HIGH HEAT FLUX EXPERIMENTS ON MOCK-UPS WITH POROUS COATING ON THE INNER SURFACE OF CIRCULAR COOLANT CHANNELS

ABSTRACT

The series of High Heat Flux experiments on circular tubes with the uniform heat loading and on rectangular samples with cylindrical cooling ducts with the one-side heating were carried out in order to study the effect of a porous coating deposed on inner cooled surfaces on the Incident Critical Heat Flux (ICHF) performance at water sub-cooled boiling regime. The stable increasing of ICHF obtained on samples with porous coating over ICHF on smooth surfaces was established for wide range of velocities (3 - 15 m/s) and subcooling (40 - 180 C). Test results have shown the heat removal ability of porous coatings to sustain an incident heat flux up to 65 MW/m^2 at the uniform circular loading on tubular channels (velocity 15 m/s and subcooling 180 C) and up to 45 MW/m^2 at one-side heating on rectangular mock-ups with tubular channels (velocity 5.6 m/s and sub-cooling 160 C). The further increasing of heating was limited by test facility.

Empirical correlation between ICHF at one-side heating condition and geometrical/operation parameters of elements of cooling design was defined.

NOMENCLATURE

ICHF - Incident Critical Heat Flux, experimental data, [MW/m^2]

CHF - Critical Heat Flux at uniform loading on circular channels, experimental data, [MW/m^2]

CHF_{Shl} - Critical Heat Flux at uniform loading on circular channels w/o porous coating, calculated data by Shlykov's correlation (1970), [MW/m^2]

PC - porous coating

P - pressure, [MPa]

A - loaded surface width for mock-up with rectangular cross section, [mm]

$F = \pi d^2/4$ - cross section of channel, [mm^2]

G - flow rate in channel, [kg/s]

$\overline{G} = G/A$ - specific flow rate per unit width of thermally loaded mock-up with rectangular cross section, [kg/s/m]

w - velocity, [m/s]

ρ' - liquid density, [kg/m^3]

ρ'' - vapour density, [kg/m^3]

t_{in} - inlet coolant temperature, [C]

Δt_{sub} - outlet sub-cooling, [C]

d - diameter of channel, [mm]

INTRODUCTION

Development of high heat flux components for fusion experimental machines has created the necessity of the heat removal enhanced technique application because of their too severe heat load conditions up to 15 - 30 MW/m^2.

Many enhanced techniques for increasing heat removal performance beginning from the trivial increasing of coolant operational parameters (inlet pressure and flow rate) and ending by both passive (swirl flow devices, extended cooled surfaces) and active methods (fluid injection, surface and fluid vibration) are well established. Unfortunately, to put into practice the most of them leads either to increasing of a device complexity or to jump drastically of the pressure drop in cooling channel and pumping power required for providing the heat removing. So, it is a common goal of researches to obtain the acceptable enhanced effect without such negative consequences.

The experimental works have been carried out during last years in Russia and U.S. by different researches have shown that just the porous coating deposed on the inner solid surface of cooling ducts meets the above requirements more completely because of: 1) their excellent heat removal enhanced performance and 2) a negligible impact on a duct hydraulic resistance (1.3-1.5 times).

The technology of this coating is established enough (Divavin et al., 1993 and Malyshenko et al., 1993) and based on baking or coagulating of tiny quasi spherical metal particles (diameter - 50-100 mkm) between themselves and with the inner cooled surface under the high temperature conditions (0.7-0.9 melting temperature) in the special vacuum furnace or in a hydrogen media.

1. PERFORMED TESTS

Either of uniform heat loading of the circular tubes and an one-side loading of the rectangular samples with a cylindrical cooling duct were a subject of the experimental investigation. The different material of mock-ups and particles were under consideration as well: stainless steel, copper and copper alloys. Detailed description of experiment procedures and facilities was done (Divavin et al., 1993 and Youchison et al., 1994).

The series experiments on circular tubes with the uniform heat loading were carried out by an omic heating of the testing tubes.

The series experiments on rectangular mock-ups with cylindrical cooling ducts with one-side heating were carried out by using special flat C-composite heater clamped between two identical testing mock-ups (Fig.1). The heater had omic heating and provided one-side heating of the two identical left and right mock-ups. The test module was equipped with 18 thermocouples y 6 in each of three crossection (Fig. 2).

