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Heat transfer enhancement from a wire to an impinging upward submerged slot jet of water in sub-cooled and saturated boiling conditions.

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ABSTRACT

Experimental data about the heat transfer from a heated 0.25 mm. in diameter nickel wire to a submerged upward flow slot jet of distilled water are reported. The tests are carried on in forced convection regime, in sub-cooled and saturated boiling conditions. The influence on the heat transfer rate of different parameters is investigated, such as the jet exit velocity, the distance between the slot exit and the wire and the heat transfer regime, in particular the sub-cooling degree. The choice of the nickel is due to the opportunity to operate in a wide range of the jet exit velocity, thanks to its mechanical and thermometric characteristics.

The main aim of this research was to find out the optimal conditions, in order to maximize the heat transfer coefficient enhancement, and moreover, to make a comparison between other results obtained in the 1990s by the same authors on horizontal circular cylinders, with the diameter one size order bigger than the present wire.

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INTRODUCTION

The problem of the heat transfer from wires, rods and tubes to a liquid in cross flow has been widely investigated both in single–phase and boiling conditions since the 1960s. Vliet and Leppert [1-2] investigated the velocity effect on the critical heat flux for water flowing around a cylinder under saturated and sub-cooled conditions; Fand et al. [3] investigated simultaneous boiling and forced convection heat flux from a cylinder to water. Yilmaz and Westwater [4] determined the complete boiling curves including nucleate boiling, critical heat flux, transition, and film boiling from saturated R113 flowing around a horizontal circular cylinder, internally heated by steam flow. They found that the boiling curves shifted up with increasing velocity and the maximum heat flux markedly enhanced as the fluid velocity was increased. Sankaran and Witte [5] studied the influence of flow velocity and liquid sub-cooling on heat flux from a horizontal cylinder to boiling water and Freon-113. They found large azimuthal temperature differences especially in the film boiling conditions. Liu et al. [6] investigated the pressure effects in a similar configuration. Huang and Witte [7] experimentally studied the forced convective heat transfer from horizontal cylinders to highly sub-cooled Freon-113. These researchers found that the fluid velocity enhances the sub-cooled boiling heat flux; however the enhancement is more noticeable in the film region than in the nucleate region. About the azimuthal wall temperature they confirmed the above-mentioned very large differences for the film boiling regime. In the 1990s Bartoli et al. [8-9] investigated experimentally the heat flux from a cylinder impinged by a submerged slot jet of distilled water in incipient and nucleate boiling conditions. The diameter of the cylinder and the width of the jet are the same, namely 3 mm.. The heat flux ranged from $4 \cdot 10^4$ to $1.5 \cdot 10^6$ W/m^2. The effects of the jet velocity and of the fluid sub-cooling degree were investigated, in the range of $0.37 \leq v \leq 1.11$ m/s and $0 \leq \Delta T_{SUB} \leq 15$°C, both in upward and downward flow. Their results showed that the influence of the above parameters is limited to the region of incipient boiling, while in fully developed boiling all the boiling curves merge into a single one, whose slope is proportional to $\Delta T_{SUB}{}^{4.1}$. Moreover, the photographic study conducted in [9] evidenced a different behaviour with respect to cylinders in uniform cross flow. Successively, Bartoli et al. reported the results of an experimental investigation concerning the heat transfer from three cylindrical heaters to a water jet in the form of correlating equations with dimensionless numbers [10]. In this paper the influence on the heat transfer of the free convection, of the geometrical ratios characterising the configuration and of the impingement direction was also examined; the investigation was completed by some visualization experiments to qualitatively clarify some aspects of the interaction between the dynamic and the thermal fields. Moreover, Bartoli and Faggiani [11] also determined the heat transfer coefficient and the temperature distribution and their dependence on the cylinder angular coordinate. In 2000, C.

Bartoli et al. studied the heat transfer from an array of circular cylinders to an impinging slot jet of water both for single – phase and two-phase conditions [12]. Different geometrical configurations were investigated in order to find which maximizes the heat transfer coefficient and the dependence on such parameters as jet velocity and sub-cooling degree was also examined. Recently Mann and Stephan [13] proposed a model which allowed them to determine the heat transfer coefficient by combination of heat transfer to bubbles adhering at the wall and heat transfer due to convection; this study refers both to horizontal cylinders and flat plates. Except the study of Bartoli et al. the above investigations were performed for channel flow. The cooling with jets in boiling conditions was studied by Monde and Katto [14]: they carried on an experimental study on the fully developed nucleate boiling at atmospheric pressure in a simple forced- convection boiling system, which consists of a heated flat surface and a small, high speed jet of water of R -113 impinging on the heated surface. A generalized correlation for burnout heat flux data for water and for R-113 is successfully evolved, and it was shown that surface tension has been an important role for the onset of burnout phenomenon, not only in the ordinary pool boiling, but also in the present boiling system with a forced flow. Similar studies were also conducted by Katto and Ishii [15], and Lienhard and Eichhorn [16]; however these studies had concerned round jets impinging on disks or plane jets on square plates.

The cooling technique by impinging slot jet of water is currently largely employed in electronics to increase the heat transfer coefficient (because of the big number of circuits per chip and the big power per circuit imposed) and in metallurgical processes [17-18]. In particular, in [17], Sparrow et al. measured the heat transfer coefficients on a circular cylinder subjected to the cross flow impingement of a slot jet. In one set of experiments, the symmetry plane of the jet was aligned with the axis of the cylinder, while in other experiments the jet was offset from the cylinder. In addition to the offset, parametric variations were also made for the width of the jet-inducing slot, the distance between the slot and the cylinder, and the Reynolds number. Supplementary flow visualization experiments showed that even in the presence of offset, the jet impinged on the cylinder, although not at the cylinder apex as in the aligned case. It was found that the heat transfer coefficient increased with the slot width and Reynolds number, but decreased with slot-to-cylinder separation distance and offset. The effect of offset is accentuated for narrow slots and at small slot-to- cylinder separation distances. The largest measured offset-related reduction in the heat transfer coefficient was slightly in excess of 50%.

In the present paper the heat transfer rate from a 0.25 mm. in diameter nickel wire, heated by Joule effect, to an impinging upward flow submerged jet of distilled water, in sub-cooled and saturated boiling conditions was investigated. In [19] are reported some preliminary experimental results, referred to a 0.2 mm. in diameter platinum wire, in order to clarify the influence of different parameters, such as the jet exit velocity and the sub-cooling degrees, on the heat transfer coefficient. The mechanical characteristics of the platinum didn't allow the authors to reach the jet velocity tested with the cylinders; for this reason the data reported here are referred to a nickel wire, 0.25 mm. in diameter. At the same time, the nickel, as well as platinum, has linear thermometric properties, even if in a more narrow thermal range, as shown in Figure 1. The aim of this paper was to find out the optimal conditions in terms of the heat transfer coefficient enhancement and so to optimize the jet exit velocity, the distance from the slot exit and the wire and the heat transfer regime. At the end, a comparison between these data and the ones referred to the cylinders (with diameter one size order bigger than the actual wire) was done.

EXPERIMENTAL SET-UP

The experimental apparatus mainly consists of two different parts: the hydraulic and the electric circuit. The hydraulic one is schematized in Figure 2. The distilled water, used as working fluid, is stored in a $3 \cdot 10^{-2}$ m^3 tank, component S_e, where it is heated and then kept at the set temperature by means by R_f (1200 W) and R_t (800W) respectively. The component R, in which water tap flows, represents the cooling circuit, that is used also to assure the constant temperature of the water. A centrifugal pump (P), in maximum flow rate 1.34 kg/s, drives the liquid through a needle valve (V) and the turbine (T). The first one is used to regulate the flow, the second one to measure the flow itself. After a filter (F), the water reaches the test section (S) passing through a slot of uniform cross section (100 x 1mm^2). The slot length, 220 mm., is such to ensure fully developed flow at the slot exit, even in laminar conditions. At the exit of the slot the water impinges directly onto the heater. This is a 0.25 mm. in diameter nickel wire, whose physical characteristics are summarized in Table 1.

