Experimental Study of Heat Transfer Coefficients in ITER Vacuum Vessel Cooling Channels

ABSTRACT

The presented work describes the results of thermal-hydraulic tests with single rectangular vacuum vessel (VV) channel models which were performed to measure heat transfer coefficients under typical VV fluid conditions. The experiments have been carried out using the thermal-hydraulic facility developed in Central Boiler & Turbine Institute (CBTI) for wide-range tests of Power Plant equipments. This facility comprises a water loop 30 m in height designed for the full-scale study of nuclear reactor passive cooling. The VV test element is a rectangular channel 0.2 m in width, 3 m in total length, 2.48 m in heated length. The channel height (δ_{ch}) is variable, i.e. δ_{ch}=12.5, 25 or 50 mm. As the results of the experiments, the heat transfer coefficients have been obtained as a function of flow rate, channel height, model inclination and heating power.

Keywords: ITER Vacuum Vessel, cooling system, mixed convection, laminar and turbulent flow.

1. INTRODUCTION

The cooling system developed for the ITER vacuum vessel is supposed to serve under various conditions of VV cooling. The laminar and turbulent water conditions as well as the forced and natural convections have effects on heat exchange intensity. Different orientations of the channels in the gravitation field provide conditions for temperature water stratification through the height of the horizontal or vertical channel. At present there are no reliable correlations describing the heat transfer for most of these conditions. The development of correlations requires experimental data that have to be obtained under conditions approximating as much as possible the real ones. The above water conditions also characterize an operation of heat passive evacuation systems in PWR's. Therefore, the solution of the problem will find wide application for calculation of the water cooling system of nuclear experimental facilities with different types of reactors.

2. EXPERIMENTAL TEST FACILITY

The facility used for thermal hydraulic tests of the vacuum vessel channel model is shown in Fig.1. This facility comprises a water loop 30 m in height designed for the full-scale study of the nuclear reactor passive cooling. Maximum design parameters of the test facility are as follows: (a) pressure - up to 30 MPa; (b) water-vapour flow rate - up to 20 m^3/h and (c) coolant temperature - up to 550°C. The water flow produced by the ten-plunger circulation pump (1) and pre-heated by the controllable electric heater (3) is delivered to the inlet of the test-model (6). Then the coolant goes to the shell-and tube heat exchanger in such a manner that the portion of water circulating through the loop is directed through the by-pass line with the valve (126) thereby providing the coarse adjustment of temperature at the tested model inlet. The test-model (see Fig.2) is a stainless steel rectangular channel 0.2 m in width, 3 m in total length, with a variable height (δ_{ch}) and different heated lengths (1.02 m and 2.48 m). The test model is thermally insulated with 100mm mineral heat insulation material of ISOVER type and electrically heated with two heaters made of stainless steel tapes 0.3 mm thick, 200 mm wide and 2.48 m long. Besides, the test-model is provided with a system of hinges, posts and cheeks welded to the middle part with which the model is installed at a specified inclination angle on the stands.

Wall temperatures are measured by a set of 40 K-type 0.2mm thermocouples mounted on the test-model, as shown schematically in Fig.3. In this case the temperature, measured by the thermocouples welded to the drilling bottom in the walls at the distance δ_{TC}=1 mm from their inner water-cooled surface, differs from the temperature on this surface by $\Delta t_{TC}=q\delta_{TC}/\lambda_w$. The water temperature is measured thanks to 7 K-type thermocouples introduced through gland seals directly into the coolant flow (Fig.3). The inlet and outlet pressures are measured using standard pressure gauges and diaphragms (three standard sizes) with differential pressure gauges give the flow rate.

The experiments were made with 3 channel heights (δ_{ch}=12.5, 25 and 50 mm), 2 heated lengths (L_{heat}=1.02 and 2.48 m) and 7 inclination angles to the horizontal (β). To cover a wide range of Reynolds numbers ($Re=wd_h/\nu$), including the traditional laminar (Re<2300), turbulent (Re>4000-6000) and the transition-to-turbulent areas ($2300\geq Re\geq 4000$) the experiments were made at water velocities from a range of 6.7 - 170 mm/s and 2 water temperatures (t_f=19-150°C) corresponding to Re=560-17900. The specific heat flux from the upper and lower sides of the channel varied in the range q=(1.3-24.4) kW/m^2. The test-model characteristics are presented in Table 1, while the test conditions are given in Table 2.

3. TEST RESULTS AND DISCUSSIONS

A total of 361 experiments were carried out. The results of the experiments are analyzed separately for the vertical, horizontal and inclined channels.

Vertical channels.

A total of 92 experiments were performed, and as a result 184 experimental heat transfer coefficients (***h***) were obtained. These data (***h***) were used to determine the generalized correlation for the vertical channels (Fig.4). To evaluate the effect of the thermo-gravitational forces on the heat transfer the technique suggested by Balunov [2] was applied. The mutual effect of the forced and natural convection forces was characterized by the ratio of heat transfer coefficients calculated by the formulas for the forced and natural convection (h_{forced}/h_{NC}) [2]. The processing of $h/h_{forced}=f(h_{forced}/h_{NC})$ provides the possibility to reflect the ultimate transitions, i.e. $h=h_{NC}$ should occur at $h_{NC}>>h_{forced}$ and $h=h_{forced}$ at $h_{forced}>>h_{NC}$. As recommended in [9], h_{NC} was defined by Mikheev formula:

$$h_{NC} = 0.135(Gr_tPr)^{1/3}\lambda_f/d_h \text{ at } Gr_tPr>2\cdot 10^7, \quad (1)$$

$$h_{NC} = 0.54(Gr_tPr)^{1/4}\lambda_f/d_h \quad \text{at } Gr_tPr<2\cdot 10^7, \quad (1a)$$

with $Gr_t=gd_h^3\beta\Delta t/\nu^2$;

whereas h_{forced} - by Petukhov formula [3]:

$$h_{forced} = Nu_0\lambda_f/d_r = \frac{(\lambda_{fr}/8)\,Re\,Pr}{k+4.5\sqrt{\lambda_{fr}}\,(Pr^{2/3}-1)}\ \lambda_f/d_h, \quad (2)$$

where k=1+900/Re; $\lambda_{fr}=(1.82\cdot lg(Re)-1.64)^{-2}$. Though in [3] correlation (2) is recommended for the range of Re=(4000 - $5\cdot 10^6$), Petukhov in [1] recommends to extend the use of formula (2), in case of the mixed convection, for lower Re values (down to Re=300) due to earlier flow turbulization under the natural convection forces. In the considered experiments with vertical channels the values of Re number (Re=770-17500) are within the above boundaries (300-$5\cdot 10^6$). Therefore, evaluation of $h/h_{forced}=\boldsymbol{f}(h_{forced}/h_{NC})$ we use the formula (3) with the correction factor 0.913 for calculation of h_{forced}. This correction factor provides the consistency $h=h_{forced}$ at $h_{forced}>2h_{NC}$ both between our own experimental data and the results of experiments in [5-8].

Fig. 4 presents all experimental data obtained in the present work regardless of the Re value for various channel heights and heated lengths. As seen from the graph, the spread in the experimental data in the zone of heat exchange worsening and at $h_{forced}>h_{NC}$ is essential. In this zone the spread is attributable to a sharp increase in ***h*** and a corresponding decrease in the temperature drop $\boldsymbol{\Delta t}$ to (5-10)°C, i.e. an increase in the relative error in defining $\boldsymbol{\Delta t}$ and ***h***. For the zone of minimum values of h/h_{forced} ratio it should be admitted that the use of formula (2) for a wide range of Re values, though contributing to universalization of calculations, makes the analytical description more approximate. Thus, the formula used to define h_{forced} at Re<4000 for the flow turbulized by the natural convection, as well as the h_{forced}/h_{NC} ratio used to characterize the interrelation between the forced and natural convection, are correlations allowable for engineering calculations only. The formulas proposed below correlate the experimental data presented in Fig.4 with an accuracy of ±10%:

$h = h_{NC}$ at $h_{forced}/h_{NC}<0.13$ (3)

$h = 0.542 \cdot h_{forced}(h_{forced}/h_{NC})^{-1.3}$ at $h_{forced}/h_{NC}=0.13\text{-}0.9$ (4)

$h = 0.66 \cdot h_{forced}(h_{forced}/h_{NC})^{0.6}$ at $h_{forced}/h_{NC}=0.9\text{-}2.0$ (5)

$h = h_{forced}$ at $h_{forced}/h_{NC}>2.0$ (6)

Thus, the mixed convection area for the vertical rectangular channels is limited by $h_{forced}/h_{NC}=0.13\text{-}2.0$. The values of h/h_{forced} are minimum at close values of h_{NC} and h_{forced}.

Horizontal channels.

Hundred experiments were carried out. Figs. 5 and 6 show the characteristic graphs of temperature distribution in channel length: water temperature (t_f), temperature of the upper and lower heating walls nearby (δ_{TC}=1mm) their inner surface (t_w), temperature of the lateral unheated wall (t_L). Graphs in Figs. 5-6 characterize the following.