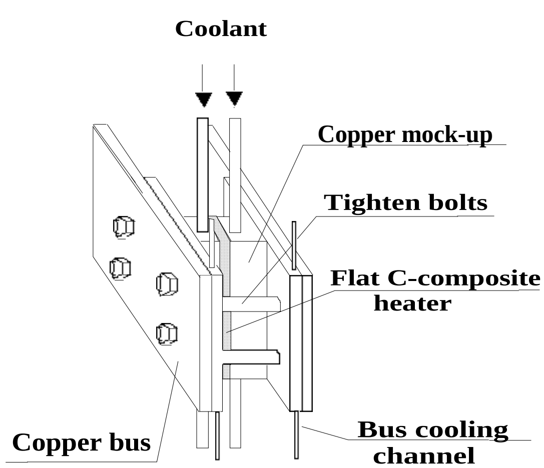

Fig.1. Experimental Module Fabricated for Testing on CBTI Facility

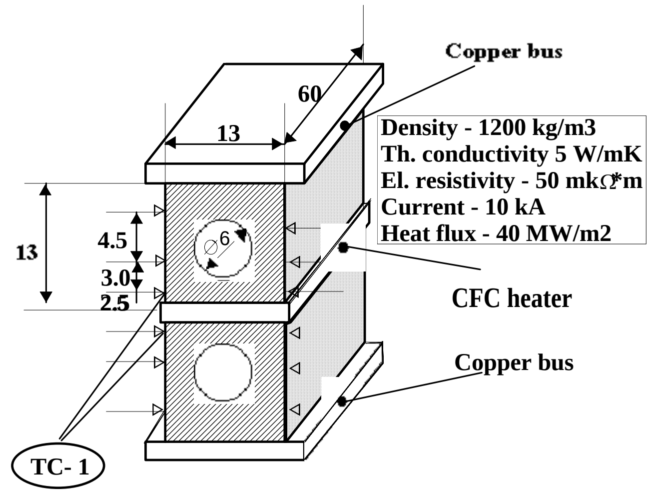

Fig.2. The Sample Cross Section with the TC's

These experiments with High Heat Flux one-side loading of rectangular samples were carried out on the CBTI facility which cooling loop appears on Fig.3.

The temperature of samples, inlet and outlet coolant temperature, flow rate, electric power and pressure were measured, controlled and recorded on computer. Heat disbalance between electric power and heat absorbed by coolant did not exceed 5%.

The ICHF was established at the lost of the sample temperature state equilibrium which was characterised by

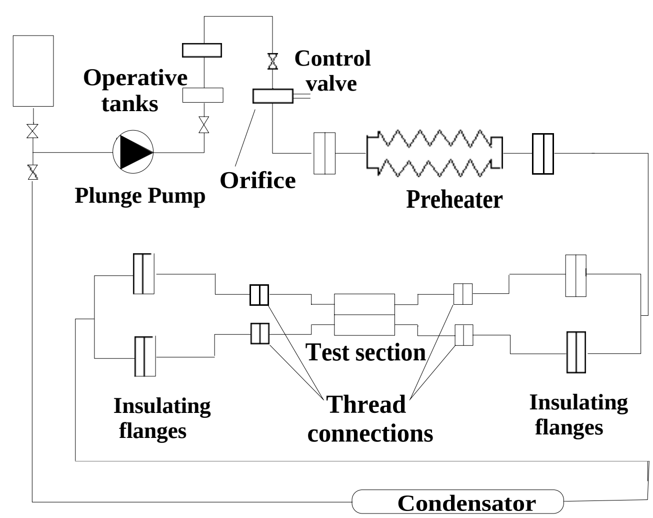

Fig.3. The Water Cooling Loop Scheme

sudden increasing of indication of thermocouples at invariable heat loading (as shown on Fig. 9 at end of paper).

A wide range of water thermalhydraulic conditions were tested:

velocity	3 - 15 m/s;
subcooling	40 - 180 C
pressure	1 - 3 MPa

2. RESULTS AND DISCUSSION

The results of the series experiments with the uniform heating by current running through long (0.4 m) copper and stainless steel samples with and without porous coating are shown on Fig.4. The heat fluxes leading to real physical burnout of the tube wall were chosen as Critical Heat Flux