Table 1

Physical characteristics of the nickel wire

Properties	**Value**
Diameter D_w	0,25 mm
Electric resistivity at 20°C (ρ_{20})	$6{,}97 \cdot 10^{-4}$ Ω·m
Density	$8{,}9 \cdot 10^{-3}$ kg/m^3
Boiling point	2730°C
Melting point	1455°C
Thermal conductivity (k_f)	90,7 W/mK

The major axes of the heater and of the slot are coplanar and parallel. Finally, the spent fluid comes out of the test section through a circular opening in the top lid above and is conveyed into the storage tank S_e. The storage tank (Figure 3) is a stainless steel circular cylinder 0.5 m. in diameter; on its lid are located the preheating resistance (B) and the cooling coil (C), whereas the thermostatic one (D) is on the lateral wall. The thermocouple (E) allows to measure the water temperature inside the tank and is connected to R_t by means of a digital display. No one watertight lock was built for the lid, because the tests are carried on at atmospheric pressure. On the external of the tank there is a plexiglass tube (A), in order to check the internal water level. The test section consists of stainless steel cylindrical vessel, (0.26 m. O.D. and 0.5 m. height) and is equipped with four glass windows, parallel by twos, to visualize the phenomenon. The slot is mounted on the bottom lid, whereas on the top lid are present the electrical connections between the heater and the DC power supply. Actually, the heater, namely the nickel wire, is warmed up by Joule effect thanks to a DC power supply of maximum current equal to 30A. Two copper wires are soldered on the wire at the distance of 100 mm (the same value of the slot width), which corresponds to the test section length, in order to measure the voltage drop across the wire. The current is measured starting from the measurement of the voltage drop across a calibrated resistance of 10^{-2} Ω. All electric connections are linked to a digital multimeter and then to a PC, as well as the thermocouple immersed in test section close to the slot exit, in order to measure the water temperature.

The wire temperature is measured thanks to the thermometric properties of the nickel; this procedure will be widely discussed in the next section. As regards the placement of the wire inside the test section, had to be modified the previous support structure used for cylinders 3 mm. in diameter.

Figures 4 (a) and (b) show the 3D design of the proper mechanical bridge, which assures the 3D control of the wire placement, the control of the wire's strain level, mechanical resistance and thermal conductivity at high temperatures and does not give problems of oxidation.

EXPERIMENTAL PROCEDURE

As regards the measurement of the wire mentioned before it was impossible to do it by means of a sliding thermocouple, as well as in the case of hollow cylinders. The present methodology is based on the thermometric characteristics of the nickel: its resistivity, and so on its electric resistance, is directly proportional to its temperature, in the range from -100 to +200°C (Figure 1). Therefore, it was possible to derive the wire temperature, by means of direct measurement of its electric resistance with 4W system. This method uses a current level low enough to ensure that there was no additional heating of the wire above the bath temperature during calibration.

The data are reported for increasing and decreasing range of temperature, without supplying electrical power to the heater, which is considered in thermal equilibrium with the working fluid. The two curves are then merge into a single one, after that the data are averaged out.

The graph in Figure 5 shows the two curves obtained for increasing and for decreasing temperature. The electric resistance is obtained by means of Ohm law (varying the current I during the tests, at constant values of voltage drop, V), and then, by means of least squares method, it is possible to obtain the wire temperature.

The heat flux is obtained thanks to Equation 1:

$$q'' = \frac{V \cdot I}{\pi \cdot D_w \cdot L} \quad (1)$$

By means of the Newton's Equation (2) it was possible obtain the heat transfer coefficient, h:

$$h = \frac{q''}{T_w - T_{H2O}} \quad (2)$$

Fundamental prerequisite of the thermometric materials is maintaining constant characteristics during the time, in order to assure the repeatability of the results. The general law which describes a thermometric material is Equation 3:

$$R = R_0\,(1+\alpha T) \quad (3)$$

where R is the electric resistance at the temperature T, R_0 is the resistance at T_0=0°C, α is called the temperature coefficient. In particular about nickel, the literature scientific, based on experimental tests, supplies a proper correlation between R and T, in the narrow range from -60 to 180°C, used in this experimental investigation, as seen in Equation 5:

$$R = R_0\,(1+AT+BT^2+CT^4) \quad (4)$$

where $A = 5.845\cdot10^{-3}$, $B = 6.650\cdot10^{-6}$ and $C = 2.805\cdot10^{-11}$.

Table 2 summarizes the comparison between the theoretical values of R obtained by means of equation 4 and the experimental values. The maximum error is about 3%, and so hugely satisfactory.

Table 2

Comparison between the experimental and the theoretical value of the nickel wire resistance, at different temperature, and the percentage error between them.

T[°C]	*R[Ω]* *Theoretical value*	*R[Ω]* *Experimental value*	***Error [%]***
50	0.1494	0.1444	3.36
54	0.1524	0.1476	3.16

58	0.1554	0.1508	2.98
62	0.1585	0.1543	2.63
66	0.1615	0.1578	2.32
70	0.1646	0.1611	2.15
74	0.1678	0.1646	1.89
78	0.1709	0.1682	1.59
82	0.1741	0.1717	1.38
86	0.1773	0.1752	1.19
90	0.1805	0.1788	0.97

Uncertainty analysis had been carried out according to [20] and is summarized in Table 3:

Table 3

Uncertainty analysis

Quantity	**Reference Value**	**Overall Uncertainty**
Water bulk temperature (T_{H2O})	95°C	±0.1 °C
Current	20A	0.3 %
Voltage	10 V	0.2%
Length measurements		<0.01 mm
Wire diameter	$0.25 \cdot 10^{-3}$ m	4%
Heater area	$7.85 \cdot 10^{-5}$ m^2	≈4%
Flow rate	1 kg/s	3%
q”	$2.55 \cdot 10^{6}$ W/m^2	4.6%
T_w	117°C	13%
T_w-T_{H2O}	17°C	13.01%
h	$3 \cdot 10^{5}$	21.7%

EXPERIMENTAL RESULTS

The tests are carried out to investigate the influence on the heat transfer of the following parameters, varying only one of them at a time, and fixing the other twos:

- the heat transfer conditions, expressed by the water temperature;
- the wire placement with respect to the slot exit;
- the jet exit velocity.

About the heat transfer conditions, we operate in sub-cooled boiling conditions, at the sub-cooling degrees, ΔT_{sub}, of 10 and 5 °C, corresponding to the water temperature equal to 90 and 95 °C, and in saturated boiling conditions, at atmospheric pressure. In order to investigate the influence of the wire placement we varied the dimensionless parameter Z/D, where Z represents the distance between the wire and the slot exit and D the slot width. This parameter was varied from 3, 5 and 8: the value of 5 corresponds to the wire placement on the apex of the potential core of the jet, whereas 3 corresponds to the wire completely inside to the potential core and 8 completely outside. With regards to the jet exit velocity, we tested the value of 0.2, 0.37, 0.6 and 0.74 m/s. These values were chosen in order to compare the results with the ones obtained with cylinder of diameter one size order of magnitude bigger than the nickel wire. In each test the heat flux was varied and then the temperature difference between the wire and the water was measured, and consequently the heat transfer coefficient. In fact the first series of graph show the trend of q” versus T_w-T_{H20}, in log-log coordinates, the second series the trend of the coefficient h versus the temperature difference wire-water and the heat flux, in log-log coordinates. Generally, we opted for not showing all the graphs, for their large number, but just only the most significant for our aims. Firstly, we investigated the exit jet velocity influence. The graphs in Figures 6-8 show the trend of the heat flux versus the difference temperature between the wire and the water, at the water temperature constant at 95°C, varying the jet exit velocity, for the three different wire placement. In all cases, the curves tend to merge into a single one: in fact the variations of the heat transfer coefficient due to the velocity fall within the experimental error. Obviously, the highest velocity, 0.74 m/s, gives the highest value of q” dissipated, even if, for energy saving reasons, is more opportune using the lowest velocity, namely 0.2 m/s, since the differences in terms of heat transfer enhancement are so small. In fact all the other graphs reported in the rest of the paper are referred to the minimum jet exit velocity. About the velocity, we have chosen to operate in the range mentioned before, because velocity lower than 0.2 m/s could not be measured accurately (it has done by means of the turbine flow meter), whereas values bigger than 0.74 m/s could have brought about the wire breakage too much frequently. Secondly, we focused on the influence of the heat transfer conditions, quantitatively expressed by the value of the water temperature with respect to the wire temperature: when the temperature of the wire was over 100°C and the water temperature was below its saturation temperature (at atmospheric pressure) we operated in sub-cooled boiling conditions, whereas when the water temperature reached the saturation value we operated in saturated boiling. The results are reported in the graphs in Figures 9-11, all referred to the minimum jet exit velocity (0.2 m/s) for the three different wire placements. Also this parameter does not seem much significant for the heat transfer. It is possible to see just a difference for the temperature of 95 °C (which corresponds to sub-cooling degree equal to 5°C), especially for Z/D = 3 and 8, whereas in the potential core the curves of the three temperatures are practically overlapped to each other. On the basis of the calculations reported in table 4 we have assumed the sub-cooling degree of 5°C as the optimal conditions, because the heat flux q" dissipated is the highest, T_w-T_{H20} being equal. Finally, the graphs in Figures 12-14 report the influence of the wire placement, at the velocity of 0.2 m/s for three water temperatures: increasing the temperature difference between the wire and the water the curves tend slightly to converge, especially in the case of T_{H20} equal to 90 and 100°C. However, in this case, the wire's placement on the apex of the potential core (Z/D=5) supplies the best results, from the point of view of the heat transfer, as the scientific literature referred to the jet theory reports. The second series of graphs report the trend of h versus the temperature difference wire-water and versus the heat flux. The graphs in the Figures 15 and 16 are referred to the value of Z/D =5 and T_{H20} =95°C, for different jet exit velocity: also the curves of the heat transfer coefficient tend to merge into a single one, varying the velocity. The graphs in Figures 17 and 18 confirm the importance of the wire's placement with respect to the slot exit for the heat transfer enhancement. In particular, it is noteworthy that increasing the temperature difference water-wire and the heat flux h remains practically constant when the wire is completely inside the jet potential core (Z/D=3) and tends to decrease when Z/D=5 and 8, especially on the apex of the potential core (the reasons of this have to be investigated later more deeply). However, in Figure 19 the trend of h versus the temperature difference wire-water shows a decrease, for each water temperature: T_w-T_{H20} increasing, q" being equal, h decreases, in agreement with Equation 2. The graphs in Figures 20-22 reports the trend q"-(T_w-T_{H20}), h- (T_w-T_{H20}), h-q" respectively in the optimal conditions, namely T_{H20}=95°C, Z/D=5 and v=0.2 m/s. Table 4 summarizes the influence of each parameter (T_{H20}, v and Z/D), by means of the calculation of the percentage enhancement of the average (calculated in the same range of q", in order to compare the values) and the maximum heat transfer coefficient, Δh_{av} and Δh_{max} for each test:

