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IIIII Development of Boiling and Two-phase Flow Experiments on Board ISS IIIII
(Review)

Development of Boiling and Two-Phase Flow Experiments on Board ISS (Condensation Section)

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Abstract

Thermal design and evaluation test for condenser system was conducted for the planned ISS experiments on two-phase boiling flow promoted by JAXA. Here perfluorohexane is planned to be used as a working fluid. In this condenser, eight rectangular copper tubes where circular channel of 6mm in diameter was installed on a flat type cold plate; same one which had be operated as thermal control system of on-orbit experimental modules. Arithmetic thermal model of condenser was established considering tube wall conductivity, heat transfer coefficient on condensation and cooling water flow. Specification of the condenser was determined for 400 W of the maximum heat transport requirements through the analysis by this arithmetic model. Evaluation tests using BBM and EM were conducted to verify thermal performance of condenser. In the BBM test FC72 was used as a working fluid. System requirement for condenser; that is, liquid subcooling at outlet is 10 K and more in consideration of suction performance of circular pump was proved to be satisfied.

Keyword(s): Condenser, Condensation, Boiling flow, Heat transfer coefficient, Arithmetic thermal model, Evaluation test
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1. Introduction

On the manned spacecraft or facility, thermal management, such as heat transport and heat rejection is one of the essential technology to be established. Especially two phase thermal management technology utilizing latent heat transport is favorably effective for the application to a large scale spacecraft system in future since the high amount heat transport and rejection are practiced by less flow rate of working fluid[1)].

The project on two-phase boiling flow experiments has been carried out as the so-called TPF "Two Phase Flow" promoted by JAXA for these several years[2)]. In this project, interfacial behaviors and heat transfer characteristics in boiling two-phase flow under microgravity condition will be investigated. The heat generation in the boiling test section including heat supplied into the working fluid must be transported to ISS thermal management system through the condenser assembled on this experimental test loop.

In order to make thermal design of condenser, we have to know condensation heat transfer coefficient under microgravity condition. Best reviews some research about reduced-gravity condensation, in the reference (1) and introduces theoretical approach and some example for reduced gravity experiments mainly by aircraft[1)]. Keshock and Sadeghipour constructed a theoretical model for shear driven condensation in externally cooled tube and difference between under 1G and 0G condition for condensation heat transfer coefficient was investigated[3)]. They showed that heat transfer coefficient in 0G is about half of 1G condition[3)].

Recently, Lee, et al. investigated thermal characteristics of flow condensation in externally cooled circular tube in microgravity condition by parabolic flight[4)]. This study was conducted as a part of NASA project of Flow Boiling and Condensation Experiment (FBCE) in ISS. In this study, observation of condensing process, measurement of heat transfer coefficient, and comparison with results by empirical equations were conducted.

As mentioned above, there are some researches investigating condensation heat transfer in microgravity, however, we can't find results by long duration experiment, e.g. on board space station or artificial satellite. Therefore it can be said that there doesn't exist reliable equations for condensation heat transfer coefficient for microgravity condition. In consideration of these situations, we made thermal design for condenser by considering

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conservative correlation factor on empirical equation for condensation heat transfer coefficient in 1G condition.

The present paper introduces the thermal design and the evaluation test for the condenser system.

2. Structure of Condenser for TPF Experimental Loop

Figure 1 shows schematic diagram of TPF experimental loop[5]. Flow visualization as well as measurement of temperature and pressure in boiling two phase flow will be conducted at transparent heated tube, metal heated tube and adiabatic tube section. Two phase flow from test section flows into condenser shown as “8” in **Fig.1** and is cooled by water in cold plate before recovering subcooling condition at pump inlet.

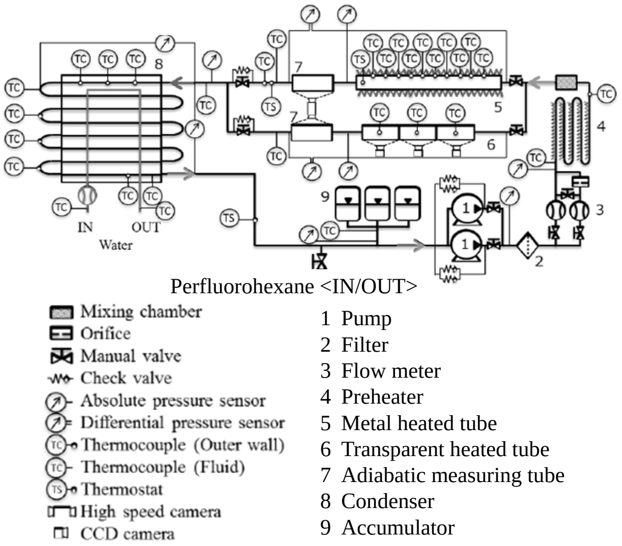


Fig.1 Schematic Diagram of TPF Experimental Loop[5]

