerations. The scaling research was later extended to applications for Moon and Mars bases (Delil, 1999a to c). Scaling to super-gravity levels was included recently, as a promising super-gravity application for two-phase heat transport systems was identified: Cooling of high power electronics in spinning satellites and military combat aircraft. During manoeuvres of such aircraft, this electronics can be exposed to transient accelerations up to 120 m/s^2, 12 g. Experimental investigation of the performance of pumped two-phase heat transport loops was started elsewhere (Kiseev, 1999; Romestant, 1999, Ku, 2000). Above overviews include all relevant issues on the scaling approach, similarity considerations, useful equations, flow patterns issues, and scaling of "classical" (capillary) pumped two-phase loops between high acceleration levels and earth gravity, including against-

(super)gravity mode performance. The following considerations discuss the aforementioned items, but now focusing on “novel” heat transport devices.

NOVEL HEAT TRANSPORT DEVICES

Various devices will be described first: Single-phase synchronised and phase-shifted forced oscillatory flow heat transfer devices, pulsating two-phase heat transfer loops, and two-phase devices like pulsating or meandering heat pipes and the flat swinging heat pipe.

Single-Phase Forced Oscillatory Flow Heat Transfer Devices

Though these devices are not two-phase but single-phase, they are discussed here, as the modelling of these devices is useful to understand the operation of pulsating/oscillating heat pipes. A very interesting device is shown in figure 1 Detailed discussions on this synchronised forced oscillatory flow heat transfer device are given in the publications of the originators (Kurzweg, 1984; 1985 a, b; 1989) and to other publications on this device and on the related phase-shifted forced oscillatory flow heat transfer devices (Nishio, 1999).

As said, the device is not a two-phase device but a single-phase one. The set-up consists of two reservoirs at different temperatures, connected by a 0.2 m long, 12.7 mm inner diameter acrylic tube, containing 31 glass capillaries with an inner diameter d = 1 mm. The open cross-sectional area of the capillary structure, including the triangular sections between the capillaries, A_l was determined to be 67 mm^2, being 53% of the total inner cross-sectional area of the tube (A = 127 mm^2). The reservoirs are equipped with flexible membranes. A variable frequency shaker is used to oscillate the incompressible working liquid inside the capillary structure. The frequency f is variable from 2 and 8 Hz. The tidal displacement Δz is variable between 20 and 125 mm. The operation is as follows. Starting with the capillary structure

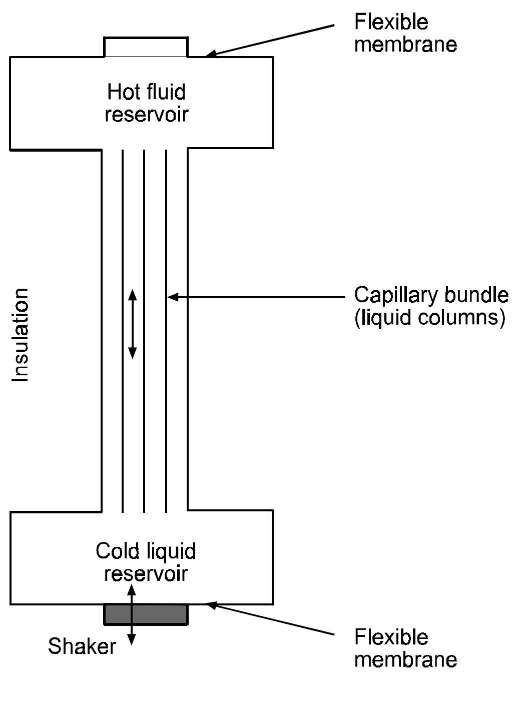

FIGURE 1. Synchronised Forced Oscillatory flow HTD

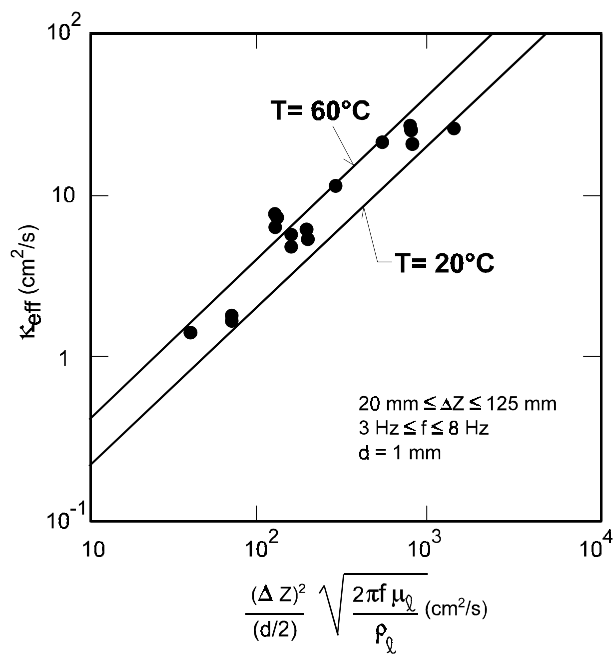

FIGURE 2. Experimental Effective Thermal Diffusivity

filled with hot liquid, this liquid is replaced, in the first half of the oscillation period, by liquid from the cold reservoir, except the thin (Stokes) boundary layer. Heat is then exchanged very effectively in radial direction between the hot Stokes layer and the cold core. Heat accepted by the core is removed to the cold reservoir in the second half of the oscillation period. The heat flow, via the liquid, between the reservoirs equals

$$Q = \lambda_{eff}\,(A_l/L)\,\Delta T = \rho_l\,Cp_l\,\kappa_{eff}\,(A_l/L)\,\Delta T, \tag{1}$$

where ΔT is the temperature difference between hot and cold reservoir. κ_{eff} is the effective thermal diffusivity , λ_{eff} is the effective thermal conductivity.

Figure 2 shows the experimentally determined effective thermal diffusivity as a function of tidal displacement and oscillation frequency, for a device with glass capillaries and water as working fluid. The solid lines are analytical predictions from the laminar theory. The figure indicates that the effective thermal conductivity via the liquid is

$$\lambda_{eff} = B\,\rho_l\,Cp_l\,\{(\Delta z)^2 / (d/2)\}\,(2\pi\,f\,\mu_l/\rho_l)^{1/2}. \tag{2}$$

The proportionality factor B, the tangent of the straight lines in figure 2, can be written as (Kurzweg, 1984,1985)

$$B = 2^{-5/2}\,Pr_l^{-1}\,\{B' + B'' - B'B''\,(1 + Pr_l^{1/2}) * \{Pr_l^{-1/2} - 2\,(1 + Pr_l)\}\}. \tag{3}$$

$$B' = Pr_l / (Pr_l - 1) \quad \text{and} \quad B'' = \{(1 - B')\,(Pr_l\,\kappa_l / \kappa_{wall})^{1/2} - B'\,\lambda_l / \lambda_{wall}\}*\{\lambda_l / \lambda_{wall} + (\kappa_l / \kappa_{wall})^{1/2}\}^{-1}. \tag{4}$$

An important conclusion for future design activities can be drawn by considering the equations (1) to (4): The highest values of proportionality factor B occur for small Prandtl number fluids and walls which are good thermal conductors (Kurzweg, 1985).

An alternative representation is obtained by defining an enhancement (proportionality) factor E for undeveloped oscillating flow in synchronised systems

$$(\lambda_{eff} / \lambda_l) - 1 = (\kappa_{eff} / \kappa_l) - 1 = (Pr_l\,\Delta z / 2d)^2\,E. \tag{5}$$

E depends on the dimensionless Womersley number

$$Wo_l = (d/2)\,(2\pi\,f\,\rho_l/\mu_l)^{1/2}. \tag{6}$$

It can be approximated by

$$E = Wo_l^4 / 24, \text{ for } Wo_l \ll 1, \quad \text{and } E = Wo_l\,(Wo_l - 2^{-1/2}), \text{ for } Wo_l \ll 1. \tag{7}$$

Figure 3 shows the enhancement factor E as a function of Womersley number, for decades of the Prandtl number: 0.1, 1 and 10 (Kurzweg, 1985; Watson, 1983).

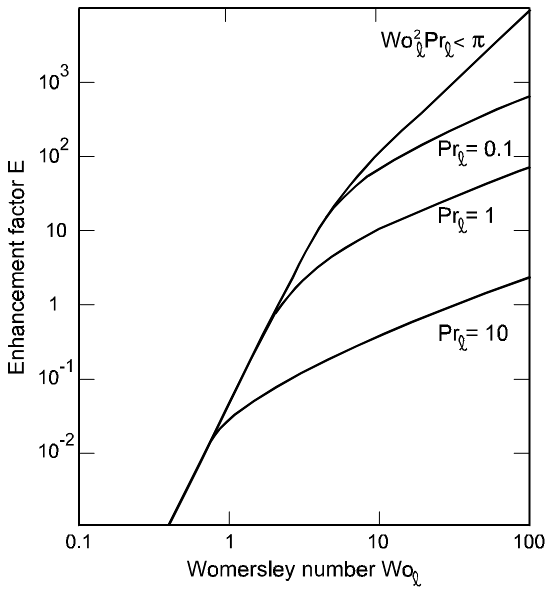

FIGURE 3. Enhancement Factor E Versus Wo and Pr

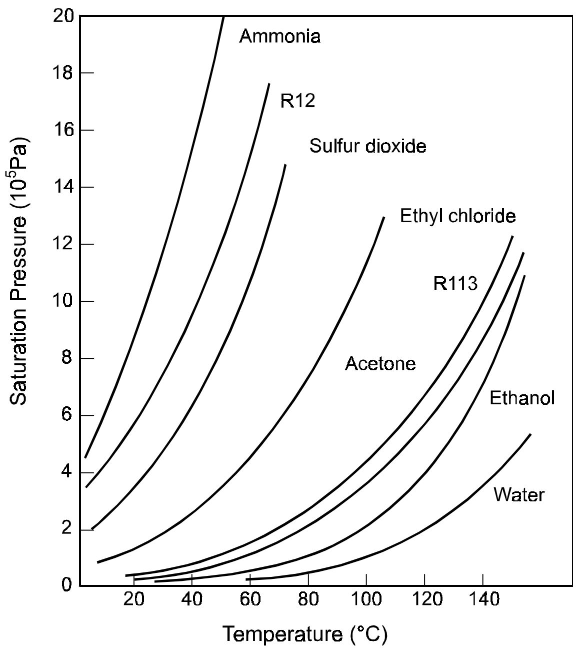

FIGURE 4. Liquid Vapour Pressure Versus Temperature

Inserting properties of water at 293 K (20 °C), d = 1 mm, Δz = 125 mm and f = 8 Hz (hence Pr_l = 6.9 and Wo_l = 3.55), yields (according to figure 3) for the above synchronised device an enhancement factor 0.14. The corresponding λ_{eff} (about $1.2 * 10^4$ W/m.K) means a power density slightly above $3 * 10^6$ W/m^2. This is confirmed by the experimental data: $2.9*10^6$ W/m^2 for a gradient of 280 K/m (Kurzweg, 1984). The total effective thermal conductivity is roughly 5000 W/m.K, if based on the total cross-sectional tube area, including insulating glass walls.

A similar expression has been derived for undeveloped oscillating flow in phase-shifted systems (Nishio, 1999):

$$\lambda_{eff} = \lambda_l \{1 + 0.707 (1 + Pr_l^{-1})^{-1} (1 + Pr_l^{-1/2})^{-1}\} * \{(\Delta z)^2/(d/2)\}\{(2\pi f \rho_l Cp_l/\lambda_l)^{1/2}\}. \qquad (8)$$

Inserting the properties of water at 20 °C, d = 1 mm, Δz = 125 mm and f = 8 Hz yields $\lambda_{eff} = 2.5*10^4$ W/m.K, being 2.5 times the value of the corresponding synchronised forced oscillatory flow heat transfer device.

Disadvantages of the concepts are poor current state of the art and power consumption of the shaker. The latter, more than 5 W in the described cases, strongly increases with oscillation frequency and tidal displacement. Disadvantage for some micro-gravity applications is the noise introduced by the shaker. Major advantage of the concepts is their variable conductance, which is adjustable via the frequency and tidal displacement from almost zero (liquid in rest in a poorly conductance structure) to values comparable to or even better than heat pipes. This makes such devices very useful for instance to drain huge amounts of thermal power from a hot vessel for nuclear reactor cooling, in an emergency case.