Table 4

Average and maximum heat transfer enhancement due to exit jet velocity, water temperature and wire placement.

	Δh_{av}	Δh_{max}
Enhancement from v=0.20m/s to v=0.74m/s	14%	21%
Enhancement from v=0.37m/s to v=0.74m/s	11%	16%
Enhancement from v=0.60m/s to v=0.74m/s	4%	8%
Enhancement from T_{H20}=90°C to T_{H20}=95°C	16%	21%
Enhancement from T_{H20}=100°C to T_{H20}=95°C	21%	28%
Enhancement from Z/D=3 to Z/D=5	36%	47%
Enhancement from Z/D=8 to Z/D=5	52%	64%

The values of h reported in Table 4 confirm again as the influence of the velocity is negligible and, whereas, as the wire placement, represented by the parameter Z/D, affects mostly the heat transfer enhancement, T_{H20} and velocity being equal.

Finally we have compared our results with the previous ones obtained by Bartoli et al. [8-11]. In particular, in [9], the test conditions were the same: the hollow cylinder of 3 mm. in diameter and 0.5 mm. in width was placed on the apex of the potential core (Z/D =5), inside or outside the potential core (Z/D =3 and 8 respectively). The velocity range was between 0.37 and 1.1 m/s. In these experimental tests the heat flux increased with the temperature increasing and decreased with the sub-cooling degree decreasing, in opposition to our present results. In Figure 23 we have compared our results with the ones referred to the cylinder, in our optimal conditions: Z/D=5, v =0.2 m/s and ΔT_{SUB}=5°C, whereas in Figure 24 the graph is referred to the saturated boiling regime, the other parameters being equal. We have opted for showing these results in function of the variable ΔT_{SAT}, which represents the superheating degree, namely the difference between the wire temperature and the water saturation temperature at atmospheric pressure. In both heat transfer conditions the values of the heat flux dissipated by the thin nickel wire are one order of magnitude bigger than the ones by the hollow cylinder. For example, in the sub-cooled case the heat flux reaches $2.8 \cdot 10^6$ W/m^2 in presence of the wire, and rounds on $2 \cdot 10^5$ and $5.2 \cdot 10^5$ W/m^2 with the cylinder. In saturated conditions the maximum value of q" is equal to $2.4 \cdot 10^6$ W/m^2. This phenomenon is probably the mostly significant, because it was demonstrated that the lowest scales could dissipated the highest values of heat flux: it should be applied in many engineering applications, such as micro heat exchangers. Successively, the investigation will be carried on in single phase conditions to clarify the phenomenon in each heat transfer conditions and, successively, we will operate with micro and nano wires.

CONCLUSIONS

This paper collects the experimental results referred to the heat transfer between a nickel wire and distilled water in sub-cooled and saturated boiling, in order to find the optimal conditions which maximize the heat flux dissipated from the heater. An upward submerged slot jet of distilled water impinges onto the heater (a nickel wire, 0.25 mm. in diameter), warmed up by Joule effect. The main aim of the experimental tests was to investigate the influence on the heat transfer of the jet exit velocity, the placement of the wire referred to the slot exit and the heat transfer conditions. The velocity was varied from 0.2 to 0.74 m/s, the placement of the heater, represented by the dimensionless parameter Z/D, was changed from 3,5 to 8. These positions corresponded to the wire located inside, on the apex and completely outside of the jet potential core. The heat transfer conditions were set in sub-cooled boiling conditions (water temperature set to 90 and 95°C) and in saturated boiling conditions, at atmospheric pressure. The most significant parameter, which affects mostly the heat transfer, was the heater's placement, whereas the temperature, and so on the heat transfer conditions, and also the velocity had a negligible influence, inside the range of the experimental error. The optimal conditions, in terms of the heat transfer coefficient enhancement, were the jet exit velocity equal to the minimum value (the curves at different velocity tend to merge into single one, so for energy saving reasons, we opted for the velocity of 0.2 m/s as the best one) , the wire located on the apex of the potential core and the water temperature equal to 95°C, in sub-cooled boiling conditions. To sum up, Z/D=5, v=0.20 m/s and T_{H20}=95°C. Finally, we compared our results with the previous ones obtained at University of Pisa by Bartoli *et al.*: the nickel wire 0.25 mm. in diameter allows to dissipate values of heat flux one order of magnitude bigger than the hollow cylinder 3 mm. in diameter used in the past. This confirms the good opportunities connected to the use of micro scales in the field of the cooling technologies.

NOMENCLATURE

Symbol	Description	Unit
D	Slot width	[m]
D_w	Wire diameter	[m]
h	Heat transfer coefficient	[W/°C m^2]
I	Current	[A]
L	Distance between the sensing wires	[m]
q”	Specific heat flux per unit area	[W/m^2]
T_{H20}	Jet temperature at outlet	[°C]
ΔT_{sat}	Superheating degree	[°C]
T_w	Wire temperature surface	[°C]
T_0	Wire temperature at 0°C	[°C]
ρ	Electric resistivity	[Ω·m]
R	Electric resistance	[Ω]
R_0	Electric resistance at 0°C	[Ω]
k_f	Thermal conductivity	[W/m°C]
ΔT_{sub}	Subcooling degree	[°C]
v	Exit jet velocity	[m/s]
V	Voltage drop	[V]
Z	Distance between the wire and the slot exit	[m]

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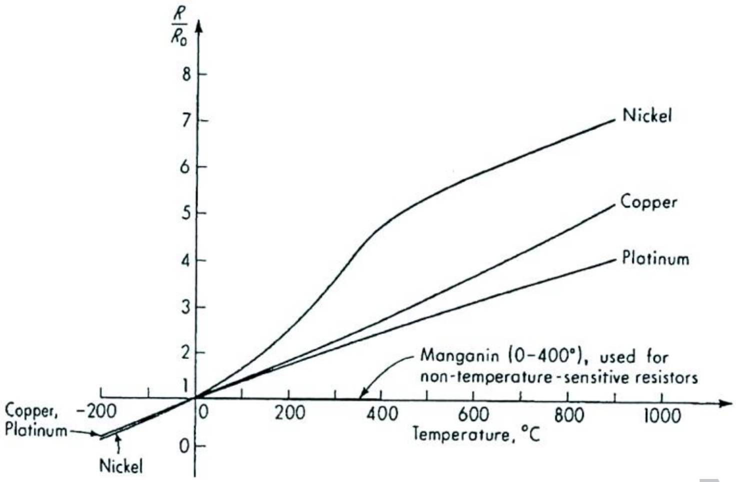


Fig.1: Thermometric curve for different materials.

