TWO-PHASE HEAT TRANSPORT SYSTEMS FOR THERMAL CONTROL APPLICATIONS

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ABSTRACT. The contents of this paper constitute the first lecture of a series of three lectures by the author on two-phase flow and heat transfer in terrestrial and aerospace-related thermal control system developments. These lectures were presented during the ICHMT Course on Passive Thermal Control (PTC-03) in Antalya, Turkey, October 22 - 25, 2003. This first lecture gives a general overview of several issues of two-phase flow and heat transfer in various gravity environments. The second lecture pertains to oscillating/pulsating heat transfer devices. The third lecture concerns mechanically pumped two-phase thermal control loops.

NOMENCLATURE & ACRONYMS

APS	Absolute Pressure Sensor
ATLID	Atmospheric Lidar
CL(H)	Capillary Link (Header)
CCPL	Cryogenic CPL
CPL/CAPL	Capillary Pumped Loop
DPS	Differential Pressure Sensor
EMP	Equipment Mounting Plate
EOS	Earth Observation Satellite
ESA	European Space Agency
GAS	Get Away Special
LFM	Liquid Flow Meter
LHP	Loop Heat Pipe
LHPFX	Loop Heat Pipe Flight Experiment
NASA	National Aeronautics & Space Administration
TEEM	Two-Phase Extended Evaluation in Microgravity
TPF	Two-Phase Flow
TPHTS	Two-Phase Heat Transport System
TPX	Two-Phase Experiment
VQS	Vapour Quality Sensor
A	area ($m^{2)}$
Bo	boiling number (-)
Cp	specific heat at constant pressure (J/kg.K)
D	diameter (m)

d	diameter of curvature (m)
Eu	Eulernumber(-)
Fr	Froude number (-)
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)
L	length (m)
Ma	Mach number (-)
Mo	Morton number (-)
$\dot{m}$	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 (-)
S	slip factor (-)
T	temperature (K) or (°C)
t	time (s)
v	velocity (m/s)
We	Weber number (-)
X	vapour quality (-)
z	axial or vertical co-ordinate (m)
α	vapour/void fraction (volumetric) (-)
β	constant in eq. (13) (-)
δ	surface roughness (m)
Δ	difference, drop (-)
μ	viscosity (N.s/m^2)
σ	surface tension (N/m)
λ	thermal conductivity (W/m.K)
π	dimensionless number (-)
ρ	density (kg/m^3)
ν	angle (with respect to gravity) (rad)

<u>Subscripts</u>

a	acceleration
c	condenser
f	friction
g	gravitation
l	liquid
m	momentum, model
o	reference condition
p	pore, prototype
t	total
tp	two-phase
v	vapour
w	water

FOREWARD

Already for more than a century two-phase heat transfer systems are frequently being applied in the power industry and the process industry. Novel mechanically pumped two-phase heat transfer system developments were started around 1980, since there was a need for a Space Station thermal control system. This system had to remove 100 kW waste heat (dissipated at various locations) over roughly 100 metres to the Space Station radiators, for dumping it to cold space. But the two-phase option was replaced by a single-phase system, as it was considered to be too risky for the Space Station. The only, currently developed, mechanically pumped two-phase thermal control systems for the International Space Station ISS are the two-phase ammonia thermal control system for the Russian module and the two-phase carbon dioxide AMS-02 Tracker Thermal Control System.

The transport capability of the classical heat pipe was foreseen not to be able to meet the high power and dedicated temperature requirements of modern and future spacecraft. Mechanically pumped single-phase loops have the drawback of large diameter, thick-walled heavy lines and large, heavy, high-power consuming and undesired vibrations producing pumps. Therefore the aerospace community started (already more than a decade ago) to investigate more powerful alternative systems, being the Capillary Pumped Loop (CPL), the miniature, reversible and ramified Loop Heat Pipe (LHP), and Vapour Pressure Driven Heat Transfer Devices (discussed in the second lecture). Also some limited research was done to develop aforementioned Mechanically Pumped Two-Phase Loops (discussed in the third lecture).

The lecture discusses the physics of two-phase flow and heat transfer in various gravity environments, some research highlights, controllability aspects and the issues of thermal-gravitational thermal modelling and scaling, including the fundamental differences between liquid-vapour (single component) and liquid-gas (two-component) two-phase flow and heat transfer.

INTRODUCTION

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 current investigations concern the 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). However, of major interest for aerospace applications are the more complicated liquid-vapour or liquid-gas flows, that are characteristic for aerospace thermal control systems, life sciences systems and propellant systems. Especially for liquid-vapour flow in aerospace two-phase thermal control systems, the phenomena become extremely complicated, because of heat and mass exchange between the two phases by evaporation or condensation. Though a huge amount of publications (textbooks, conference proceedings and 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 microgravity.

The various heat and mass transfer research issues of two-phase heat transport technology for space applications are discussed. It is focused 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 straightforwardly be derived from this liquid-vapour case, as various terms in the constitutive equations can be set zero.

The discussions start with the background of the research, followed by a short general description of two-phase flow and heat transfer phenomena. The impact of the gravity level will be assessed.

The discussions focus next on development supporting theoretical work: thermal/gravitational scaling of two-phase flow and heat transport in the different sections of two-phase thermal control loops, including the various aspects of gravity level dependent two-phase flow pattern mapping and condensation. Outcomes of the theoretical activities are compared with results of various experiments, carried out both on earth (one-g) and in micro-gravity environment.

The discussions include also a brief overview of past and near-future in-orbit technology demonstration experiments, including a survey of the current research and development status, plus some recommendations for future investigations.

Since the majority of the above systems are currently considered to have reached maturity, they are already applied in various spacecraft. Many applications in the near and far future are foreseen. This will be illustrated in a brief survey. Some of the challenging applications will be discussed.

BACKGROUND

Thermal management systems for future large spacecraft have to transport large amounts of dissipated power (up to say hundreds kW) over large distances (up to say 100 metres). This can be realised byConventional single-phase heat transport 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 arranged in a pure series, pure parallel, or a hybrid configuration. ESA’s R-114 Two-Phase Heat Transport System TPHTS (Fig. 1) is a parallel system.

The schematic shows the advantage of the parallel concept: a modular approach, in which branches with dissipating stations (evaporators/cold plates) or heat demanding stations (condensers/radiators) simply can be added or deleted. A drawback is the relatively complex system controllability, as compared to a series system.

As already said: Two very important near-future two-phase heat transport system applications are:
- The two-phase ammonia thermal control system of the Russian segment of the ISS [1-5].
- The two-phase carbon dioxide AMS-02 Tracker Thermal Control System [6-11].

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 (Fig. 2):
- The western-heritage Capillary Pumped Loop CPL, derived from the first loop proposed in 1966 by Stenger [12], shown in Figure 3.
- The Russian-heritage Loop Heat Pipe LHP [13].