1. For the upper heating surface the character of distribution of its temperature (t_w) in channel length and water temperature in channel height was different:

- practical coincidence of temperatures of the upper and lower heat releasing surfaces and the filled symmetrical profile of water temperature through the channel height. These experiments were characterized by relatively high water velocities (w=0.13-0.15 m/s) and an excess of h_{forced} over h_{NC}. The heat transfer coefficients are rather high (h>1.8 kW/m^{2}K) and exceed the calculated values by a factor of ~1.2 in case of the turbulent forced convection (see formulas (3)), i.e. the heat exchange is intensified to a certain extent under the combined action of the forced and natural convection forces;
- nonmonotonous change (Fig.6) or a drop in temperature (t_w) along the heated length of the channel. These experiments performed at a water velocity less than (0.1-0.032) m/s were characterized by a strictly asymmetric water temperature profile through the channel height (the difference in temperatures of the heat releasing surfaces amounted to 55K) and a linear change in the water temperature on the upper near-wall part of the channel up to half of channel height. The latter is characteristic only for the stagnant liquid layer, which is confirmed by similar values of heat flux on the upper heating surface (q_w) and heat flux transferred through water only due to thermal conductivity. The heat transfer coefficients for the upper heat releasing surface are low (h=62-110 W/m^{2}K) and are increased but slightly with a rise in the flow velocity. The above refers to the channels of different height and to different bulk water temperatures, which is characterized by a wide range of Reynolds numbers (for example, Re=750-4000 at w≈0.015 m/s). A drop in temperature of the heat releasing surface along the channel length might be explained by the presence of a weak counter-flow of water nearby this surface. Note also that the flow temperature stratification increased the period of the flow temperature stabilization by a factor of 3-8, as compared with the vertical channels;
- at w≈0.04 m/s noteworthy are the experiments where the value of dt_f/dy derivative on the near-wall upper part 3 mm in height is lower than on the part located below. These experiments are characterized by moderate values of h=170-210 W/m^{2}K and for them the values on the upper heating surface are considerably higher than the maximum values obtained from $q_\lambda=\lambda\Delta t_\delta/\delta$.

2. For the lower heating surface the temperature difference $\Delta t=t_w-t_f$ remained unchanged along the channel length (within the measurement error), i.e. for this surface, as for the vertical channel, the length of the thermal stabilization of the flow did not exceed 0.356 m.

Thus, for the lower and upper heating surfaces the heat exchange mechanisms are noticeably different, so they are considered below separately.

<u>For the lower heating surface</u> the heat transfer intensity is determined mainly by the natural convection forces with a certain action of the inertial-viscous forces (natural convection forces). Therefore, the experimental data were processed practically in the same coordinates, as for the vertical channels: h/h_{NC}=f(h_{forced}/h_{NC}). The results of this processing are presented in Fig.7. Note that:

- at $h_{forced} < h_{NC}$ the experimental ***h*** values exceed the calculated h_{NC} values by a factor of ~1.3, the reasons of which are presented above;
- at $h_{forced} > h_{NC}$ the heat transfer is intensified to a certain extent, as compared with the calculation by formulas (2, 3) for the natural or forced convection. But in this area the experimental data are obviously not sufficient for any generalization. It is only possible to state that the zone of impaired heat transfer is absent at close values of h_{forced} and h_{NC}, which occurred in the vertical channel.

<u>For the upper heating surface</u> while taking into account the dependence of the heat transfer coefficient only on the flow velocity and its independence on the channel height and water temperature (Re and Pr values), the primary processing of the experimental data was made in the coordinates h=f(ρw). Fig.8 shows the examples of this processing for the channels δ_{ch}=(12.5, 25, 50) mm in height with the heated channel length L_{heat}=2.48 m. The graphs show the range of Re values, at which the experiments were made, and curves calculated by formulas:

curve 1	$h=0.755(\rho\, w)^{1.45}$	(8)
curve 2	$h=30.5(\rho w)^{1/3}$	(9)
curve 3	by formula (1)	

The first two formulas generalize the presented experimental data (though with a considerable spread), and the last formula defining the heat transfer intensity under the forced turbulent convection (at t_f=80°C) is chosen as a scale. Before analyzing the experimental data presented in Fig.8 note the lack of effect of the heat flux (no less than four-fold range of changes during the experiments), water temperature (three-fold change in Pr number) and channel height on the character of the dependence h=f(ρw). In this case different Re values corresponded to a certain value of ρw. Thus, for example, Re=880-9600 corresponded to the bend point on the «generalizing curve»(ρw=29 kg/m^{2}s).

The above noted dependence of ***h*** on the geometric and regime parameters in the experiments (independence from everything except for the flow velocity) made necessary the processing of Nu=f(Pe) when representing the «characteristic length (L)» in both criteria as a function of physical constants varying slightly with a change in the flow temperature. Therefore, the «temperature-conducting length» $L_a=(a^2/g)^{1/3}$ was used, and the processing took the final form: $Nu_a=f(Pe_a)$, where $Pe_a=wL_a/a= w/(a\cdot g)^{1/3}$. The results of this processing for all experimental data are shown in Fig. 9. The bulk of the presented experimental data (more than 90%) are generalized with a spread of ±25% by the following correlations:

$$Nu_a=1.44\cdot10^{-3}(Pe_a)^{1/3} \quad \text{at } Pe_a<2.6 \qquad (10)$$

$$Nu_a=4.95\cdot10^{-4}(Pe_a)^{1.45} \quad \text{at } Pe_a>2.6 \qquad (11)$$

or

$$\max\{Nu_a=1.44\cdot10^{-3}(Pe_a)^{1/3} \text{ and } Nu_a=4.95\cdot10^{-4}(Pe_a)^{1.45}\} \qquad (12)$$

The spread in the experimental data (±25%) is, of course, essential, but it is attributable to a number of objective reasons:

- error in defining q_{upper} [$\delta(q_{upper})\approx6\%$];
- certain stochasticity of the process characteristics due to parameters of secondary backward flows can vary within certain limits under random factors;

The limited amount of experimental data, with no flow stratification, makes possible only qualitative estimate of the flow stratification point in the horizontal channel. The flow temperature stratification was absent only at a flow velocity more than 0.13 m/s occurring only in experiments with a channel 12 mm high. It was characterized by an excess of h_{forced} over h_{NC}.

In this case the transition from stratified to unstratified flow regime was stepwise with a two-three-fold increase in h/h_{forced} in going through $h_{forced} \approx h_{NC}$. This was most evident in experiments with the inclined channels, where the processing for both heating surfaces was made as for the vertical channels in the form of $h/h_{forced}=f(h_{forced}/h_{NC})$.

Inclined channels.

169 experiments were performed. In all experiments the temperature of the heat releasing wall (t_w) changed monotonously along the length of the channel, and the length of the thermal flow stabilization, did not exceed 0.356 m, as for the vertical channel. Therefore, the indications of the thermocouples (t_f; t_w; t_S; t_L) located in the third measured cross-section (x=2.12 m) were also used for processing of the experimental data, and the processing itself was made in the coordinates $h/h_{forced}=f(h_{forced}/h_{NC})$ used for the vertical channels. In this case h_{forced} and h_{NC} were also calculated by correlation (3) and correlations (2; 2a), respectively. The examples of such a processing of the experimental data at the channel inclination angle from 5° to 45° are presented in Figs. 10-14.

As follows from these figures:

- the flow temperature stratification, as in the case of the horizontal channels, is absent, provided the condition $h_{forced}>h_{NC}$ is fulfilled, and the stratification-to-unstratification transition is followed by a two-three-fold increase in h/h_{forced} for the upper heating surface;
- for the lower heating surface at h_{forced}/h_{NC} <1.0 the heat transfer coefficient corresponds to the calculated value for the natural convection (2, 2a). True, with the channel inclination angle decreased, a certain rise in ***h*** is observed caused by an increase in the flow temperature stratification and a relevant increase in the bulk water temperature, as compared with its temperature in the lower part of the channel which actually determines the value of $\Delta t=q/h_{NC}$. At an inclination angle of 45° in the vicinity of h_{forced} =(0,3-0,9)h_{NC} the heat transfer is slightly impaired, which is characteristic of the vertical channels. But the above-mentioned improvement and deterioration are moderate and fit in the limit (±15%) of the proposed recommendations. At $h_{forced}/h_{NC}>1.0$ the heat transfer coefficient corresponds to the calculated value for the forced convection (see formula (3) witht he introduced additional correction factor equal to 0.913).
- for the upper heating surface the calculation curve described by the correlation given below is drawn in Figs. 11-15:

$$h = 0.23\, e^{0.88\sin\beta}\, h_{forced}\, (h_{forced}/h_{NC})^{-1,42} \qquad (13)$$

At h_{forced}/h_{NC} <0.85 the difference of minimum ***h*** values from those described by correlation (13) amounts to no more than 10%. For the vertical channel (β=90°) the calculations by formulas (5 and 13) yield identical results, i.e. it is proposed to use correlation (13) both for the vertical and inclined channels at β≥5°.

4. CONCLUSION

The experimental study has been made to define the heat transfer coefficients for a wide-range of the operational parameters of the Vacuum Vessel cooling system. On the whole 361 experiments were made.

On the basis of the analysis of the experimental results the completing correlations were developed describing the heat transfer coefficient (*h*) at the parameters described in Tables 1- 2. Thus, for the heat releasing surfaces of the Vacuum Vessel cooling channels it is recommended to define ***h*** values by the following correlations:

- correlations (4-7) for the vertical channels;
- correlations (2; 3) for the horizontal channels and for the surfaces cooled from above (lower heating surfaces) by choosing the highest out of the obtained ***h*** values;
- correlation (12) for the horizontal channel and for the surfaces cooled from below (upper heating surface);

- for horizontal channels no more than 12 mm in height the water temperature stratification is absent, if the water velocity therein exceeds the value corresponding to the condition h_{forced}=1,3h_{NC}, where h_{forced} and h_{NC} are to be calculated by correlations (2 and 3), respectively. With this condition fulfilled, ***h*** is to be calculated by common formula (3) for the upper and lower heating surfaces;
- correlations (2, 2a) for inclined channels and for the lower heating surface at $h_{forced}/h_{NC}<1.0$ and formula (3) (with the introduced additional correction factor equal to 0.913).at $h_{forced}/h_{NC}>1.0$;
- correlation (13) for the inclined channels and for the upper heating surface.

