Numerical Simulation and Analyses of Flow Rate Exchange between Accumulator and Main loop

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Abstract. In order to study the dynamic behaviors of heat and mass transfer between accumulator and mechanically pumped two-phase loop (MPTL) system, a transient numerical model is developed by using the time-dependent Navier-Stokes equations. By comparison between simulation and test results, it is found that the error of numerical model is in the range of ±10%, which verifies the validity and accuracy of the model. Simulation results show that the accumulator will exchange fluid with the main loop in response to heat load variations. In this case, the temperature and pressure of two phase fluid in accumulator, and the total system flow resistance will be affected. The rate of mass transfer between accumulator and main loop will increase along with the charge amount of working fluid, and also for the variation trend of temperature and pressure of two phase fluid in the accumulator. The model can be used to study the operating state, flow and heat characteristics of MPTL system.

Nomenclature

Symbols

A	Area (m^2)
C_p	Specific heat at constant pressure (J/(kg·°C))
e	Internal energy (kJ/kg)
f	Friction coefficient
$\mathbf{g}$	Gravity acceleration (m/s^2)
h	Specific enthalpu (kJ/kg)
h_{lv}	Latent heat of vaporizaiton (kJ/kg)
l	length (m)
m	mass (kg)
$\dot{m}$	Mass flow rate (kg/s)
$\mathbf{n}$	Normal vector
p	Pressure (N/m^2)
q	Heat flux (W/m^3)
$\dot{Q}$	Heat power (W)
T	Temperature (°C)
t	Time (s)
$\mathbf{u}$	Velocity vector
U	Coefficient of heat transfer (W/(m^2•°C))
V	Volume (m^3)
x	Quality
Δz	Height (m)

Greek symbols

Δ	Difference
ρ	Density (kg/m^3)

Subscripts/ superscripts

a	Accumulator
al	Liquid phase in accumulator
av	Vapour phase in accumulator
f	Fluid
i	Mesh
j	Junction
l	Liquid
sat	Saturated
v	Vapour

1 Introduction

With the development of space technologies, thermal control systems for satellite are faced with several difficulties and challenges, such as temperature control in micro dimensional room, high thermal control precision and stable temperature, heat collection in large area, high heat flux, et al [1]. Several research and flight cases demonstrated the excellent performances of heat transfer and temperature control of MPTL system. As a most advanced technology in spacecraft thermal control field, MPTL system possessing the characteristics of accurate temperature control of multi-heat sources, high stability level and large transport distance can solve the mentioned problems better, which is of interest for the thermal control of large power laser, optical pay-loads and active antenna.

MPTL system requires an accumulator to perform gas and liquid management, to adjust the working fluid distribution and to control the saturation temperature of the system[2]. The accumulator is one key component which acts like the brain of system. In this paper, one two phase temperature controlled accumulator (short for accumulator) is used. The accumulator requires a fixed volume tank whose temperature is controlled by external heating and cooling. The fluid condition within the accumulator must be in two-phase in order to maintain

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MPTL system saturation control. However, MPTL system involves complicated coupled processes of heat and mass transfer between accumulator and main loop, for the purpose of realizing the functions of temperature control and fluid management.

Compared with the experimental investigations, a few numerical studies of MPTL systems have been conducted so far. Hence, an accurate model is needed to study the thermal and flow performances. The MPTL system involves complex heat and mass transfer processes, rendering the mathematical modeling highly challenging and more difficult. Huang et al. [3] carried out the simulations to investigate the coupling behaviors between accumulator and the loop for MPTL system, which revealed the mutual disturbance between them. van Genrner et al. [4, 5] developed a software for transient two-phase system with working fluid of CO_2 and R134a, and model numerically solves the one-dimensional time dependent Navier-Stokes equations, and the factors that affects fluid exchange behaviors were analyzed. However, the present developed models, studied for the dynamical thermal and flow behaviors between accumulator and main loop, simplified for the physical model. Particularly, the mutual affect for the vapor and liquid phases within the accumulator and the dynamical behaviors of main loop in response to the mass transferring processes between accumulator and main loop are not considered.

In this paper, a transient numerical system model is established, which is used to study the behaviors of heat and mass transfer between accumulator and main loop. In the following, the MPTL system composition and mathematical model are presented first, followed by the comparison between test data and simulation results, and the discussion and analyses of heat and mass transfer processes between the accumulator and the main loop. Finally, some conclusions are drawn.