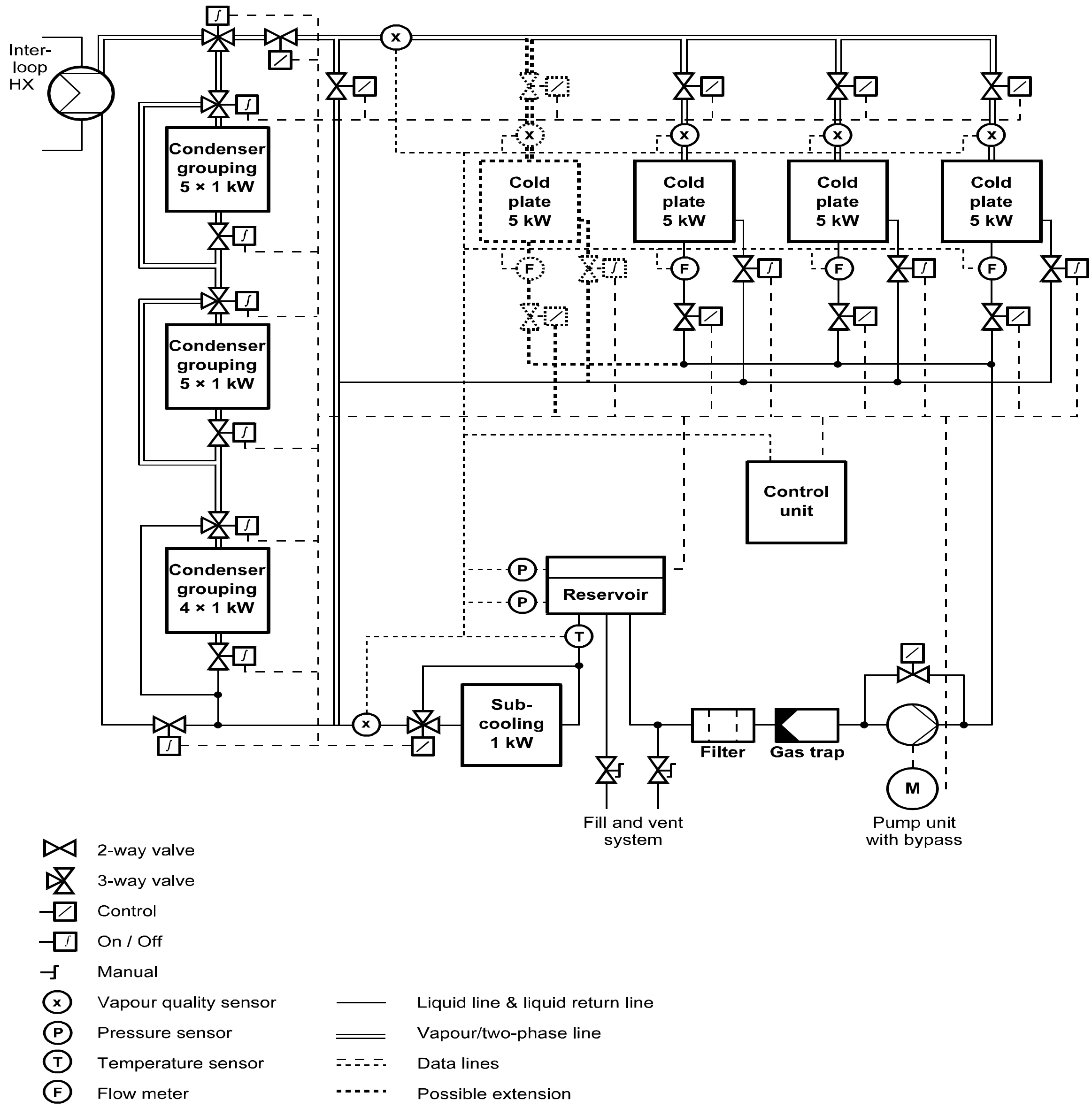

Figure 1. Schematic of the ESA TPHTS

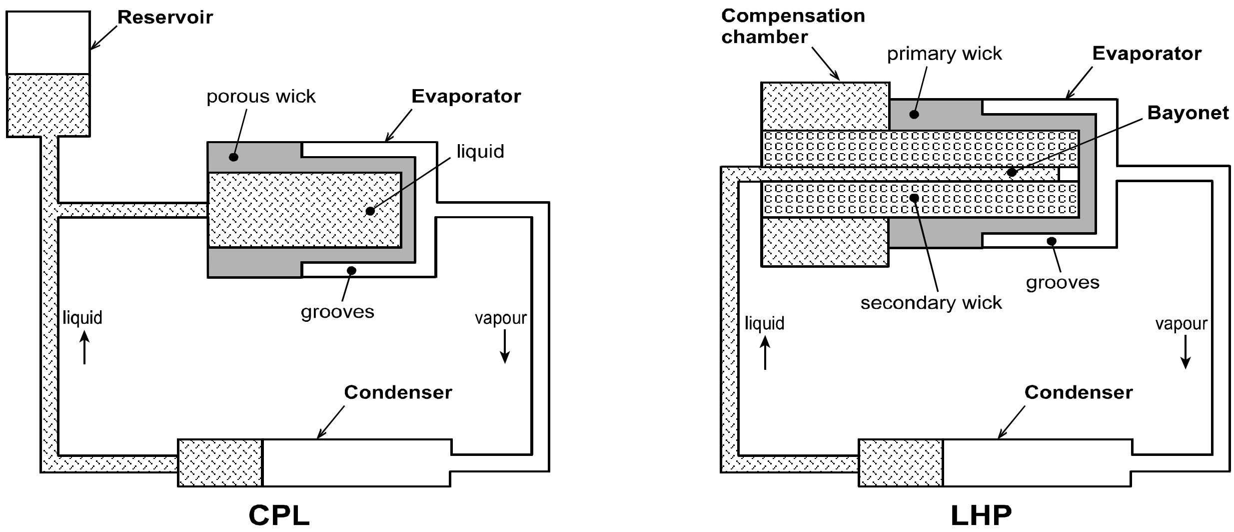

Figure 2. Schematics of a Capillary Pumped Loop and a Loop Heat Pipe

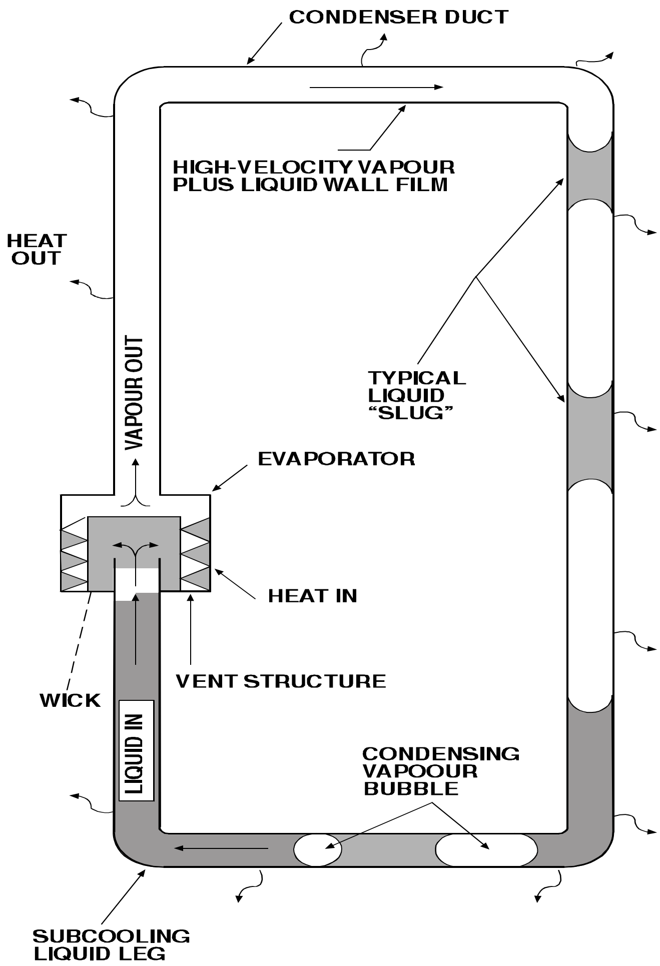

Figure 3. Stenger's original Capillary Pumped Loop [13]

Active loop temperature setpoint 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 CPLs and LHPs were planned for several future spacecraft missions, not only low-orbit or geo-synchronous satellites, but also for missions to planets [14]. Examples are the American Earth Observation Satellite EOS-AM, the European Atmospheric Lidar earth observation spacecraft ATLID, the Russian spacecraft OBZOR, the Hubble Space Telescope retrofit mission, the US COMET spacecraft, the Hughes 702 satellites, and other commercial geo-synchronous communication satellites.

Since two-phase flow and heat transfer is essentially different in earth gravity, reduced gravity (on the Moon and Mars) and micro-gravity (on orbiting spacecraft), two-phase heat transport system technology is to be demonstrated in space. Therefore in-orbit experiments have been carried out, e.g. ESA's Two-Phase eXperiment TPX I&2 [15, 16], NASA's CApillary Pumped Loop experiments CAPL 1&2&3 [17, 18], the Loop Heat Pipe Flight eXperiment LHPFX [19], the all US Loop Heat Pipe with Ammonia ALPHA, the Cryogenic Capillary Pumped Loop CCLP [20], and the Two-Phase Flow experiment TPF [21]. Others were planned for future flights: the Two-phase flow Extended Evaluation in Microgravity TEEM [22], and Granat [23].

Figure 4 depicts the schematics of the ESA in-orbit technology demonstration experiments TPX I & II. Figure 5 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 kW.hr 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.