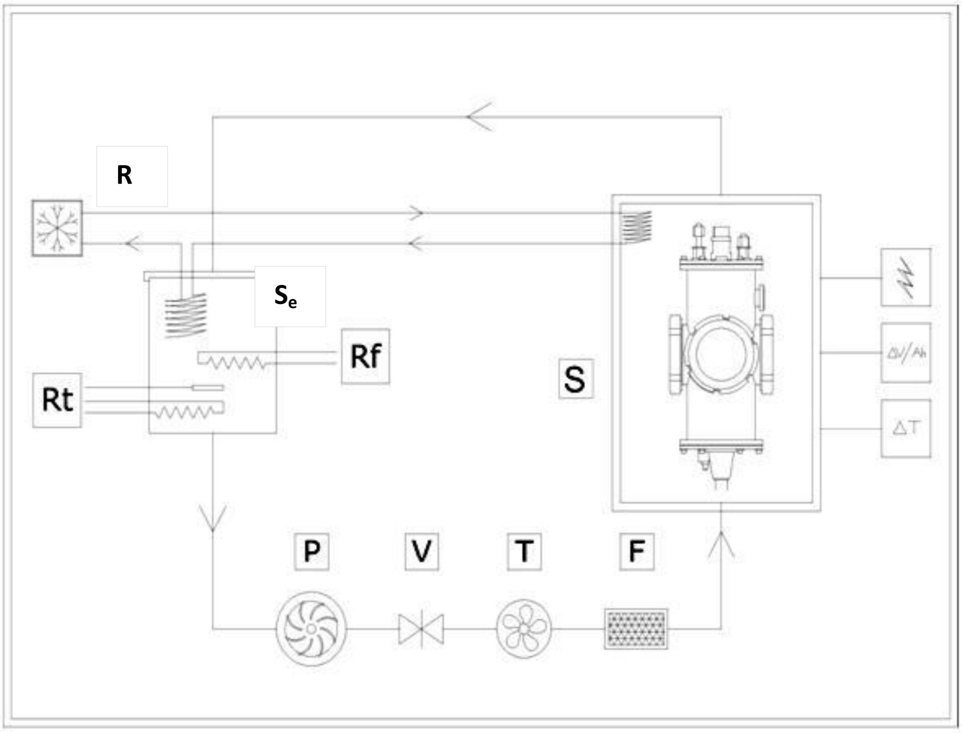
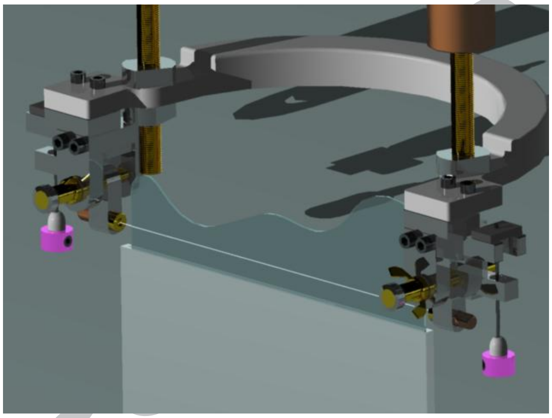
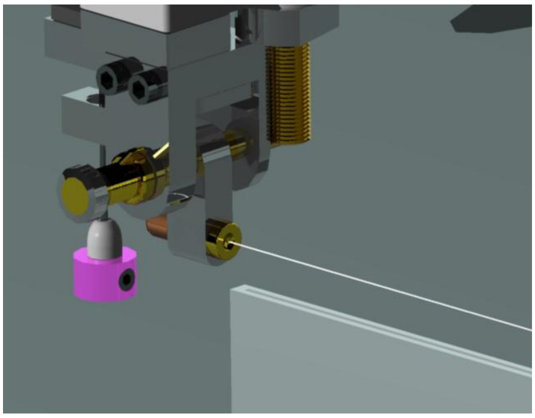


Figure 2: Plumbing circuit of the experimental apparatus, S_e storage tank, R_f preheating resistance, R_t thermostatic resistance, P centrifugal pump, V needle valve, T turbine flow meter, F filter, S test section.

Figure 3: Picture of the test section, with inside the slot and the hydraulic and electric connections on the upper lid.



(a)



(b)

Figure 4: (a) and (b) 3D design of the support structure built for the nickel wire.

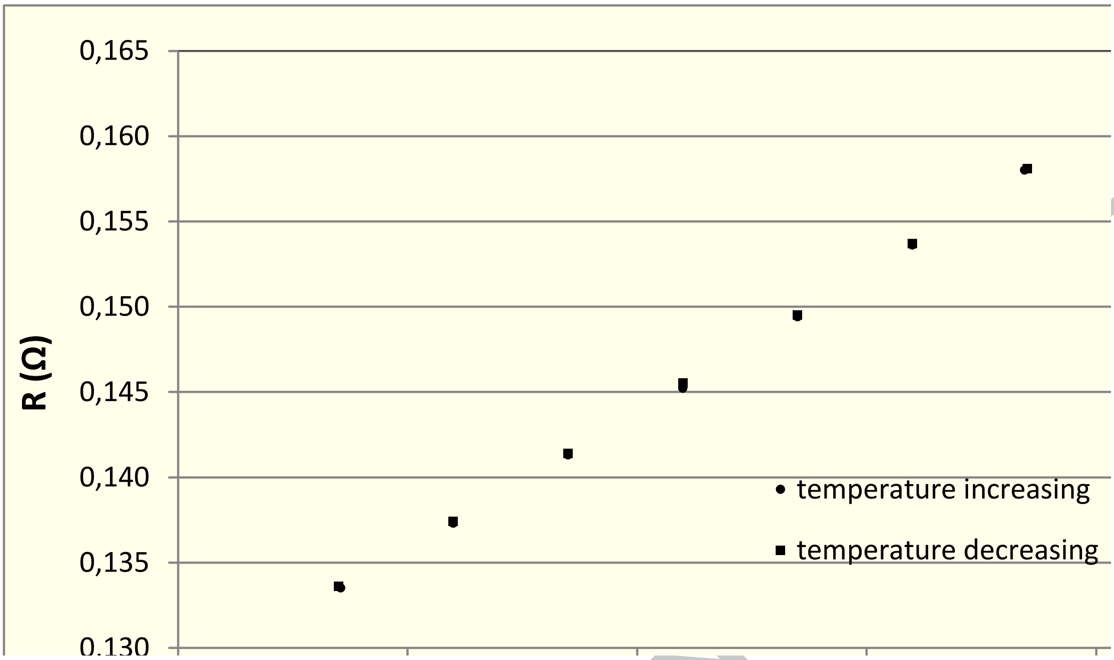


Fig5: Trend of the electric resistance versus temperature for the nickel wire.

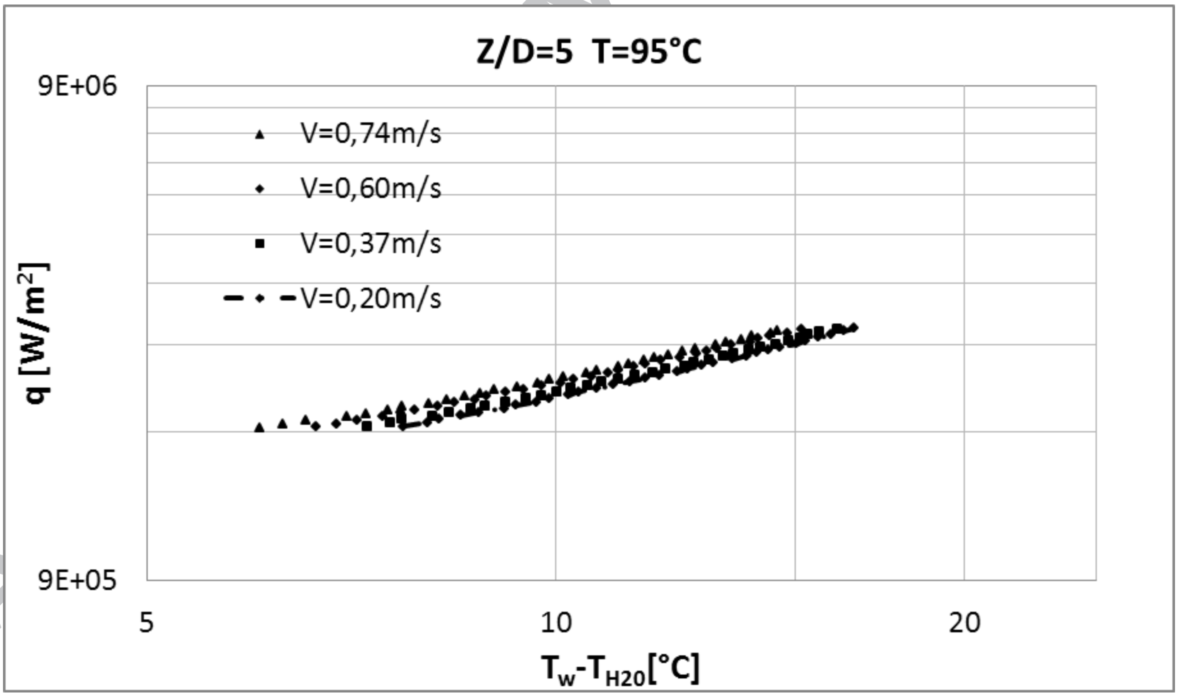


Fig.6: Trend of q'' versus T_w-T_{H20} at Z/D=5, T_{H20}=95 °C, at different exit jet velocity.