FIGURE CAPTIONS

Fig. 1. Flow diagram of the CBTI experimental facility.

Fig. 2. General view of the Test-model.

Fig. 3. Scheme of the thermocouples location
○ - thermocouples t_w; t_s; t_{lat}; ○- - thermocouples t_f

Fig. 4. Generalization of experimental data on the vertical channel.

δ_{ch} = 12.5, 25, 50 mm; L_{heat} = 2.48, 1.02 m

Fig. 5. Temperature distribution in the tested model for horizontal channel.
δ_{ch} = 50 mm, L_{heat} = 2.48 m
w = 0.0612 m/s; N_{upper} / N_{lower} = 6.41 / 7.22 kW

Fig. 6. Temperature distribution in the tested model for horizontal channel.
δ_{ch} = 50 mm, L_{heat} = 2.48 m
w = 0.0315 m/s; N_{upper} / N_{lower} = 2.98 / 3.09 kW

Fig. 7. Generalization of experimental data on horizontal channel (lower heating).
L_{heat} = 2.48 m
$h_{NC} = 0.135 \cdot (GrPr)^{0.333} \cdot \lambda_f / d_h$ at $GrPr > 2 \cdot 10^7$
$h_{NC} = 0.54 \cdot (GrPr)^{0.25} \cdot \lambda_f / d_h$ at $GrPr < 2 \cdot 10^7$
$h_{forced} = (0.125 \varepsilon_{fr} RePr)/(k+4.5(\varepsilon_{fr} \cdot (Pr^{0.67}-1))^{0.5}) \cdot \lambda_f / d_h$

Fig. 8a. Heat transfer coefficient (h) vs. flow rate (ρw).

δ_{ch} = 50 mm, L_{heat} = 2.48 m, upper heating

1, 2 – generalizing curves; 3 – calculated curve

$1 - h = 0.755 \cdot (\rho w)^{1.45}$

$2 - h = 30.5 \cdot (\rho w)^{0.333}$

$3 – h = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4} \cdot \lambda_f / d_h$

Fig. 8b. Heat transfer coefficient (h) vs. flow rate (ρw).

δ_{ch} = 25 mm, L_{heat} = 2.48 m, upper heating

1, 2 – generalizing curves; 3 – calculated curve

$1 - h = 0.755 \cdot (\rho w)^{1.45}$

$2 - h = 30.5 \cdot (\rho w)^{0.333}$

$3 – h = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4} \cdot \lambda_f / d_h$

Fig. 8c. Heat transfer coefficient (h) vs. flow rate (ρw).

δ_{ch} = 12.5 mm, L_{heat} = 2.48 m, upper heating

1, 2 – generalizing curves; 3 – calculated curve

$1 - h = 0.755 \cdot (\rho w)^{1.45}$

$2 - h = 30.5 \cdot (\rho w)^{0.333}$

$3 – h = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4} \cdot \lambda_f / d_h$

Fig. 9. Experimental data processing $Nu_a = f(Pe_a)$.

Horizontal channel, L_{htr} = 2.48 m, upper heating

1, 2 – generalizing curves:

$1 - Nu_a = 1.44 \cdot 10^{-3} (Pe_a)^{0.333}$

$2 - Nu_a = 4.95 \cdot 10^{-4} (Pe_a)^{1.45}$

Fig. 4.a. Обобщение экспериментальных данным по наклонным каналам (угол наклона β =45 °).

Нижняя теплоотдающая поверхность

δ_{ch} = **12.5, 25, 50 mm; L_{heat} = 2.48, 1.02 m**

1 – обощающая кривая:

$$h = 0.23 \cdot \exp(0.88 \cdot Sin\beta) \cdot h_{forced} \cdot \left(h_{forced} / h_{NC}\right)^{-1.32}$$

Fig. 4.б. Обобщение экспериментальных данным по наклонным каналам (угол наклона β =45 °).

Верхняя теплоотдающая поверхность

δ_{ch} = **12.5, 25, 50 mm; L_{heat} = 2.48, 1.02 m**

1 – обощающая кривая:

$h = 0.23 \cdot \exp(0.88 \cdot Sin\beta) \cdot h_{forced} \cdot (h_{forced} / h_{NC})^{-1.32}$

Fig. 5. Обобщение экспериментальных данным по наклонным каналам (угол наклона β=5° и 7°).

δ_{ch} = **12.5, 25 mm; L_{heat} = 2.48 m**
1 - β=5° ; 2 - β=7°
1 ; 2 – generalizing curves:
$h = 0.23 \cdot \exp(0.88 \cdot Sin\beta) \cdot h_{forced} \cdot (h_{forced} / h_{NC})^{-1.32}$

Fig. 12. Generalization of experimental data on the inclined channel (inclination angle β=30 °).

δ_{ch} =**25 mm; L_{heat} = 2.48 m**
○ - lower heating plate
○ - upper heating plate
1 – generalizing curve:
$h = 0.23 \cdot \exp(0.88 \cdot Sin\beta) \cdot h_{forced} \cdot (h_{forced} / h_{NC})^{-1.32}$

Fig. 13. Generalization of experimental data on the inclined channel (inclination angle β=15 °).

δ_{ch} = **12.5, 50 mm; L_{heat} = 2.48 m**
1 – generalizing curve:
$h = 0.23 \cdot \exp(0.88 \cdot Sin\beta) \cdot h_{forced} \cdot (h_{forced} / h_{NC})^{-1.32}$

Table 1. Test-model characteristics

Channel shape	rectangular
Inner width, a (m)	0.2
Total length, l (m)	3
Hight, δ_{ch} (mm)	12.5 ; 25 ; 50
Heated length, L_h (m)	1.02 ; 2.48
Inclination angle, β (°)	0, 5, 7, 15, 30, 45, 90

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Table 2. Test matrix (361 tests)

Pressure, P, MPa	1.0
Water inlet temperature, t_{in}, °C	20, 100
Water velocity, w, m/s	0.0067 ÷ 0.17
Heat flux, q, kW/m^2	1.3 ÷ 24.4

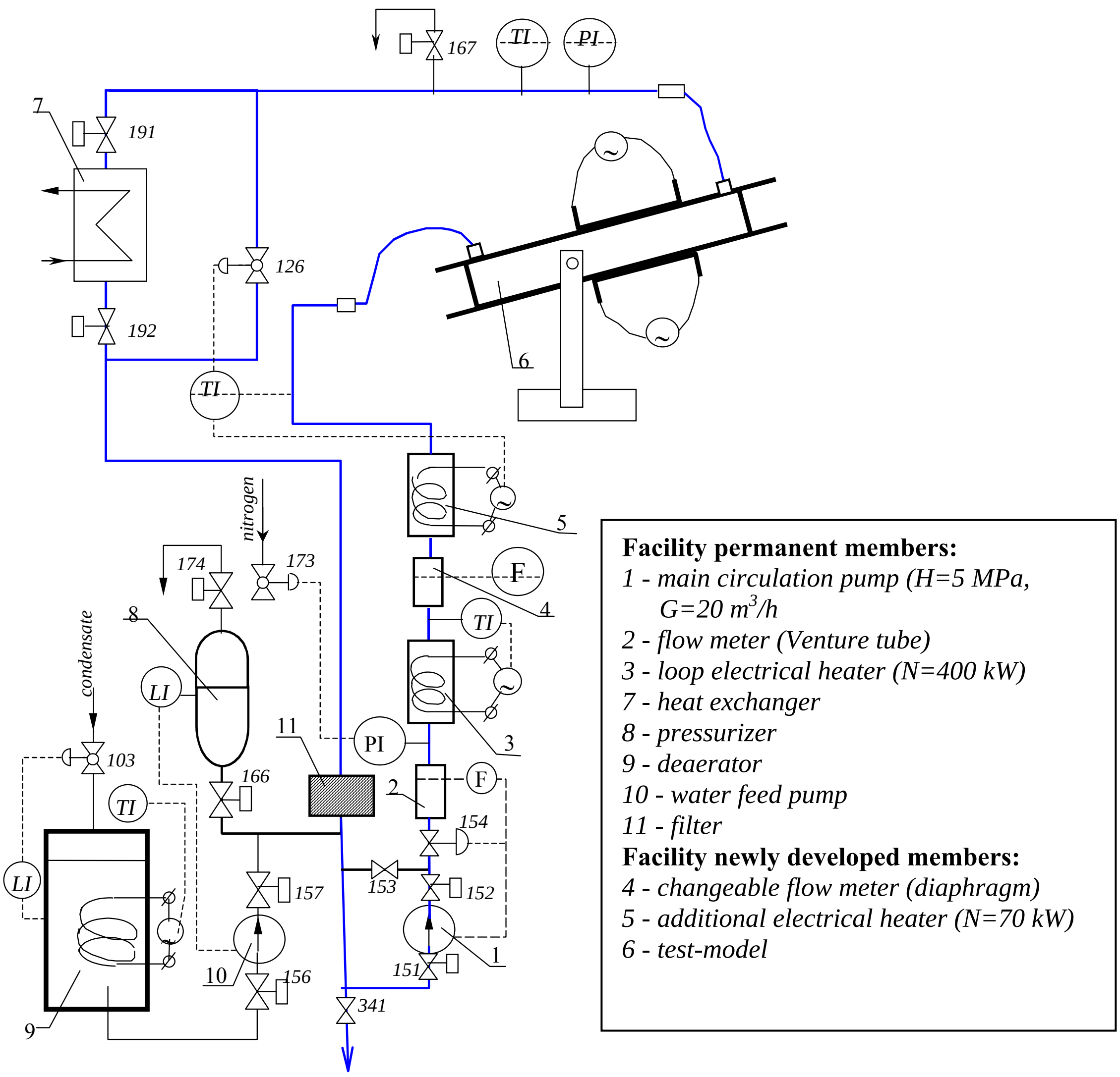


Fig. 1.
(preferable 1/6pp)