CL	Capillary Link	CLH	CL Header	Q_{COND}	Condenser Imbalancing Power

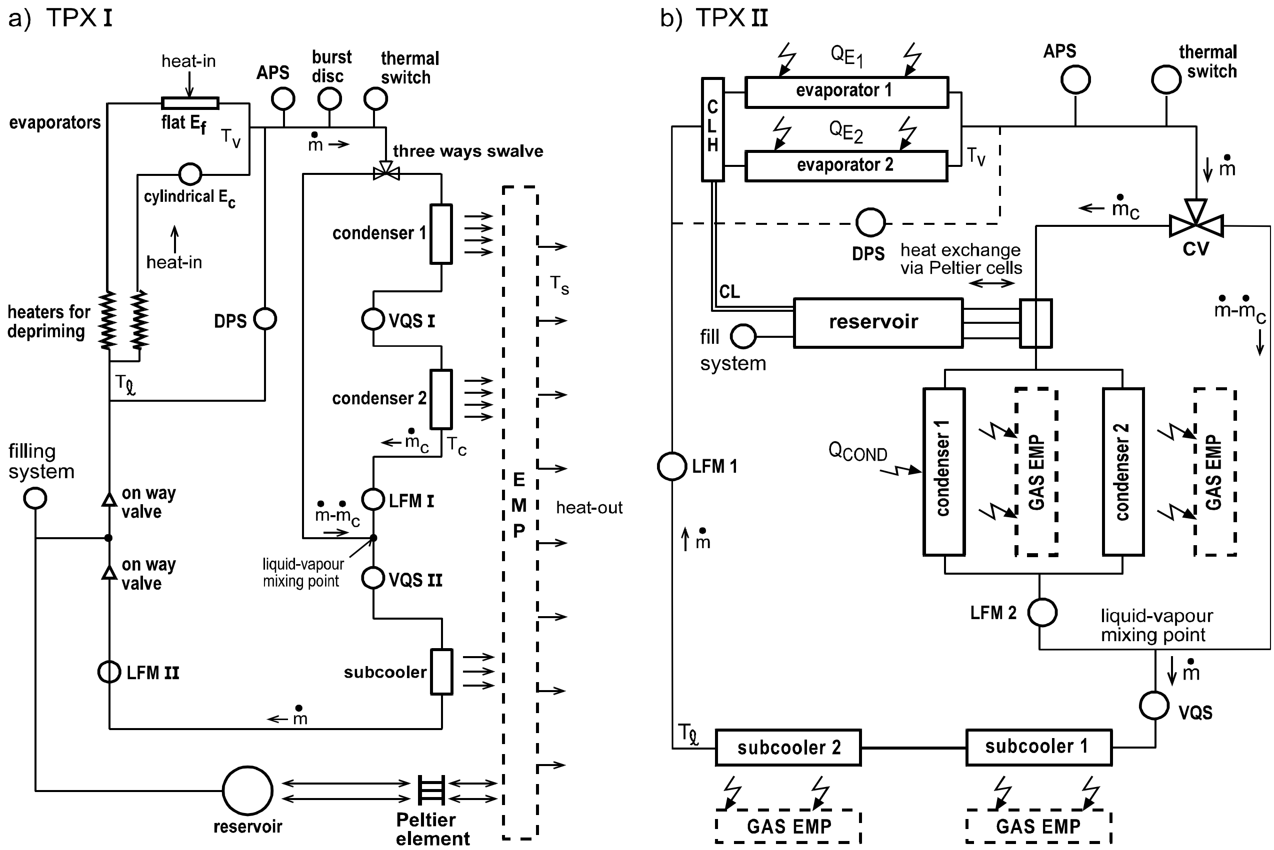

Figure 4. Schematics of TPX I & II

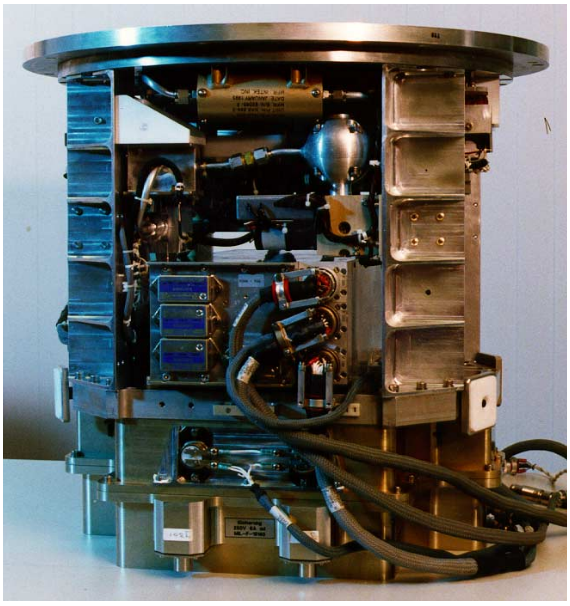

Figure 5. TPX I hardware

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 Some experiments were done in drop towers or during Microgravity Science Laboratory missions on the Space Shuttle [22, 23]. Many others were executed during low-g aircraft flights [24-34].

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 & experiments in a system can be used in another, if they concern flow phenomena only, hence without 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 can not 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 more complicated than the equations for single-phase flows. The complications are due to the fact that inertia, viscosity and buoyancy effects can be attributed both to the liquid phase and to the vapour phase, and also due to the impact of surface tension effects.

An extra, and major, complication is the spatial distribution of liquid and vapour, the so-called flow pattern. Figure 6 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 7. Figure 8 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.

Various text books on two-phase flow and heat transfer [35-38] 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.

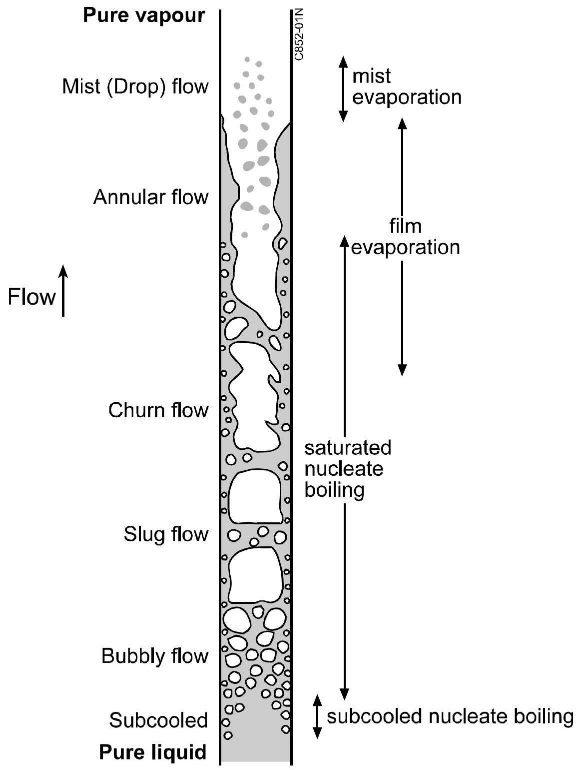

Figure 6. Flow patterns and boiling mechanisms for up-flow in a vertical line on earth

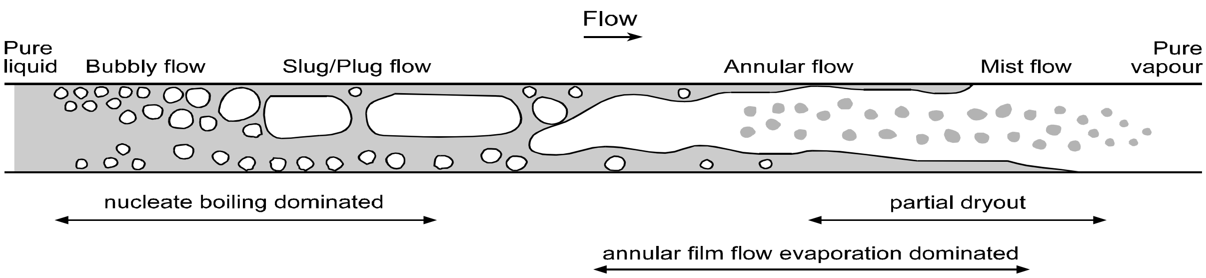

Figure 7. Horizontal evaporator line on earth

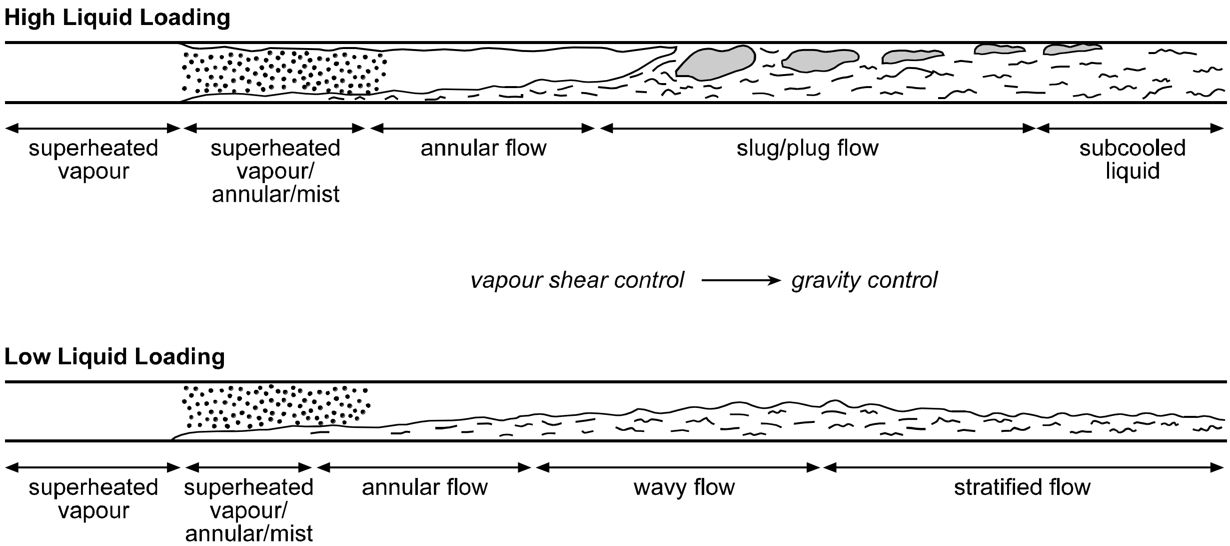

Figure 8. Horizontal condenser line on earth

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 9 and 10. An alternative way to derive these dimensionless numbers is by dimension analysis, constituting a useful baseline for similitude in engineering approaches, discussed in specialised text books [40].

