Study of the heat transfer capacity of thermosyphon cooling system under the inclined condition

K Yamaguchi1, E Shaanika1, M Miki1, M Izumi1, Y Murase2, T Oryu2, T Yanamoto2

1Tokyo University of Marine Science and Technology, 2-1-6 Etchu-jima, Koto-ku, Tokyo 135-8533, Japan
2Kawasaki Heavy Industries, Ltd. 1-1 Kawasaki-cho, Akashi, Hyogo, 673-8666, Japan
y.kouta0215@gmail.com

Abstract. Machine applications with superconductors attract much interest in the industry. It is crucial to maintaining the operating temperature range between 28 K and 40 K for high-temperature superconductor (HTS) field poles. A thermosyphon (TS) is a suitable choice for HTS applications' cooling system. The cryo-mechanical system composition is simple, lightweight and based on natural convection therefore not using any mechanical pump. Considering the rotating application, an adiabatic tube is necessary to carry the cryogen from the vertical condenser portion toward the horizontal evaporator part. However, this tube shape prevents the natural convection when in an inclined configuration which can occur for applications such as ship propulsion and electrical aircraft motors or generators. In this work, we focused on and investigated the heat transfer capacity of TS for inclinations. The results show that the TS cooling system employing the coaxial tubes architecture maintained the natural convection even under inclined conditions

1. Introduction
The sophisticated cooling system is required for the large-scale high-temperature superconductor (HTS) rotating machines. A series of 1 to 36.5 MW HTS rotating machines have been developed with sophisticated cooling systems [1-5]. Adequate cooling is necessary to suppress the thermal load on the order of several hundred W, and an adiabatic structure is inevitable. A thermosyphon (TS) cooling system which based on the natural convection is one cooling technique for HTS rotating machines [6-8]. This TS cooling system is simple, as it is composed of a cryocooler, condenser, and adiabatic tube guided to the evaporator. It can use a higher heat transfer rate due to the utilization of latent heat and eliminate circulator. From the standpoint of currently available commercial HTS performance, the temperature of the HTS field pole on the rotor of the HTS machine should be maintained under at least 40 K. The operating temperature of the TS cooling system depends on the cryogen, the TS cooling system using neon as a cryogen can give the operating temperature under the 40 K.

We have studied the TS cooling systems which could provide enough cooling power for HTS machines since 2006 [6-12]. We adopted neon or mixed neon with helium for a suitable cryogen and realized an HTS field pole operating temperature range from 28 K to 40 K [9]. Thus, we focus on the utilization of TS cooling system and peripheral structure design so that the HTS rotating machines can be operated with enough survivability under the actual operating situation. The performance of TS cooling systems at sea has been studied [10-11]. Utilizing the latent heat, once the boiling form changes from nucleate
boiling to film boiling, the heat transfer rate decreased significantly [13]. We also investigated the relationship between the saturated vapor pressure and liquefied neon quantity in TS cooling system [12]. The TS cooling system for HTS rotating machine has the structure of adiabatic tube which composed vertical and horizontal tube for supplying cryogen from the rotating shaft. Considering the inclined operation such as a ship or aircraft, it is easy for condensed cryogen to get stuck at the elbow. To solve this problem, we employed the coaxial tube for TS cooling system. In the present paper, we report our study of the performance of a 100 W-grade TS system using neon as a cryogen with 30 degrees inclination. We also investigated each heat transfer capacity using the various inner tube.

2. Experiment

The experimental TS cooling system is shown in figure1. The TS cooling system was composed of the condenser, adiabatic coaxial tube and the evaporator which can rotate. For supplying the cryogen to inside rotate part, this system has a static part which is a cryocooler and turn part which is a model of the motor. The cryogenic rotary joint is used for connecting rotate part and static part [14-15]. We employed the GM-type cryocooler (AL330, Cryomech) which has an 85 W cooling power at 30 K. The condenser is attached to a heat exchange plate to the GM-type cryocooler. Promoting the heat exchange, there is an optimal fin array which made by oxygen free copper in the condenser. In the heat exchange plate, a silicon diode sensor (DT-670-CU, Lakeshore) and a cartridge heater (E2A72, Watlow) are embedded. The condenser temperature is maintained by the temperature controller (Model 331S, Lakeshore). The adiabatic coaxial tube is divided into a vertical tube and horizontal tube. The vertical tube and horizontal tube lengths are 451 mm and 517 mm, respectively. It is noted that the coaxial tube is composed of the outer tube and inner tube. The outer tube is a 1/2-inch tube. We employed a 1/4-inch tube and a 1/8-inch tube as the inner tube. Protecting from the ingress of the condensed cryogen in the condenser side, the parasol shape cap was equipped on the condenser side inner tube. The effective cross-sectional area of the single 1/8-inch tube, twin 1/8-inch tube and 1/4-inch tube are 3.73 mm², 7.47 mm² and 14.9 mm², respectively.