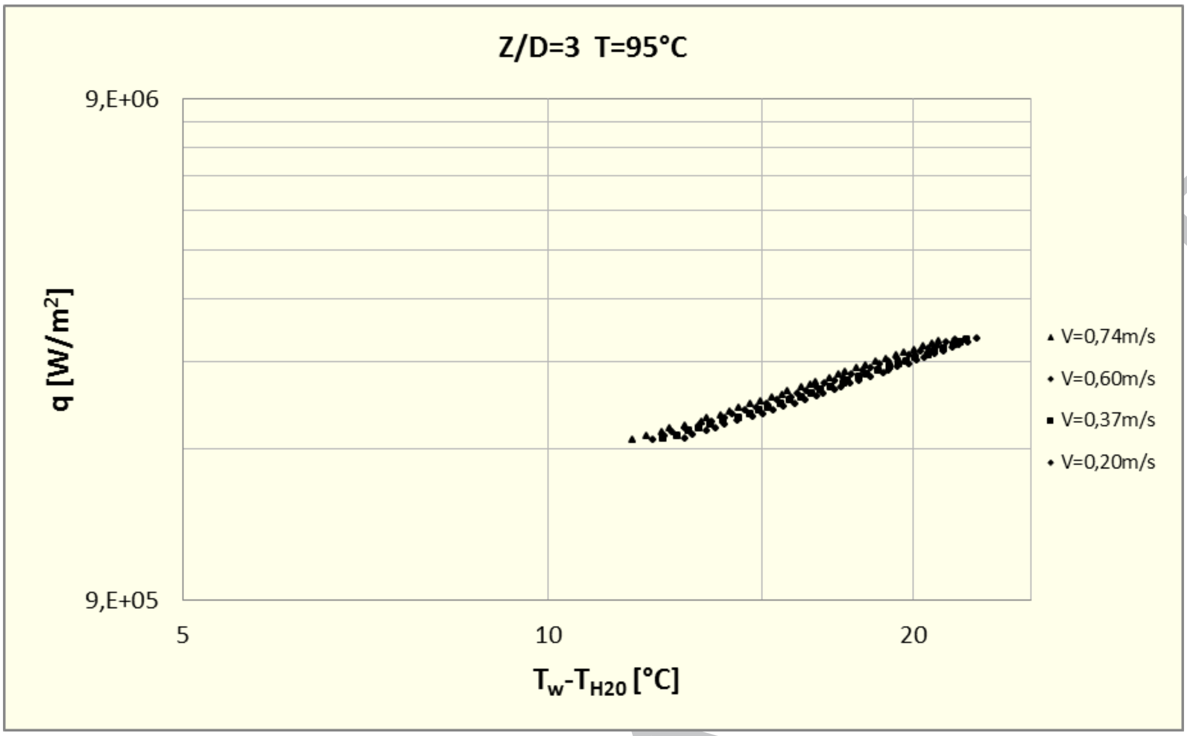


Fig.7: Trend of q'' versus T_w-T_{H20} at Z/D=3, T_{H20}=95°C, at different jet exit velocity.

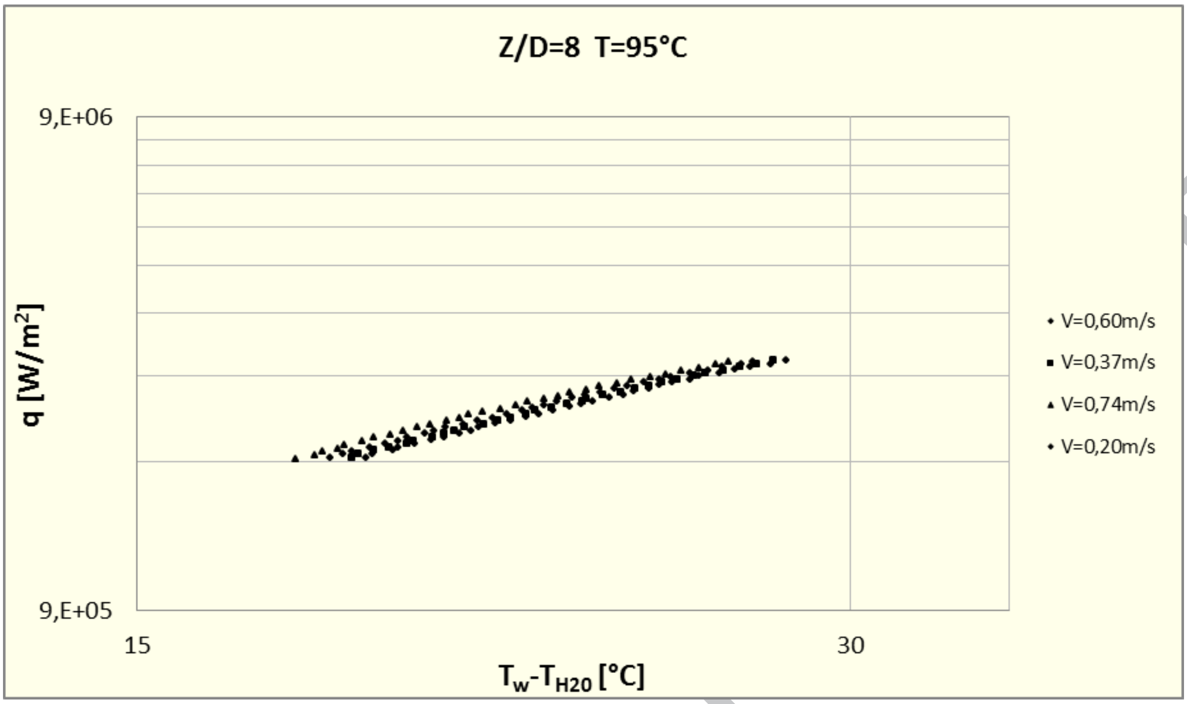


Fig.8: Trend of q'' versus T_w-T_{H20} at Z/D=8, T_{H20}=95°C, at different jet exit velocity.