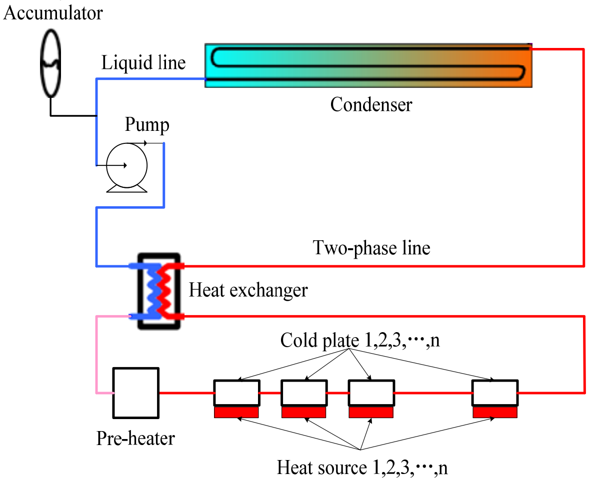

Fig. 1 The flow schematic of MPTL system

2 Analysis and modelling method

MPTL system is a thermal management system, which includes the processes of heat collection, transport and rejection, as shown in Fig. 1. The pump is used to circulate fluid in the loop, and the accumulator controls the system working pressure. The saturated liquid in the cold plates is used to absorb the heat from the heat sources by the latent heat. Then the liquid and vapor mixture flows into the heat exchanger, in which heat from the two-phase fluid is used to warm the cold liquid that comes from the pump. In the condenser, the vapor-liquid two phase fluid condenses back into cold liquid. In the heat exchanger and preheater, the cold liquid is warmed to saturation temperature. And that cycle repeats.

2.1 Mathematical Model

The fluid flow in MPTL system can be modeled with the integral form of Navier-Stokes equations [6].

$$\int_V \frac{\partial \rho}{\partial t} dV + \oint_A \rho \mathbf{u} \cdot \mathbf{n} dA = 0 \quad (1)$$

$$\int_V \frac{\partial (\rho \mathbf{u})}{\partial t} dV + \oint_A \rho \mathbf{u}\mathbf{u} \cdot \mathbf{n} dA = -\int_V \nabla p dV + \int_V \rho \mathbf{g} dV \quad (2)$$

$$\int_V \frac{\partial (\rho e)}{\partial t} dV + \oint_A \rho \mathbf{u} e \cdot \mathbf{n} dA = \int_V q dV - \oint_A p \mathbf{u} \cdot \mathbf{n} dA + \int_V \rho \mathbf{u} \cdot \mathbf{g} dV \quad (3)$$

The staggered gid method is used to discretized the computational model. Fig. 2 shows the schematic of mesh discretization of loop. All the scalar parameters and physical properties, including pressure, temperature, quality, et al., are defined on the nodes. The vector parameters, u, are located on the junctions, which are staggered half step lengths from main nodes.

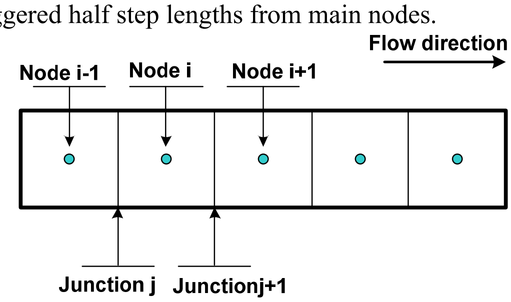

Fig. 2 The schematic of mesh discretization for main loop

By the integration of the equations (1)-(3), the following equations are obtained,

$$\left[V \frac{\partial \rho}{\partial t} \right]_i = \dot{m}_j - \dot{m}_{j+1} \quad (4)$$

$$p_i - p_{i+1} = \frac{l_{i+1,i}}{A_{j+1}} \ddot{m}_{j+1} + \frac{f_{j+1}}{2} \cdot \rho_i \cdot \left(\frac{\dot{m}_{j+1}}{\rho_i A_{j+1}} \right)^2 - \rho_i g \Delta z_i \quad (5)$$

$$\left[V \left(\frac{\partial e}{\partial t} \rho + e \frac{\partial \rho}{\partial t} \right) \right]_i = \dot{m}_j h_{i-1} - \dot{m}_{j+1} h_i + \dot{Q}_i \quad (6)$$

In the accumulator, there is a non-equilibrium thermodynamic state, which differs from the single equilibrium thermodynamic state. In the single equilibrium thermodynamic state, the heat transfer and mixing processes are assumed to occur over a negligibly small time scale. To model the phasic non-equilibrium state, in which the vapor and liquid within a two-phase control volume may not be at the same temperature (and in some instances not the pressure), two control equations for vapor and liquid phase are required from literature [7].