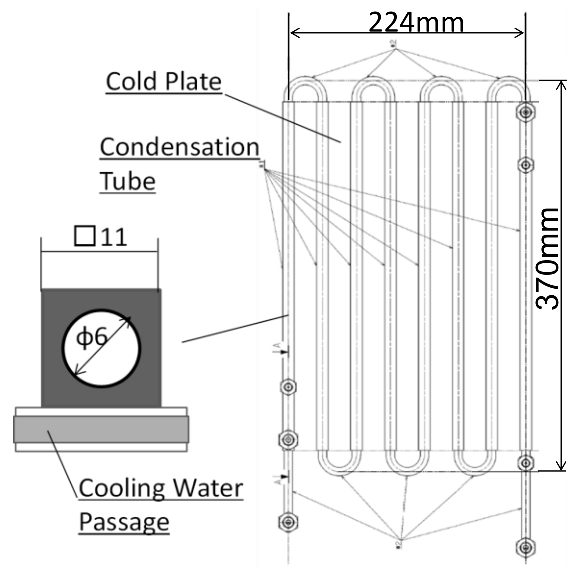


Fig.2 Structure of Condenser for TPF Experimental Fluid Loop

Figure 2 shows structure of the condenser. The condenser tube was assembled on a flat type cold plate which had employed for on-orbit system and high reliability in practice. The condenser tube is square rod of 11 mm × 11 mm with a circular flow path of 6 mm in diameter. Each rod is connected by U-shape tube. The square tube is made of copper in order to realize less thermal resistance by the high thermal conductivity of wall section. High heat conductive adhesion is used for the joint surfaced between the condenser tube and the cold plate. Measurement points for temperature and pressure are shown in **Fig. 1**. At the first straight tube from condenser inlet, three measurement points for fluid temperature as well as wall temperature are located. Fluid temperature at U-bend and most downstream straight tube was also installed. From these temperature, we plan to estimate local and overall condensation heat transfer coefficient under microgravity condition.

3. Arithmetic Thermal Model of Condenser

In order to make thermal design for condenser, we constructed arithmetic thermal model. **Figure 3** shows an arithmetic thermal model for the condenser. It is considered that one dimensional thermal path is applied from fluid in the condenser tube to cold water in the cold plate through the condenser tube wall, the adhesion, and the cold plate wall to obtain thermal resistance. Heat resistances by thermal conduction are considered on solid area and heat resistances by heat transfer are calculated in fluid area in cold plate and condenser. The condensation heat transfer coefficient under microgravity condition is given as a half value from the empirical equations for the ground based condition. We utilized empirical equations in commercial code for heat exchanger design of HTRI Xchanger Suite[6] by Heat Transfer Research Inc. The half value is assumed from the fact that the condensation length under microgravity is twice as long as under 1G condition by the numerical calculation[1)3)]. Thermal resistance through condenser tube wall is calculated by two dimensional thermal conduction analysis. Fluid temperature and quality are calculated via above thermal resistances. Local pressure is obtained by calculating local pressure loss considering enlarging effect by two phase flow. Fluid temperature in saturated condition is calculated by relationship

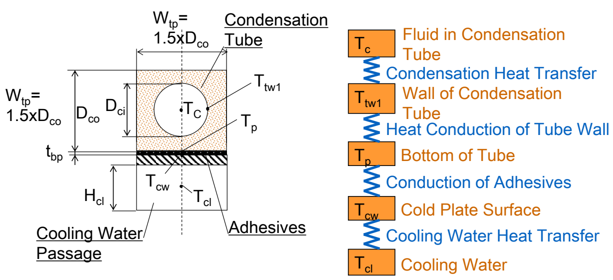


Fig. 3 Arithmetic Thermal Model of Condenser

Table 1 Data for numerical calculation

Fluid	Hot Side	FC72
	Cold Side	Water
Hot Side	Inlet Pressure	0.103MPa
	Inlet Temperature	55.7 deg C
	Inlet Quality	1.0
	Mass Flow Rate	0.0042kg/s
Cold Side	Inlet Temperature	23 deg C
	Mass Flow Rate	45kg/h
	Heat Transfer Coefficient	2000 W/m^2/K
Condenser Tube	Inner Diameter	ϕ4mm ϕ6mm
	Length	2m (ϕ6mm)