Figure1. (A) Schematic view of a rotor-scale model of an HTS rotating machine. The closed thermosyphon (TS) cooling system is composed of a condenser connected to the cold head of a GM-type cryocooler via heat exchange plate, adiabatic coaxial tube and evaporator in the rotor. (B) The adiabatic coaxial tube is composed of the inner tube and the outer tube. (C) The parasol shape cap is equipped at the condenser side end of the inner tube. (D) A pair of cartridge heater and a silicon diode sensor are attached on the surface of the evaporator.
The evaporator shape is cylindrical with an inner diameter and length of 120 mm and 26 mm, respectively. A silicon diode sensor (DT-670-CU, Lakeshore) and a pair of cartridge heaters were placed on the surface of the evaporator. The condensed neon falls from the condenser beneath the cold head down to the evaporator through coaxial tubes. In the evaporator, the condensed neon experiences the heat load from the cartridge heater and vaporizes.

In the inclined condition, the condensed cryogen gather in the coaxial tube and the cryogen flow is separated by coaxial tube. If the gaseous neon flows through the inner tube and liquid neon flows the outer tube in the inclined condition, the amount of liquid neon in the evaporator changes with the difference of the inner tube. We adjust the supplied neon gas quantity so that the amount of liquid in the evaporator at the 30 degrees inclined state would be the same. The filled neon gas quantity is shown in table 1. It is countered by a mass flow controller (SEC-E40, HORIBA STEC Co. Ltd.).

| Inner tube       | Filled quantity of neon gas [NL] |
|------------------|----------------------------------|
| Single 1/8 inch  | 133                              |
| Twin 1/8 inch    | 125                              |
| Single 1/4 inch  | 110                              |

We conducted the heat load tests with different at both 0 degrees and 30 degrees inclined state. The heat load was applied to the evaporator from 5 W to 65 W with 5 W increments using a regulated constant power supply (ZX-400LA, TAKASAGO). During the heat load test, the condenser temperature was maintained at 30.0 K and the rotating speed of the rotor was kept at 18 min\(^{-1}\). We measured the temperature of the evaporator and saturated vapor pressure under the heat load. After applying the heat load, the steady state temperature and saturated vapor pressure were recorded.

3. Results and Discussion

Figure 2 exhibit the steady state temperature with each inner tube with 0 degrees and 30 degrees. We applied heat load until 65 W for each inner tube with 0 degrees. When we applied the heat load 70 W, the temperature of the condenser could not maintain at 30 K for each condition. We also measured the total heat invasion including the thermal radiation and conduction for this TS cooling system. The full heat invasion is 11-15 W in this system. Comparing results of each condition, there is no difference of heat transfer capacity and the steady-state temperature because both of gaseous and liquid neon flow is not separated by coaxial tube under the inclined condition. It’s indicated that there is no effect of the coaxial tube with 0 degrees.

We also conducted the heat load test at 30 degrees inclined state. Thanks to the coaxial tube, the TS cooling system could operate even we inclined at 30 degrees, the temperature of the evaporator stabilized. When we add the 15 W for TS cooling system using the 1/8-inch inner tube, the temperature of the evaporator raised and could not maintain. This means the boiling form changed from nucleate boiling to film boiling in the evaporator. The same phenomenon was also observed under twin 1/8-inch tubes when we applied 40 W heat load. In the result using the 1/4-inch inner tube, we could not observe this phenomenon. Comparing the results between with 0 degrees and 30 degrees, the heat transfer capacity was reduced using single and twin 1/8-inch inner tube because the gaseous flow is limited by the size of inner tube diameter under the inclined condition. Figure 3 shows the velocity of gaseous neon in the inner tube. The velocity is calculated by

\[
v = \frac{H}{h\rho_v A}
\]

where \(v\), \(H\), \(h\), \(\rho_v\), and \(A\) are the Flow velocity of neon in the inner tube, total heat load which is the amount of heat load and heat invasion, latent heat of neon, the density of gaseous neon, the cross-sectional area of the inner tube, respectively. From the calculation result, the maximum of gaseous flow
velocity shows 4.1 m/s in the inner tube. This indicates the 4.1 m/s is a limitation of heat transfer of TS cooling system with 30 degrees inclined state. And the 1/4-inch tube has enough heat transfer capacity for this TS cooling system.