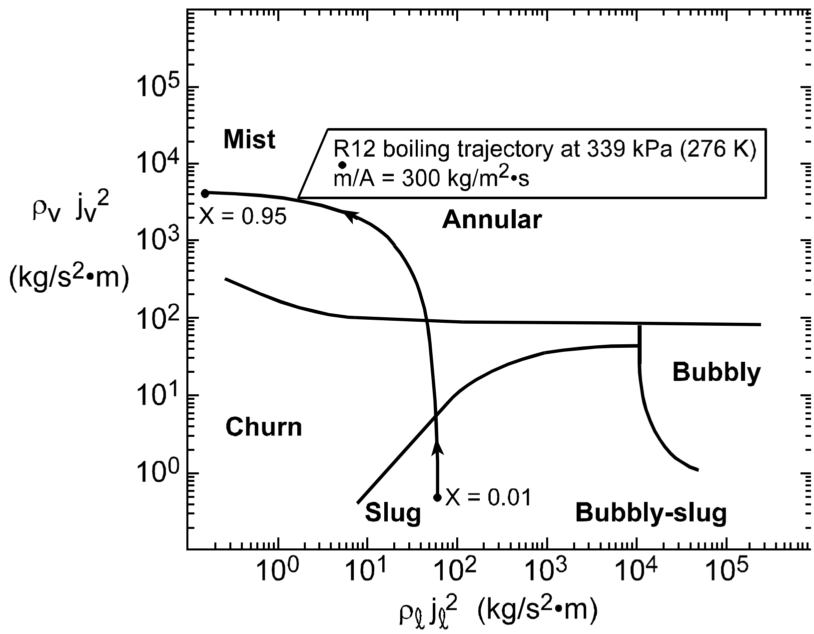

Figure 9. Flow pattern map for vertical flow [35]

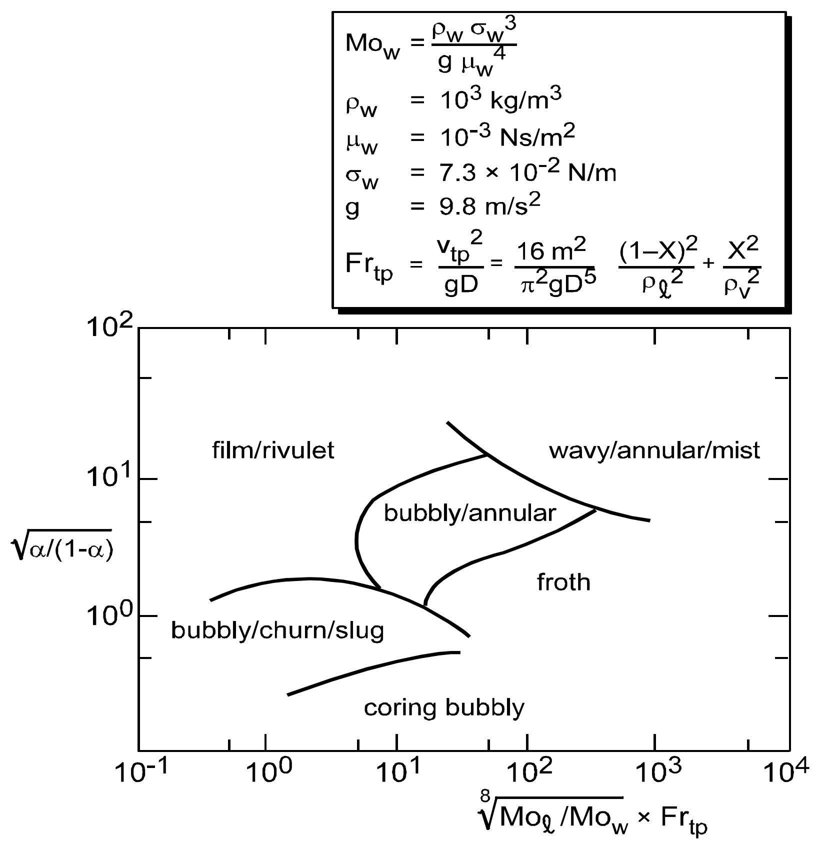

Figure 10. Flow pattern map for vertical downward flow [39]

It is remarked that the discussions here will be based on dimension-analytical considerations, assuming that:
- Lines have a circular cross section, the problem is circle-symmetric, hence one-dimensional.
- Surface tension is not influenced by surfactants. Presence of the latter is to be avoided for proper loop operation.
- 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.

THERMAL-GRAVITATIONAL MODELLING AND SCALING

Development supporting theoretical work, like thermal-gravitational modelling and scaling of two-phase heat transport systems [41-43], is being done for:
- A better understanding of the impact of gravitation level on two-phase flow and heat transfer phenomena.
- Providing means for comparison and generalisation of data.
- Developing tools to design space-oriented two-phase loops (components), based on terrestrial tests, to reduce costs.

Scaling of the 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.
- To define in-orbit experiments to isolate phenomena to be investigated, e.g. excluding gravity-induced disturbing buoyancy effects on alloy melting, diffusion and crystal growth, for a better understanding of the phenomena.

The magnitude of the gravitational scaling varies with the objectives:
- From 1 g to 10^{-6} g (random direction) for the terrestrial scaling of orbiting spacecraft.
- From 1 g to 0.16 g for Moon base and to 0.4 g for Mars base systems.
- From 10^{-2} or 10^{-6} g to 1 g for isolating gravity induced disturbances on physical phenomena under investigation.
- From low-g to another or the same low-g level for in-orbit technology demonstration.

One g is not the upper limit in scaling. Higher values (pertaining to larger planets) can be simulated during special aircraft flight trajectories or in centrifuges.

Even in single-phase systems scaling is anything but simple, since flow and heat transfer are equivalent in model and prototype <u>only if</u> 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:
- In addition to the above fields, the spatial density distribution (void fraction, flow pattern) is to be considered.
- Geometric scaling often makes no sense since some characteristic dimensions, *e.g.* bubble size and surface roughness, hardly depend on the system dimensions.
- Of the proportion problem at high power density levels, typical for two-phase flow boiling heat transfer.

SIMILARITY CONSIDERATIONS AND DIMENSION ANALYSIS

Similarity considerations [41] 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. These18 π-numbers are listed in the first column of Table 1. It is remarked that detailed dimension analyses on the boiling crises and on bubbles and drops can be found in literature (e.g. [44]).

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

Relevant π-numbers for the thermal-gravitational scaling of two-phase loops	Liquid Parts		Evaporators Swirl & Capillary	Non-liquid: Vapour/2-Phase	Condensers
	Adiabatic	Heating/Cooling			
$\pi_1 = D/L$ = geometry	•	•	•	•	•
$\pi_2 = Re_l = (\rho v D/\mu)_l$ = inertia/viscous	•	•	•	•	•
$\pi_3 = Fr_l = (v^2/gD)_l$ = inertia/gravity	•	•	•	/•	•
$\pi_4 = Eu_l = (\Delta p/\rho v^2)_l$ = pressure head/inertia	•	•	•	•	•
$\pi_5 = \cos \nu$ = orientation with respect to g	•	•	•	/•	•
$\pi_6 = S$ = 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 [40], if the relative magnitudes of the different 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.

Further it can be remarked that in scaling two-phase heat transport systems:
- Geometric distortion is not permitted to study boundary layer effects and boiling heat transfer, as identity of surface roughness in prototype and model is to be guaranteed.
- Geometrical distortion is a must when the length scaling leads to impractical small (capillary) conduits in the model, in which the flow phenomena basically differ from flow in the full size prototype.

It can be convenient to replace vapour quality X by the volumetric vapour or void fraction α. X is the ratio of the vapour mass flow and the total mass flow. The naming originates from steam engineering: X stands for the wetness of steam; X=1 is "dry steam", hence pure vapour, no liquid at all). This can be done using the relation

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

It is clear that the presented set of π-numbers is rather arbitrary, e.g. several numbers contain only liquid properties. These can be easily transferred into vapour properties containing numbers using π_6 to π_8. Similarly π_1 can be used to interchange characteristic length (duct length, bend curvature radius) and a characteristic diameter (duct diameter, hydraulic diameter, but also surface roughness or bubble diameter). Sometimes it will even be convenient to simultaneously consider two geometric π_1-numbers. One concerns the overall channel (channel diameter versus length or bend curvature radius). The second pertains to other parameters as the ratio of surface roughness and bubble diameter to investigate boiling heat transfer, or the ratio of surface roughness and channel diameter to study friction pressure drop.

The best scaling approach is to choose combinations of π-numbers that optimally suit the problem under investigation:

- The Morton number $$\pi_{15} = Mo_l = Re_l^4 Fr_l/We^3 = \rho_l \sigma^3/\mu_l^4 g \qquad (2)$$

  useful for scaling two-phase flow with respect to gravity, as it contains g, liquid properties and surface tension only.

- The Mach number $$\pi_{16} = Ma = v/(\partial p/\partial\rho)^{1/2} \qquad (3)$$

  important if compressibility effects are important, as choking depends on the vapour quality of a two-phase mixture.

- The boiling number $$\pi_{14} = Bo = Q/\dot{m}\, h_{lv} = \Delta H/h_{lv} \qquad (4)$$

  Q is the power fed to the boiling liquid. This number appears in the expression for the dimensionless enthalpy of the mixture at any z in a line of diameter D, radially heated via the outside wall (q is the radial heat flux):

  $$H(z)/h_{lv} = H(0)/h_{lv} + Q/\dot{m}\, h_{lv} = H(0)/h_{lv} + \pi D\, z\, q/\dot{m}\, h_{lv} \qquad (5)$$

  For sub-cooled/heated liquid this is $$\pi_{14} = Q/\dot{m}\, Cp_l\, \Delta T \qquad (6)$$

  ΔT being the temperature drop. The above implies that, if the dimensionless entrance enthalpies are equal for different fluids flowing in a similar geometry, equality of the boiling number ensures equal non-dimensional enthalpies at all similar axial locations. For thermodynamic equilibrium conditions this means equal qualities at similar locations, and similar sub-cooling and boiling lengths.