between saturated temperature and pressure for working fluid FC72. Water temperature in cold plate are calculated by considering heat amount through above mentioned heat path from condenser to cold plate. In this thermal model, thermal conductance toward condenser tube axis isn't considered, which may not provide so unreasonable solution in saturated area, since temperature gradient toward tube axis is not so large. However, thermal conduction has to be considered in subcooling region since temperature gradient in this area is not so small. Construction of thermal model considering thermal conduction toward tube axis will be done in next step of research.

Numerical calculation was conducted by the arithmetic thermal model on the following conditions shown in **Table 1**. Microgravity condition was simulated by adopting half value of ground based condition as condensation heat transfer coefficient.

We provided the additional following condition as required specification for the condenser; outlet liquid subcooling is 10 K at 400 W for the equivalent mass flow rate. This condition was determined in consideration of measured suction performance of mechanical pump.

Figure 4 shows the distribution of thermal resistance on flow path. The thermal resistance by condensation increases toward downstream section since the condensation liquid film becomes thicker as shown in **Fig. 4**. We can see that other major thermal resistance is achieved in the cold water passage, and that the resistance at the tube wall and the adhered portion between the condensation tube and the cold plate are sufficiently small. It is found that the thermal resistance on condensation side is dominant at the downstream.

Figure 5 shows distribution of fluid temperature and subcooling with tube length in the hot side passage. It is confirmed in the experiments that 9 K in liquid subcooling for the 6mm-tube and 14 K for the 4mm-tube are obtained, which indicated achievement of our target. We had already confirmed that the cold plate had enough area to install the condensation tube with required length.

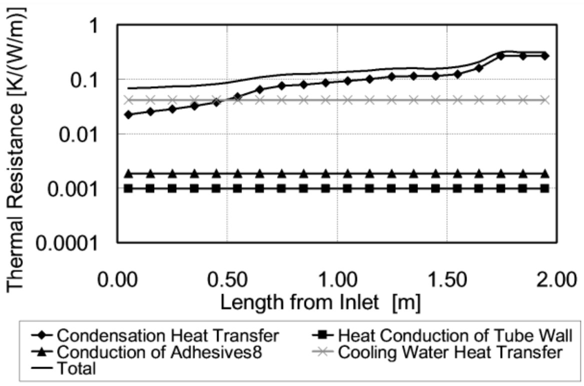
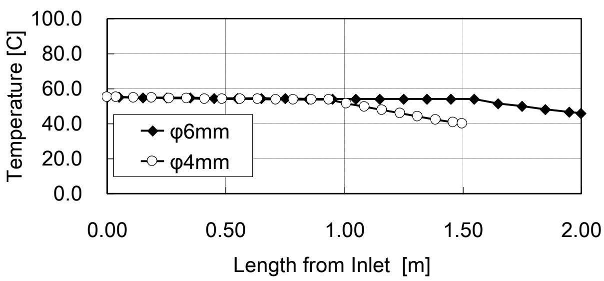
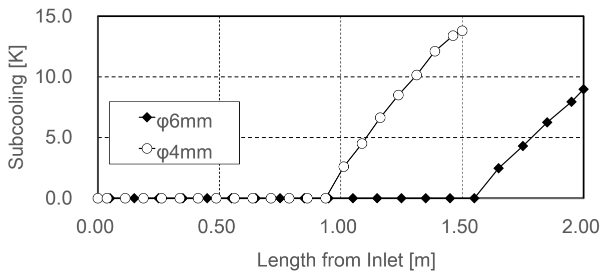


Fig.4 Thermal resistance at local point



(a)Temperature in condenser tube



(b) Subcooling in condenser tube

Fig.5 Temperature and sub-cool temperature resistance at local point

4. Results of Evaluation Test for BBM Condenser on Ground Based Condition

BBM (Bread Board Model) test on ground based condition was carried out to confirm performance of condenser.