Pulsating Loops & Other Oscillating/Pulsating Two-Phase Heat Transfer Devices

Two oscillating two-phase heat transfer devices can be distinguished: Pulsating two-phase heat transfer loops and devices like the pulsating or meandering heat pipe or flat swinging heat pipe. These systems have in common that the operation is driven only by vapour pressure differences induced by the heat to be transported: They don't need an additional power source. Figure 4 depicts the temperature dependent saturation pressure of some candidate working fluids. It illustrates that when designing a device, one has to select a fluid with a high saturation pressure gradient (dp/dT) in the operating (temperature) range, as higher system pumping pressures correspond with higher dp/dT.

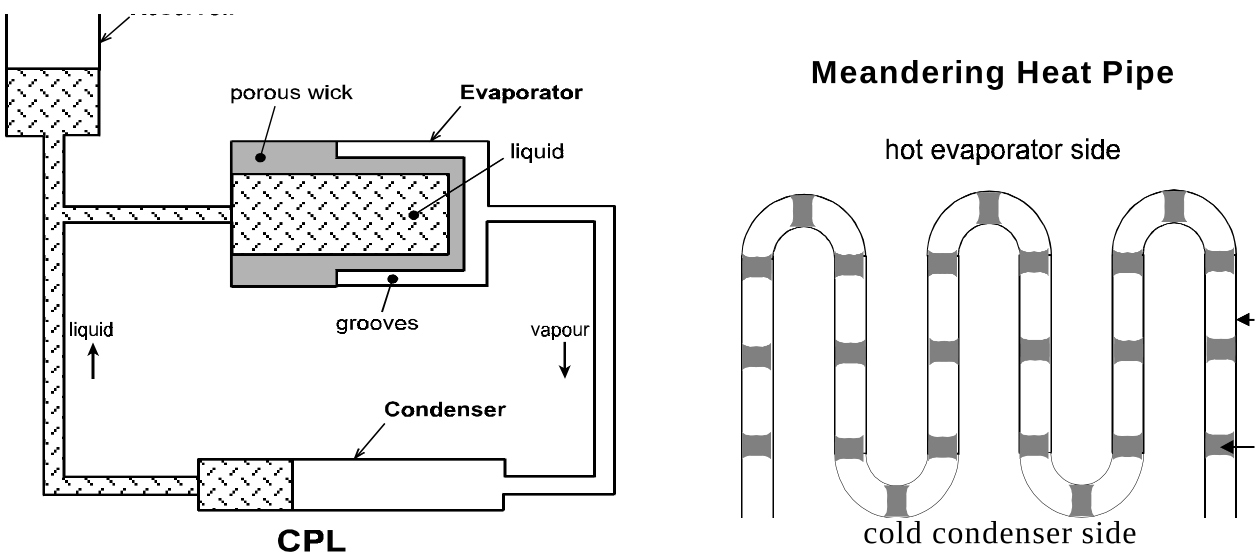

FIGURE 5. Schematics of a Capillary Pumped Loop (left) and a Meandering Heat Pipe (right)

Pulsating two-phase heat transfer loops already were proposed many years ago (Tamburini, 1978; Lund, 1993; Borodkin, 1995). These systems are just two-phase loops driven by vapour pressure differences, instead of mechanical or capillary pumping action. The vapour pressure pumping action is realised by incorporating in a normal loop (schematically depicted in figure 5) two one-way valves: one at the entrance, the other at the exit of any evaporator. Power fed to the system increases the internal vapour pressure in the section between the two valves till the exit valve will open and the loop starts to run, also opening the second valve to let sub-cooled liquid flow into the pumping section. After some pressure decay the valves will close re-starting the process. A careful design will certainly lead to a properly performing heat transfer device. Advantages of such loops are the driving mechanism heat is being transferred, without additional power source), high heat transport capability, self-priming capability and capability to work against gravity. A disadvantage is the pulsating operation, as pulsating heat and mass transfer, accompanied by temperature variations and possibly also vibrations (g-jitter), will make the system not attractive for some micro-gravity applications.

The majority of the remarks on two-phase thermal-gravitational modelling and scaling made in the summarizing articles are directly applicable for pulsating two-phase loops. The only new issue to be accounted for is the proper design of the desired driving pressure, which is directly coupled to the corresponding difference of the saturation temperatures in evaporator and condenser. It is obvious that for substantial temperature differences a constant fluid property approach will not be maintainable. This considerably complicates the equations and their solution.

Other oscillating/pulsating devices have many names, like bubble-driven heat transfer device (Nishio, 1999), pulsating or meandering heat pipe (Hosoda 1999; Nishio, 1998), capillary heat pipe or capillary tunnel heat pipe (Wong, 1999), flat swinging heat pipe (Es, 2000), spirally wound or serpentine-like heat pipe (Terpstra, 1987). Figure 5 also shows a schematic of a section of such an oscillating device, called looped (Maezawa, 1995) or closed-loop (Hosoda, 1999), if the two legs at each end are not dead ends but interconnected, thus creating a closed loop configuration. If the latter configuration has a spring-like geometry like the arrangement discussed by Maezawa (1995) and Smirnov (1999), the operation of the device has been frequently observed to stabilise after a certain start-up period, producing a periodic pumping of almost constant frequency into one direction (Smirnov, 1999). Such behaviour is very similar to the behaviour of the pulsating heat transfer loops, which similarity suggests that the function of valves in the pulsating heat transfer loops is now delivered by stick-slip conditions of the slug-plug distribution of the working fluid in the closed-

Relevance of π-numbers for thermal-Gravitational scaling of two-phase loops	Liquid Sections		Evaporator Swirl & Capillary	Vapour & 2-Phase Sections	Condenser
	Adiabatic	Heating/Cooling			
π_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	•	•	•	/•	•
π_6 = S = slip factor = v_v/v_l			•	•	•
π_7 = density ratio = ρ_v/ρ_l			•	•	•
π_8 = viscosity ratio = μ_v/μ_l			•	•	•
$\pi_9 = We_l = (\rho v^2 D/\sigma)_l$ = inertia/surface tension			•	/•	•
$\pi_{10} = Pr_l = (\mu Cp/\lambda)_l$		•	•		•
$\pi_{11} = Nu_l = (hD/\lambda)_l$ = convection/conduction		•	•		•
$\pi_{12} = \lambda_v/\lambda_l$ = thermal conductivity ratio			•		•
$\pi_{13} = Cp_v/Cp_l$ = specific heat ratio			•		•
$\pi_{14} = \Delta H/h_{lv}$ = Boil =enthalpy nr. = X = quality		•	•	•	•
$\pi_{15} = Mo_l = (\rho_l \sigma^3/\mu_l^4\ g)$ = capillarity/buoyancy			•	/•	•
$\pi_{16} = Ma = v/(\partial p/\partial \rho)_s^{1/2}$			•	•	•
$\pi_{17} = (h/\lambda_l)(\mu_l^2 g)^{1/3}$			•		•
$\pi_{18} = L^3 \rho_l^2\ g\ h_{lv}/\lambda_l \mu_l (T-T_o)$			•		•

loop spring-like meandering structure, whose heating and cooling sections have also a certain periodicity. Anyhow, slug-plug distribution is essential for these oscillating devices, which can be equipped with or without valves.

The velocities of bubbles and slugs in a tube are governed by buoyancy, liquid inertia, liquid viscosity and surface tension forces, in the general case of bubbly or slug-plug flow in a gravity field. This means that properly chosen dimensionless groups can be very helpful to discuss the aspects of slug-plug flow in oscillating devices (Wallis, 1969). These groups can be used together with the groups, shown in the table above and in the figures 6 and 7, taken from dimension-analytic considerations discussed in the overview articles and their references (Delil, 2000b, 2001, 1989).

The condition of slug-plug distribution determines the maximum inner tube diameter (Wallis, 1969; Bretherton, 1961).

$$D_{max} = 1.836\, g^{-1/2}\, [\sigma/(\rho_l - \rho_v)]^{1/2} \approx 1.836\, g^{-1/2}\, (\sigma/\rho_l)^{1/2}\,, \qquad (9)$$

for $\rho_l >> \rho_v$. Consequently, the thermal-gravitational scaling of the inner tube diameter can be derived from figure 7.

The slug-plug condition also sets the lower limit of the slug (bubble) size: It shall be at least 0.6 times the tube diameter (Wallis, 1969; Hosoda, 1999). This extra requirement has an impact on the liquid filling ratio $(1 - \alpha)$. If not fulfilled, the bubbles will be too small to maintain the slug flow pattern, characterised by high heat transfer densities. The resulting bubbly flow leads to a far less efficient heat transport.

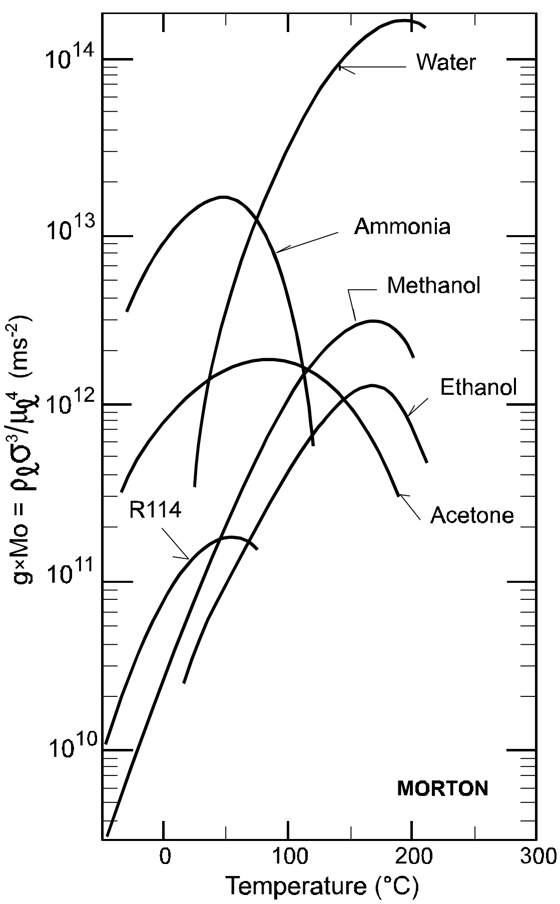

FIGURE 6. $\rho_l.\sigma^3/\mu_l^4$ Versus Temperature for Six Fluids

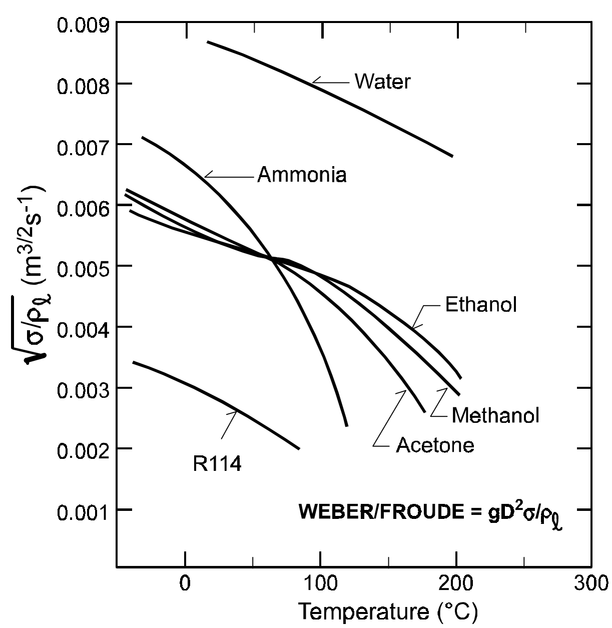

FIGURE 7. $(\sigma/\rho_l)^{½} = D.g^{½}/(We/Fr)^{½}$ Versus Temperature

Other useful plots of dimensionless numbers (Wallis, 1969) are shown in the figures 8 and 9. Figure 8 shows the dimensionless velocity v* as a function of Morton number Mo (see Table) and Eötvös number Eö or Bond number Bo:

$$v^* = v\,[g\,D\,(1 - \rho_v/\rho_l)]^{-½} \approx v\,/\,(g\,D)^{½}\,, \tag{10}$$

$$Eö = 4\,Bo = We/Fr = g\,D^2\,(\rho_l - \rho_v)/\,\sigma \approx.\ g\,D^2\,\rho_l\,/\sigma. \tag{11}$$