- The condensation number, in which h is the local heat transfer coefficient,

$$\pi_{17} = (h/\lambda_l)\,(\mu_l^2/g\,\rho_l^{\,2})^{1/3} \tag{7}$$

- The vertical wall condensation number, with T_o as local sink, T as local saturation temperature.

$$\pi_{18} = L^3\rho_l^2\, g\, h_{lv}/\mu_l\,\lambda_l(T-T_o) \tag{8}$$

A first step in a practical approach to scale two-phase heat transport systems is identification of important phenomena, to obtain π-numbers for which identity in prototype and model must be required to realise perfect scaling according to the so-called Buckingham Pi theorem (crucial in similarity considerations). Distortion will be permitted for π-numbers pertaining to less important phenomena. Important phenomena and the relevant π-numbers will be different in different parts of a system. The relevance of the π-numbers in the various loop sections is indicated by • in Table 1 (π-numbers for thermal gravitational scaling of two-phase loops), given earlier in this section.

For refrigerants, like ammonia and R114, forced convection heat transfer overrules conduction completely. Therefore π_{10}, π_{11} and π_{12}, are not critical in gravitational scaling. π_{16} can be neglected also as the system maximum power level and line diameters correspond with flow velocities far below the sonic velocity in all parts of a system.

Considering 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 low-g prototype and terrestrial model, for a horizontal arrangement of these sections. Also it can be remarked that, in the porous (liquid) part of a capillary evaporator, surface tension forces (σ/D_p) are dominant over inertia ($\pi_9 \rightarrow 0$): hence the evaporator exit quality will approach 1 (pure vapour). This means that gravity is unimportant for the vapour part of the evaporator and the vapour line connecting evaporator and condenser.

Several important conclusions can be drawn now:

- 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 can be derived from the flow pattern map for downward two-phase flow (Fig. 10). In this figure [39], water properties at 20 °C must be used to determine the scale of the abscissa. The Froude number for two-phase flow used in this figure is defined as:

$$Fr_{tp} = (16\,\dot{m}^2/\pi^2 D^5 g)[X^2/\rho_v^{\,2} + (1-X)^2/\rho_l^{\,2}]\,. \tag{9}$$

Comparing low-g and vertical downward terrestrial flow one has to correct the latter for the reduction of the slip factor by the gravity forces assisting the liquid layer lowing down. Anyhow, vertical down flow is the preferred two-phase line orientation in the terrestrial model because of the 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.

QUANTITATIVE EXAMPLES

Consequences of scaling are elucidated by the figures 11 and 12, depicting the temperature dependence of the groups $g.Mo_l = \rho_l.\sigma^3/\mu_l^4$ and $(\sigma/\rho_l)^{½} = D.g^{½}/(We/Fr)^{½}$.

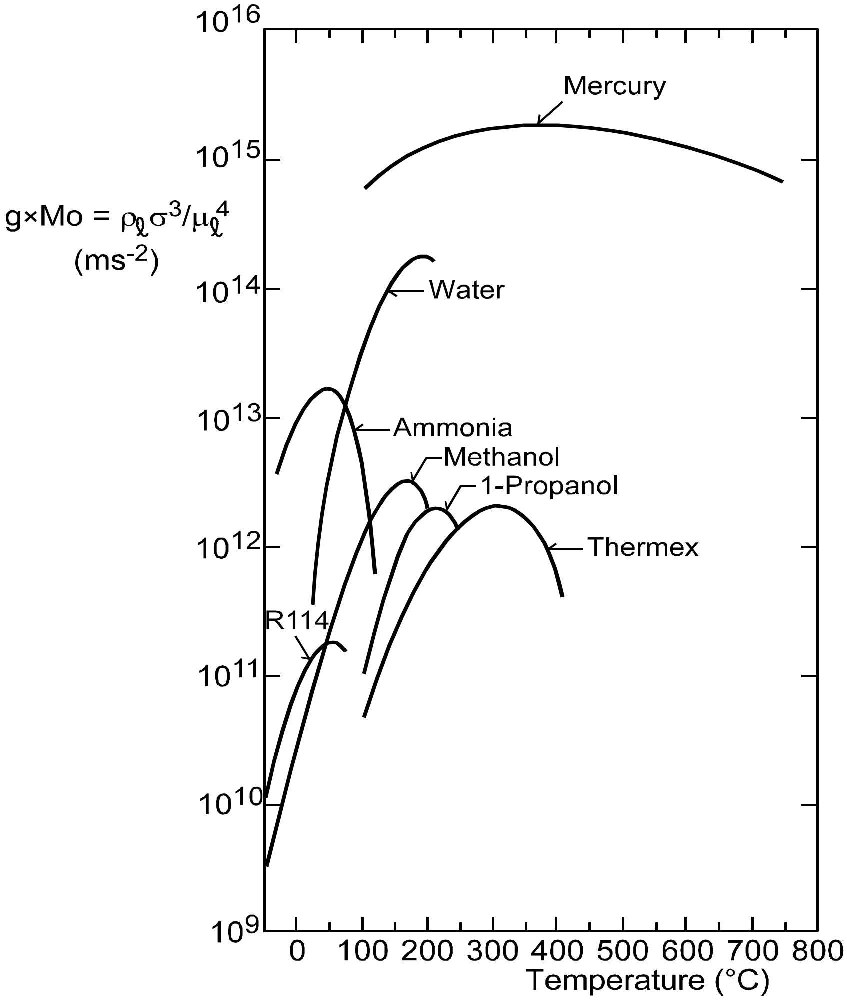

Figure 11. $\rho_l.\sigma^3/\mu_l^4$ versus temperature for various fluids

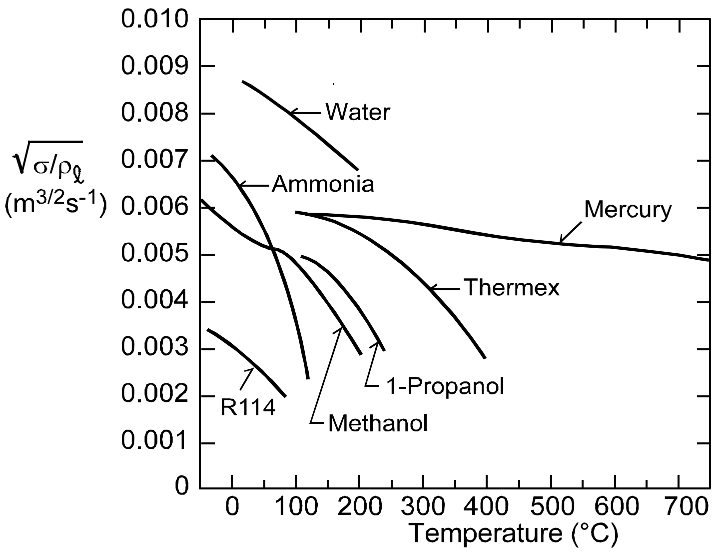

Figure 12. $(\sigma/\rho_l)^{½} = D.g^{½}.(We/Fr)^{-½}$ versus temperature, for various fluids

Scaling at the same gravity level

First, it can be seen in figure 11 that the value $\rho_l\sigma^3/\mu_l^4 = 2*10^{12}$ m/s^2 can be realised by seven systems: 115°C ammonia, 115°C methanol, 35°C water, 180°C propanol, 235°C propanol, 250°C thermex and 350°C thermex. Requiring, in addition to Morton Number identity, also the identity in $(We/Fr)^{½}$, in other words $D/(\sigma/\rho_l)^{½}$, the length scales of the seven systems derived from the corresponding (σ/ρ_l)-values in figure 12, turn out to be proportional to each other with ratios 2.5 : 4.5 : 8.4 : 4.2 : 3.0 : 5.0 : 3.6. Hence the maximum scaling ratio obtainable equals 8.4/2.5 ≈ 3, indicating that geometry scaling at the same gravity level can cover only a limited range.

Second, the scaling of high pressure (at say 110 °C) ammonia system parts by low pressure (at say -50 °C) ammonia system parts might be attractive for safety reasons or to reduce the impact of earth gravity in vertical two-phase sections. Similarly, it follows from figure 12, that the length scale 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 $L_p/L_m = [(\sigma/\rho_l)_p/\sigma/\rho_l)_m]^{1/2} \approx 0.4$. For ammonia such a scaling can be attractive only for sections without heat transfer, since otherwise it will certainly lead to unacceptable high power levels in the model system evaporators and condensers.

Scaling with respect to gravity

Figure 11 shows that the scaling with respect to gravity is restricted to say two decades, if the fluid in prototype and model is the same. As an example, a 10^{-2} g, 80 °C thermex prototype can be scaled well by a 300 °C thermex terrestrial model. The geometric scaling clairly is, according to figure 12, $L_p/L_m=D_p/D_m=(g_m/g_p)^{½}(\sigma/\rho_l)_p^{½}/(\sigma/\rho_l)_m^{½} \approx 14$.