Test loop for the condenser system is shown in **Fig. 6**. Working fluid, FC72, circulates the condenser loop from a constant temperature bath shown in **Fig. 6**. Steam of FC72 is supplied into the condenser tube from a steam generator at a quality of 1.0. Cooling water of 23 ºC flows into the cold plate and heat is exchanged between the cold plate and the condenser tube. Heat loss from the condenser tube is calculated by the fluid flow rate and the temperature difference between inlet and exit of the old plate. The working fluid is condensed completely and turns to liquid at the exit of condenser tube. Heat gain of the cold plate is also calculated by water flow rate and temperature difference between inlet and exit of the cold plate.

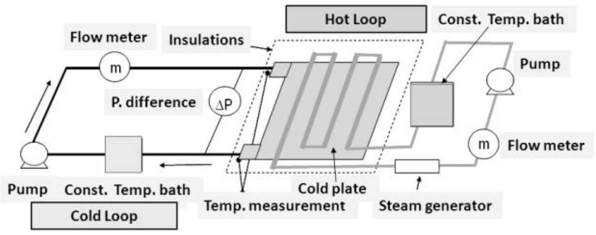


Fig.6 Ground Test Loop of Condenser System

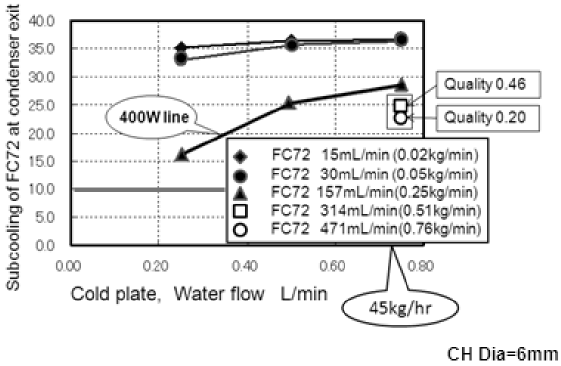


Fig.7 Liquid Subcooling of FC72 at Exit of Condenser Tube

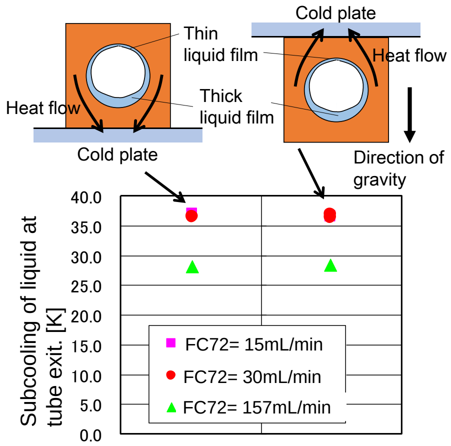


Fig.8 Effect of Orientation of Condenser on Thermal Performance

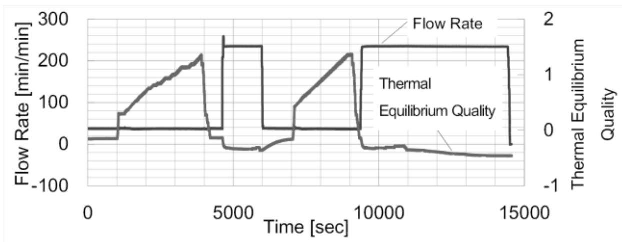
Liquid subcooling at the condenser tube exit is shown for the flow rate of cooling water in the cold plate in **Fig. 7**. According to the test results, the flow rate of working fluid FC72, is given as 0.25 kg/min in the present experiment on boiling for the maximum heat transport of 400 W.

Here, the liquid subcooling of working fluid is 28 K at the condenser tube exit for the flow rate of cooling water of 45 kg/h (0.75 L/min), and this satisfies sufficiently the requirements of ISS experiment. For the higher fluid flow rate of FC72 0.51 kg/min and 0.76 kg/min for example, it is unable to condense completely in the condenser tube. We confirmed that subcooling more than 20 K at condenser outlet was performed when inlet quality was 0.46 and 0.2 for 0.51 kg/min and 0.76 kg/min in fluid flow rate of FC72 as shown **Fig. 7**.