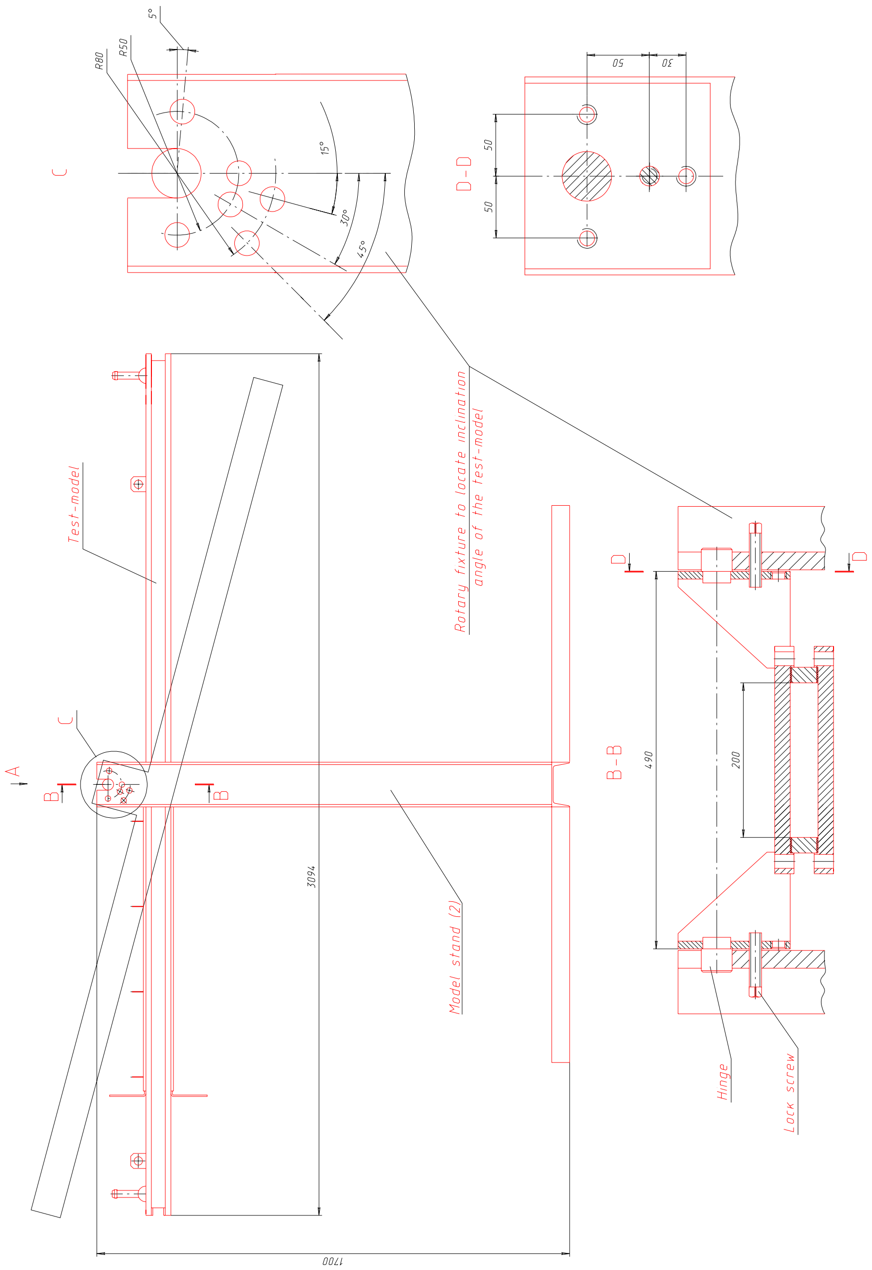
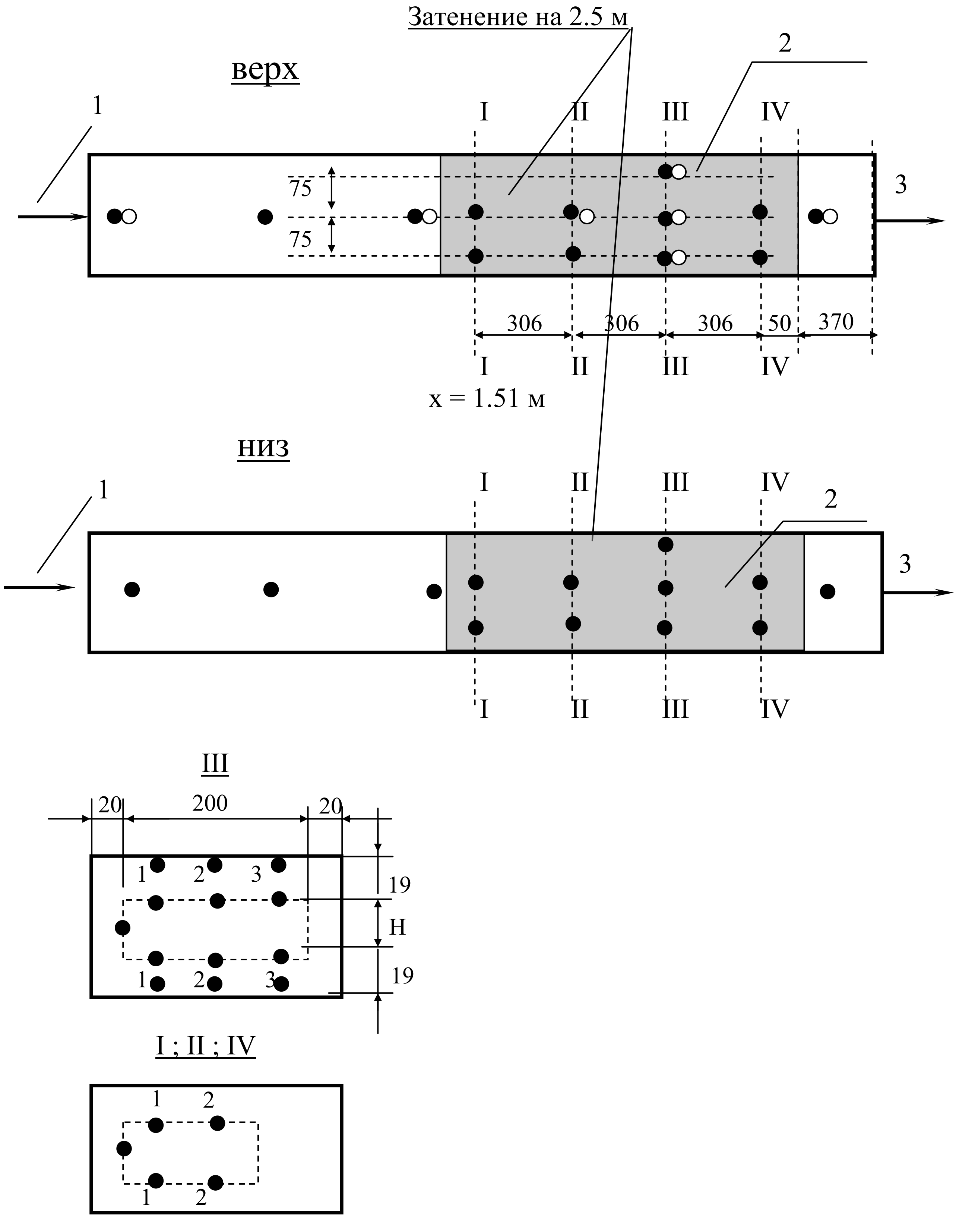


Fig. 2.
(preferable 1/6pp)

Обобщение экспериментальных данных по вертикальным каналам


Затенение на 2.5 м
верх
1
2
3
I
II
III
IV
75
75
306
306
306
50
370
x = 1.51 м
низ
III
20
200
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19
1
2
3
I ; II ; IV