Figure 9 depicts experimental data in the alternative plotting (Wallis, 1969): v* as a function of the inverse viscosity number Mu, for different values of Archimedes number Ar. The dimensionless numbers Mu and Ar are given by:

$$Mu = \mu_l\,[g\,D^3\,(\rho_l - \rho_v)\,\rho_l]^{-1/2} \approx \mu_l\,(g\,D^3\,\rho_l^2)^{-1/2}, \tag{12}$$

$$(Ar)^2 = Mo = (\rho_l\,\sigma^3/\mu_l^4\,g)\,/\,(1 - \rho_v/\rho_l] \cong \rho_l\,\sigma^3/\mu_l^4\,g\,. \tag{13}$$

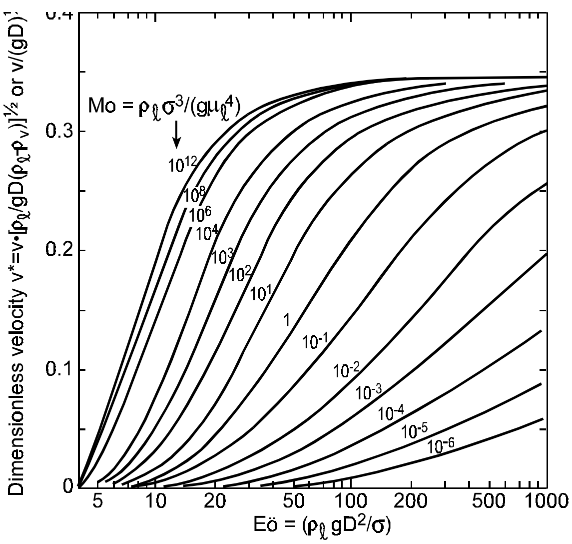

FIGURE 8. The Dimensionless Velocity v* as a Function of the Morton Number Mo and the Eötvös Number Eö

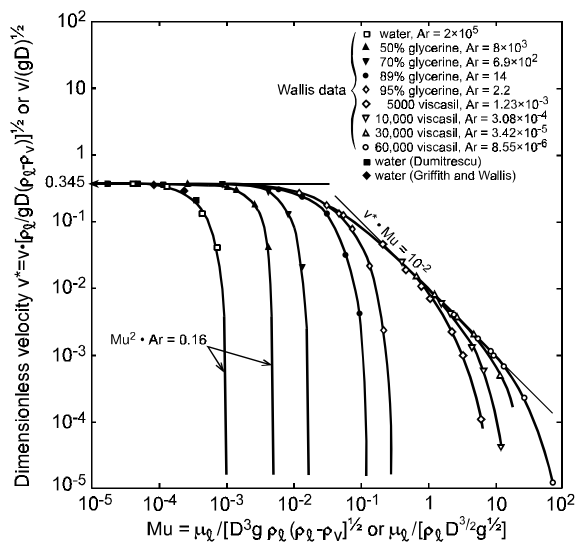

FIGURE 9. Experimental Data for th Dimensionless velocity v* as a Function of Dimensionless Inverse Viscosity Mu

The three asymptotes shown are: v* = 0.345 for Eö > 100 and Mu < 10^{-3} (the inertia dominant domain), v*. Mu = 10^{-2} for Eö > 100 and Mu > 0.5 (the viscosity dominant domain), and Mu^2. Ar = 0.16 and Eö < 3.37 (the surface tension dominant domain). The last domain is the most important for the oscillating devices considered here, as Eö < 3.37 straightforwardly leads to the maximum tube diameter (equation 9) and dominating surface tension means that plug-slugs do not move, if there is no (thermal) power input. In this domain slug flow is guaranteed by surface tension, if slug bubbles have diameters of at least 0.6 times the tube diameter.

Some quantitative considerations on these pulsating two-phase heat transfer devices are presented next, in order to get a certain feeling for or a better understanding of how to design, scale and test such devices. Geometrical data and some performance figures of the earlier described single-phase oscillating heat transfer device will be used.

EXPERIMENTAL DATA FROM LITERATURE

Several recently published results of experiments confirm the remarks made up to now. They will be summarised first, before being used to establish a simple quantitative model for oscillating (pulsating) heat transfer devices.

Super-gravity experiments were done on a 4.5 m diameter centrifuge table with a non-looped device, consisting of 23 turns of 0.42 m long, 1.1 mm internal diameter, stainless steel capillaries (Kiseev, 1999). The working fluid was acetone, the liquid filling ratio (1–α) was 0.6. The length of the sections was 120 mm for the heating and the adiabatic section, 180 mm for the cooling section. Experiments were done in both gravity-assist and anti-gravity conditions. Experimental data confirms that gravity influences the pressure drop and the corresponding temperature drop across such devices. For example, while continuously transporting the maximum power 40 W, the finally reached stable heater section temperature increased from 403 K (130 °C) in the 6-g-assist case, via 433 K at 0-g and 458 K at 6-g anti-gravity, to 473 K (200 °C) at 12-g anti-gravity. There was no evaporator dry-out for all acceleration conditions specified. The above tests, and vibration tests (frequency 0 to 16 kHz, acceleration 0 to 15 g, amplitude 0 to 7 mm, inclination 0 - 180°) showed that these oscillating devices are not sensitive for acceleration fields. It is to be noted that the centrifuge table tests properly simulate high-g conditions in aircraft and spinning satellites. But the fluid in lines oriented in a radial direction, experiences an assisting or counteracting acceleration gradient (as a function of radial position and rotation speed. This gradient is absent in the (super)gravity environment of planets.

Experiments with a 4-turn, 396 mm long, glass, closed-loop meandering device showed good performance also (Nishio, 1999), even for capillary diameters larger than the maximum diameter D_{max}, which follows from equation (9). But poor performance was observed for diameters below 0.6 D_{max}. Results of experiments with a 2.4 mm diameter tube test device showed for all liquids a maximum performance at a liquid filling ratio ($1-\alpha$) around 0.35. For filling ratios up to 0.8, the performances are slightly below these maximum performances, above 0.8 they strongly decrease. Typical values of the performance are about 1.25 W/K for water and 0.8 W/K for ethanol and R141b, for a temperature drop of 50 K, within the working range 10 - 100 °C (283 – 373 K). The total wall resistance of the heating and cooling sections were calculated to be 1.25 K/W per tube.

A typical periodic vapour-plug propagation phenomenon was reported to occur in a 10-turn water system (Hosada, 1999): 2.5 Hz for a temperature drop of 30 K at the very high liquid filling $1-\alpha = 0.95$.

DEFINITION OF PULSATING DEVICES FOR TESTING

A first step in a logic approach to develop pulsating two-phase test devices is to dimension these such that test outcomes can be compared directly to the performance data of the described synchronised oscillating single-phase device.

A second step is to assume that the single-phase device (Fig. 1) consists of 85 identical cylindrical channels with an average internal diameter d = 1 mm. This represents the actual configuration of 31 glass capillaries (1 mm ID) and 54 triangular channels present between these capillaries, thus yielding the total liquid cross-sectional area $A_l = 85*\pi/4 \approx 67$ mm^2. For simplicity reasons the tube length is assumed to be equal to the displacement length: $L = \Delta z = 125$ mm. The power transported by each capillary can be calculated from the data presented. For the maximum transport case, being for frequency f = 8 Hz and temperature difference $\Delta T = 56$ K, this becomes $(\pi/4)\ d^2\ 2.9\ 10^6 = (\pi/4)\ 10^{-6} \approx 2.3$ W.

The third step is the simplification to consider the pulsating two-phase device (Fig. 5 to be, in essence, a configuration of identical, parallel elements, each one transporting the same amount of power, driven by the vapour pressure difference between the heat input (evaporator) section and the cooling (condenser) section. In addition, the working fluid and dimensioning of the two-phase and single-phase devices are identical: The working fluid is water, the capillary diameter d = 1 mm, the evaporator and condenser length are $L_e = L_c = L = 125$ mm. In the first approximation, it is assumed that there is no adiabatic section. The main differences between the two devices pertain to the driving mechanisms, the heat transfer processes and the heat transfer locations.

A mechanical actuator is the driver of the oscillating axial movement of the liquid in the single-phase device. Only specific heat is exchanged over the entire capillary tube length L in two sequential radial (conduction) steps. In the first half of the period, heat is transferred in radial direction from the hot fluid in the core to the thin Stokes layer (and the tube wall). In the second half of the period, this heat is moved back to the cold fluid brought into the core.

In the two-phase device heat is simultaneously exchanged in radial direction, mainly by conduction, plus some convection, at two different locations. The heat is fed via the wall to the working fluid in the hot input (evaporator) section. The heat is extracted from the fluid via the wall in the cold section (condenser). This heat transfer, via the specific heat of the liquid, looks more or less identical in the two systems. The transfer difference (i.e. two-step sequential at one location, respectively simultaneous heat addition and extraction at two different locations) suggests that it is reasonable to assume that the transported power in the two-phase case is, for $\Delta T = 56$ K, twice the value for a single-phase capillary (4.6 W). Alternatively, it can also be assumed that $\Delta T = 28$ K only, at a power transport of 2.3 W.

However, there is an additional latent heat transfer contribution in the two-phase device: The heat transported via the vapour bubble that grows in the heat input section (evaporator) by evaporation of a liquid micro-layer (Ref. 38). This bubble collapses in the condenser, releasing its latent heat. The pressure difference, between the (super-heated) vapour in the evaporator and the saturated vapour in the condenser, is the driving force moving the hot liquid slug from evaporator to condenser, plus moving at the same moment a similar cold slug from condenser back to the same or a neighbouring evaporator. The power transported by latent heat can be obtained by calculating the energy needed to create 8 bubbles, of length L and diameter d, per second. Consequently one obtains $8\ (\pi/4)\ d^2\ L\rho_v\ h_{lv} = 8\ (\pi/4)\ 10^{-6}\ (0.125)\ (0.2)\ (2.25 * 10^6 \approx$ 0.45 W, which constitutes a minor, but non-negligible contribution.

The pressure head across the capillary single-phase water system can be calculated as follows. The displacement of 125 mm at frequency 8 Hz yields a liquid velocity v = 2 m/s. For water around 300 K, the Reynolds number Re_l is around 2000, which means laminar flow. Consequently the required pressure drop is 8 kPa, according to the equation

$$\Delta p = 4 * (16 / Re_l)\ (L / D)\ (\rho_l\ v^2 / 2). \tag{14}$$

In the corresponding two-phase device the required pressure difference has to be far larger, because of several reasons. In the first place twice the single-phase device mass (a hot and a cold slug) has to be moved. Secondly, this double mass has to be forced through a 180 degrees bend instead a straight channel. Further, the length of the adiabatic section (L_a) is of course, in reality, never equal to zero. Finally, the process concerns all except fully developed flow, hence there is a liquid acceleration term to be added.

To get a feeling for the magnitude of these pressure enlarging effects, the length L_a is taken to be also 125 mm (L_a = L), as example. This has impact on the contribution of the power transport via the latent heat of evaporation. This contribution will be around 0.9 W, since the length of each of the 8 vapour bubbles, being generated and collapsing in one second, is L_a + L (hence 2L, instead L). The average liquid velocity becomes v = 4 m/s. For water around 300 K, the Reynolds number Re_l now lies around 4000, which means turbulent flow. Consequently the required pressure drop has to be calculated according to

$$\Delta p = 4\ (0.0791)\ Re^{-1/4}\ (L / D)\ (\rho_l\ v^2 / 2). \tag{15}$$

As discussed in textbooks, the effect of the two bends can be accounted for by adding an extra length of 50 D. The pressure drop can now be calculated, according to equation (15), by inserting the different parameter values and by replacing L by 2 (2L + 50 D). The result is 480 kPa. To be complete an inertia term, accounting for the acceleration of the slugs eight times per second, has to be added: $8\ (\rho_l\ v^2 / 2)$, hence 16 kPa, yielding about 500 kPa for the pressure difference required. Consequently, it can be concluded from the water curve in figure 4, that the two-phase device has to operate at a hot section temperature of at least 350 K (80 °C), to be able to deliver the pressure drop required. Figure 4 makes also clear that more or less comparable power can be transported by ammonia, R12, acetone, etc. Though at comparable ΔT's, this will be realised at far lower operating temperatures, as these fluids show a steeper dp/dT-relation.

