J. Jpn. Soc. Microgravity Appl. Vol. 28 No. 2 2011 (S109–S111)

8th Japan-China-Korea Workshop on Microgravity Sciences for
Asian Microgravity Pre-Symposium

Thermal Control System for Space Experiment on Two-Phase Boiling Flow -I ; Analysis and Design of Condenser

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Abstract

Design of condenser planned to be installed on boiling flow experiment on ISS was conducted. In this condenser, rectangular vapor tube was installed on flat type cold plate which had experienced on-orbit experiments. Arithmetic thermal model was constructed, which considered tube wall conductivity, heat transfer coefficient on condensation and cooling water flow. Specification of condenser which can condensate vapor of 400W was determined through analysis by this arithmetic model.

1. Introduction

On the manned spacecraft, thermal management, such as heat transport and heat rejection is important technology to be established. Especially two phase thermal management technology utilizing latent heat is favorably applied to large scale spacecraft system in future since high amount heat transport and rejection are realized by less flow rate of working fluid. Currently, boiling flow experiment onboard ISS Kibo is being planned. In this paper, thermal design on condenser is reported.

2. Structure of Condenser for ISS Boiling Flow Experiment

In this condenser, condensing tube was installed on the flat type cold plate which has experience on orbit system by considering cost saving and shortening development period. **Figure 1** shows the structure of condenser. Condenser tube was square rod having circular cross sectional flow path. Each rod was connected by U-shape tube. Square tube was selected in order to realize less thermal resistance due to thermal conductivity of wall. High heat conductivity adhesion (Emerson Cuming ECCOBOND®) with 7.5W/m/K in thermal conductivity was used to join the condensing tube surface on cold plate. In this condenser, fluid temperature at inlet/outlet of hot side as well as tube wall are planned to be measured in order to obtain local and overall condensation heat transfer coefficient. Through these measurements, valuable data will be obtained under micro gravity condition.

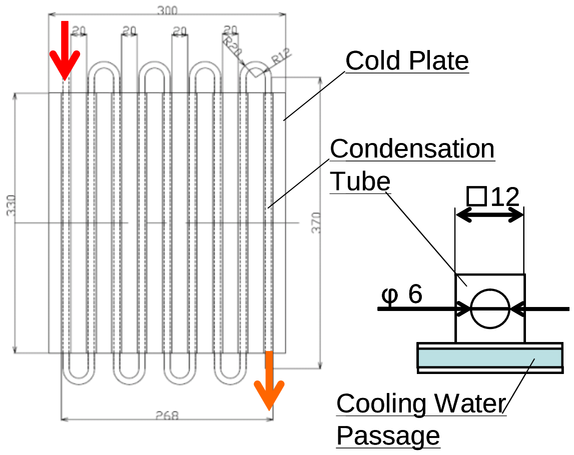

Fig.1 Structure of condenser

3. Arithmetic Thermal Model of Condenser

Figure 2 shows arithmetic thermal model for condenser. One dimensional thermal path from fluid in condenser tube to cold water in cold plate through condenser tube wall, adhesion, and cold plate wall was considered to obtain thermal resistance due to thermal conductivity and heat transfer. Condensation heat transfer coefficient under microgravity condition was given

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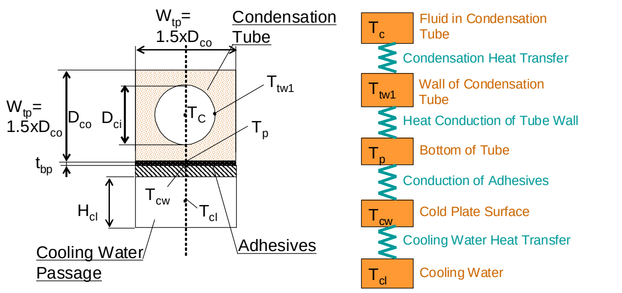


Fig.2 Arithmetic thermal model of condenser Structure of condenser

as a half value from empirical equations for ground based condition. Half value was assumed from the fact that condensation length under microgravity is twice as long as under 1-g condition by numerical calculation[1]. In this study, condensation heat transfer coefficient was calculated by numerical calculation tool for heat exchanger; HTRI[2]. Thermal resistance through condenser tube wall was calculated through two dimensional thermal conduction analysis. Fluid temperature, quality, and local pressure were calculated via above thermal resistances.