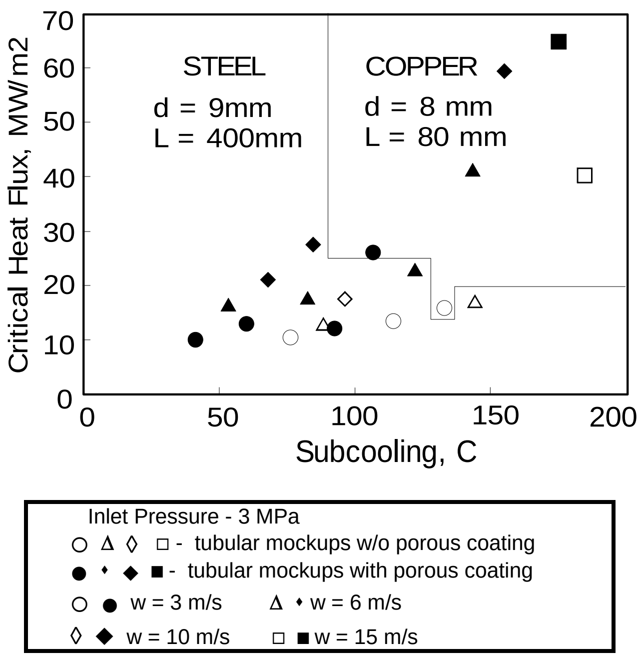

Fig.4. Porous Coating Impact on Critical Heat Flux at Uniform Heat Loading

(CHF) values. The analysis of the presented results shows that the porous coating effect on CHF values grows with the increasing of the water subcooling and the ratio of the CHF on surfaces with porous coating to CHF on smooth ones can reach value 1.3-2.0. The copper coating gives more appreciable benefits than stainless steel one. The most interesting result of these tests: ICHF increasing with coolant velocity rise is more considerable at PC using than w/o PC.

The results of tests with one-side loading of rectangular samples appear on Fig.5-8. As it was reported in our previous works (Divavin et al., 1993, Divavin et al., 1994, Divavin et al., 1990a&b) it is incorrect to use the CHF definition applied to the inner cooled surface in case of one-side heat loading because of the strong non-uniform distribution of the heat removed from the inner side due to its dependence from the local inner surface temperature distributed non-uniformly as well. Our experimental investigation together with numerical modelling (Divavin et al., 1993, Divavin et al., 1990a, Divavin et al., 1990b,) convinces that achieving of the heat removal crisis condition in a local point on the inner cooled surface not leads to burnout of the cooling channel fatality. There is a possible simultaneous coexistence of all cooling regimes on the wet surface: single phase forced convection, fully developed sub-cooled boiling, transient boiling and even film boiling. That is why it was recommended in case of the one-side heating to put into consideration the definition Incident Critical Heat Flux (ICHF), i.e. the heat flux density absorbed by the receiving surface and leading to the total heat removal crisis on the cooled wall (physical burnout of the sample). The value ICHF depends either on the sample geometrical parameters such as lateral wall thickness, ratio of a heated surface to the cooled one and from the cooled surface condition. So, ICHF 's presented on Fig. 5 demonstrate the s ignificant

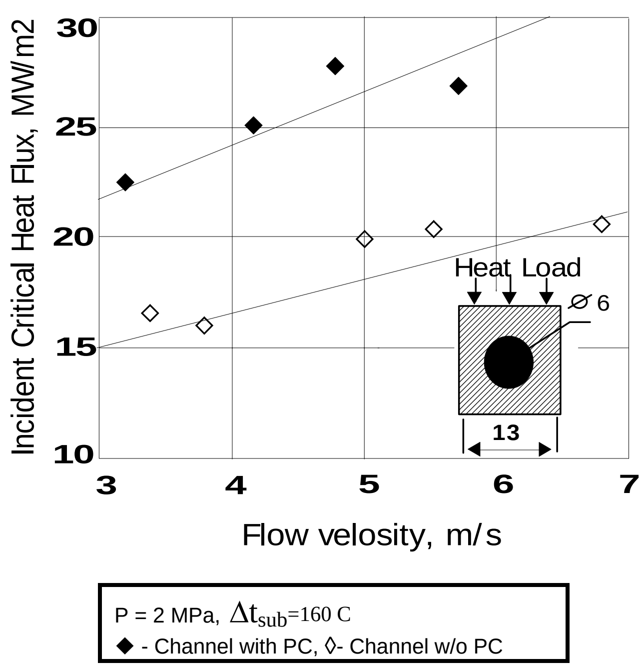

Fig.5 Porous Coating Impact on Incident Critical Heat Flux at the One-Side Loading.

increasing their values at the porous coating deposition on the wet surface. The enhancement coefficient exceeds 1.5. The sample with the width A = 13 mm and the duct diameter d=6 mm sustains the heat load q = 28 MW/m2 at the moderate velocity w = 5 m/s.

Table 1. Experimental Dates.