The results of the above simple approach and of the detailed modelling of the physical processes, including mass-spring simulations, currently is compared and will be compared in the near future to experimental data, resulting from further experimenting at NLR. The experimental activities include many high-acceleration experiments on a rotation table. The experiments have been, are, and will be carried out both with all-metal devices, and with all-glass devices (Es, 2000). Additional experiments will be executed using a helical (spring-like) configuration of transparent (PTFE or polyethylene) flexible tubing, equipped with a simple one-way valve to influence direction and frequency of the periodic behaviour of a closed-loop configuration. Experiments pertain to various working fluids and different locations of hot and cold sections, to different lengths of adiabatic section, to various orientations, and to different acceleration levels in various directions.

CONCLUDING REMARKS

In conclusion it is remarked that the presented information summarises the approach for the thermal-gravitational scaling of two-phase systems in general. Results of similarity considerations are given. Modelling results are discussed, including comparison with experimental data. The discussions include gravity-assist system aspects and issues of operation against gravity or super-gravity. The systems considered are capillary pumped and mechanically pumped two-phase loops, pulsating two-phase loops, and other single-phase and two-phase oscillating (pulsating) devices. A simple rationale to get a (quantitative) feeling for the development of pulsating devices for useful experiments is given.

Finally two very critical issues are to be stressed, since they are not mentioned at all in literature (most probably because one did not recognise these). First, planetary super-gravity has a constant magnitude felt in each part of any (two-phase heat transfer) device. This principally differs from the “super-g” accelerations in spinning satellites, in military combat aircraft and on turntables. In the latter, the g-vectors have gradients across a device. Those gradients depend on local position and orientation with respect to the rotation axis. Second, in pulsating (pressure driven) two-phase loops heat transfer is by latent heat of evaporation/condensation. This means that the working fluid selection will be based on high latent heat, in addition to high a dp/dT to deliver a minimum temperature drop driving force for the system. The other pulsating two-phase devices also require a fluid with a high dp/dT. But as it was shown that in the latter devices the majority is by caloric (specific) heat of the liquid, it is clear that a high specific heat, high dp/dT fluid will be preferred. The question is even to be raised whether there exists an ideal high specific heat, high dp/dT fluid, having in also a low latent heat. This because low latent heat means that, as the driving bubble growth is fast, the pulsation frequency and heat transport efficiency will increase. If such a fluid does not exist, there has certainly to be looked at a fluid with an optimum combination of the above properties.

NOMENCLATURE

A	area ($m^{2)}$
Ar	Archimedes number = $Mo^{½}$ (-)
B	proportionality factor (-)
Bo	Bond number = $g\ D^2\ \rho_l\ /4\sigma$ (-)
Boil	boiling number = $\Delta H/h_{lv}$ = Boil (-)
C	conductance (W/K)
Cp	specific heat at constant pressure (J/kg.K)
D	diameter (m)
d	diameter of capillary or of curvature (m)
E	enhancement factor (-)
Eö	Eötvös number = $g\ D^2\ \rho_l\ /\sigma$ (-)
Eu	Euler number = $\Delta p/\rho v^2$ (-)
f	frequency (Hz)
Fr	Froude number = v^2/gD (-)
g	gravitational acceleration (m/s^2)
H	enthalpy (J/kg)
h	heat transfer coefficient ($W/m^2.K$)
h_{lv}	latent heat of vaporisation (J/kg)
j_l	superficial liquid velocity = $v_l\ (A_l\ /A_t)$ (m/s)
j_v	superficial vapour velocity = $v_v\ (A_v\ /A_t)$ (m/s)
k	thermal conductivity (W/m.K)
L	length (m)
Ma	Mach number = $v/(\partial p/\partial \rho)_s{}^{1/2}$ (-)
Mo	Morton number = $\rho_l\sigma^3/\mu_l^4\ g$ (-)
Mu	Inverse viscosity number $\approx \mu_l\ (g\ D^3\ \rho_l^2)^{-1/}$ (-)
$\dot{m}$	mass flow rate (kg/s)
Nu	Nusselt number = hD/λ (-)
p	pressure (Pa = N/m^2)
Pr	Prandtl number = $\mu Cp/\lambda$ (-)
Q	power (W)
Q	heat flux (W/m^2)
Re	Reynolds number = $\rho vD/\mu$ (-)
S	slip factor, ratio of velocities of the phases (-)
T	temperature (K = 273 + °C)
t	time (s)
v	velocity (m/s)
v*	dimensionless velocity = $v\ /\ (g\ D)^{½}$ (-)
v	velocity (m/s)
We	Weber number = $\rho v^2 D/\sigma$ (-)
Wo	Womersley number = $(d/2)\ (2\pi\ f\ \rho_l\ /\ \mu_l\ )^{1/2}$ (-)
X	vapour quality = vapour mass fraction (-)
z	axial or vertical co-ordinate (m)
α	vapour/void fraction (volumetric) (-)
Δ	difference, drop (-)
δ	surface roughness (m)
κ	thermal diffusivity (m^2/s)
λ	thermal conductivity (W/m.K)
μ	viscosity ($N.s/m^2$)
ν	angle (with respect to gravity) (rad)
π	dimensionless number (-)
ρ	density (kg/m^3)
σ	surface tension (N/m)

Subscripts

a	acceleration, adiabatic, axial
ch	capillary channel
c	condenser, cold
e	evaporator
eff	effective
f	friction
g	gravitation
h	hot
I	inner
l	liquid

m momentum
M model
o reference condition, or outer
p pore
P prototype
r radial, or radius (m)
s entropy
t total
tp two-phase
v vapour
w water

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Appendix

Compilation of Slides

D127-01N

Issues of Various Two-Phase Heat Transfer Devices in Gravity Environments Ranging from Micro- Gravity to Super-Gravity

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Abstract

- *A number of earlier publications describe research carried out at NLR on the thermal-gravitational modelling and scaling of two-phase heat transport systems for spacecraft applications. They dealt with mechanically and capillary pumped two-phase loops. The scaling activities, based on dimension-analytical similarity considerations, pertained to pure geometric, pure fluid to fluid, or hybrid scaling between a prototype system and a model at the same gravity level, and between a prototype in micro-gravity and a model on earth. Recent publications also include the scaling aspects of a prototype loop for a Moon or a Mars base application and a terrestrial model.*
- *The extension of the work, also discussed here, was carried out in the last couple of years. It concerns scaling to super-gravity levels, and was done because a promising super-gravity application for (two-phase) heat transport systems can be the cooling of high power electronics in spinning satellites and in military combat aircraft. In such aircraft, power electronics can be exposed, during manoeuvres, to transient accelerations up to say 12 g (120 m/s^2). The discussions include pulsating two-phase loops and oscillating (pulsating) heat transfer devices, for which promising applications in different gravity environments have been identified*

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Contents

- *Π-Numbers for Two-Phase Heat Transport Loops*
- *Thermal-Gravitational Scaling Possibilities*
- *Equations for Two-Phase Flow and Heat Transfer*
- *Constituents of the Total Pressure Gradient*
- *Results for Adiabatic and Condensing Annular Flow*
- *Flow Pattern Map Issues and their Consequences*
- *Super-Gravity and Anti-Gravity Issues*
- *Pulsating Two-Phase Loops*
- *Single-Phase and Two-Phase Oscillating Devices*
- *Possibilities for Experimental Verification*
- *Concluding Remarks*
- *References*

3

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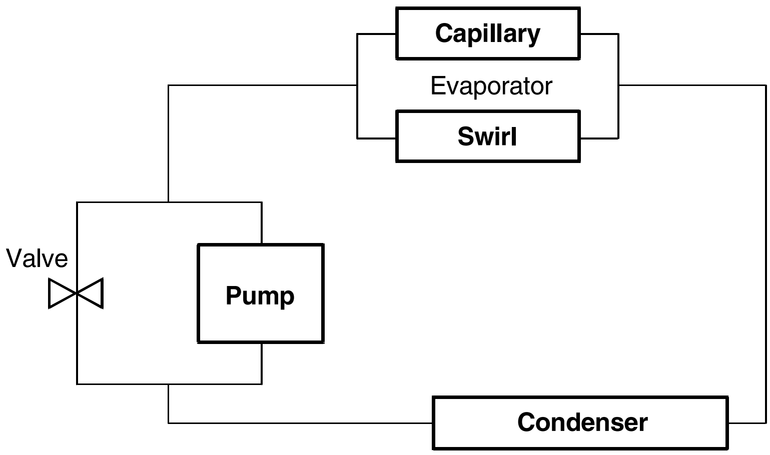
Two-Phase Loop Schematic



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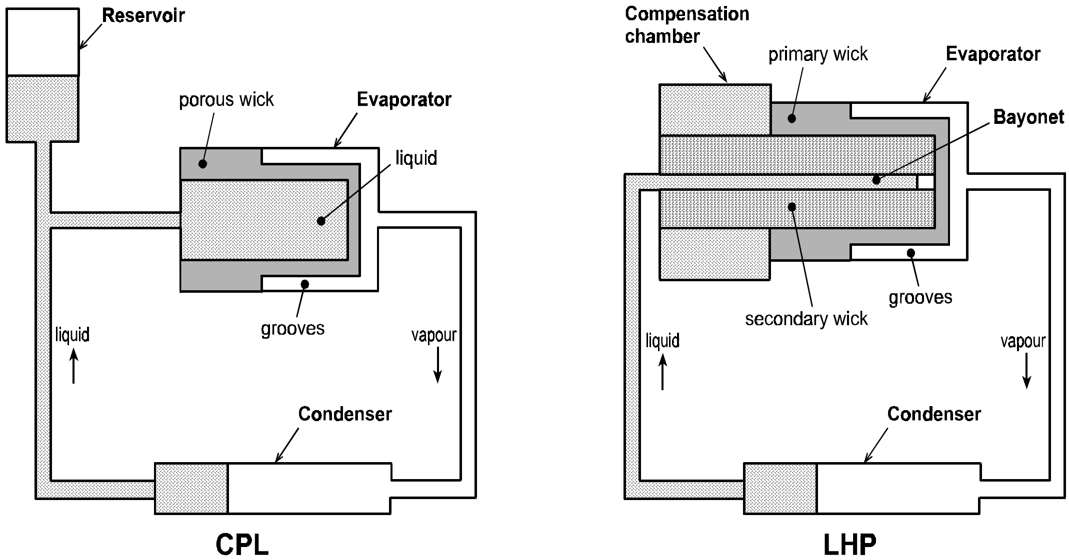
Schematics of Capillary Pumped Loop and Loop Heat Pipe



5

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Dimension Analysis & Similarity Considerations