Far more interesting is fluid to fluid scaling: e.g. alkali metal terrestrial prototypes can be scaled by various model systems in space, e.g. a 400°C mercury prototype:

- At 10^{-2} g, by a 35°C ammonia model ($L_m/L_p \approx 11$) or 80°C water model ($L_m/L_p \approx 14$).
- At 10^{-4} g, by methanol at 35°C ($L_m/L_p \approx 95$), or thermex at130°C ($L_m/L_p \approx 100$), or R114 at 30°C ($L_m/L_p \approx 45$).

It is obvious that space-oriented mercury systems must be scaled by other fluid systems in centrifuges on earth.

In addition it can be said that a 25°C R114 prototype at 10^{-2} g can be scaled by a 25°C 1 g ammonia model ($L_p/L_m \approx 5$)., important for the developments discussed next.

Finally it is remarked that the scaling of Moon or Mars base prototype systems by terrestrial models with the same or a scaled working fluid is very well possible. The g-ratios between these planets and earth (0.16 and 0.4 respectively) lead to geometric sizes that do not differ very much in prototype and model.

Definition of useful experiments

In order to support ESA two-phase activities, experiments had to be carried out using the NLR two-phase test rig. This ammonia rig, having approximately the same line diameter as the TPX I loop [15] was used to develop, test and calibrate TPX components, and to scale low-gravity adiabatic and condensing flow. As it will be discussed in the following sections, terrestrial low temperature vertical down flow minimises the impact of gravity, hence simulates low-gravity conditions the best.

In addition it is recalled that the full size low-gravity ($< 10^{-2}$ g) mechanically-pumped R114 ESA TPHTS (Fig. 1) can be adequately scaled by the above 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 leads to a length scaling $D_p/D_m = (g_m/g_p)^{½} (\sigma/\rho_l)^{½}{}_p/(\sigma/\rho_l)^{½}{}_m \approx 4.5$ to 6.5, in agreement with the ratio of actual diameters: 21 mm for the R114 space prototype and 4.93 mm for the terrestrial ammonia model.

Concluding remarks

Scaling two-phase heat transport systems is very complicated. Only distorted scaling offers some possibilities, when not the entire loop but only loop sections are involved. Scaling with respect to gravity is hardly discussed in literature. Some possibilities can be identified, for typical and very limited conditions only.

The mechanically pumped two-phase ammonia test rig developed offers some opportunities to scale a TPX ammonia loop and a very promising application: the terrestrial scaling of a mechanically pumped R114 flight unit.

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.

MODELLING AND EXPERIMENTS

As stated, 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. Therefore we will concentrate on pressure drops in condensing and adiabatic flow and restrict the discussion to straight tubes.

Modelling equations

The total local (z-dependent) pressure gradient for annular flow is the sum of friction, momentum and gravity gradients:

$$dp(z)/dz)_t = (dp(z)/dz)_f + (dp(z)/dz)_m + (dp(z)/dz)_g \tag{10}$$

The elaborate publication [45] presents in detail the straightforward derivation of the three constituents of equation (10). As the very complicated derivations are beyond the scope of this book, only the results will be given here. The contribution of friction, can be written [46], deleting the z-dependence to shorten the notation, as:

$$(dp/dz)_f = -(32\dot{m}^2/\pi^2\rho_v D^5)(0.045/Re_v^{0.2})[X^{1.8}+5.7(\mu_l/\mu_v)^{0.0523}(1-X)^{0.47}X^{1.33}(\rho_v/\rho_l)^{0.261}+8.1(\mu_l/\mu_v)^{0.105} \; * \\ *(1-X)^{0.94}X^{0.86}(\rho_v/\rho_l)^{0.522}] \tag{11}$$

X is local vapour quality X(z). Re_v is Reynolds number

$$Re_v = 4\dot{m}/\pi D\mu_v \tag{12}$$

The fluid properties μ_l, μ_v, ρ_l and ρ_v are assumed to be independent of z, since they depend only on the mixture temperature, which usually is almost constant in adiabatic and condensing sections.

The same publication derives that the momentum constituent can be written as:

$$(dp/dz)_m = -(16\dot{m}^2/\pi^2D^4)\{[2X(1-\alpha)/\rho_v\alpha^2 - \beta(1-X)/\rho_l\alpha+(1-\beta)(1-X)/\rho_l(1-\alpha)+(1-X)/\rho_l(1-\alpha)](dX/dz)+ \\ -[X^2(1-\alpha)/\rho_v\alpha^3+(1-X)^2/\rho_l(1-\alpha)^2](d\alpha/dz)\} \quad (13)$$

α is the z-dependent local void fraction $\alpha(z)$. $\beta = 2$ for laminar liquid flow, 1.25 for turbulent flow.

The gravity constituent can be simply approximated by:

$$(dp/dz)_g = (1-\alpha)(\rho_l-\rho_v)g \cos \quad (14)$$

$g \rightarrow 0$ for microgravity 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. α is eliminated in (13) and (14) by inserting (1).

The slip factor S is to be specified. The principle of minimum entropy production [47]

$$S =[(1+1.5Z)(\rho_l/\rho_v)]^{1/3} \quad (15)$$

This is for annular flow, in which the constant Z (according to experiments) is above 1 and below 2.

$$S = \{(\rho_l/\rho_v)[1+Z'(\rho_v/\rho_l)(1-X)/X]/[1+Z'(1-X)/X]\}^{1/3} \quad (16)$$

for real annular-mist flow, annular flow with a mass fraction Z' of liquid droplets entrained in the vapour. Z' is between 0 (zero entrainment) and 1 (full entrainment). In the limiting cases $Z \rightarrow 0$ and $Z' \rightarrow 0$, (15) and (16) reduce to:

$$S = (\rho_l/\rho_v)^{1/3} \quad (17)$$

This represents ideal annular flow. It will be used here for simplicity reasons and since it allows comparison with the results of calculations found in literature. The influence of $Z \neq 0$ and $Z' \neq 0$ is interesting for future investigations.

Inserting (17) into (1) and (11, 13, 14), yields

$$(dp/dz)_m = -(32\dot{m}^2/\pi^2\rho_vD^5)(D/2)(dX/dz) \,.\, [2(1-X)(\rho_v/\rho_l)^{2/3} +2(2X-3+1/X)(\rho_v/\rho_l)^{4/3} + \\ + (2X-1-\beta X)(\rho_v/\rho_l)^{1/3} +(2\beta- \beta X-\beta/X)(\rho_v/\rho_l)^{5/3} + 2(1-X-\beta+\beta X)(\rho_v/\rho_l)] \quad (18)$$

$$(dp/dz)_g = (32\dot{m}^2/\pi^2\rho_vD^5)\{1-[1+(\rho_v/\rho_l)^{2/3} (1-X)/X]^{-1}\} \,.\, [\pi^2D^5g \cos\nu(\rho_l-\rho_v)\rho_v/32\dot{m}^2] \quad 19)$$

To solve (11, 18, 19) an extra relation is necessary, defining the z-dependence of X. A relation often used:

$$dX/dz = -X_{entrance}/L_c \quad (20)$$

(L_c is the condensation length), means uniform heat removal (linear quality decrease along the duct), which is unrealistic. It is better to use

$$\dot{m}\, h_{lv}(dX/dz) = - h\pi D[T(z)-T_s] \quad (21)$$

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

$$h = 0.018(\lambda_l\rho_l^{1/2}/\mu_l)Pr_l^{0.65}|-(dp/dz)_t|^{1/2} D^{1/2} \quad (22)$$

assuming [45] that the major thermal resistance is in a laminar sub-layer of the turbulent condensate film.

As already mentioned the two-phase flow path is almost isothermal, which implies constant temperature drop $T(z) - T_s$ (for constant sink temperature T_s), constant fluid properties and constant Prandtl number, defined by

$$Pr_l = Cp_l \, \mu_l / \lambda_l \tag{23}$$

The total condensation pressure drop is

$$\Delta p_t = \int_0^{L_c} (dp/dz)_t dz \tag{24}$$

The equations (10, 11, 18, 19, 21) and (22) 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

$$F(dX/dz, X) = 0 \tag{25}$$

Results for adiabatic flow

Figure 13 compares the pressure gradient constituents at two temperatures. The curves prove that at low temperature the gravity constituent is overruled by the other contributions. This confirms the earlier statement that low-gravity behaviour can be investigated in terrestrial tests at low temperature.