Рис. 1

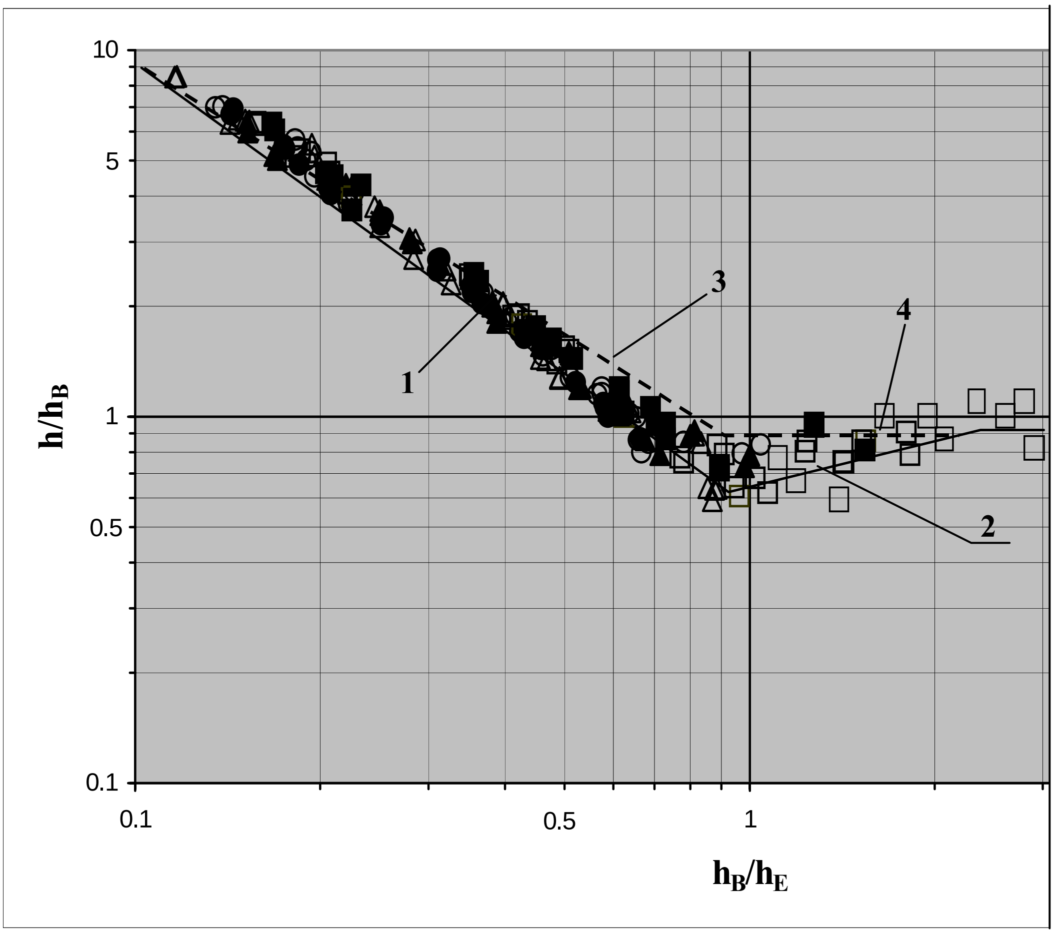
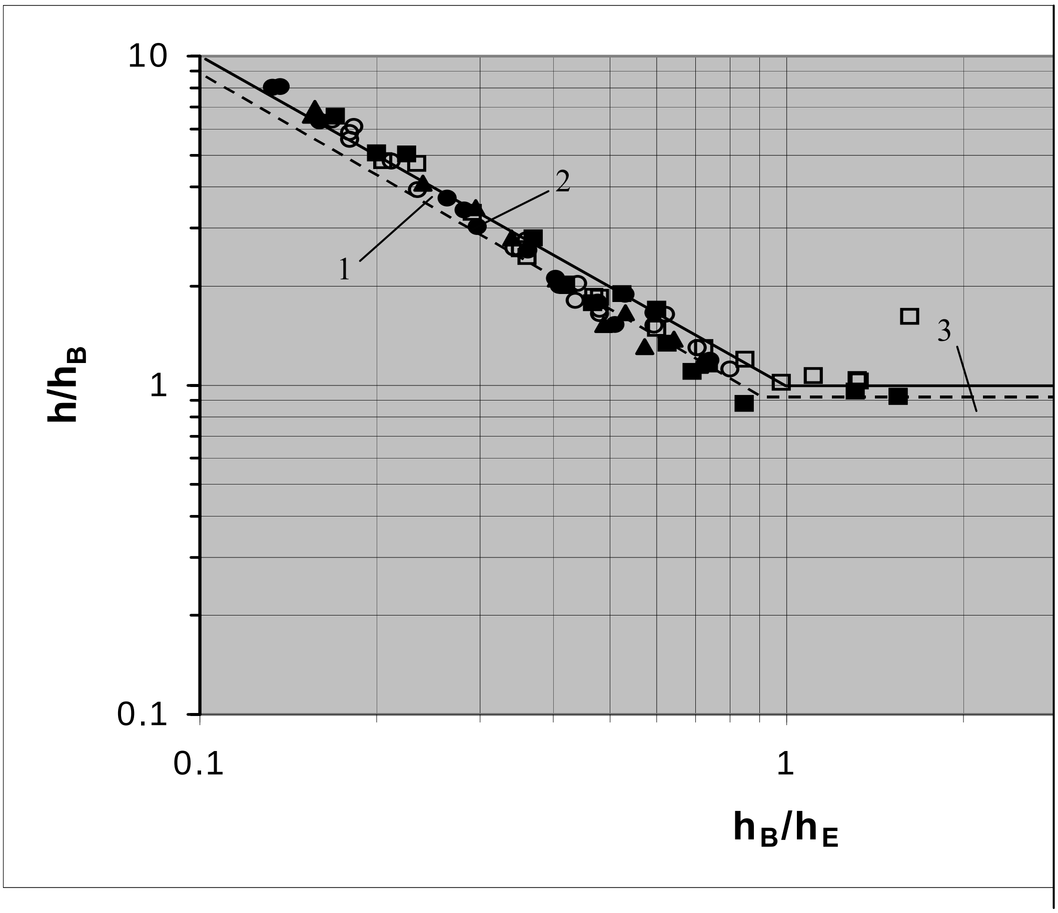
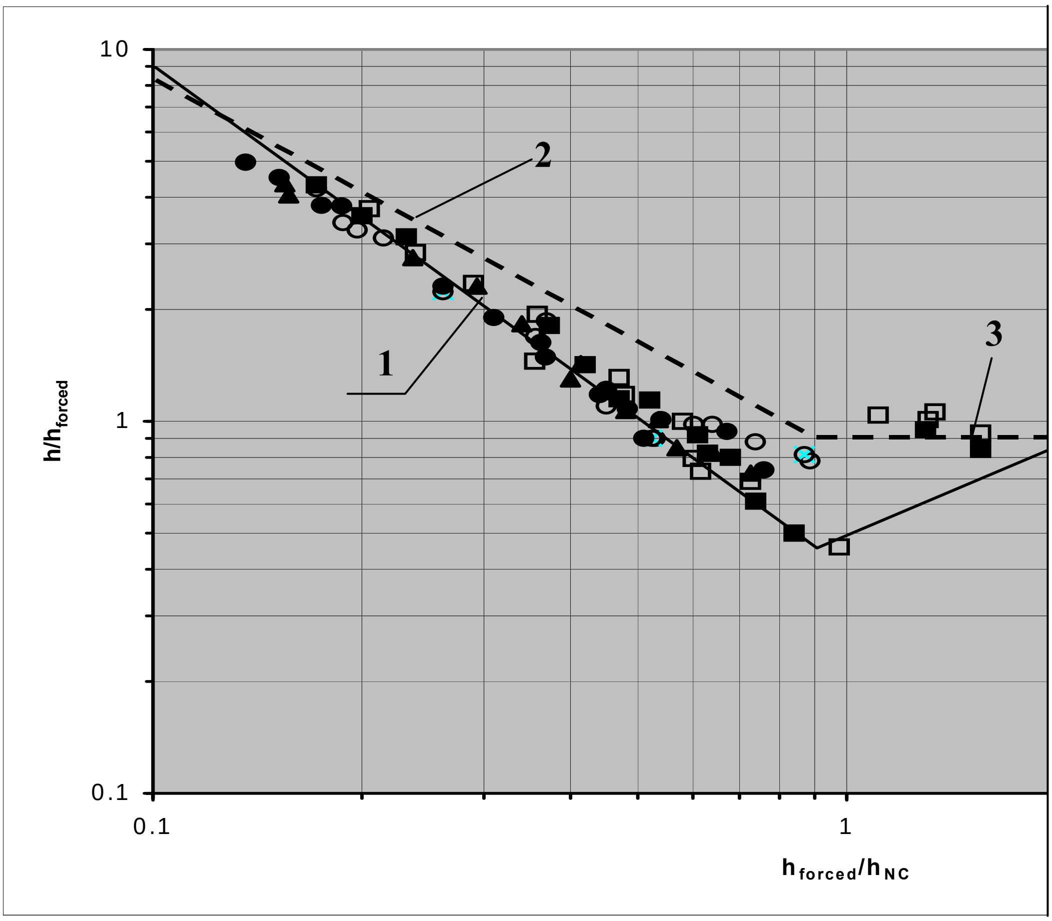


Рис. 2



а

Рис. 4.



б

Рис. 4.