π-number relevance for thermal-gravitational scaling of two-phase loops

	Liquid Parts		Evaporators Swirl & Capillary	Non-liquid Lines Vapour/2-Phase	Condensers
	Adiabatic	Heating/Cooling			
π_1 = D/L = geometry	•	•	•	•	•
π_2 = $Re_l = (\rho v D/\mu)_l$ = inertia/viscous	•	•	•	•	•
π_3 = $Fr_l = (v^2/gD)_l$ = inertia/gravity	•	•	•	/•	•
π_4 = $Eu_l = (\Delta p/\rho v^2)_l$ = pressure head/inertia	•	•	•	•	•
π_5 = cos ν = orientation with respect to g	•	•	•	/•	•
π_6 = S = slipfactor = v_v/v_l			•	•	•
π_7 = density ratio = ρ_v/ρ_l			•	•	•
π_8 = viscosity ratio = μ_v/μ_l			•	•	•
π_9 = $We_l = (\rho v^2 D/\sigma)_l$ = inertia/surface tension			•	/•	•
π_{10}= $Pr_l = (\mu Cp/k)_l$		•	•		•
π_{11}= $Nu_l = (hD/k)_l$ = convective/conductive		•	•		•
π_{12}= k_v/k_l = thermal conductivity ratio			•		•
π_{13}= Cp_v/Cp_l = specific heat ratio			•		•
π_{14}= $\Delta H/h_{lv}$ = enthalpy number = X = quality		•	•	•	•
π_{15}= $Mo_l = (\rho_l \sigma^3/\mu_l^4\, g)$ = capillarity/buoyancy			•	/•	•
π_{16}= $Ma = v/(\partial p/\partial \rho)^{1/2}{}_s$			•	•	•
π_{17}= $(h/k_l)(\mu_l^2 g)^{1/3}$			•		•
π_{18}= $L^3 \rho_l^2\, g\, h_{lv}/k_l \mu_l (T-T_o)$			•		•

6

List of Symbols

A	area (m^2)
Ar	Archimedes number (-)
B	proportionality factor (-)
Bo	Bond number (-)
Boil	boiling number (-)
C	conductance (W/K)
Cp	specific heat at constant pressure (J/kg.K)
D	diameter (m)
d	diameter of capillary or of curvature (m)
E	enhancement factor (-)
Eö	Eötvös number (-)
Eu	Euler number (-)
Fr	Froude number (-)
f	frequency (Hz)
g	gravitational acceleration (m/s^2)
H	enthalpy (J/kg)
h	heat transfer coefficient ($W/m^2.K$)
h_{lv}	latent heat of vaporisation (J/kg)
j	superficial velocity (m/s)
k	thermal conductivity (W/m.K)
L	length (m)
Ma	Mach number (-)
Mo	Morton number (-)
Mu	Inverse viscosity number (-)
m	mass flow rate (kg/s)
Nu	Nusselt number (-)
p	pressure (Pa = N/m^2)
Pr	Prandtl number (-)
Q	power (W)
q	heat flux (W/m^2)
Re	Reynolds number (-)
T	temperature (K or °C)
r	radius (m)
S	slip factor (-)
t	time (s)
v	velocity (m/s)
v*	dimensional velocity (-)
We	Weber number (-)
Wo	Womersley number (-)
X	vapour quality (-)
z	axial or vertical co-ordinate (m)

7

Greek Symbols & Subscripts

Greek

α	vapour/void fraction (volumetric) (-)
β	constant in eq. (13) (-)
δ	surface roughness (m)
Δ	difference, drop (-)
κ	thermal diffusivity (m^2/s)
λ	thermal conductivity (W/m.K)
μ	viscosity ($N.s/m^2$)
ν	angle (with respect to gravity) (rad)
π	dimensionless number (-)
ρ	density (kg/m^3)
σ	surface tension (N/m)

Subscripts

a	acceleration, adiabatic, axial
c	condenser, cold
ch	(capillary) channel
e	evaporator
eff	effective
f	friction
h	hot
I	inner
g	gravitation
l	liquid
m	momentum, model
o	reference condition, outer
p	pore, prototype
r	radial
s	entropy
t	total
tp	two-phase
v	vapour
w	water

8

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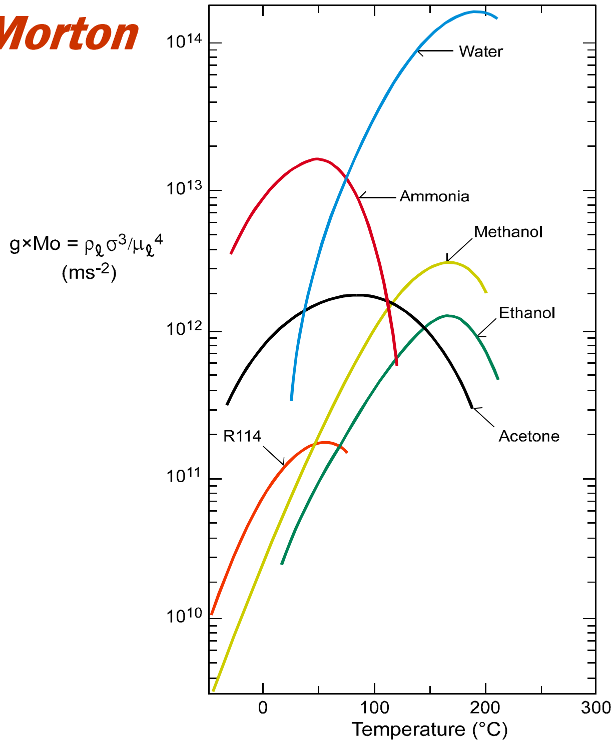
Morton



$\rho_l \cdot \sigma^3/\mu_l^4$ versus temperature for some fluids

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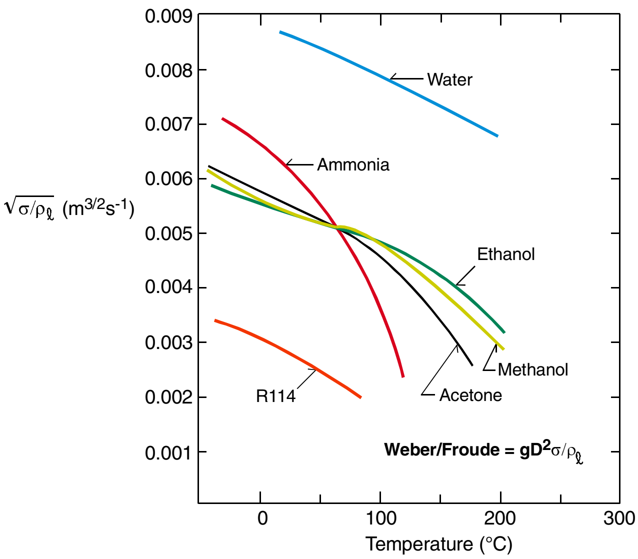
$(\sigma/\rho_l)^{1/2}$ = D.G$^{1/2}$ • (We/Fr)$^{-1/2}$ versus temperature for various fluids



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Identified Possibilities for the Thermal-Gravitational Scaling of Two-Phase Heat Transport Systems

- ***At the same gravity level***
- ***Between earth gravity and low-g environments***
- ***Between different low-g environments***
- ***For Moon and Mars base applications***
- ***For high-g accelerations applications***

11

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Constitutive Equations for Two-Phase Annular Flow

- *Total Pressure Gradient* $(dp/dz)_t = (dp/dz)_f + (dp/dz)_m + (dp/dz)_g$
  - *friction* $(dp/dz)_f = -(32m^2/\pi^2\rho_v D^5)(0.045/Re_v^{0.2})[X^{1.8}+5.7(\mu_l/\mu_v)^{0.0523}(1-X)^{0.47}X^{1.33}(\rho_v/\rho_l)^{0.261}$ $+8.1(\mu_l/\mu_v)^{0.105}(1-X)^{0.94}X^{0.86}(\rho_v/\rho_l)^{0.522}]$
  - *momentum* $(dp/dz)_m = -(32m.^2/p^2\rho_v D^5)(D/2)(dX/dz)[2(1-X)(\rho_v/\rho_l)^{2/3} + 2(2X-3+1/X)(\rho_v/\rho_l)^{4/3}$ $+(2X-1-\beta X)(\rho_v/\rho_l)^{1/3} + (2\beta-\beta X-\beta/X)(\rho_v/\rho_l)^{5/3} + 2(1-X-\beta+\beta X)(\rho_v/\rho_l)]$
  - *gravity* $(dp/dz)_g = (32m.^2/p^2\rho_v D^5)\{1-[1+(\rho_v/\rho_l)^{2/3}(1-X)/X]^{-1}\}[p^2D^5 g \cos\nu\,(\rho_l-\rho_v)\,\rho_v/32m^2]$
- *Void - Quality Relation* $(1-\alpha)/\alpha = S(\rho_v/\rho_l)X/(1-X)$
- *Simplified Zivi-Correlation for Slip Factor* $S = (\rho_l/\rho_v)^{1/3}$
- *Condensation Heat Transfer* $m.h_{lv}(dX/dz) = -h\pi D[T(z)-T_s]$
- $h = 0.018(\rho_l\rho_l^{1/2} D^{1/2}/\mu_l)Pr_l^{0.65}|-(dp/dz)_t|^{1/2} + R(4\rho_l/D)/\ln[1+(\rho_v/\rho_l)^{2/3}(1-X)/X])$ and $0 < R < 1$
- *Combination Yields* $F(dX/dz, X) = 0 \Rightarrow X(z)$ & Δp_t integrating $(dp/dz)_t dz$ from 0 to L_C

12

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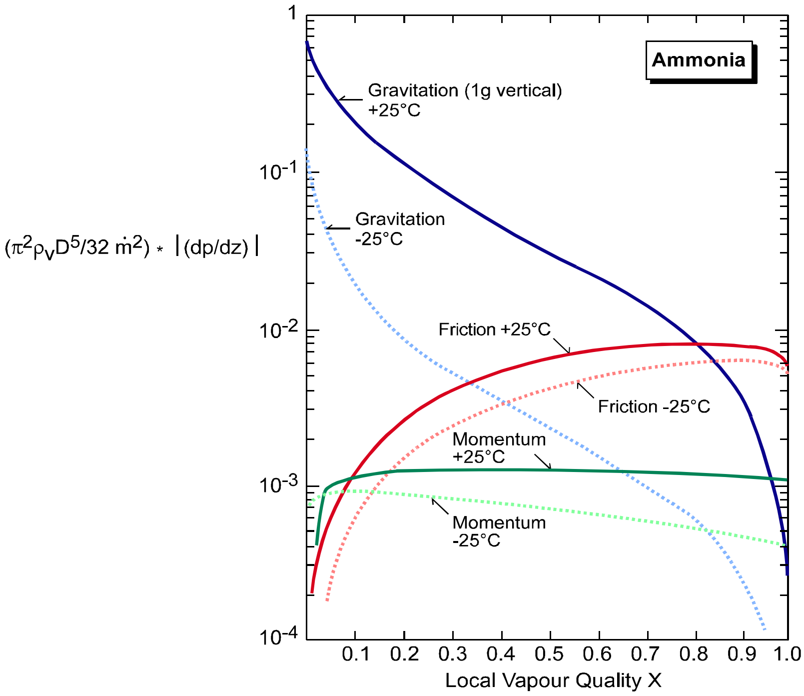
Pressure Gradient Constituents



13

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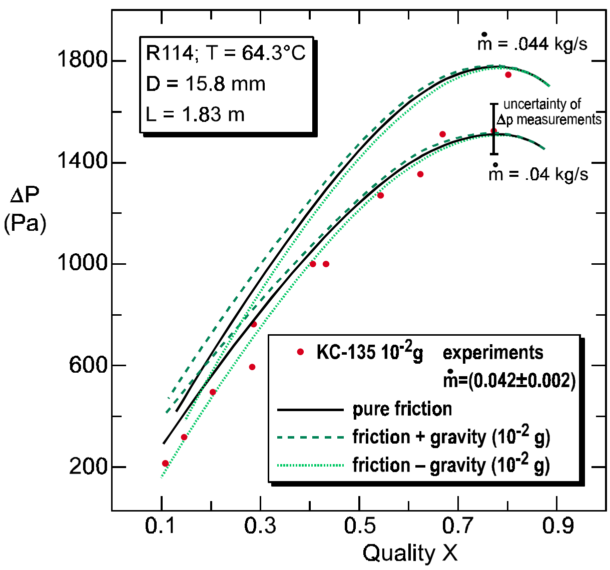
Low-Gravity Adiabatic Flow