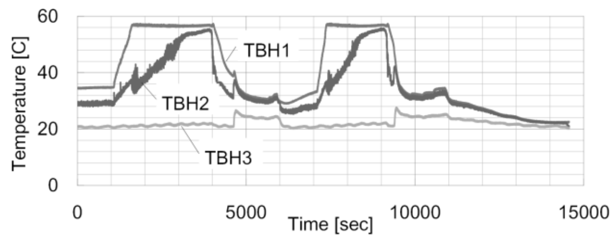
In order to investigate the effect of gravity on performance of condenser, we measured thermal characteristics with changing orientation of condenser. **Figure 8** shows relationship between orientation of condenser and subcooling of liquid at condenser exit for 15, 30 and 157 mL/min in flow rate of FC72. Orientation was set to be cold plate on bottom and top side of condenser tube respectively. We can see that there are no remarkable difference between these orientations, which implies thermal performance of condenser won't be affected by gravity. However, detail investigation concerning effect of gravity will be made through the results by ISS experiments.

5. Results of qualification test for EM

Engineering Model (EM) test on ground based condition had been carried out. EM had almost same schematic as Flight Model (FM) shown in **Fig 1**. Condenser was installed on the test loop prepared as EM for the experiment chamber onboard "KIBO". Thermal characteristics of condenser were investigated in the series of system verification test. **Figure 9** indicate example for results of EM test. **Figure 9(a)** shows history of flow rate and thermal equilibrium quality at condenser inlet. This quality was calculated by considering applied heat rate at test section and preheater, specific enthalpy at preheater inlet obtained from measured temperature and pressure, and measured mass flow rate. Flow rate was set to be low and high value alternatively as shown in **Fig. 9(a)**. Quality at condenser inlet was more than 1.0 in low flow rate as shown in **Fig. 9(a)**, therefore vapor flow entered to condenser. We installed temperature sensor at condenser inlet, outlet of first condensation tube and condenser outlet in order to measure fluid temperatures in condensation tube. **Figure 9(b)** shows



(a) History of flow rate and thermal equilibrium quality at condenser inlet



(b) History of temperature in condenser tube
(TBH1: Inlet of condenser, TBH2: Outlet of first condensation tube, TBH3: Outlet of condenser)

Fig.9 Results of EM test

history of temperature in condenser tube. From this figure, we can see that temperature at condenser outlet was about 20 ºC in low flow rate condition. In this case liquid subcooling was sufficiently higher than 10 K; system requirement for condenser.

Thermal performance in microgravity condition was investigated by revised arithmetic thermal model on which EM test results were reflected. Thermal resistance between bottom of condenser tube and cold plate surface was adjusted in such a way that results by arithmetic model were consistent with EM test results. By using this thermal resistance, thermal performance of condenser in flight test condition was confirmed. As an example for this discussion, **Figure 10** shows temperature distribution in condenser and cold plate for 200 $kg/m^2/s$ in mass flux of working fluid and 0.73 in inlet quality. It was found that subcooling at condenser exit was higher than 10 K; system requirement for condenser.

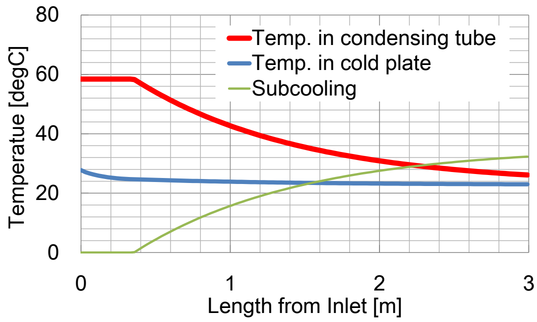


Fig.10 An example for calculated result of temperature distribution in condenser and cold plate on flight test condition
(Mass flux: 200 $kg/m^2/s$, Inlet quality: 0.72)

6. Conclusion

Thermal design and evaluation test on condenser for TPF; ISS Boiling Flow Experiment were carried out. Results are summarized as follows;

(1) Arithmetic thermal model for condenser on flat type cold plate was created. By this analytical model, the specification of condenser which satisfies the requirements of ISS experiments, that is, 10 K in outlet liquid subcooling could be determined.

(2) Ground based experiment for BBM condenser shows that heat of 400 W was transported to the cold plate from the at 45 kg/h of cooling water flow. The liquid subcooling ofFC72 was higher than 10 K at the exit of condenser tube.

(3) Results of BBM and EM tests proved that the present condenser satisfied the requirements in the TPF experimental system.

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