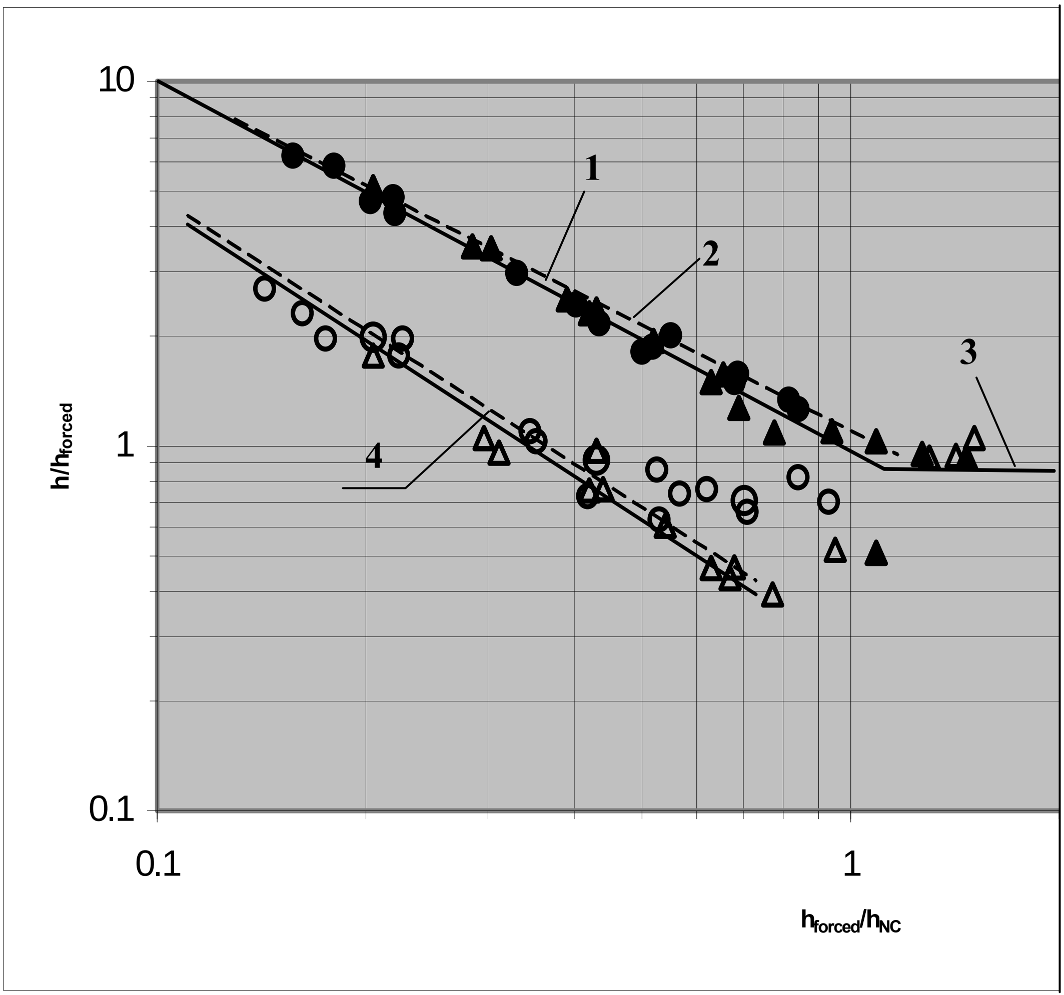


Рис. 5.

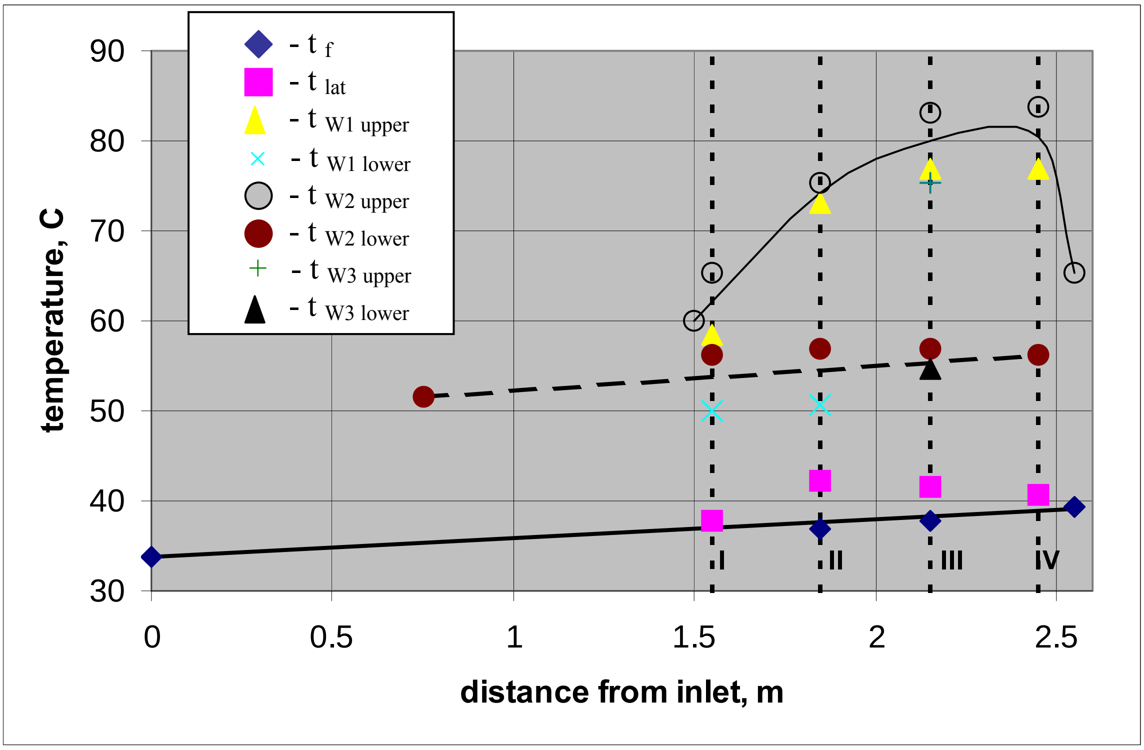


Fig. 5.
(preferable 1/6pp)

δ, mm	β, degree	upper side	lower side
25	7		
12.5	5		

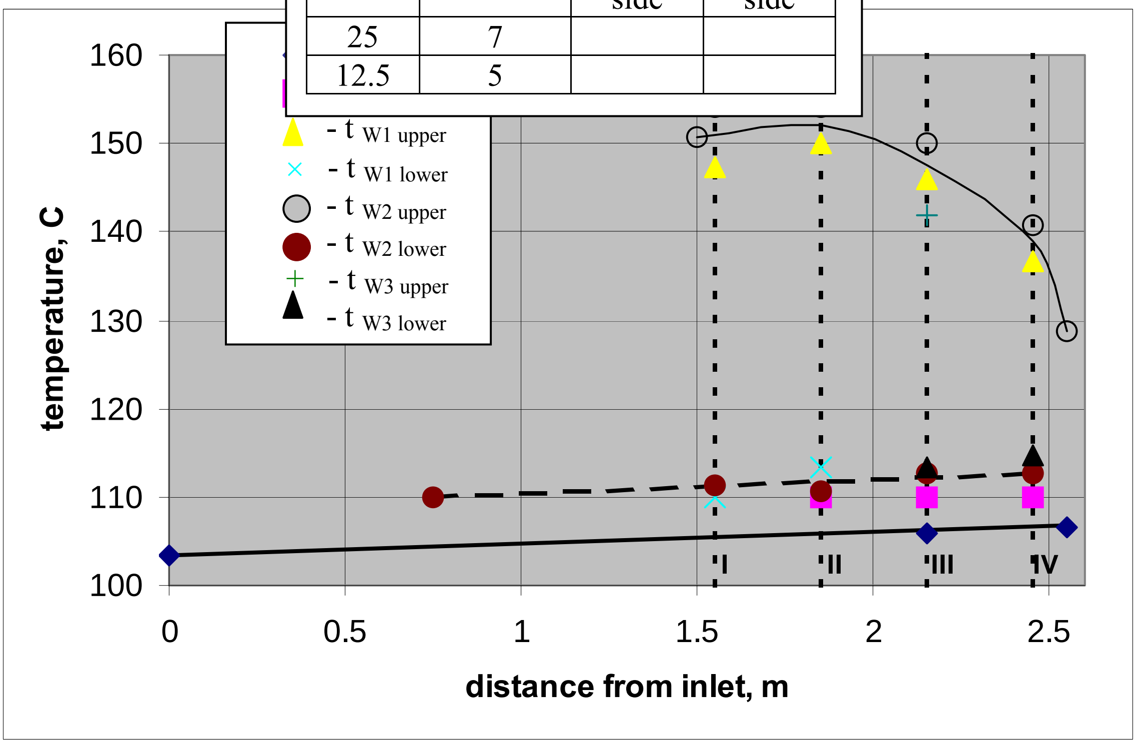


Fig. 6.
(preferable 1/6pp)

