Development of Enhanced Heat Transfer Coolant Channels for the SPring-8 Front End Components

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1. Introduction

Designing carefully masks, absorbers and slits has been required to withstand the extremely high power and power density produced at third-generation synchrotron radiation facility. One of the solutions to the thermal problem is to increase the ability of heat transfer of cooling channel. Early cooling channel design for SPring-8 front end components employed wire-mesh blazed coolant channel [1], which has been developed at Advanced Photon Source and realized five to ten times higher value of the heat convection coefficient compared with the conventional plain channel. However, the wire-mesh has some difficulties as follows: (1) there are limitations in use because of the large pressure loss of coolant flow. (2) very skilled techniques are required in blazing the wire-mesh to the inside wall of channel. Therefore, it is important to develop another cooling method lowering the pressure loss. We investigated an inserted wire-coil [2] and a grooved inside wall as an alternative cooling design. The purpose of this study is to measure the pressure loss and the heat transfer coefficient for the corresponding test piece provided, and to supply necessary data for cooling design of components suffering high heat load.

2. Experimental

2.1 Tube-like Models for Heat Convection Test

Test tubes for the heat transfer experiment are shown in Fig. 1. There are three types of test tubes made from copper, *e.g.*, plain tubes, grooved tubes and wire-coil inserted tubes. The plain tubes have the inside diameter of 8 mm and 10 mm with wall thickness of 1 mm. The grooved tubes inside wall were machined to be spline or screw geometry. The wire-coils with various pitches and diameters are manufactured. The length of every test tube is 600 mm and its center is rolled up with the rubber heater of 400 mm in breadth. The heater has a capacity of 1 kW at 100 V AC. Also, seven thermocouples are blazed with silver-copper alloy at 50 mm intervals on the outer surface of the test tube. The parameters about the test tubes are presented in Table 1.

2.2 Experimental Unit

The heat transfer coefficient and pressure loss of coolant was measured by the experimental unit. The unit consists of several parts, that is, test section, water reservoir, electric panel and electric power supply. The test section consists of the test tube, the coolant temperature and pressure sensors and the flow control unit. The test tube is attached to the transportation line made with stainless steel tubes. The diameter of transportation tube is decreased by reducer so as to match the diameter of test tubes. Portholes are prepared upstream and downstream of the test section, where thermal sensors are inserted. The thermal sensor, which has two probes and common ground line, is made especially for the temperature difference measurement. The coolant temperature of the test tube is measured upstream by another thermocouple. The pressure loss at the test section is measured by a differential pressure sensor. The coolant pressure is measured by an upstream pressure sensor. The flow control section consists of a flow meter, a flow control valve, a PID controller as well as a reservoir vessel. The water reservoir section consists of a water tank, a water circulation pump, rotor meter and a heat exchanger. The electric power section contains a power supply for heaters.

2.3 Data Processing

Electric signals corresponding to the temperature distribution along to the test tube are obtained from seven thermocouples on the outer surface of the tube. Translating the signals to the temperature value, we used equation (1) as follows.

$$E = \sum_{i=0}^{8} b_i T^i + 125 \exp\left[-\frac{1}{2}\left(\frac{T-127}{65} \right)^2 \right] \qquad (1)$$

where E and T are the electromotive force and the temperature, respectively, and b_i is the constant given on the standard table related to the thermocouple. The inside wall temperature of the test tube is estimated with heat flux, wall thickness and thermal conductivity of the test tube.

The heat flux q transporting from the heater to the coolant is proportional to both temperature increase ΔT_f and the mass flow rate of the coolant G as follows.

$$\dot{q} = c_p G \Delta T_f \qquad (2)$$

where c_p is the specific heat at constant pressure. The heat convection coefficient h is calculated from the ratio of the heat flux on the unit area to the temperature difference between the coolant and the wall. Therefore,

$$h = \frac{\dot{q}}{T_{wi} - T_f} \tag{3}$$

where T_{wi} and T_f are the inner wall temperature and the bulk temperature of coolant, respectively. The Nusselt number *Nu* is represented by

$$Nu = \frac{hd}{\lambda} \tag{4}$$

where *d* and λ are the characteristic length and the thermal conductivity of coolant. The experimental correlation of Nusselt number *Nu* was already given, as the Petukov-Gnielinski correlation [3]. That is

$$f = (3.64 \log_{10} \mathrm{Re} - 3.28)^{-2} \tag{5}$$

$$Nu = \frac{f/2(\mathrm{Re} - 1000)\,\mathrm{Pr}}{1 + 12.7\sqrt{f/2}(\mathrm{Pr}^{2/3} - 1)} \tag{6}$$

Where *Re* and *Pr* are Reynolds number and Prandtl number, respectively.