The pressures of vapor and liquid phase in the accumulator are the same, which are determined by the vapor phase. The pressure of accumulator can be calculated as follows

$$p_m - p_{al} = \frac{l_{al,m}}{A_q}\ddot{m}_q + \frac{f_{al,m}}{2\rho_{al}}\frac{\dot{m}_q^2}{A_q^2} \tag{7}$$

Accordingly, the continuum, momentum, and energy conversation equations of the node m, which connects with accumulator, are given by

$$\left[V\frac{\partial\rho}{\partial t}\right]_i = \dot{m}_p - \dot{m}_{p+1} - \dot{m}_q \tag{8}$$

$$\begin{aligned} p_{al} - p_{m+1} = \frac{l_{al,m}}{A_q}\ddot{m}_q + \frac{f_{m,al}}{2\rho_n}\left(\frac{\dot{m}_q}{A_q}\right)^2 + \frac{l_{m,m+1}}{A_{p+1}}\ddot{m}_{p+1} \\ + \frac{f_{m,m+1}}{2\rho_m}\left(\frac{\dot{m}_{p+1}}{A_{p+1}}\right)^2 + \rho_m g\Delta z_m \end{aligned} \tag{9}$$

$$\left[V\left(\frac{\partial e}{\partial t}\rho + e\frac{\partial\rho}{\partial t}\right)\right]_m = \dot{m}_m h_p - \dot{m}_{m+1}h_{p+1} - \dot{m}_a h_{al} \tag{10}$$

Besides, two-phase flow phenomenon exists in the components of pre-heater, cold plates and condensers. The relative enthalpy can be used to determine the condition of liquid or saturated fluid for these components. Eq. (11) is used as follows

$$h_{l\_sat} - h = \begin{cases} \dot{m}\cdot C_{pl}\cdot\left(T_f - T_{sat}\right) & h_{l\_sat} \ge h \\ -x\cdot h_{lv} & h_{l\_sat} < h \end{cases} \tag{11}$$

2.2 Parameters and conditions

2.2.1 Model parameters

Table 1 gives the main parameters of each component. Flat-plate heat exchangers with small channels are used as the preheater and cold plates. The rectangle section of inner flow channel is 2 mm×1.5 mm, which is equivalent to hydraulic diameter of 1.7 mm. The heat load loaded on the preheater and cold plates is distributed on each node uniformly. The contact heat transfer coefficients between pipe and condenser and between accumulator and sink are tested by the thermal contact resistance tester in the lab. The values are 1500 W/(m^2°C) and 1000 W/(m^2°C), respectively.

Table 1. The parameters of model and test

Component	Description
Pump	Flow rate：1g/s
Accumulator	Volume：200ml; Filling amount：89 g Temperature range: 20°C±0.3°C
Pre-heater	Power: 50W
Cold plates	Power: 100W
Condenser	Temperature range: 10°C±0.5°C
Tubes	OD: 0.003m; ID: 0.002m

2.2.2 Initial and boundary conditions

Initial conditions: T=20°C; quality x=0; pressure p=0.86MPa.

Boundary condtions: (1) the wall of accumulator T_a=20°C; (2) condenser T_c=10°C±0.5°C; (3) the wall of pipe (except condenser pipe) $\partial T/\partial a = 0$.

2.2.3 Property of working fluid

The working fluid for MPTL system is ammonia, possessing the characteristics of large latent heat of evaporation, high efficiency of heat transfer, stable property. Ammonia is the common working fluid of two phase heat transfer devices (such as heat pipe, loop heat pipe) for spacecraft thermal control system. In this model, all the relevant thermo-physical parameters of the working fluid, such as vapor or liquid density, viscosity, saturated pressure, latent heat of evaporation, thermal conductivity of the liquid and vapor, etc., are obtained by the calling program REFPROP for each volume node and junction area, at each time step.

2.2.4 Computation method

A set of coupled nonlinear equations are obtained from Eq. (4)-(11). The time terms are using an implicit finite difference scheme, and the main variables, pressure, flowrate, specific enthalpy, are using the fully implicit scheme, and other parameters are using the explicit scheme.