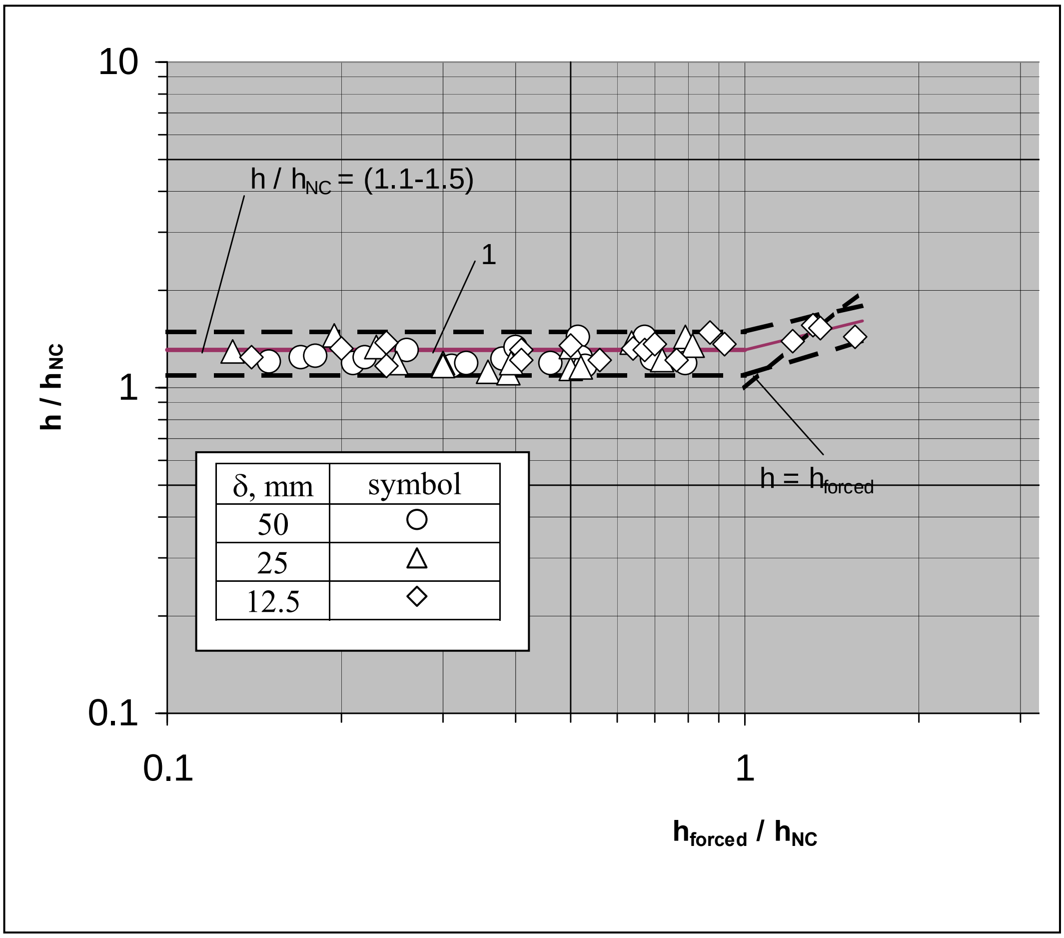


Fig. 7.
(preferable 1/6pp)

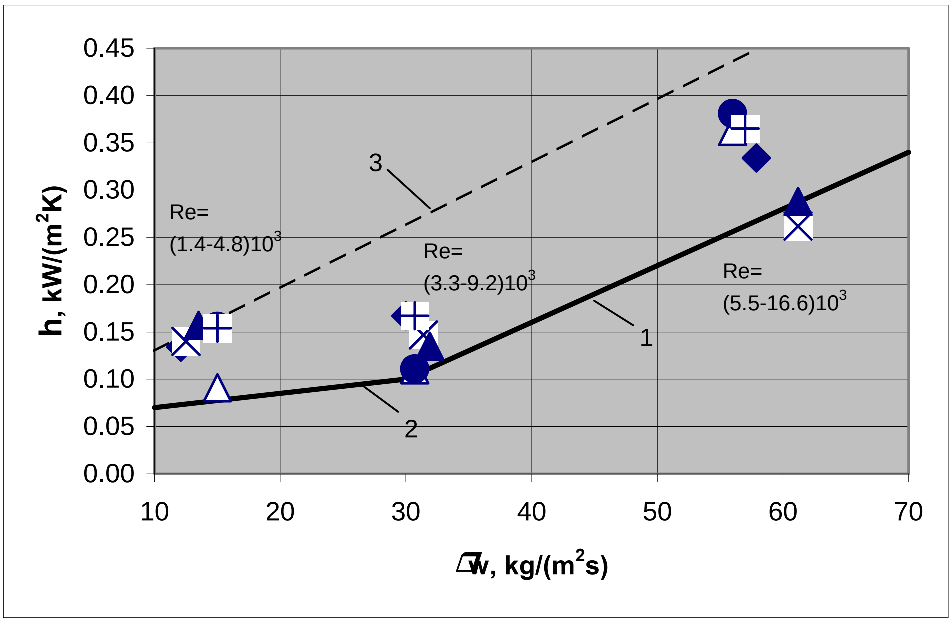


Fig. 8a.
(preferable 1/6pp)

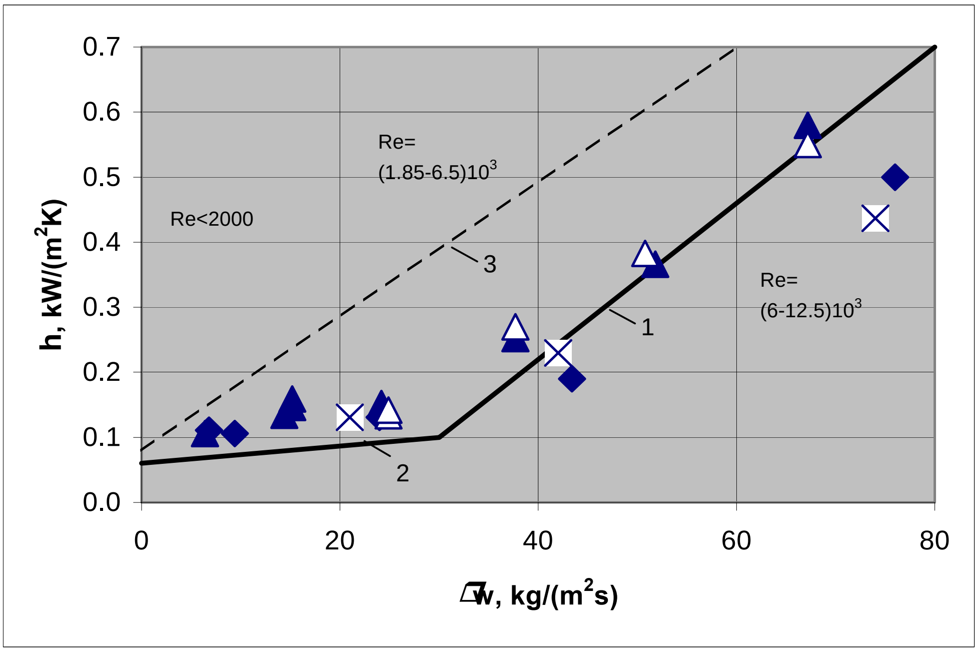


Fig. 8b.
(preferable 1/6pp)

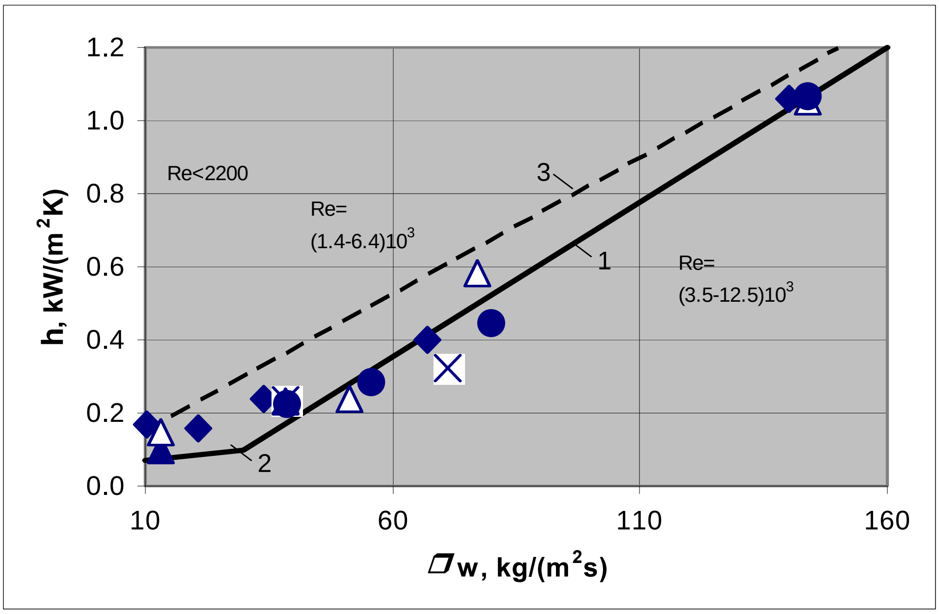


Fig. 8c.
(preferable 1/6pp)