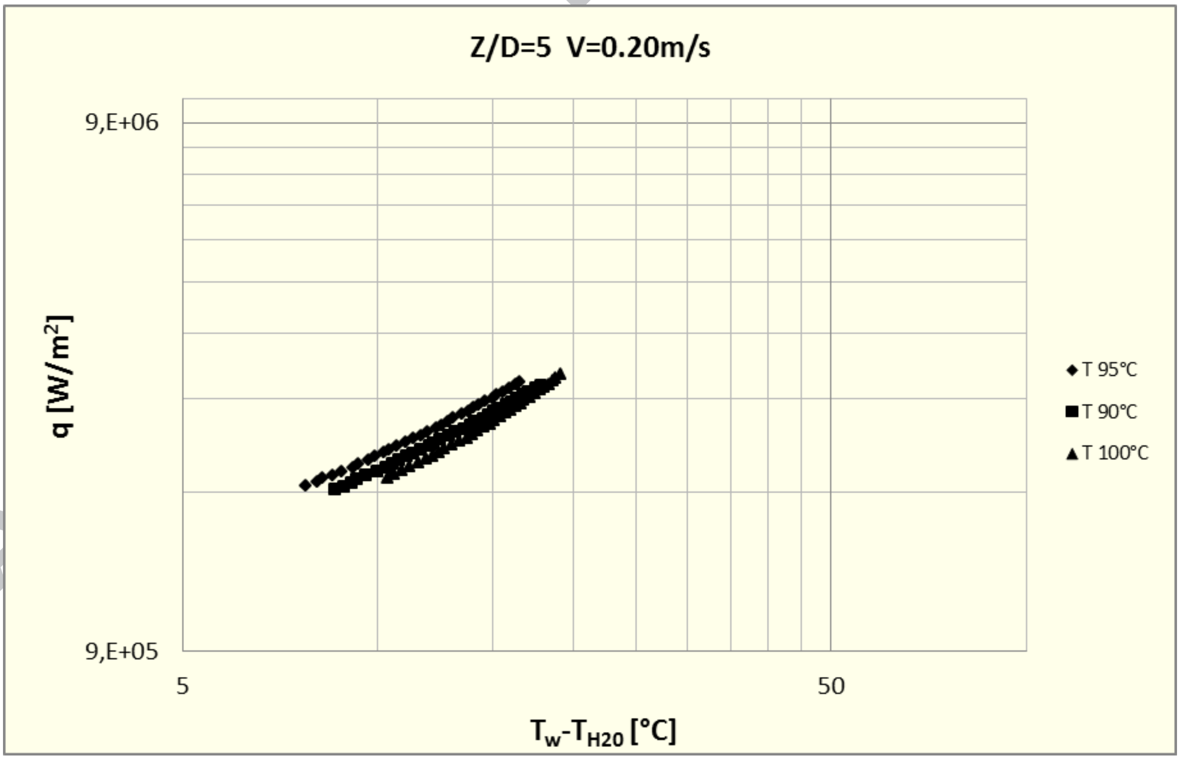


Fig.9: Trend of q'' versus T_w-T_{H20} at Z/D=5, v=0.20 m/s, at different water temperature.

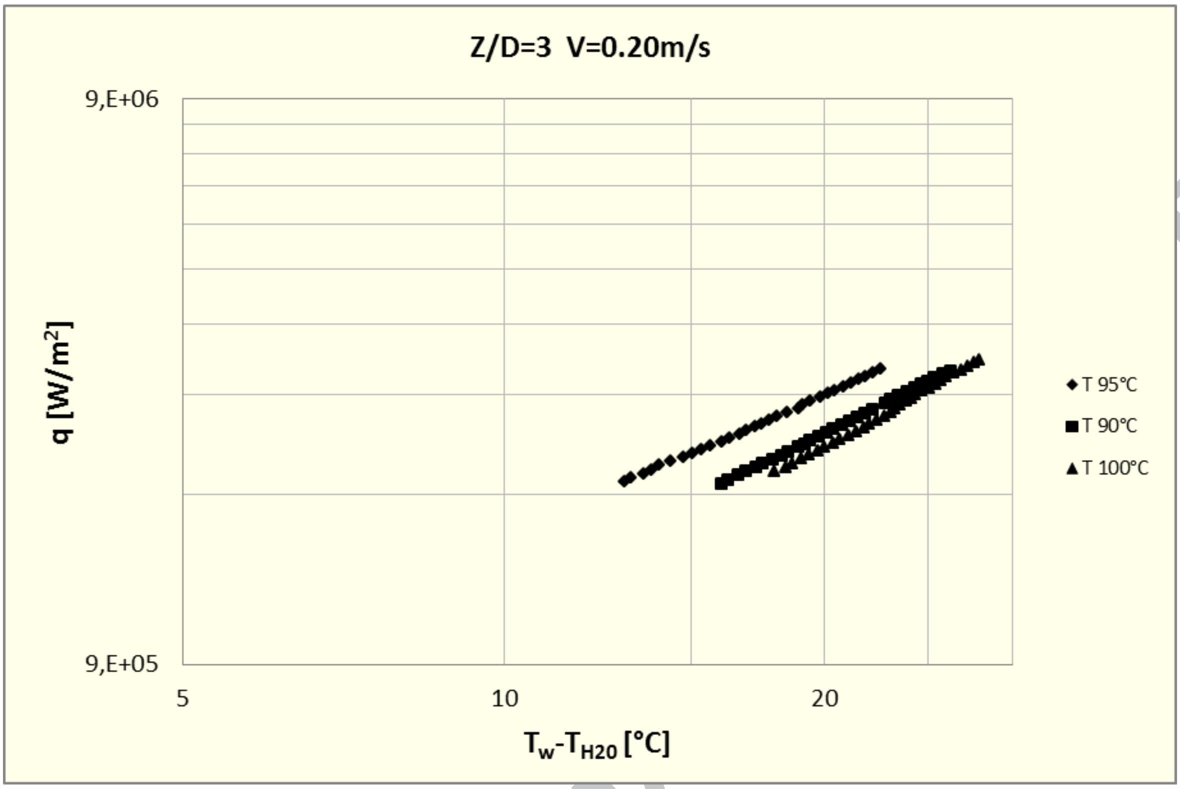


Fig.10: Trend of q'' versus T_w-T_{H20} at Z/D=3, v=0.20 m/s, at different water temperature

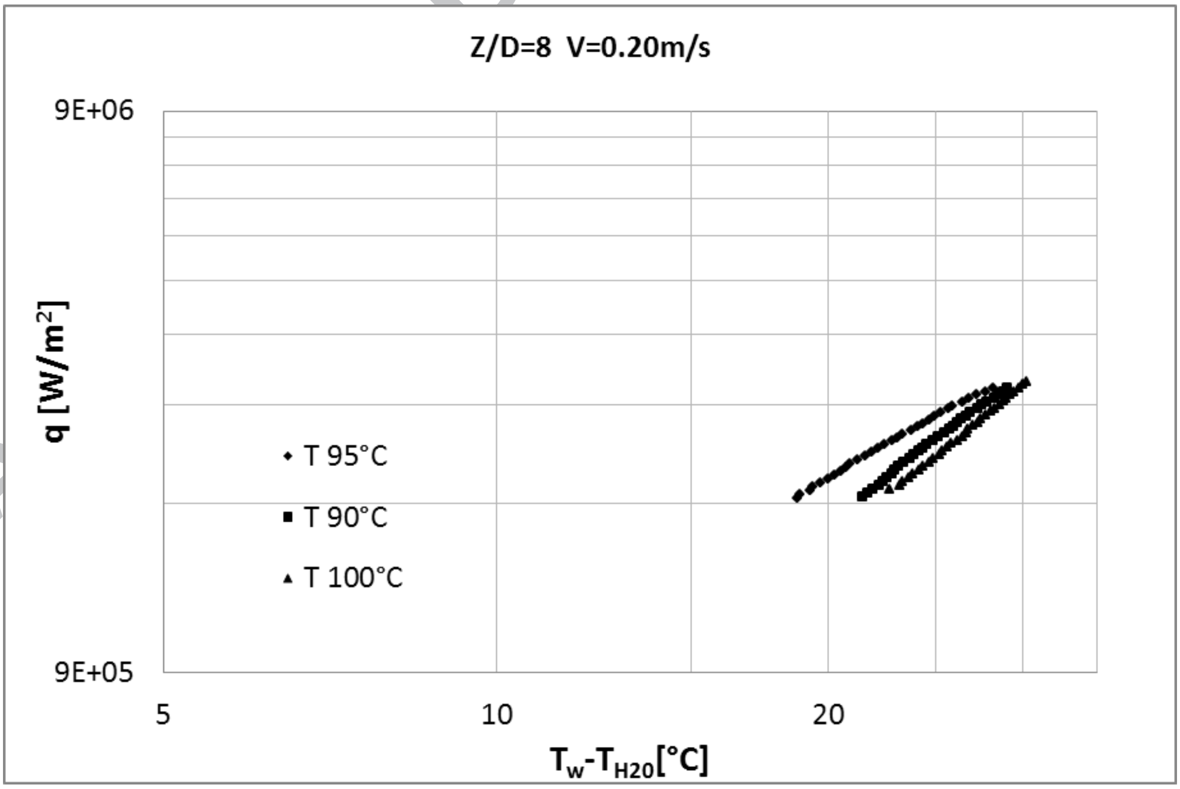


Fig.11: Trend of q'' versus T_w-T_{H20} at Z/D=8, v=0.20 m/s, at different water temperature.