3. Results

The higher Nusselt numbers were observed at the inlet of the tube. The trend suggests that there is an entrance flow region near the inlet of tube. Therefore, we picked up data from the measurement points of 5, 6 and 7 which are in a fully developed region, in order to evaluate the mean value of Nusselt number for the tube.

3.1 The Influence of Tube Geometry to Nusselt Number and Pressure Loss

The comparison of Nusselt number and pressure loss for each test tube is indicated in Fig. 2. In general, the higher Nusselt number was obtained by the wire-coil inserted tube than by grooved tube. In comparison between the wire-coil inserted tube No. 3 and No. 6, the effect of double wire-coil tube is found in Nusselt number. However, the pressure loss for wire-coil tube No. 3 is larger than wire-coil tube No. 6, and the merit of the double wire-coil seems to be diminished. The increase of heat transfer coefficient might be responsible for the thermal boundary layer disturbed by the outer wire-coil. The inner coil seems not effective to the disturbance of the thermal boundary layer. For the relatively large coil pitch as wire-coil tube No. 4, the high Nusselt number was obtained under the low pressure loss. The pressure loss is also large for wire-coil inserted tubes, similar to the case of Nusselt number. The pressure loss is especially large for the wire-coil tube of No. 2 and No. 3. The effect of the coil-pitch to the variation of Nusselt number is clear under the condition of the large diameter of coil.

The relation between the Nusselt number and the coil-pitch is shown in Fig. 3 as a parameter of wire-rod diameter. The Nusselt number increases with decreasing of the coil-pitch. The Nusselt number shows a maximum at 1.5 mm of the coil wire-rod diameter. The fact suggests that, disturbing the thermal boundary layer on the inside wall of test tube, the smaller coil-pitch is more effective and the value of 1.5 mm in rod diameter is most suitable. The value of the rod diameter is thought to be equal nearly the thickness of the thermal boundary layer. Figure 4 shows the variety of pressure loss against the coil-pitch, where the parameter is the coil diameter. It is found that the larger pressure loss becomes, the smaller diameter of coil is. In the mock-up design of cooling systems, it is necessary to decide the geometry of the coolant channel considering a balance between the Nusselt number and acceptable pressure loss in a facility.

Finally, we selected geometric parameters of wire-coil inserted as follows, (1) We selected single wire-coil inserted as the double wire-coil has no remarkable merit. (2) The diameter of coil-rod was determined to be 1.5 mm, by which relatively high Nusselt number will be obtained. (3) The coil-pitch was determined to be 10 mm, by which the relatively low pressure loss is obtained compared with the permitted loss.

4. Conclusion

In the present report, the heat convection test was performed for the tube-like models such as grooved tubes and wire-coil inserted tubes, in order to obtain design data for the coolant channel of masks and absorbers in use of the components exposed to high heat flux. The results are summarized as follows.

(1) The wire-coil inserted tubes are capable of enhancing the heat transfer coefficient, while no clear effect of grooved tube is recognized in comparison with the plain tube.
(2) The smaller coil-pitch becomes, the higher Nusselt number is in the wire-coil tubes. But no trend is indicated between Nusselt number and the coil diameter.
(3) The pressure loss increases with both decreasing of the coil-pitch and increasing of coil-rod diameter.
(4) A better choice is the tube with wire-coil of 10 mm pitch and 1.5 mm wire-rod diameter.

References

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Table 1. Geometric parameters of test tubes

Test tube No.	Configuration	Inside Dia. mm	Outside Dia. mm	Wire Dia. mm	Inside coil pitch mm	outside coil pitch mm
1	plain tube	8.0	10.0	-	-	-
2	grroved tube (spline)	8.0	10.0	-	-	-
3	grroved tube (screw)	6.0	8.0	-	-	-
4	wire coil No. 1	8.0	10.0	1.0	-	5.0
5	wire coil No. 2	8.0	10.0	2.0	-	5.0
6	wire coil No. 3	8.0	10.0	2.0	5.0	10.0
7	wire coil No. 4	8.0	10.0	2.0	10.0	20.0
8	wire coil No. 5	8.0	10.0	1.0	10.0	20.0
9	wire coil No. 6	8.0	10.0	2.0	-	10.0
10	wire coil No. 7	8.0	10.0	1.5	-	5.0
11	wire coil No. 8	8.0	10.0	1.5	-	10.0