Measured and calculated adiabatic pressure drops in a R114 duct



14

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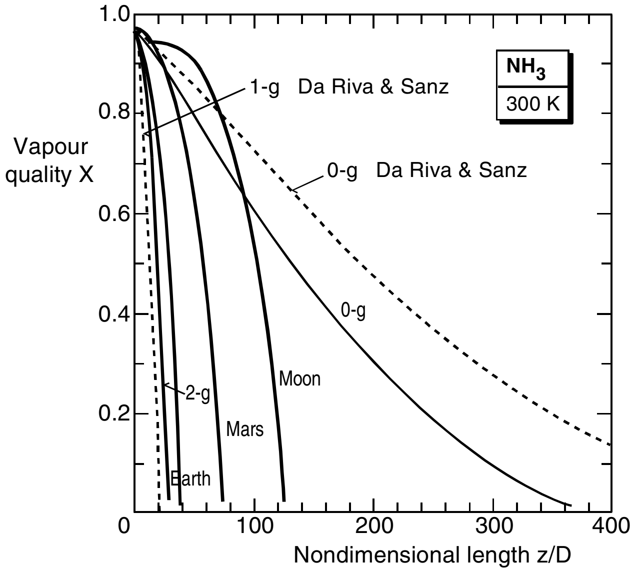
Gravity Dependent Condensation Length

Vapour quality along the reference duct



15

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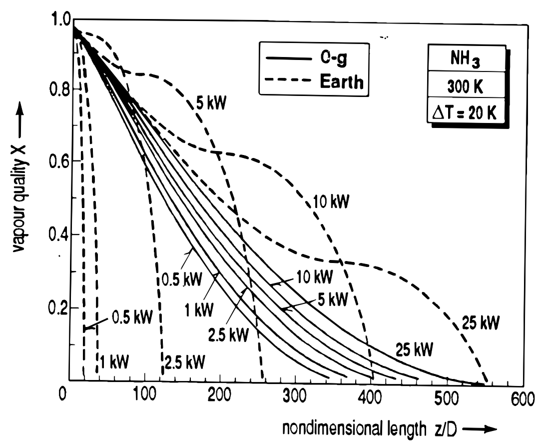
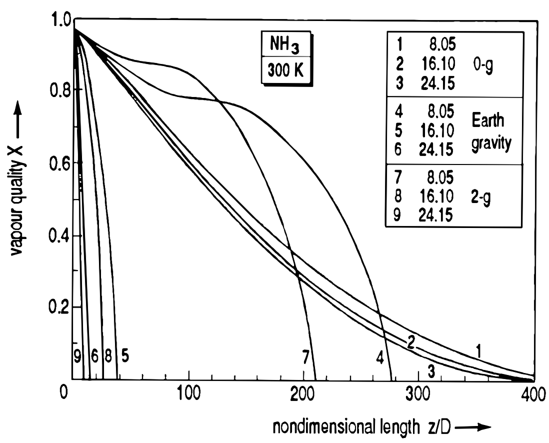
Gravity Dependent Condensation Length

Power dependence of NH_3 quality



Duct diameter dependence of NH_3 quality



16

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Gravity Dependent Condensation Lengths

- ***Condensation lengths needed considerably increase with decreasing gravity.***
- ***Differences between condensation lengths at different gravity levels become less pronounced with decreasing temperature and line diameter, increasing power, and heat transfer coefficient fine-tuning or enhancement. This coefficient, which varies along the condensation trajectory through the various flow patterns, is to be derived from experiments in different gravity environments.***

17

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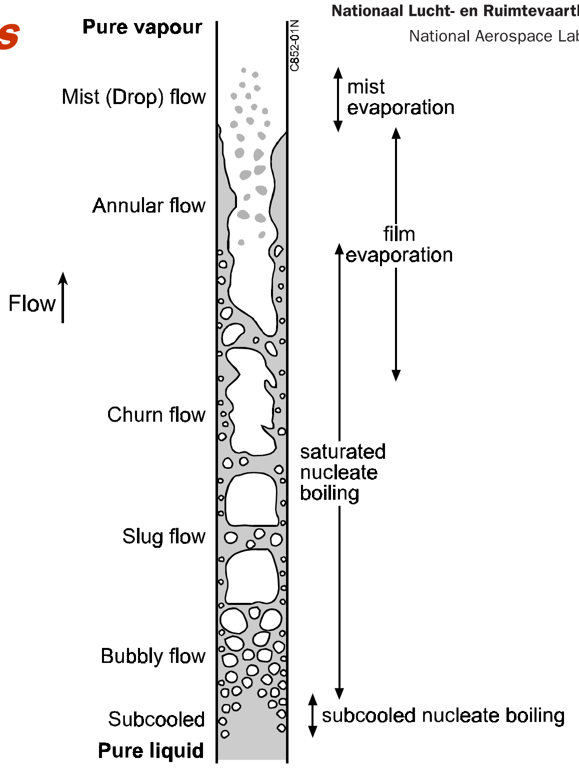
Up-Flow Regimes in Heated Circular Tube



18

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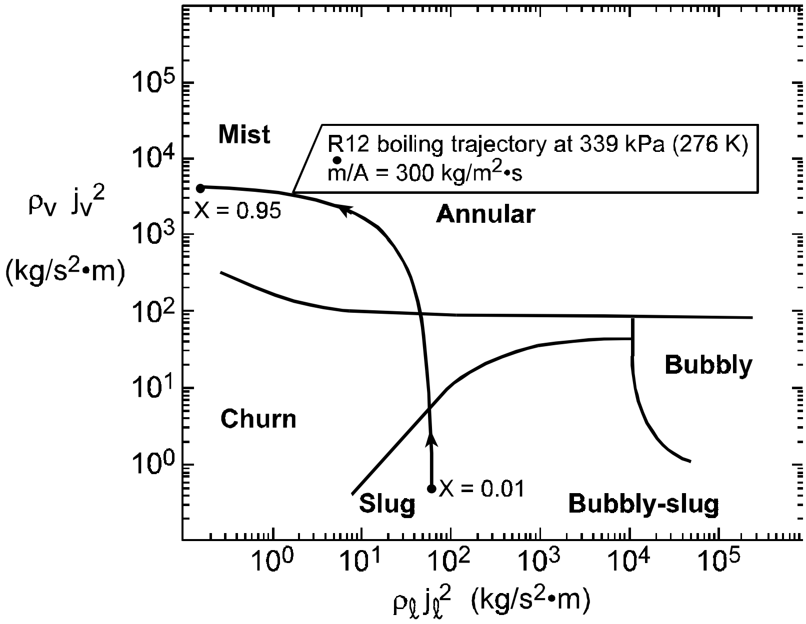
Flow Pattern Issues
Vertical Flow Boiling on Earth



19

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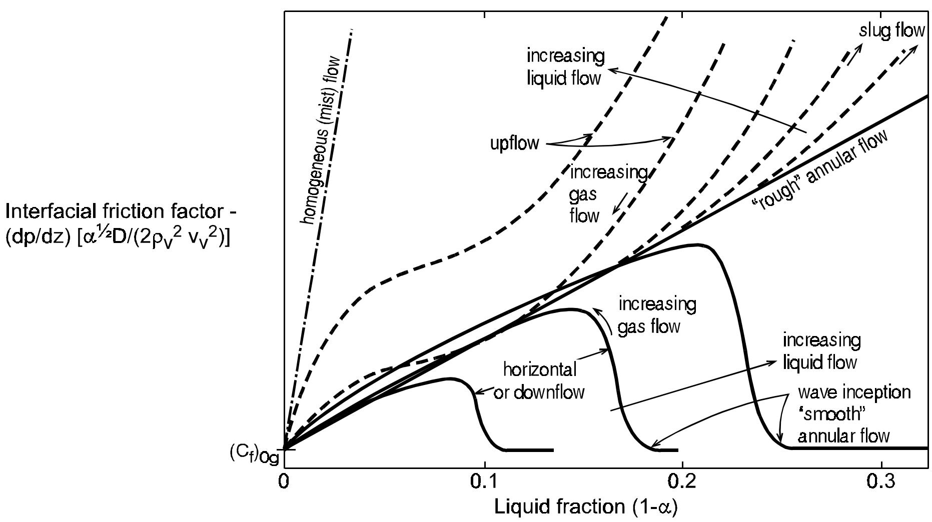
Flow Pattern Issues

Qualitative aspects of pressure gradient for various flow patterns



20

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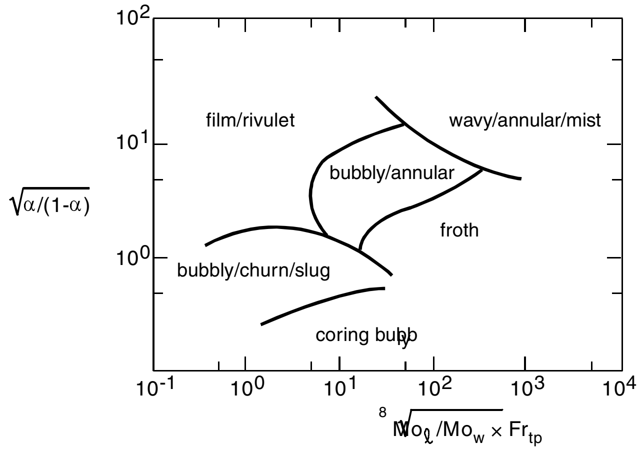
Generalised Flow Pattern Map

Flow pattern map (vertical downflow)



$Mo_w = \frac{\rho_w\, \sigma_w^3}{g\, \mu_w^4}$

$\rho_w = 10^3$ kg/m^3

$\mu_w = 10^{-3}$ Ns/m^2

$\sigma_w = 7.3 \times 10^{-2}$ N/m

$g = 9.8$ m/s^2

$Fr_{tp} = \frac{v_{tp}^2}{gD} = \frac{16\, m^2}{\pi^2 g D^5} \left(\frac{(1-X)^2}{\rho_\ell^2} + \frac{X^2}{\rho_v^2} \right)$

21

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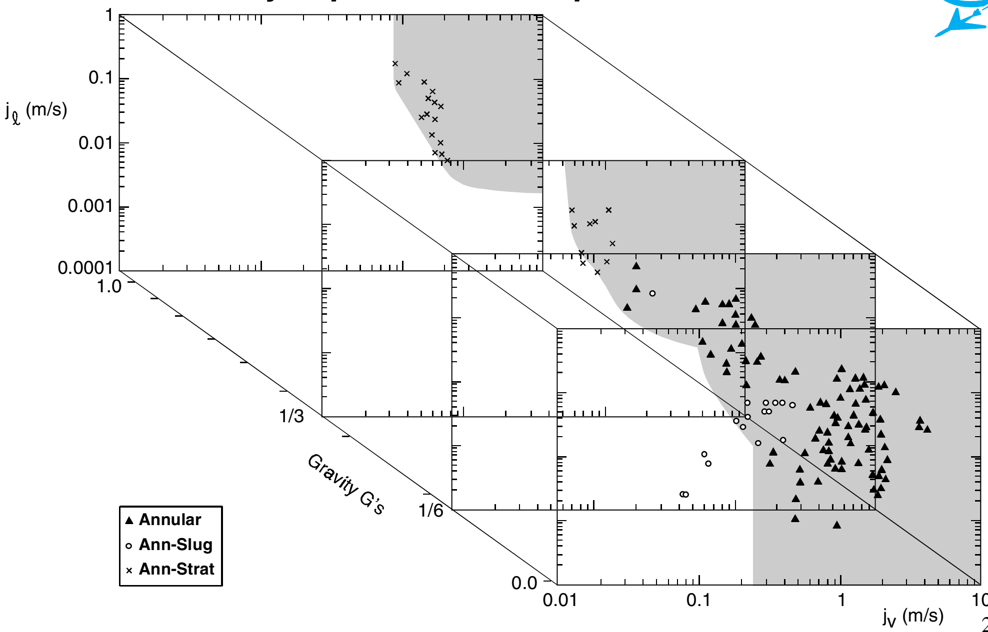
3-Dimensional Flow Pattern Mapping