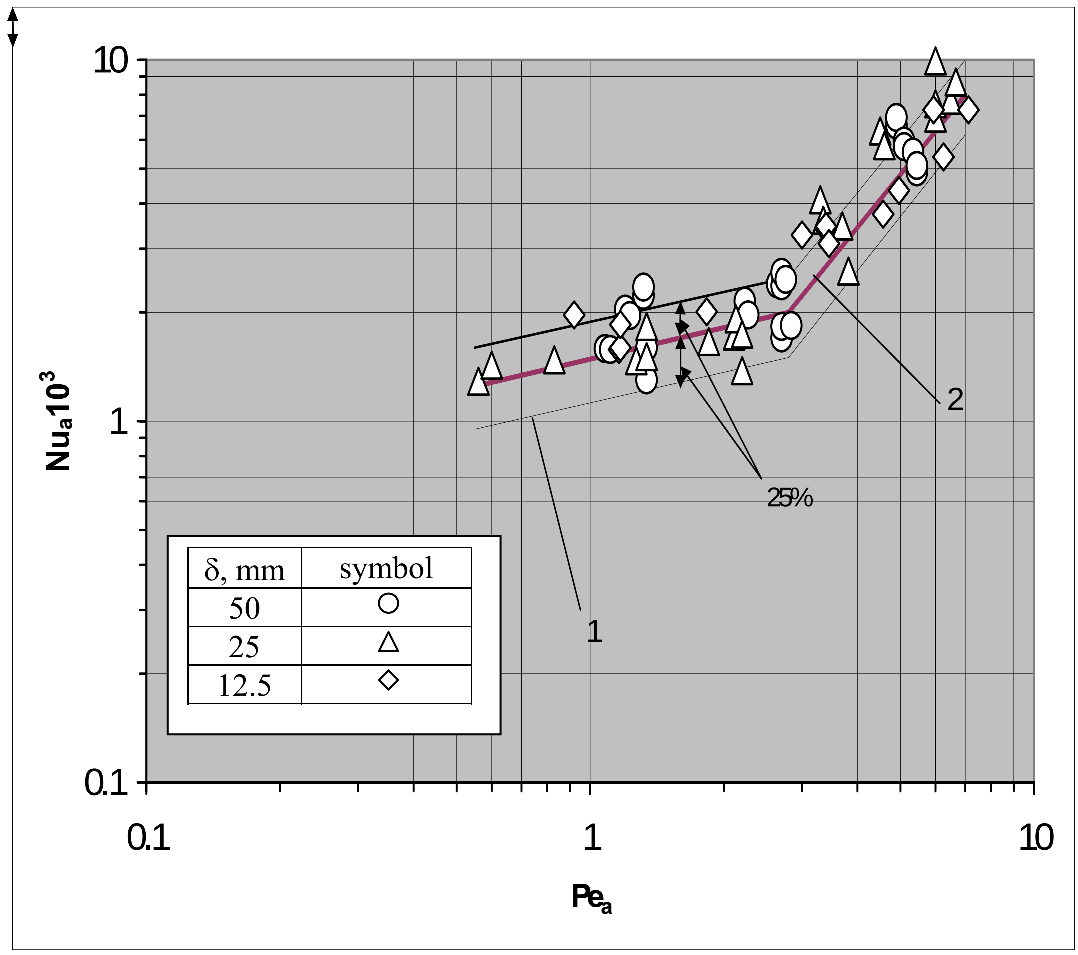


Fig. 9.
(preferable 1/6pp)

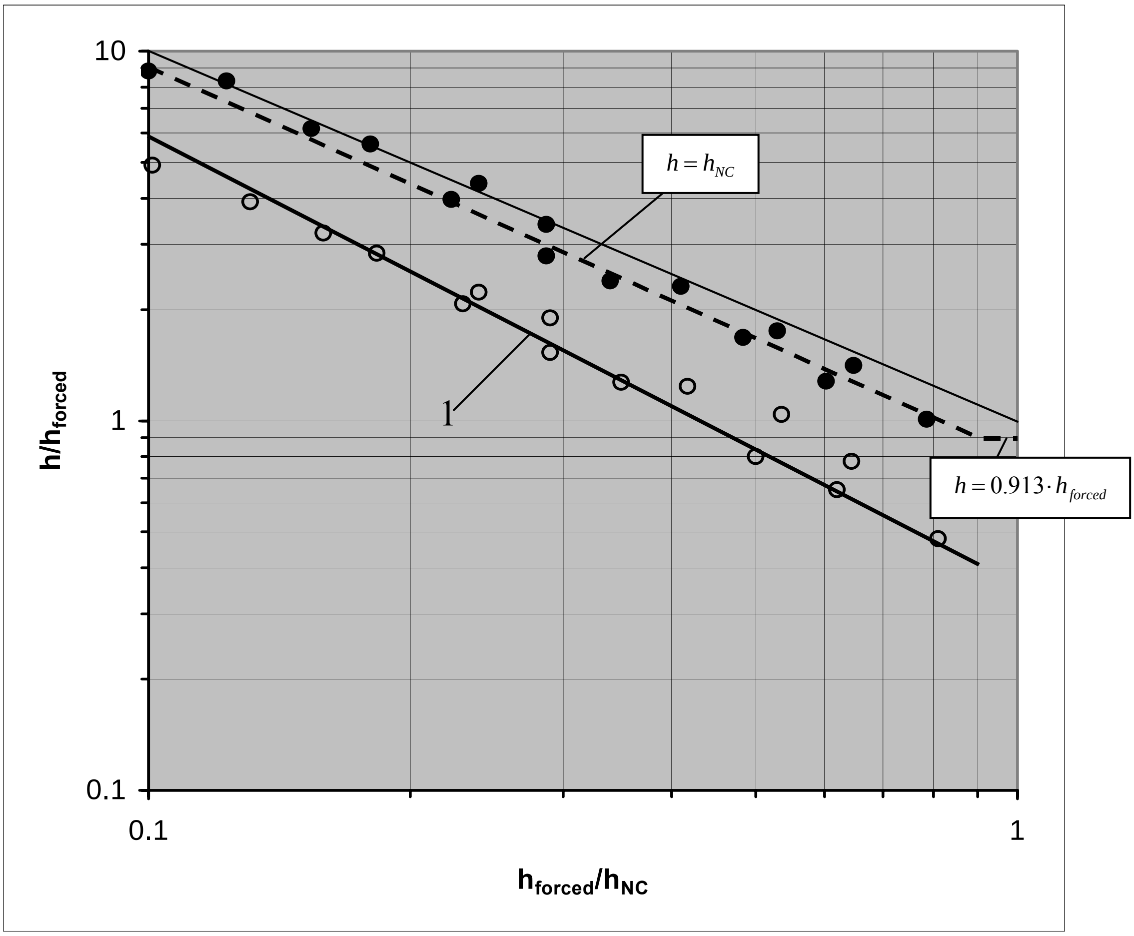


Fig. 12.
(preferable 1/6pp)

L_{heat}, m δ, mm	2.48	1.02
50		
25		
12.5		

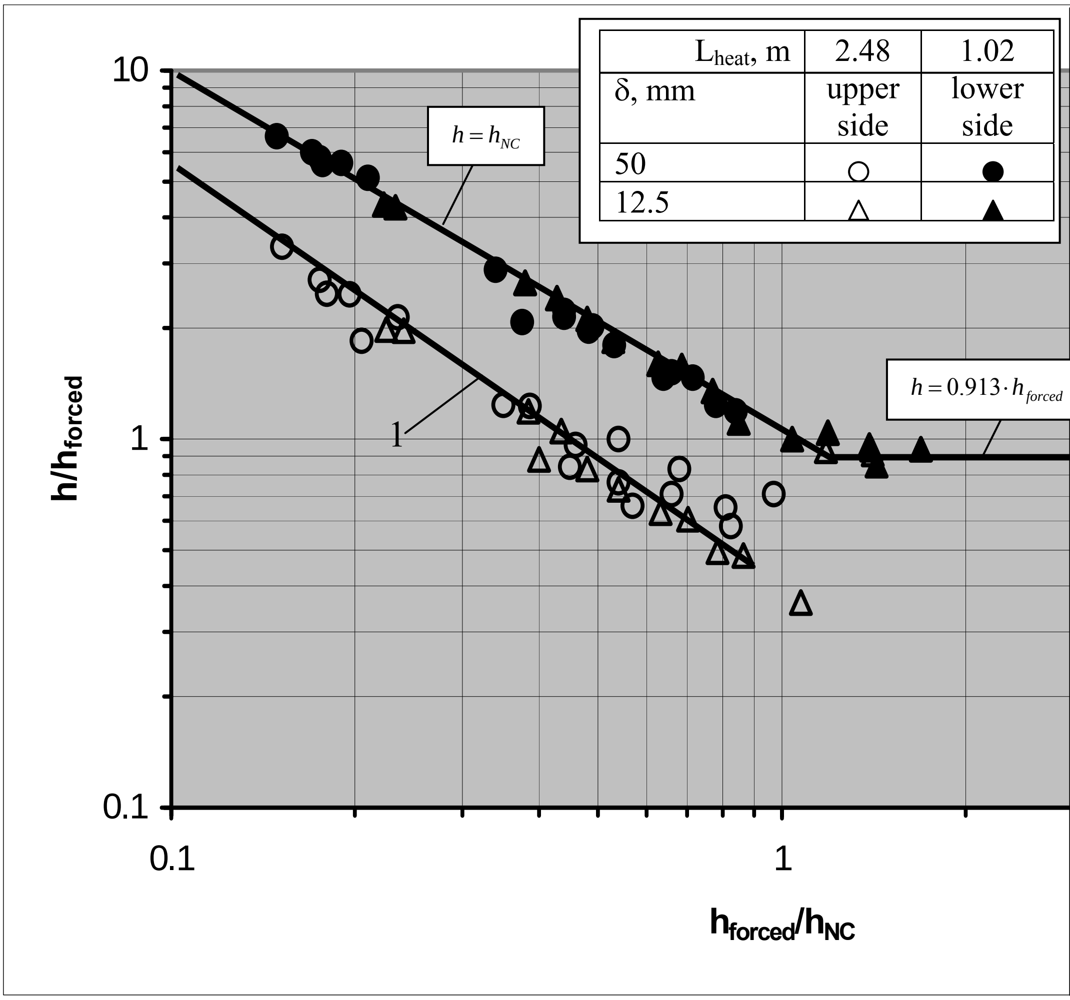


Fig. 13.
(preferable 1/6pp)

1; 2 – обобщенные кривые;

1 - $h = 0.542 \cdot h_{forced} \cdot \left(h_{forced} / h_{NC}\right)^{-1.3}$

2 - $h = 0.66 \cdot h_{forced} \cdot \left(h_{forced} / h_{NC}\right)^{0.6}$

L_{heat}, м / δ, мм	2.48	1.02
50		
25		
12.5		