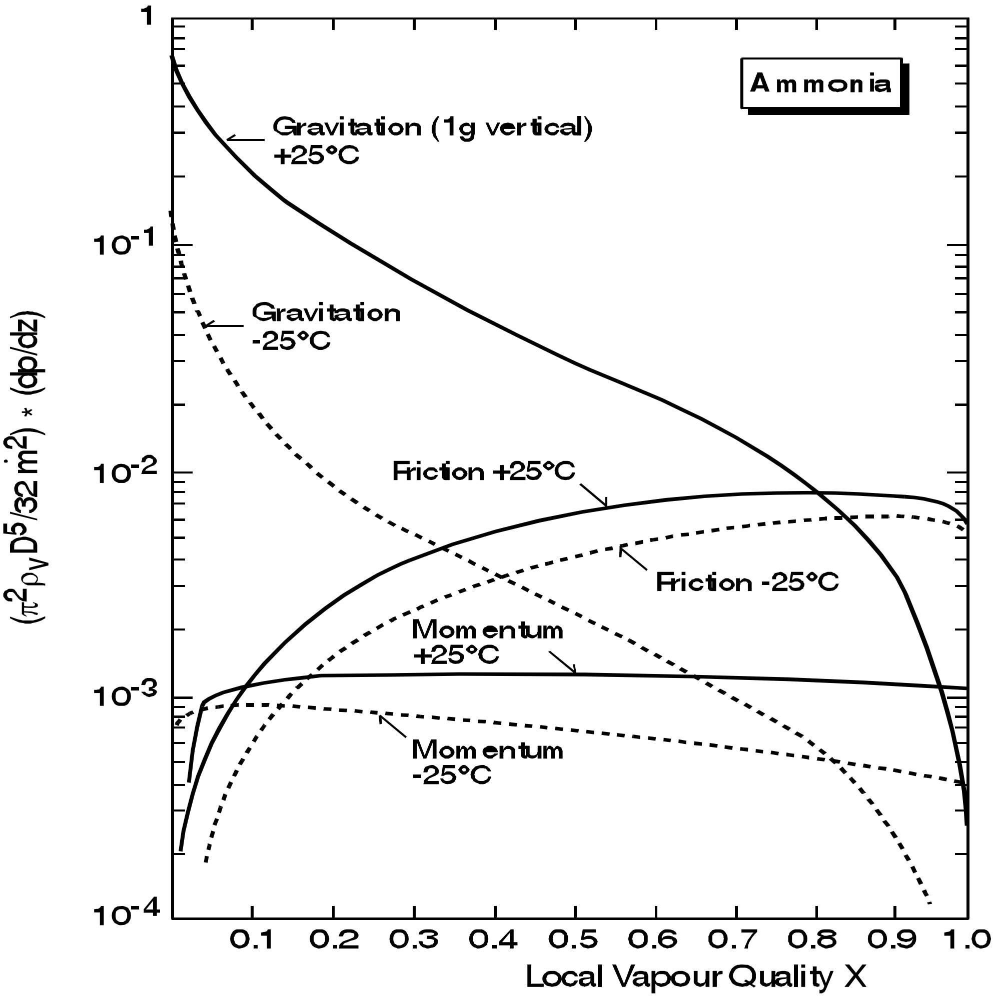

Fig. 13: Friction, momentum and gravity contributions to the local pressure gradient as a function of the vapour quality

Figure 14 shows curves calculated [41], 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 [24] an average g of the order 10^{-2}-g . These measured data lie within the boundaries of the calculated curves.

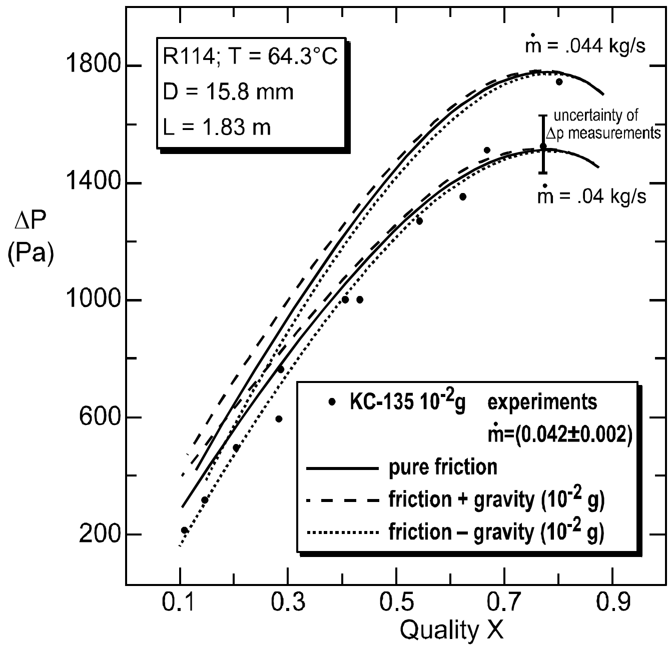

Figure 14. Measured versus predicted adiabatic pressure drops for a R114 duct

Condensation lengths

Modelling and calculations were extended from adiabatic to condensing flow in a straight duct [46], 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 in earth gravity [47], 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 in [47] was chosen as the baseline. Main characteristics are power 1 kW, line diameter 16.1 mm, ammonia temperature 300 K and temperature drop to sink 10 K. The other parameter values are shown in Table 2.

Table 2
Parameter Values

T	(K)	300	243	333
h_{lv}	(J/kg)	$1.16*10^6$	$1.36*10^6$	$1.00*10^6$
$\dot{m}$	(kg/s)	$8.64*10^{-4}$	$7.36*10^{-4}$	$9.98*10^{-4}$
μ_l	(Pa.s)	$1.40*10^{-4}$	$2.47*10^{-4}$	$0.94*10^{-4}$
μ_l/μ_v	(-)	12.30	30.66	8.54
ρ_l	(kg/m^3)	600	678	545
ρ_l/ρ_v	(-)	72.46	652.4	26.6
λ_l	(W/m.K)	0.465	0.582	0.394
Pr	(-)	1.42	1.90	1.25

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 macro-gravity level 19.6 m/s^2.

Illustrative results of calculations [46] are shown in figure 15, depicting the calculated 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 [47]. 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. Consequently the data of [47] can be considered as extremes.

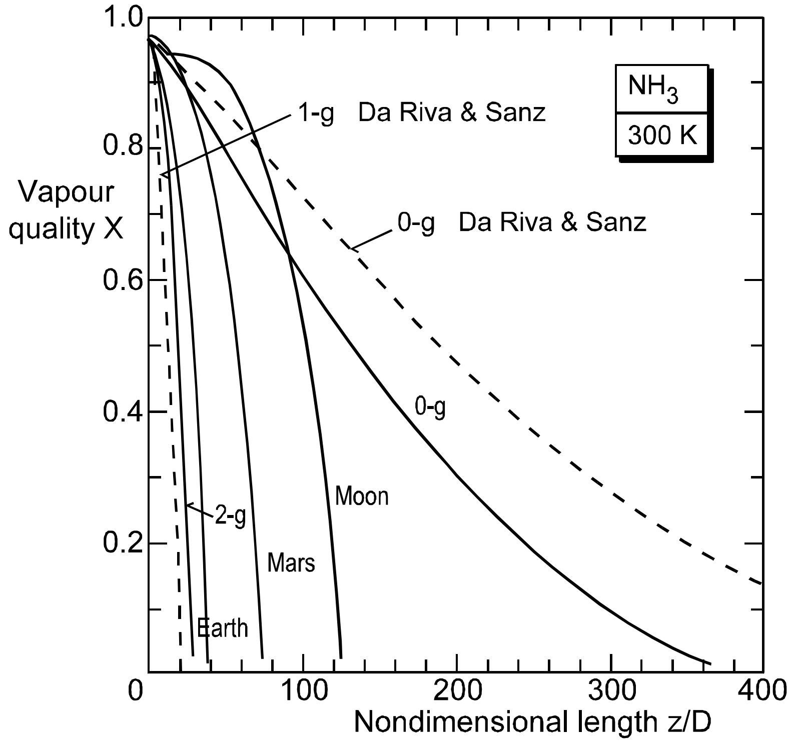

Figure 15. Vapour quality along the 16.1 mm duct for ammonia at 300 K, 1 kW, for all gravity levels

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 [46]. The calculations show that the full condensation length increases with the temperature for zero-g conditions, but decreases with temperature for the other gravity levels. This implies that the differences between earth gravity 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 [46] 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).
- Under earth gravity conditions, power and full condensation length are strongly interrelated: from L_c = 554 D at 25 kW to only 19 D at 500 W.
- The gravity dependence of the full condensation length decreases with increasing power, until the differences vanish at roughly 1 MW condenser choking conditions. The latter value is an upper limit, calculated (using the Zivi relation) 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 absolute duct lengths L_c(m) needed for full condensation under zero-g and one-g respectively [46]. It has been 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 for differences in gravity levels as compared to larger diameter systems. This was confirmed by TPX I flight data [15].

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. A complication is the lower boundary of the annular-wavy-mist flow pattern. In addition, flow pattern transitions occur at quality values, which strongly depend on temperature and line diameter.

The preceding paragraphs can be summarised by:

- The information presented confirms the results of other models i.e. 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 model equations given are useful for a better understanding of the problems that can be expected: problems related to two-phase flow and heat transfer (the necessary lengths of condensers for space applications).
- Equations and results of the calculations suggest that hybrid scaling exercises, which combine geometric and fluid-to-fluid scaling, can support the design of space-oriented two-phase heat transport systems and their components.
- With respect to the local heat transfer equation used, equation (22), it is remarked that it has a wrong lower limit $h \rightarrow 0$ for $(dp/dz)_t \rightarrow 0$, which disappears by incorporating conduction via the liquid layer. Preliminary calculations indicate that incorporation of pure conduction will lead to somewhat shorter full condensation lengths, both for zero and for non-zero gravity conditions. This implies quantitative changes only, hence the conclusions presented above remain valid.