Annular flow: Gravity dependent 3-D map



22

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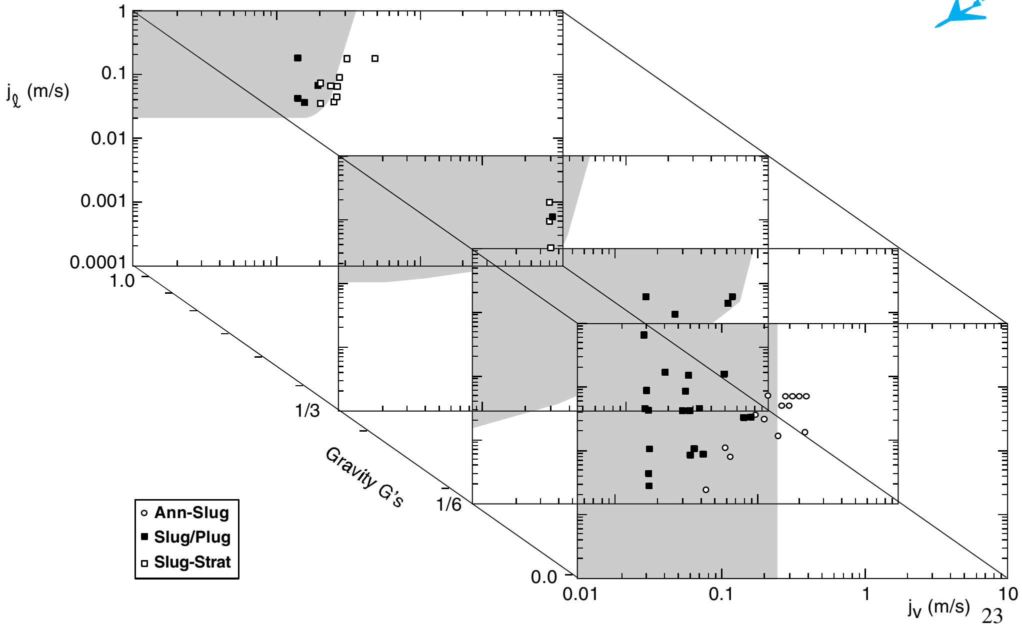
3-Dimensional Flow Pattern Mapping

Slug/plug flow: Gravity dependent 3-D map



23

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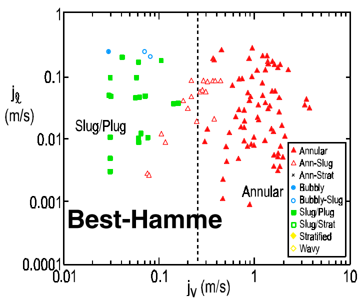
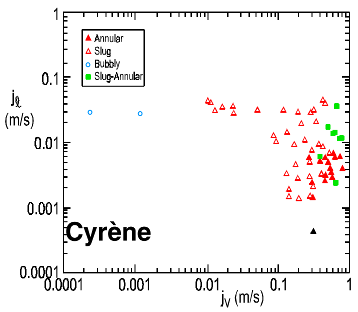
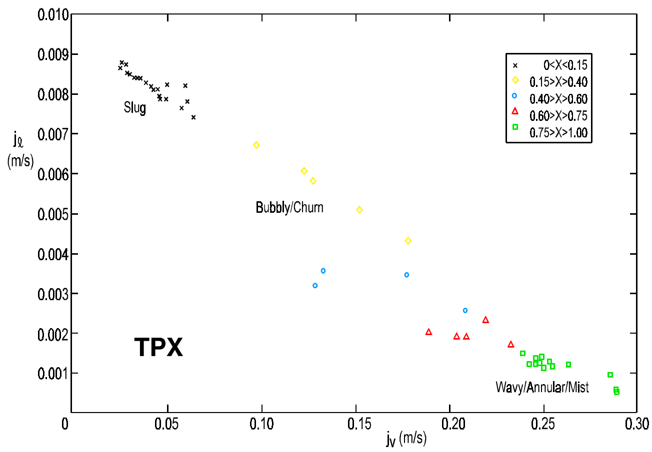
Low-G Flow Pattern Data







24

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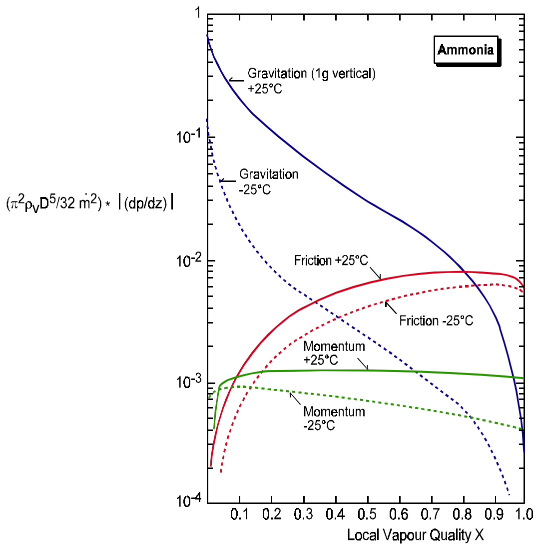
Anti-Gravity Issues



- *Anti-g constituent overrules sum of other gradients, below X=0.76 at 25 °C and X=0.38 at -25 °C.*
- *Annular liquid layer starts to break and fall back below these quality values.*
- *Shear-governed flow becomes (pulsating) slug, plug or churn flow, described by different equations and slip factors.*
- *Driving pressure (temperature) drops will strongly increase.*
- *Anti-g flow pattern maps are to be created.*

25

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Oscillating & Pulsating Systems

Two-Phase Loops

- *T-System*
- *Vapour Pressure Pumped Loop*
- *Loop Pumped by Evaporation and Condensation*

Single-Phase Devices

- *Synchronised Forced*
- *Phase-Shifted Forced*

Two-Phase Devices

- *Bubble-Driven HTD*
- *Pulsating, Meandering, Capillary (Tunnel), Flat Swinging & Serpentine Heat Pipes*

26

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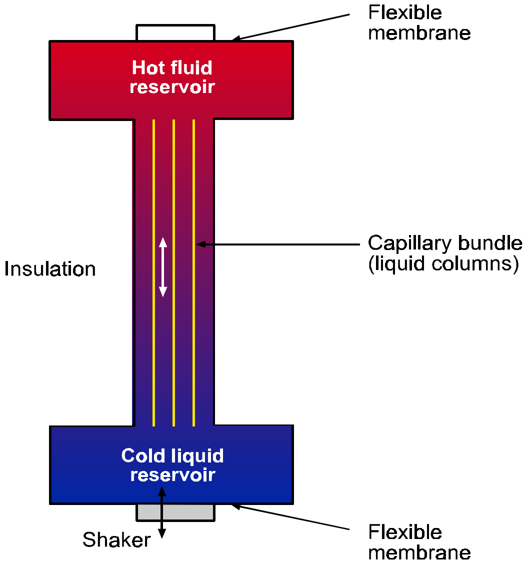
Single-Phase Oscillating Heat Transfer

Synchronised oscillatory heat transfer device



27

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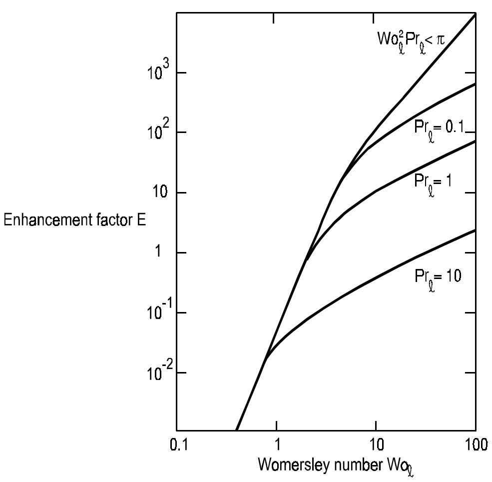
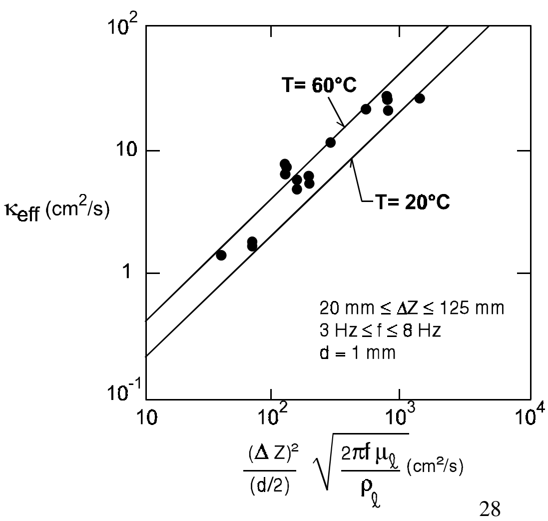
Single-Phase Oscillating HTD

Effective thermal diffusivity

Enhancement factor E versus Womersley Wo_l and Prandtl Pr_l





28

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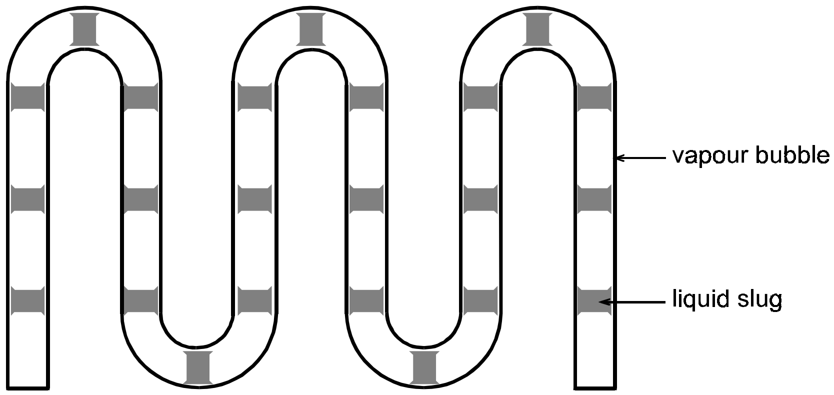
Two-Phase Oscillating or Pulsating HTD

Schematic of an oscillating device section



29

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Two-Phase Oscillating or Pulsating Heat Transfer Devices

- *Single-phase oscillating heat transfer device data, equations and dimensionless numbers, can be used for a better understanding of two-phase oscillating/pulsating devices.*
- *Slug/plug flow characteristics, dimensionless numbers, and (constitutive) equations, can be used to assess the performance and to design two-phase oscillating/pulsating devices.*
- *Two-phase oscillating or pulsating heat transfer loops and other devices are essentially non-isothermal. They need as the driving mechanism, like normal two-phase loops in the anti-(super)-gravity operation mode, a drop in working fluid saturation temperature that corresponds to an appreciable pressure drop. The latter plays a crucial role in selecting the working fluid.*

30

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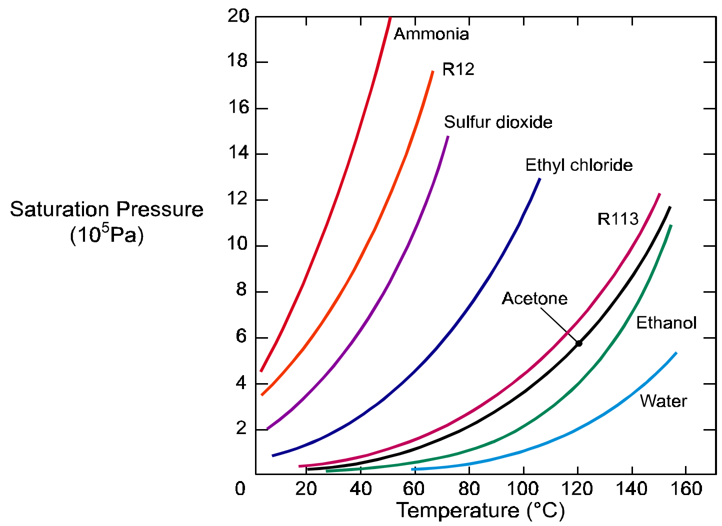
Working Fluids for Two-Phase Oscillating or Pulsating HTD