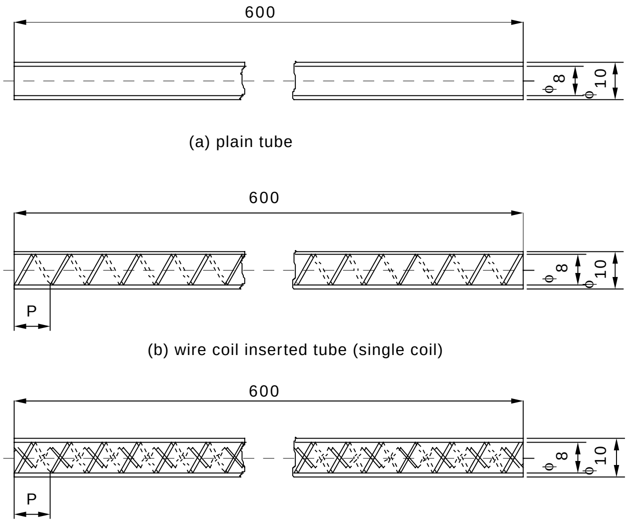

Fig. 1. Sketch of test tubes.

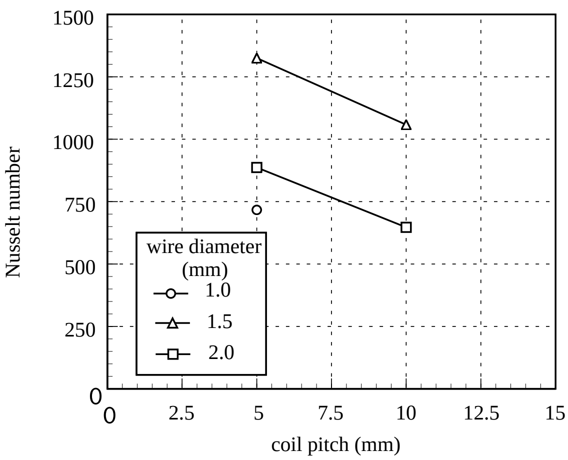

Fig. 3. Nusselt number vs. coil pitch, flow rate = 10 1/m.

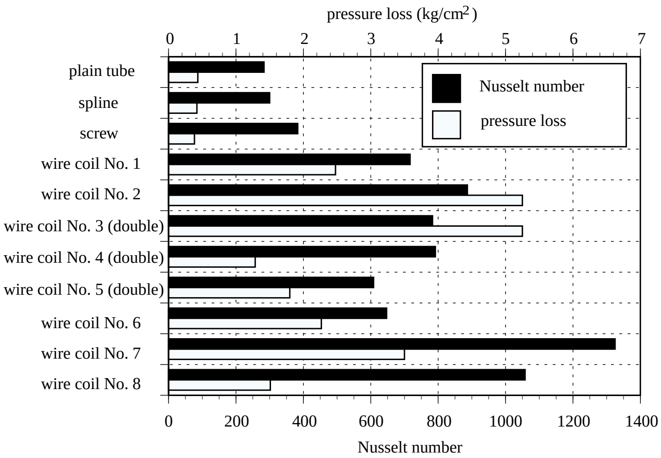

Fig. 2. Comparison of Nusselt number among various test tubes, flow rate = 10 1/m.

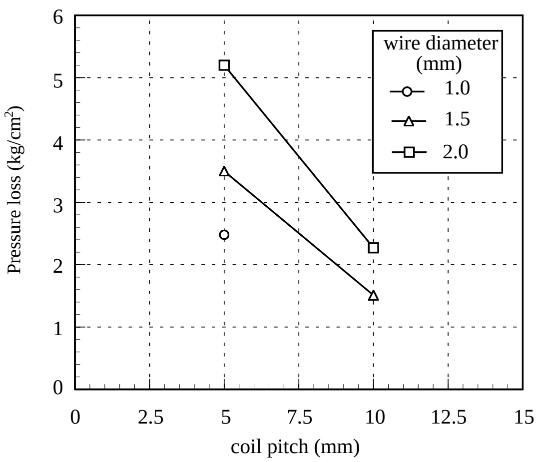

Fig. 4. Pressure loss vs. coil pitch, flow rate = 10 1/m.