The main variables of nodes for main loop include p, $\dot{m}$, h, and the main variables of accumulator are Tal, Tav, $\dot{m}_q$. The other variables are obtained from REFPROP database and the function relations with the main variables. The number of nodes for the model is N+1 (N is the number of nodes for main loop, and 1 represents the accumulator), and the number of equations on each node is 3. Hence, the number of the discrete algebraic equations is 3(N+1). The nonlinear equations are linearized by Newton-Raphson method firstly, and then they are solved by the Gaussian iteration method.

2.3 Experimental verification setup

Fig. 3 shows the photo of the experimental verification setup. Two-phase temperature control accumulator is used in this setup. There is a capillary structure in the accumulator, which can meet the applying requirements for both micro-gravity and one-g environments. Heater

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and cooling devices are applied on the wall of accumulator, which can control the temperature of accumulator. Two preheaters and four cold plates are also flat-plate heat cold plates, which are connected in series. Gear micro-pump (Model: Micropump GA-V21) is used in the test setup. Two mass flow meters (Model: SKSH DMF-1) are installed in the locations at the downstream of pump and condenser, respectively. The difference value of two mass flowmeters is used to obtain the exchange rate between accumulator and main loop. In order to reduce the heat leakage from environment to MPTL system, all the pipes are covered by heat insulation materials.

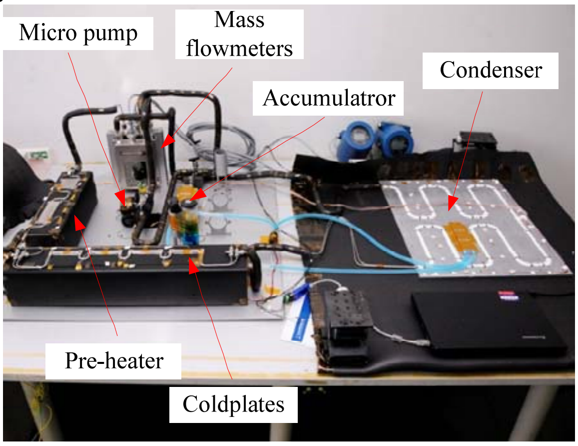
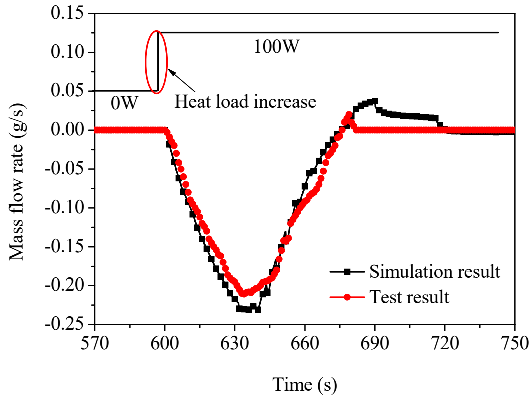


Fig. 3 Practicality picture of MPTL system

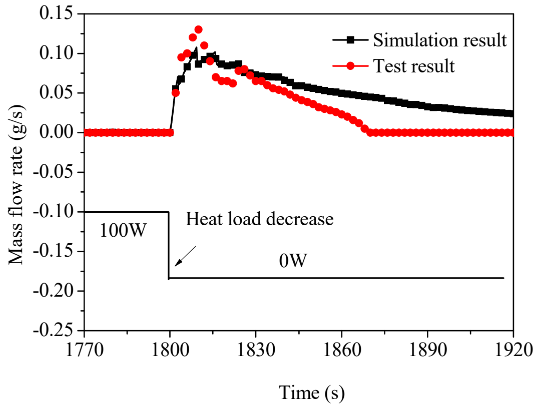
3 Results and discussion

Fig. 4 shows the comparison between the simulation results and test data. It is indicated that the numerical results are in consistent with the experiment results by studying the exchange rate between accumulator and main loop when heat load changes. The variation trends for the two results are basically agreed. The errors for predicted numerical results are in the range of ±10%. An explanation for this is that in the model, the thermal resistance between heater and the wall of accumulator and the thermal capacity of accumulator are neglected, and as a result, the fluid in the accumulator will heat up slightly quicker in the simulation than in the test, which results in a higher flowrate peak into the accumulator.