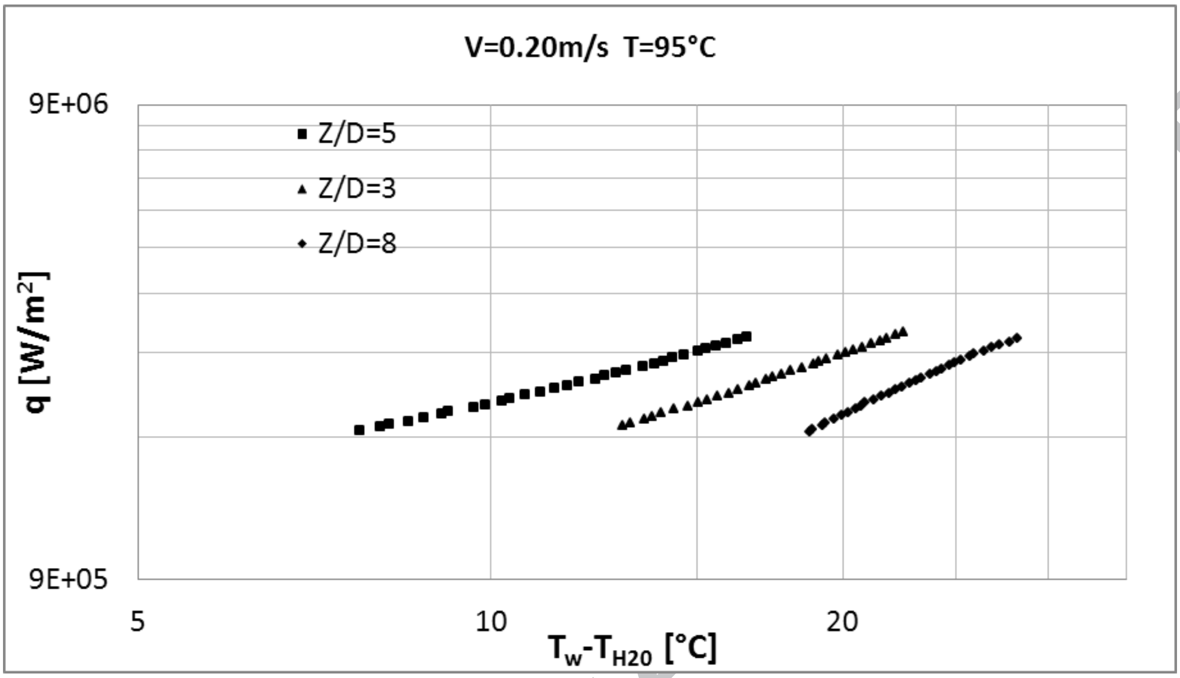


Fig.12: Trend of q'' versus T_w-T_{H20} at v=0.20 m/s, T_{H20}=95 °C, at different Z/D.

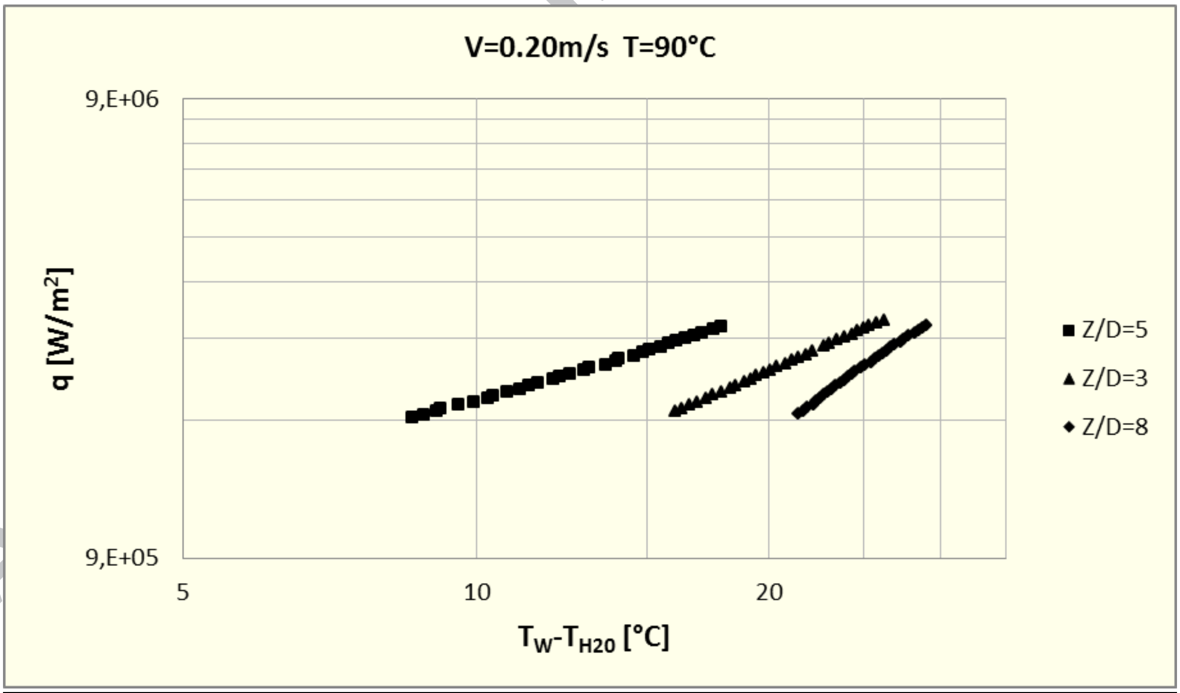


Fig.13: Trend of q'' versus T_w-T_{H20} at v=0.20 m/s, T_{H20}=90 °C, at different Z/D.

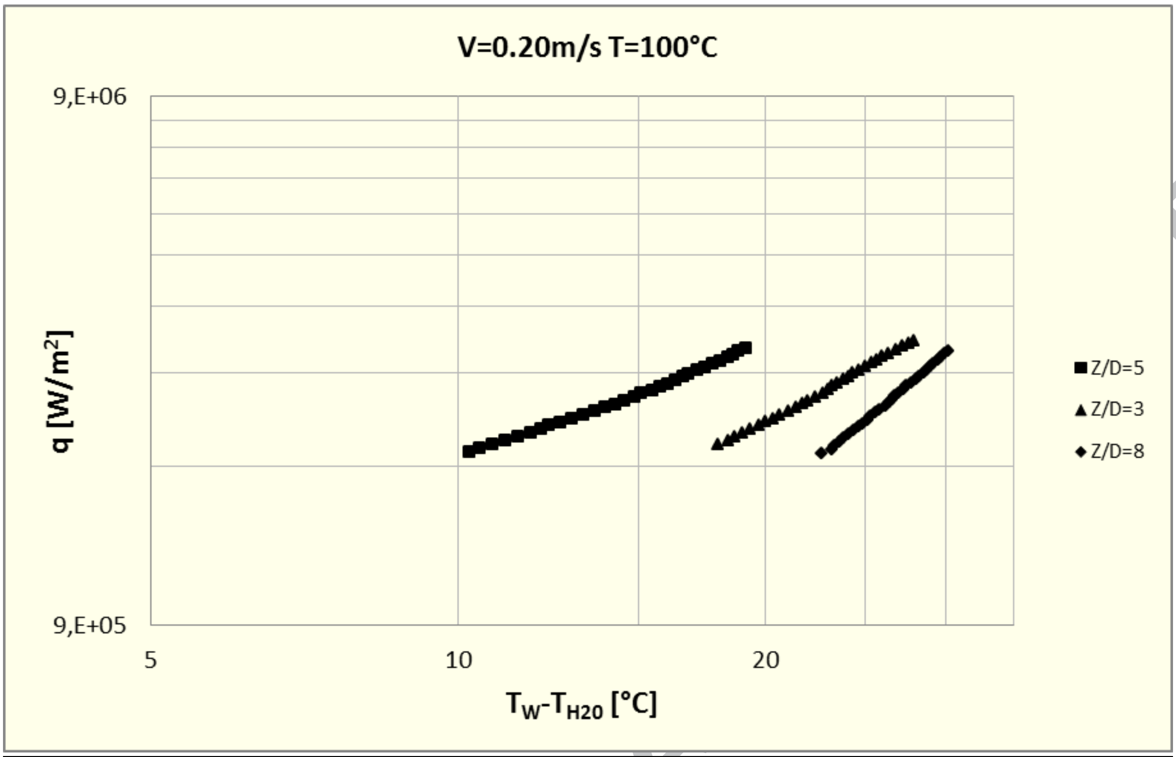


Fig.14: Trend of q'' versus T_w-T_{H20} at v=0.20 m/s, T_{H20}=100 °C, at different Z/D.