	A	d	A/d	w	$\overline{G}$	P	t_{in}	ICHF	CHF	$\frac{ICHF}{CHF}$
	mm	mm	-	m/s	kg/s/m	MPa	C	MW/m^2	MW/m^2	-
Experiments on Russian facility	8.5	6	1.42	3.0	9.97	20	50	37.5	11.7	3.2
				4.2	13.96			39.5	13.8	2.86
				5.6	18.6			45.5	16.0	2.84
	10	6	1.67	3.2	9.04	20	50	32.0	12.0	2.66
				4.2	11.86			34.0	13.8	2.46
				5.6	15.82			36.5	16.0	2.28
	13	6	2.17	3.2	6.95	20	50	22.0	12.0	1.83
				4.2	9.13			25.5	13.8	1.84
				5.8	12.6			29.5	16.2	1.82
Experiments on SNLA facility	20	15	1.33	0.43	3.79	10	20	11.0	4.5	2.44
				0.86	7.59			16.2	6.3	2.57
				1.72	15.8			20.0	8.9	2.25
	20	8	2.5	1.5	3.77	10	20	12	8.3	1.44
				3.0	7.53			16	11.8	1.35
				6.0	15.1			19.5	16.7	1.167

The influence of the lateral wall thickness h = (A - d)/2 on the ICHF was studied during this campaign as well. It was obtained the increasing of the ICHF up to q = 45 MW/m^2 at reducing A/d from 2.17 to 1.42 (Fig.6). Our joint with SNLA studies carried out on EBTI faculty confirmed such ICHF dependence (Fig.7).

There is question about choice of the step between cooling circular channels allocated near heat loaded surface in design of

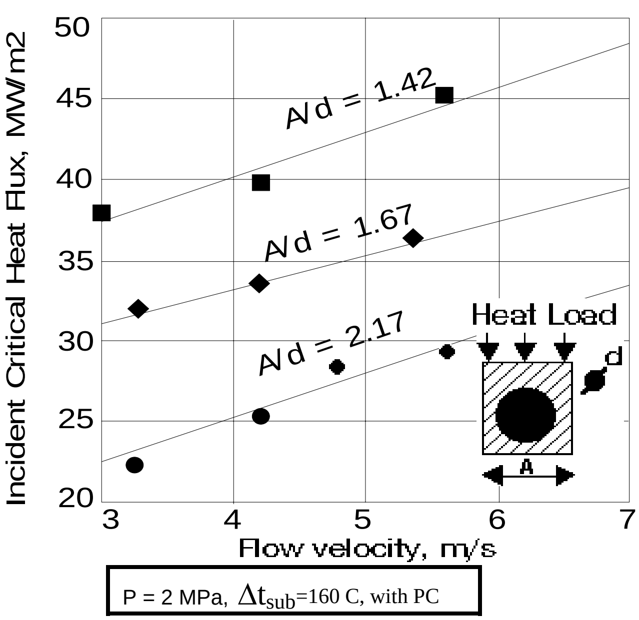

Fig. 6. A/d Impact on Incident Critical Heat Flux at One-Side Loading.

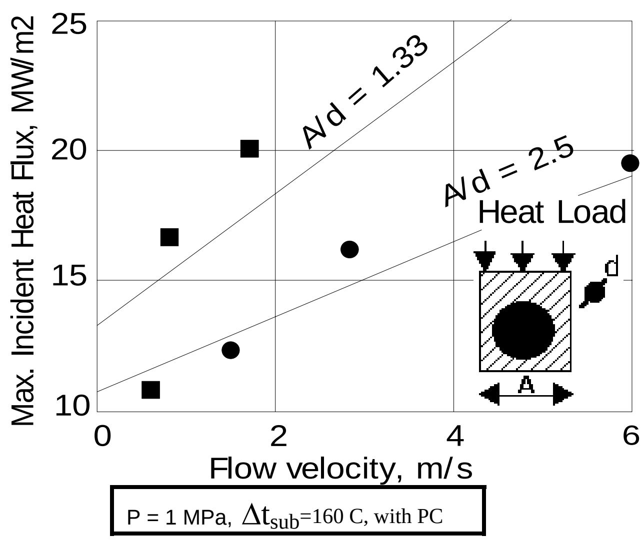

Fig. 7. A/d Impact on Max. Incident Heat Flux at One-Side Loading. Tests in SNLA

one-sided heat loading samples. (The same problem is particularly in design of HHF (15-30 MW/m^2) components as a divertor plates for ITER tokamak reactor. That’s why there is necessivity of prediction of ICHF due cooling channels sizes and step between channels by numerical methods.