FLOW PATTERN MAPPING ISSUES

Accurate knowledge of the gravity level dependent two-phase flow regimes is crucial for modelling and designing two-phase heat transport systems for space, as flow patterns directly affect thermal hydraulic characteristics of two-phase flow and heat transfer. Therefore flow pattern (regime) maps are to be created, preferably in the non-dimensional format of figure 10.

The three-dimensional flow pattern maps, shown in the figures 16 and 17, were created by using many K135 aircraft flight data obtained with a R12, 10.5 mm line diameter experiment [33]. The data were obtained at various g-levels, realised during the flights. The figures clearly show the gravity level dependency of the shifts in transitions from annular flow to slug flow or to stratified flow, and from slug/plug flow to annular flow and stratified flow. Figure 18 summarises the 0-g data. It is a cross-section at 10^{-2}-g of the figures 16 and 17. Figure 19 depicts the data of the low-g aircraft experiments with Cyrène, an two-phase ammonia system with a 4.7 mm line diameter [34]). Figure 20 shows the 0-g map, derived from TPX I (ammonia, 4.93 mm lines) VQS flight data, measured in the Shuttle cargo bay [15].

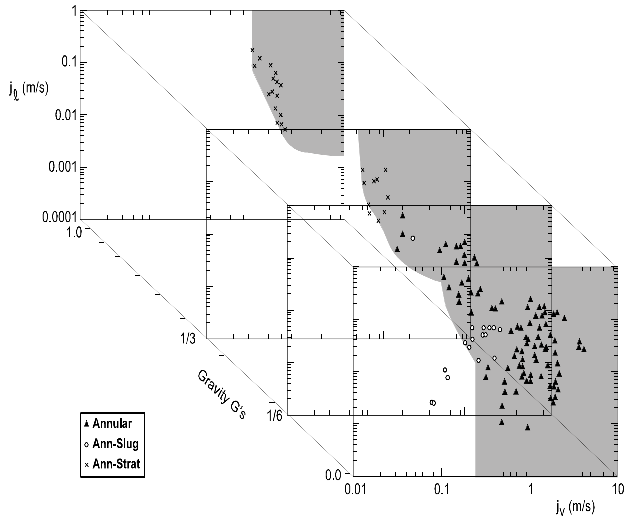

Figure 16. Annular flow: Gravity dependent three-dimensional flow pattern map [33]

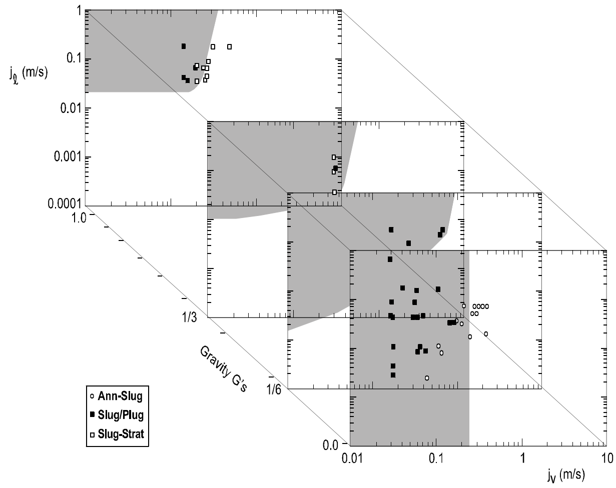

Fig. 17: Slug-plug flow: Gravity dependent three-dimensional flow pattern map [33]

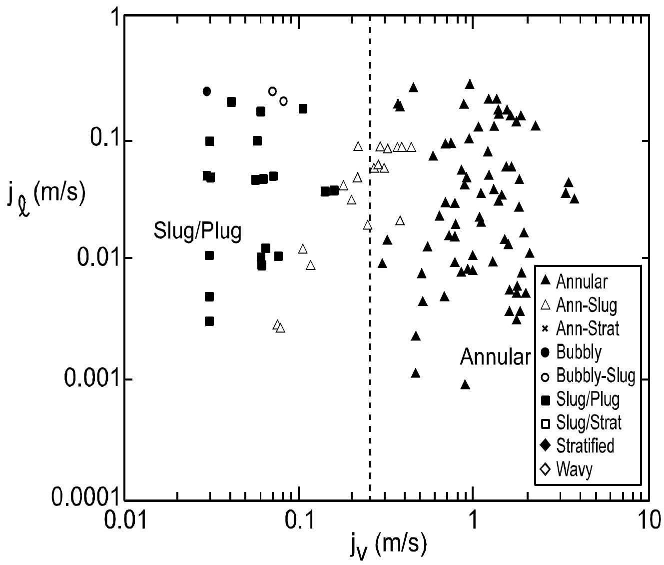

Figure 18. Flow pattern map in microgravity

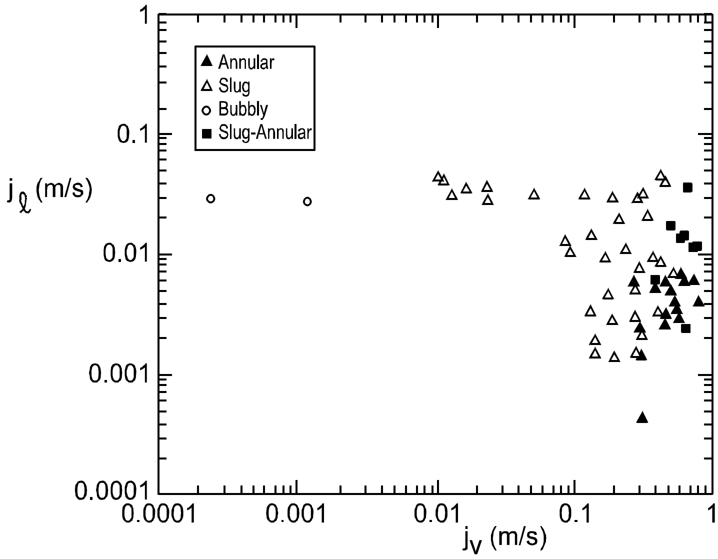

Figure 19. Cyrène flow pattern map [34]

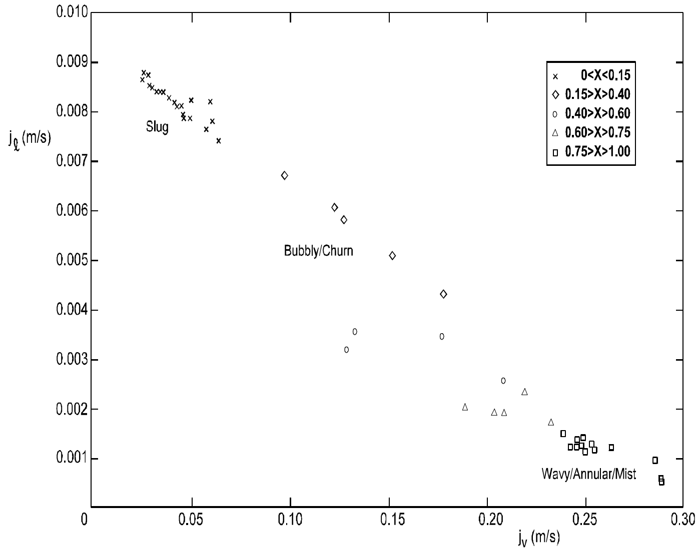

Figure 20. Flow patterns derived from TPX I vapour quality sensor data [15]

The above maps partly contradict each other. A comparison between the figures suggests that the transition to annular flow occurs in these three systems more or less at the same j_V-value 0.2-0.25 m/s, but at different j_l-values. This can be caused either by the different working fluids (R12/ammonia/ammonia) or the different inner line diameter (10.5 mm/4.7 mm/4.93 mm). More data are to be gathered to draw 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 10 or in the very good alternative three-dimensional j_v - j_l - g format, given in the figures 16 to 20. 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.

SOME FINAL REMARKS

To conclude this first lecture some remarks are made with respect to some issues, which can be important for the development of two-phase thermal control systems.

First: One must be careful in using data obtained during experiments in drop towers and low-g aircraft. Apart from the fact that the duration of these experiments is too short to yield reliable information for the steady state or slow transient heat transfer processes in real thermal control systems, many of these experiments were done with liquid-gas systems. The latter systems do not represent the real physical behaviour of the liquid-vapour flow and heat transfer in thermal control systems, being [49]:

- The phenomenon of flashing: The change of (vapour) quality induced by e.g. frictional pressure gradients, not by heat addition or withdrawal. The flashing issue will be discussed in detail in the third lecture.
- The phenomena of heat transfer by evaporation and condensation.

Second: It must be stressed (again and again), since it is not adequately discussed in literature (most probably because one did not recognise the issue), that 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.

Third: Many two-phase systems are either already in orbit or are scheduled for (near) future launch. An excellent survey is given in [50].

Fourth: To support the design of two-phase thermal control systems for future base developments on the Moon and on Mars, flow pattern maps are currently being created for reduced gravity environments by KC-135 flights, flying during up to 20 seconds a Mars (0.4 g)or Moon (0.14 g) trajectories [51].

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