![Figure 2. Steady-state evaporator temperature as a function of heat load with 0 degrees (open mark) and 30 degrees (closed mark).](image)

![Figure 3. Calculation results of the velocity of the gaseous neon in the inner tube with 30 degrees inclined state.](image)

4. Conclusion
For a large-scale HTS rotating machine, suitable refrigeration and cooling system is a prerequisite. Considering the sip propulsion and electrical aircraft application, the cooling system is required maintaining the temperature under the inclined condition. We study the performance of TS cooling system which employs the coaxial tube with 0 degrees and 30 degrees.

The TS cooling system succeeded to operate during the 30 degrees inclined condition. Moreover, we observed that the heat transfer capacity depends on the cross-sectional area of the inner tube. From the calculation results, we found the coaxial tube is adequate for TS cooling system under inclined operation and the heat transfer limit was 4.1 m/s in the inner tube with 30 degrees inclination.
References

[1] Umemoto K, Aizawa K, Yokoyama M, Yoshikawa K, Kimura Y, Izumi M, Ohashi K, Numano M, Okumura K, Yamaguchi M, Gocho Y and Kosuge E 2010 Development of 1MW-class HTS motor for podded ship propulsion system J. Phys. Conf. Ser. Vol. 234 032060

[2] Gamble B, Snitcher G and MacDonald T 2011 Full power test of a 36.5 MW HTS propulsion motor IEEE Trans. Appl. Supercond. Vol. 21 no. 3 pp. 1083–1088

[3] Nick W, Grundmann J and Fraunhofer J 2012 Test results from Siemens low-speed, high-torque HTS machine and description of further steps towards commercialization of HTS machines Physica C Vol. 482 pp. 105-110

[4] Terao Y, Sekino M and Ohsaki H 2013 Comparison of conventional and superconducting generator concepts for offshore wind turbines IEEE Trans. Appl. Supercond. Vol. 23 no. 3 5200904

[5] Yanamoto T, Izumi M, Umemoto K, Oryu T, Murase Y and Kawamura M 2017 Load Test of 3-MW HTS Motor for Ship Propulsion IEEE Trans. Appl. Supercond. Vol. 27 no. 8 5204305

[6] Felder B, Miki M, Tsuzuki K, Izumi M and Hayakawa H 2010 Optimization of a condensed–neon cooling system for a HTS synchronous motor with Gd-bulk HTS field magnets J. Phys. Conf. Ser. Vol. 234 032009

[7] Felder B, Miki M, Deng Z, Tsuzuki K, Shinohara N, Izumi M and Hayakawa H 2011 Development of a cryogenic helium-neon gas mixture cooling system for use in a Gd-Bulk HTS synchronous motor IEEE Trans. Appl. Supercond. Vol.21 no. 3 pp. 2213-2216

[8] Felder B, Miki M, Tsuzuki K, Shinohara N, Hayakawa H and Izumi M 2012 A 100-W grade closed-cycle thermosyphon cooling system used in HTS rotating machines AIP Conf. Proc. Vol. 1434 pp. 417–424

[9] Sato R, Felder B, Miki M, Tsuzuki K, Hayakawa H and Izumi M 2013 Helium-neon gas mixture thermosyphon cooling and stability for large scale HTS synchronous motors IEEE Trans. Appl. Supercond. Vol. 23 no. 3 5200704

[10] Yamaguchi K, Sato R, Miki M, Yamagata K, Ikeda T, Izumi M, Murase Y, Umemoto K and Yokoyama M 2015 Study of the thermosyphon cooling system with a vessel in the sea states Physics Procedia Vol. 67 pp. 245-249

[11] Yamaguchi K, Miki M, Yamagata K, Ikeda T, Kashima H, Izumi M, Murase Y, Yanase E and Yamamoto T 2016 Study of HTS machine system cooling with a closed loop thermosyphon: Stability of unsteady heat load and transient conduction IEEE Trans. Appl. Supercond. Vol. 26 no. 3 5204405

[12] Yamaguchi K, Miki M, Izumi M, Murase Y, Oryu T and Yamamoto T 2017 The effect of condensation area and operating temperature on heat transfer capacity of a closed loop thermosyphon cooling system for HTS machinery IOP. Conf. Ser.:Mater. Sci. Eng. 278 012024

[13] Astuc J M and Perroud P, “Pool boiling heat transfer in liquid neon 1966 Adv. Cryogen. Eng. Vol. 12 pp. 387–394

[14] Miki M, Felder B, Tsuzuki K, Izumi M and Hayakawa H 2010 Development of the cryo-rotary joint for a HTS synchronous motor with Gd-bulk HTS field-pole magnets J. Phy, Conf. Ser. Vol. 234

[15] Cryo-rotary joint, by Izumi M, Miki M, Kitano M (2013, Dec.31). Patent US 8616587B2.