With regard to the phenomenon of mass exchange, as a result of the increase or decrease in the cold-plates' heat input, liquid will flow into or out of the accumulator. When the heat load increases, a large amount of vapor in the cold-plates is suddenly generated and then the volume of pipe located the downstream cold-plates will be occupied by the generated vapor, which will give rise to flowing from main loop to the accumulator, as shown in Fig. 4(a). When the heat load decreases, vapor in the cold-plates will be reduced quickly and the pipe located the downstream cold-plates will be re-filled by the fluid from the accumulator, which will cause backflow from the accumulator to the main loop, as shown in Fig. 4(b). In the latter case, the fluid from the accumulator must be pure, which is important for the safety of pump running continuously.



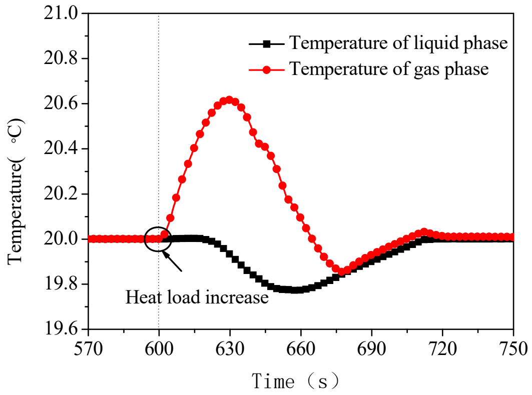
(a)



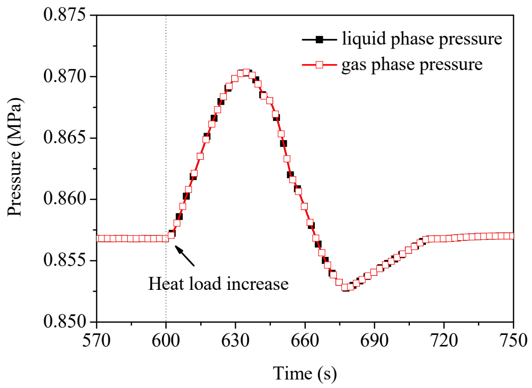
(b)

Fig. 4 Comparison between simulation and test results

Fig. 5 shows the temporal evolution of temperature and pressure for vapor and liquid phases in the accumulator when the heat load increases. The temperature and pressure in the accumulator will change as a result of the increasing power. The variation trends of temperature for gas and liquid phases have large discepancy while the pressure difference between them is very small. The explanation of changing processes of temperature for gas and liquid phases is as follows. The cold liquid from the main loop flows into the accumulator and then mixes with the liquid in it. Compressing the space of gas phase quickly will give rise to the temperature rising rapidly. The maximum value of temperature increase is 0.6 °C as the flow rate reaches the peak value. After that, the flow rate decreases gradually, and the temperature of gas phase drops until it is the same as the liquid phase due to the heat exchange at the gas and liquid interphase. Influenced by the larger thermal capacity of liquid phase in the accumulator, the temperature of liquid phase drops is delayed and slower. The peak value of temperature drop for liquid phase is 0.25°C. Fig. 5(b) shows that the pressure variations of gas and liquid phases are in accordance, and the changing trend of temperatures are the same as the trend of temperature of gas phase in the accumulator, which indicates that the working temperature in accumulator is decided by the gas phase.



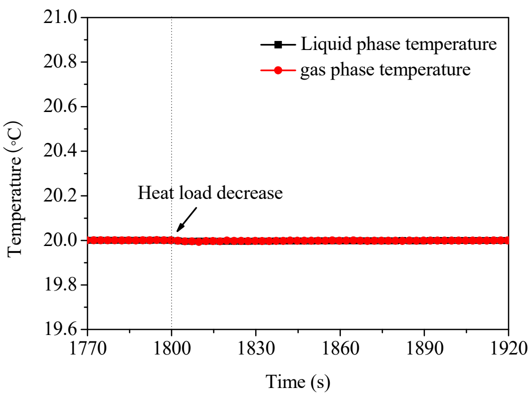
(a)



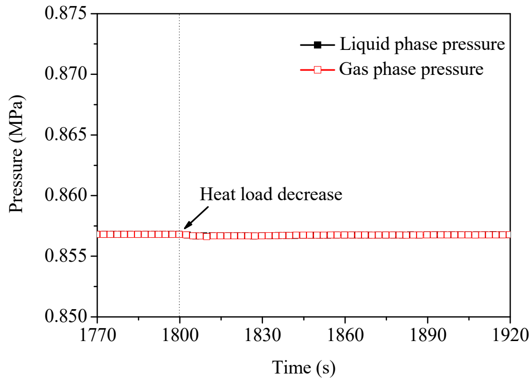
(b)

Fig. 5 The temporal evolution of temperature and pressure of two phase fluid in accumulator in response to heat load increase

Fig. 6 gives the temporal evolution of temperature and pressure for vapor and liquid phases in accumulator when the heat load decreases. Unlike the variation trend of parameters in the case of power increasing, the differences for temperature and pressure between gas and liquid phase is small and the temperature and pressure change with a consistent between them. An explanation for this is that in this case, the backflow rate from accumulator to the main loop is low relatively, and thus the decreasing rate of volume for gas phase changes is small, which leads to the low variations of temperature and pressure for gas and liquid phase.