Vapour pressures of candidate working fluids, as a function of the temperature



31

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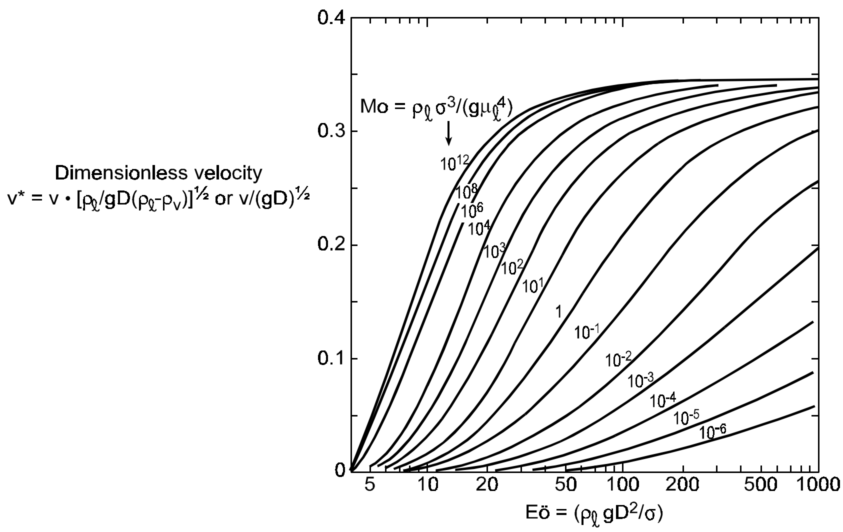
Dimensionless Numbers for Two-Phase Oscillating & Pulsating Heat Transfer

Dimensionless velocity v* as a function of Morton number Mo and Eötvös number Eö



32

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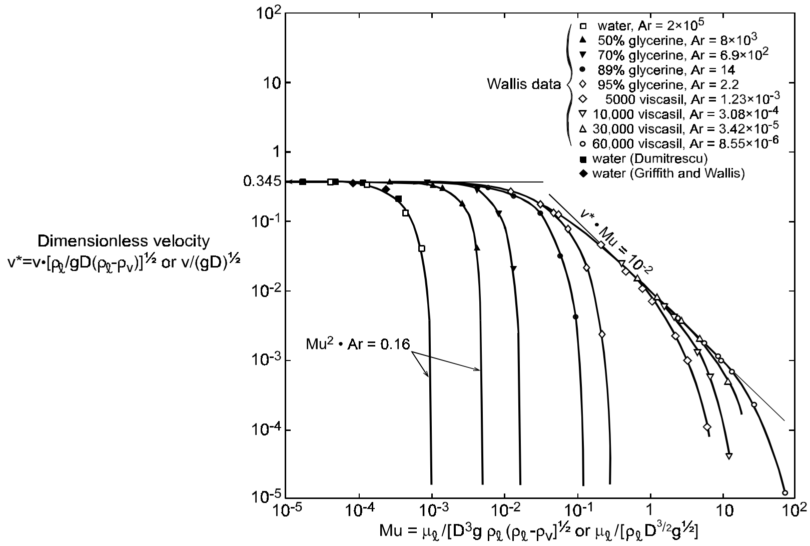
Dimensionless Numbers for Two-Phase Oscillating & Pulsating Heat Transfer

Dimensionless velocity v* versus dimensionless inverse viscosity Mu



33

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Experimental Verification Approach

- *Re-assessment of existing 1-g, low-g & super-g experimental data of:*
  - *TPX I & II, LHPFX, CAPL 1 & 2, TPF, CRYOFOD.*
  - *Special aircraft trajectories: Texas A&M, Cyrène.*
  - *Super-g turntables: NLR, ENSMA, Ural state Univ.*
- *Execution of additional, dedicated experiments:*
  - *CIMEX-3 in the Fluid Science Laboratory on ISS.*
  - *During special aircraft trajectories or on turntables.*

34

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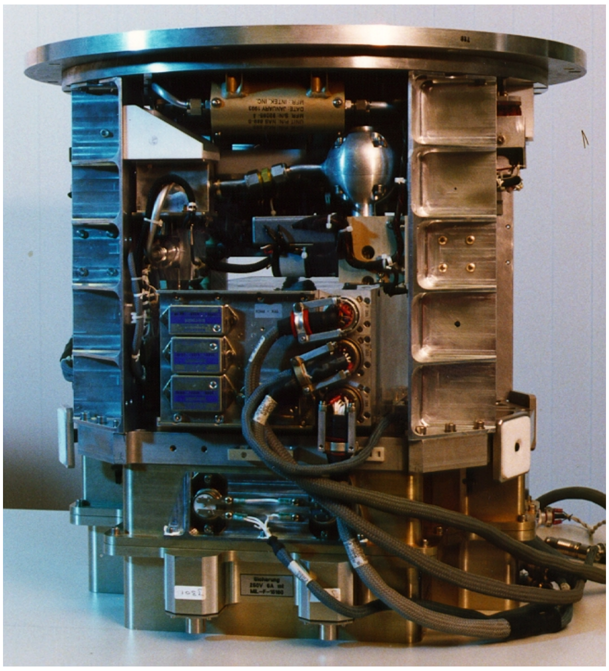
TPX I



35

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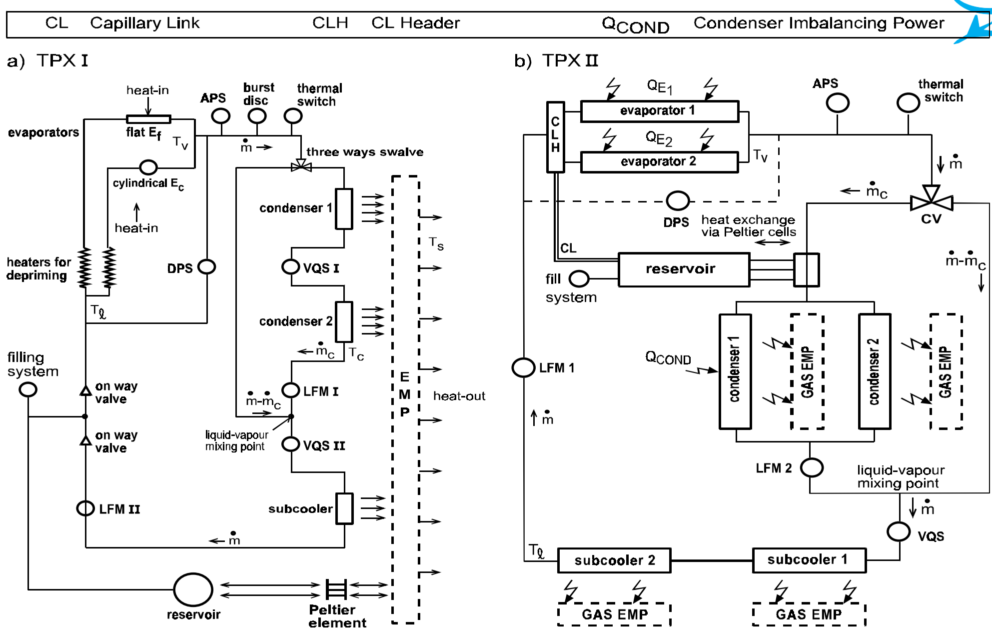
Schematics of TPX I & 2



36

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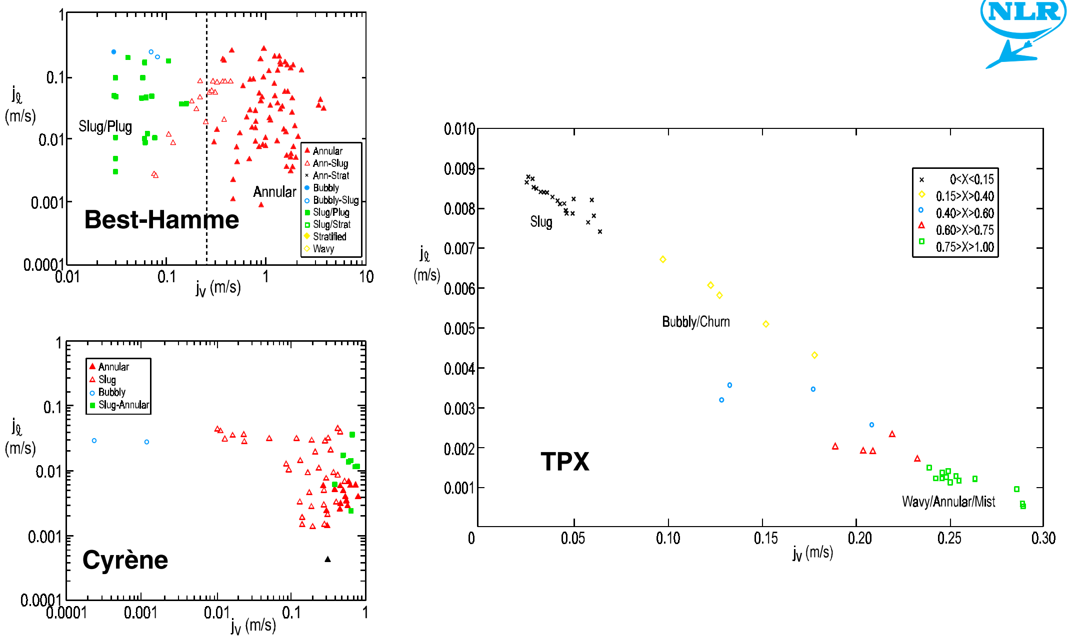
Low-G Flow Pattern Data



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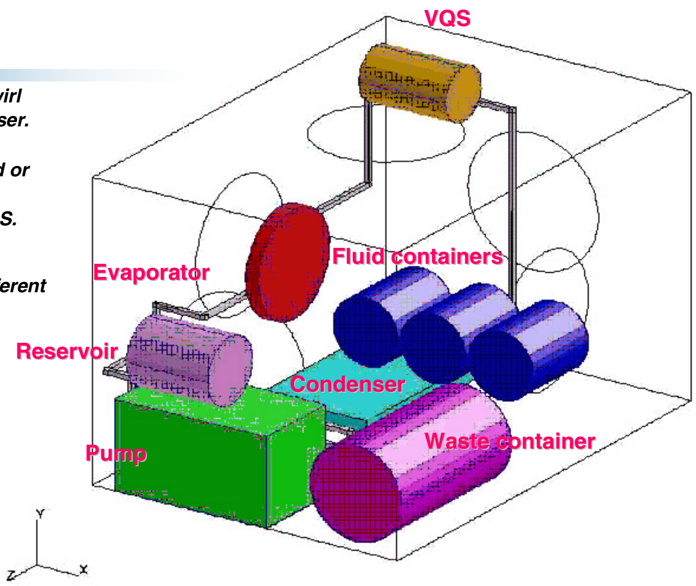
CIMEX-3
Versatile Two-Phase Loop Baseline

Experiment objectives:

- Development of (transparent) swirl evaporators and HELPD condenser.
- Measuring flow patterns and void or mass fraction in a (transparent) adiabatic section to calibrate VQS.
- Demonstration of mechanically pumped two -phase loop for different working fluids or mixtures.



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Concluding Remarks

- *The dimension-analytic approach offers possibilities for thermal-gravitational scaling between various gravity environments: 1-g on earth, reduced-g on moon and mars, low-g in special aircraft and rockets, micro-g in spacecraft, super-g on several planets and in spinning satellites and military combat aircraft.*
- *Equations for adiabatic and condensing annular two-phase flow are given.*
- *Flow pattern aspects, including super-gravity and anti-gravity issues are discussed.*
- *Promising applications exist for pulsating two-phase loops and single- and two-phase oscillating devices.*

39

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Ignored or Not Recognised Critical Issues

- *Planetary Gravity is constant in magnitude across a device. This is not valid for combat aircraft, spinning satellites and turntables.*
- *Pulsating two-phase systems need working fluids with a high dp/dT, plus:*
  - *In "classical" pulsating loops also high latent heat.*
  - *But in "novel" pulsating devices low latent (to enhance pulsation rate and transfer efficiency) and high caloric heat of the liquid. The latter transfers the majority of the heat, but needs also dT. Hence optimisation is a must.*

40

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Advice to Those Scaling & Metering Two-Phase Flow and Heat Transfer

- ***Best rule for engineers proposing to meter two-phase flow is the classical advice to those about to be married . . . . Don't do it!***
- ***Maybe the same can be said to those trying to thermally model and to scale two-phase heat transport systems with respect to gravity.***
- ***But who will listen, since both a marriage and those two-phase activities are very attractive and challenging, as they include endeavour, risk, steady and transient modes, fuzz and maybe even some chaos?***

41

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42

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43

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44

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45

D127-01N

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46