4. Arithmetic Thermal Model of Condenser

Numerical calculation was conducted by arithmetic thermal model on the following conditions;

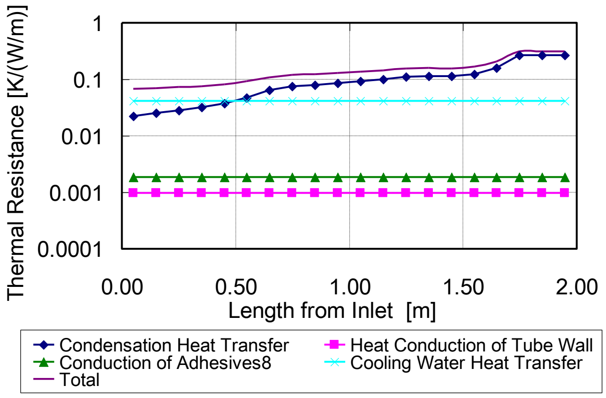
Fluid	Hot Side	FC-72
	Cold Side	Water
Hot Side	Inlet Pressure	0.103MPa
	Inlet Temperature	55.7 C
	Inlet Quality	1.0
	Mass Flow Rate	0.0042kg/s
Cold Side	Inlet Temperature	23C
	Mass Flow Rate	45kg/h
	Heat Transfer Coefficient	2000 W/m2/K
Condenser Tube	Inner Diameter	ϕ4, ϕ6mm
	Length	2m (ϕ6mm) 1.5m (ϕ4mm)

We provided following condition as required specification for condenser;

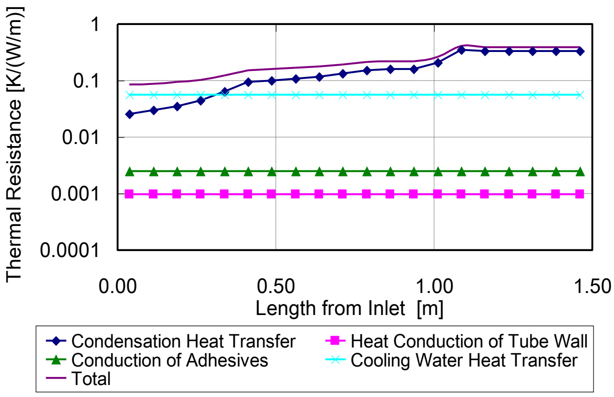
Outlet sub-cool temperature is 10K at 400W equivalent mass flow rate

This condition was determined in consideration of measured suction performance of mechanical pump.

Figure 3 shows the distribution of thermal resistance on flow path. This figure shows that thermal resistance by condensation increases toward downstream since condensation liquid film gets thicker. We can see that other major thermal resistance is achieved in cold water passage, and that resistance at tube wall and adhered portion between condensation tube and cold plate are sufficiently small. It is found that thermal

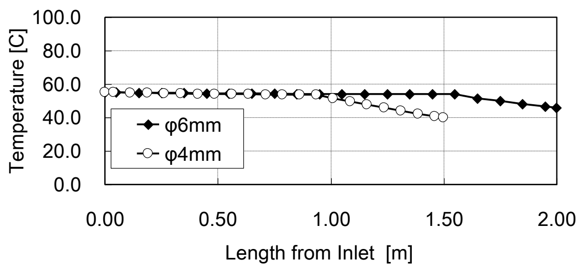


(a) 6mm in tube diameter

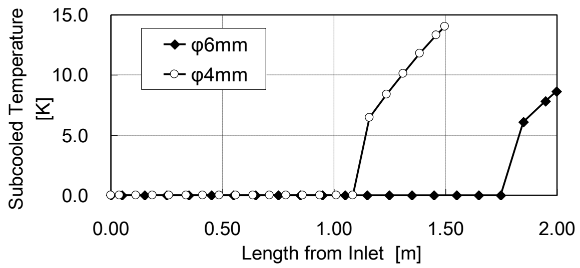


(b) 4mm in tube diameter

Fig.3 Thermal resistance at local point



(a)Temperature in condensing tube



(b)Sub-cooled temperature in condensing tube

Fig.4 Temperature and sub-cooled temperature resistance at local point resistance on condensation side is dominant at the downstream.

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

5. Conclusion

Connectional design of condenser for ISS Boiling Flow Experiment was conducted. Arithmetic thermal model for condenser which installs square rod tubes on flat type cold plate was constructed. From this analytical model, specification of condenser which satisfying requirement, that is, 10K in outlet sub-cool temperature at 400W vapor mass flow rate, could be determined.

References

1) Krotiuk W. J., Thermal-hydraulics for Space Power, Propulsion、 and Thermal Management System Design, Vol. 122, Progress in Astronautics and Aeronautics, AIAA, 1990.
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(Received 24 Sept. 2010; Accepted 12 Sept. 2011)