(a)



(b)

Fig. 6 The temporal evolution of temperature and pressure of two phase fluid in accumulator in response to heat load increase

Fig. 7 shows the profiles of mass flowrate of main loop along flow distance at two time instants, which are corresponding to the moment of maximum values when the flowing into and out of the accumulator are peak. After the heat load increasing or decreasing, the flowrates from the pump exit to cold plates' inlet are the same while the rates in other locations are affected by the power change. When heat load increases, the mass flowrates of two-phase between cold plates' outlet and the condenser's inlet are increasing linearly. In this case, with the increase of quality of fluid out of the coldplates, the propotion of gas phase increases. The fluid with increased quality travels through the MPTL system with increased fluid velocity. Downstream of this increased quality front, the fluid adopted the velocity of front with higher density is speeded up. When the fluid flows into condenser, the accerlation process will be stopped. When heat load decreases, the quality of fluid decreases, and thus the velocity reduces. When the fluid passes through the condenser, the quality will be decreased again. As the quality reduces to zero, the mass flow rate decreses to the minimum value.

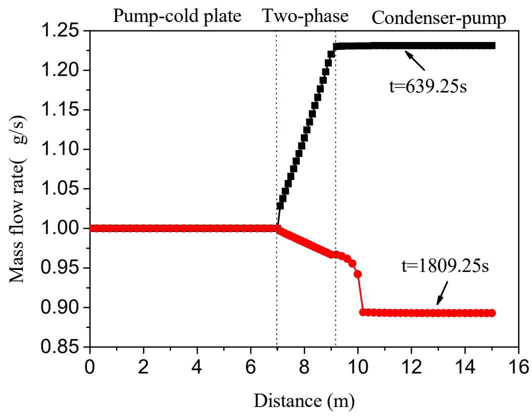


Fig.7 Profiles of mass flow rate along flow distance

4 Conclusions

Numerical simulations have been carried out for a better understanding of the phenomenon of heat and mass transfer between accumulator and main loop, temperature and pressure variations of gas and liquid phase in the accumulator and the variation of mass flow rate of main loop. The main conclusions from the numerical simulations are given as follows. By comparison between simulation and test results, it is found that the error of numerical model is in the range of ±10%, which verifies the validity and accuracy of the model. When the heat load changes, the temperature and pressure of two phase fluid in accumulator, and the total system flow resistance will be affected. The model can be used for predicting the transient characteristics of MPTL system, and for understanding and analyzing MPTL performance.

References

1. van Es, J., Pauw, A., van Gerner, H. J., et al., AMS02 tracker thermal control cooling system commissioning and operational results, 43rd International Conference on Environmental system, Colorado, USA (2013).
2. Laudi, E., AMS-02 Tracker thermal control system: development of new technologies for manufacturing of two-phase cooling system (Ph. D. thesis), University of Perugia, Italy (2016).
3. Huang, Z. and Z. He, et al., Coupling Between an Accumulator and a Loop in a Mechanically Pumped Carbon Dioxide Two-Phase Loop. Microgravity Sci. Technol, 2011. 21(Supp1): p. S23-S29.
4. van Gerner, H. J. and Braaksma, N., Transient modelling of pumped two-phase cooling systems: Comparison between experiment and simulation, 46th International Conference on Environmental Systems, Vienna, Austria (2016).
5. van Gerner, H. J., Bolder, R., and van Es, J., Transient modelling of pumped two-phase cooling systems: Comparison between experiment and simulation with R134a, 47th International Conference on Environmental Systems, Charleston, South Carolina, USA (2017).
6. Batcheler, G. K., An introduction to fluid dynamics, Cambridge University press, (1967).
7. Leonid, V. J., Marco, M. and Claudio, F., Advanced design of a “low-cost” loop heat pipe, SAE paper 2009-01-2519, (2009).