List of above mentioned experimental dates (Fig.5-7) is summarised in Table 1. In this table will be seen that the value of specific flow rate $\overline{G}=\pi\rho' wd^2/(4A)$ hasn't confidential correlation with ICHF value. Mathematics process of experimental data was carried out as a function R=f(r), here:

$$R = \frac{ICHF}{CHF_{Shl}} \tag{1}$$

$$r = \frac{A}{1.4\pi d} \tag{2}$$

CHF value calculated by Shlykov's correlation (1970) was choiced as a base magnitude:

$$\mathbf{CHF}_{Shl} = 4.12 \cdot 10^{-2} \sqrt{\square G}\ \Delta t_{sub}^{0.33} \left(1 - \frac{\square\square}{\square}\right)^{1.8} \tag{3}$$

Coefficient 1.4 in equation (2) corrects for average (at different coolant velocity) value of heat removal increasing due using PC at one-sided heating (Fig.4).

On Fig.8 , which indicate all experimental points from Table 1, calculated approximation was shown as:

$$\frac{ICHF}{CHF_{Shl}} = 3.95 - 4.75\left(\frac{A}{1.4\pi d}\right) \tag{4}$$

It will be seen that this approximation has 15% accuracy for all experimental and operation parameters.

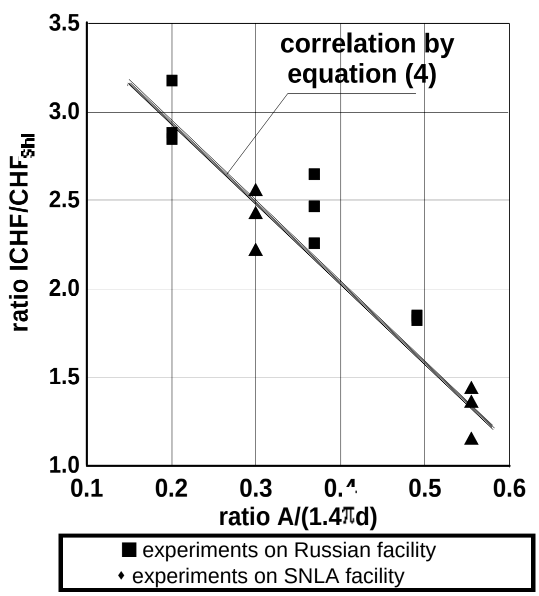

Fig. 8. Influence of Geometric and Operating Parameters on ICHF at One-Side Thermal Loading of Mock-ups.

The temperature curves on Fig.9 illustrate another important feature of the porous coating - more "soft' crisis development. Actually, a rate of temperature increase at crisis regimes not exceeds 5 - 7 C/s, what gives enough time for prevent of the cooling duct burnout in case of any sudden off-normal events.

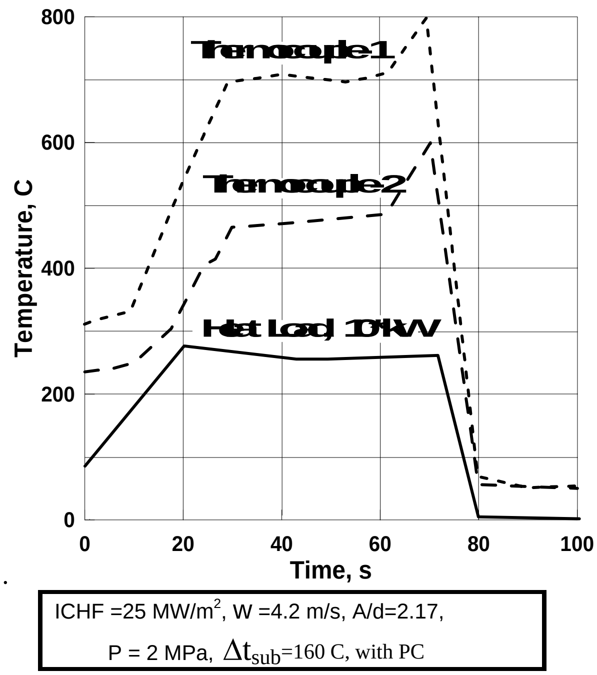

Fig.9. Temperature Response by TC`s on ICHF

3. CONCLUSION

The presented results of the experiments convince that the porous coating deposed on the inner wet surface of cooling ducts is the one of the most perspective heat remove enhanced technique, because of its:

- relatively simplicity of manufacture technology;
- obtained high ICHF without significant increasing of the hydraulic resistance for the wide range of the coolant parameters;
- "soft" crisis development with the temperature increasing rate ~5-7 C/s;
- reducing of the temperature gradient along the wet perimeter.