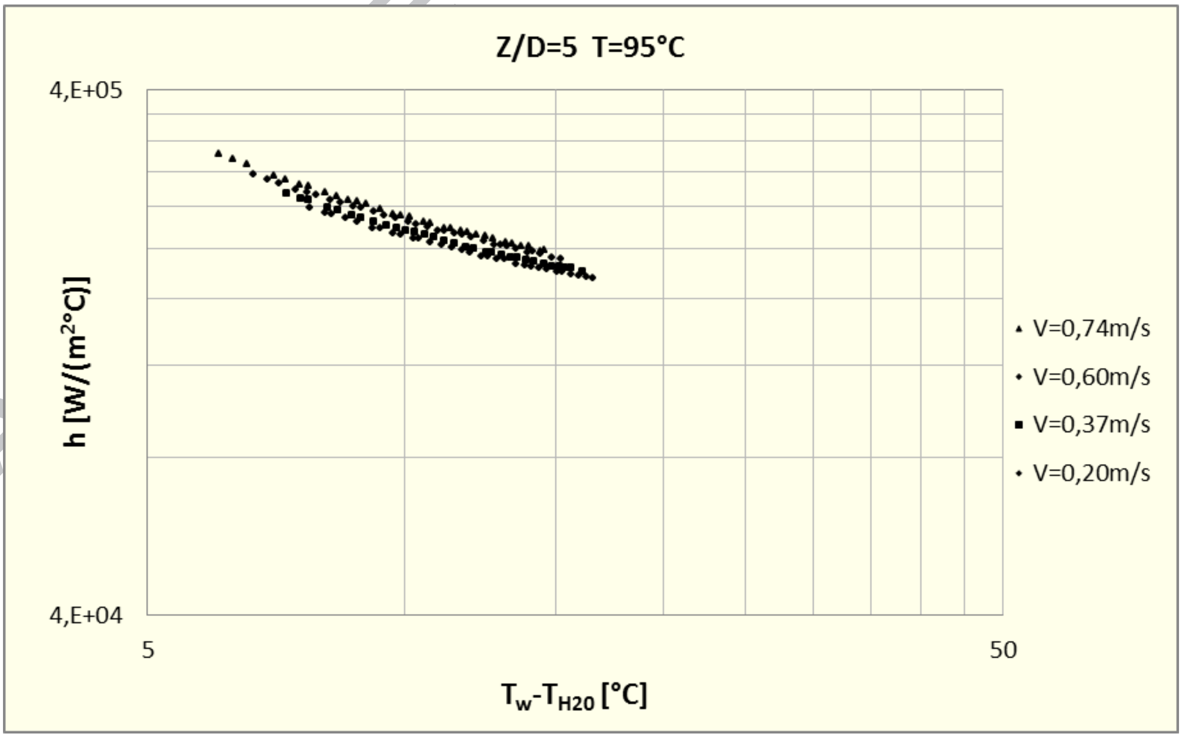


Fig.15: Trend of the heat transfer coefficient versus T_w-T_{H20} at Z/D=5, T_{H20}=95°C, at different exit jet velocity.

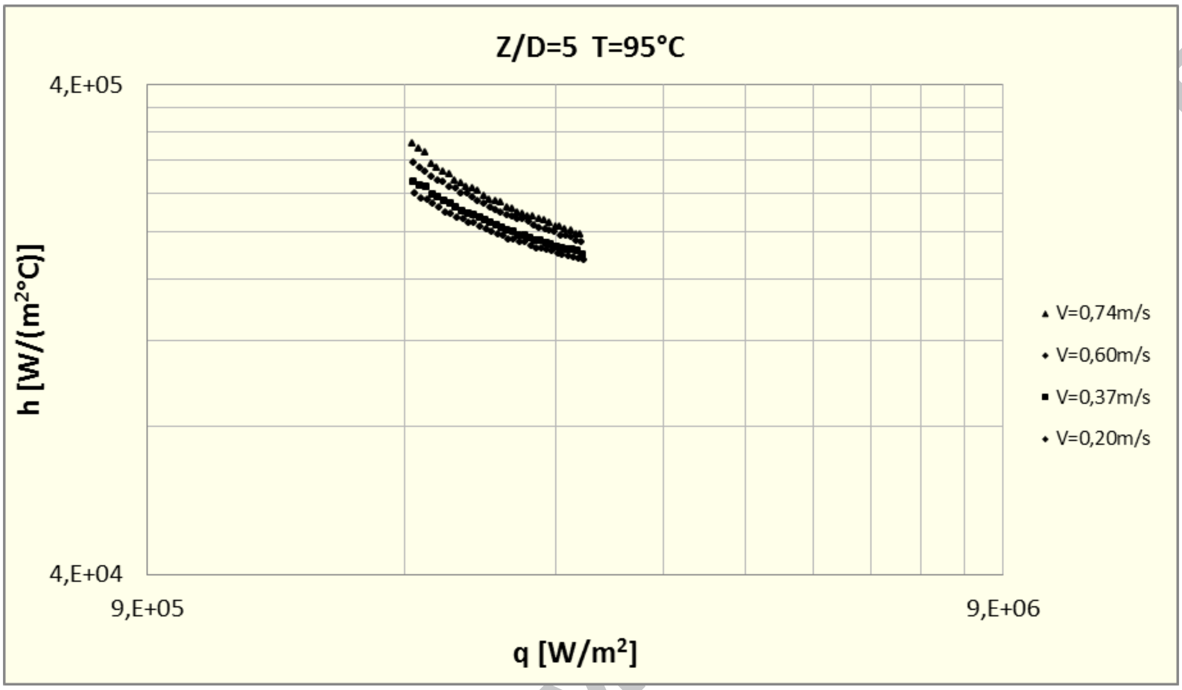


Fig.16: Trend of the heat transfer coefficient versus q'', at Z/D=5, T_{H20}=95°C, at different exit jet velocity.

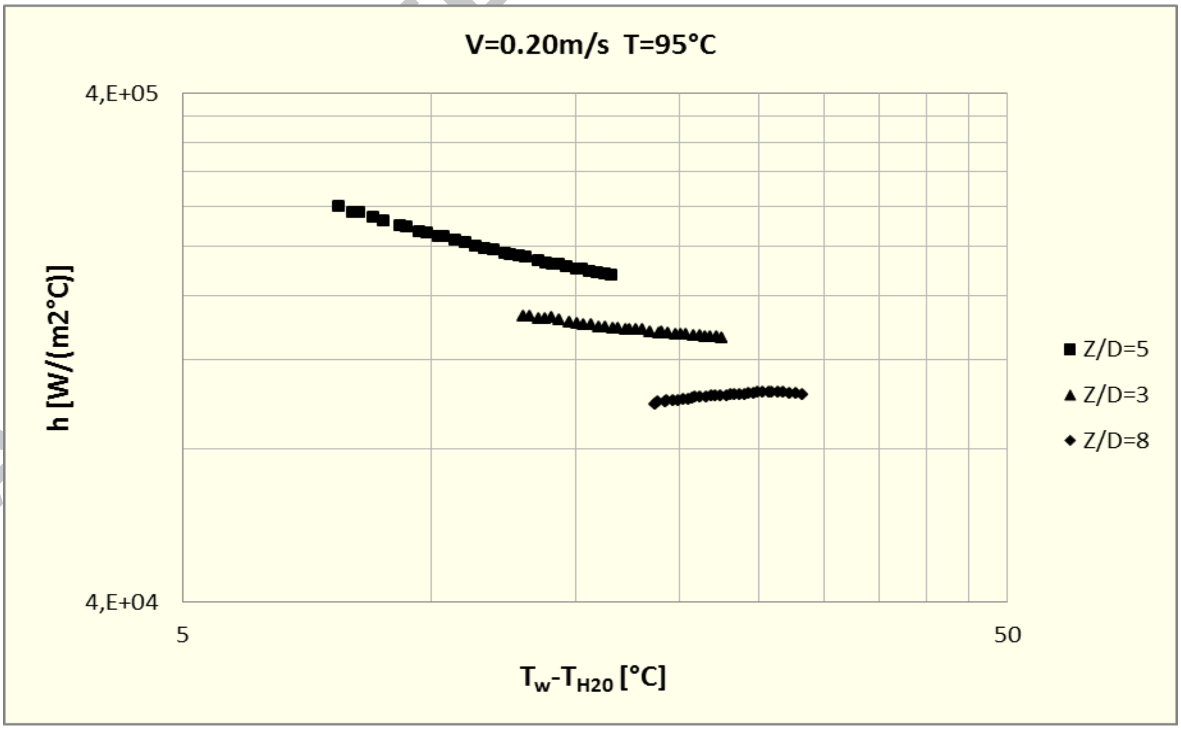


Fig.17: Trend of the heat transfer coefficient versus T_w-T_{H20}, at v=0.2 m/s, T_{H20}=95°C, at different Z/D.

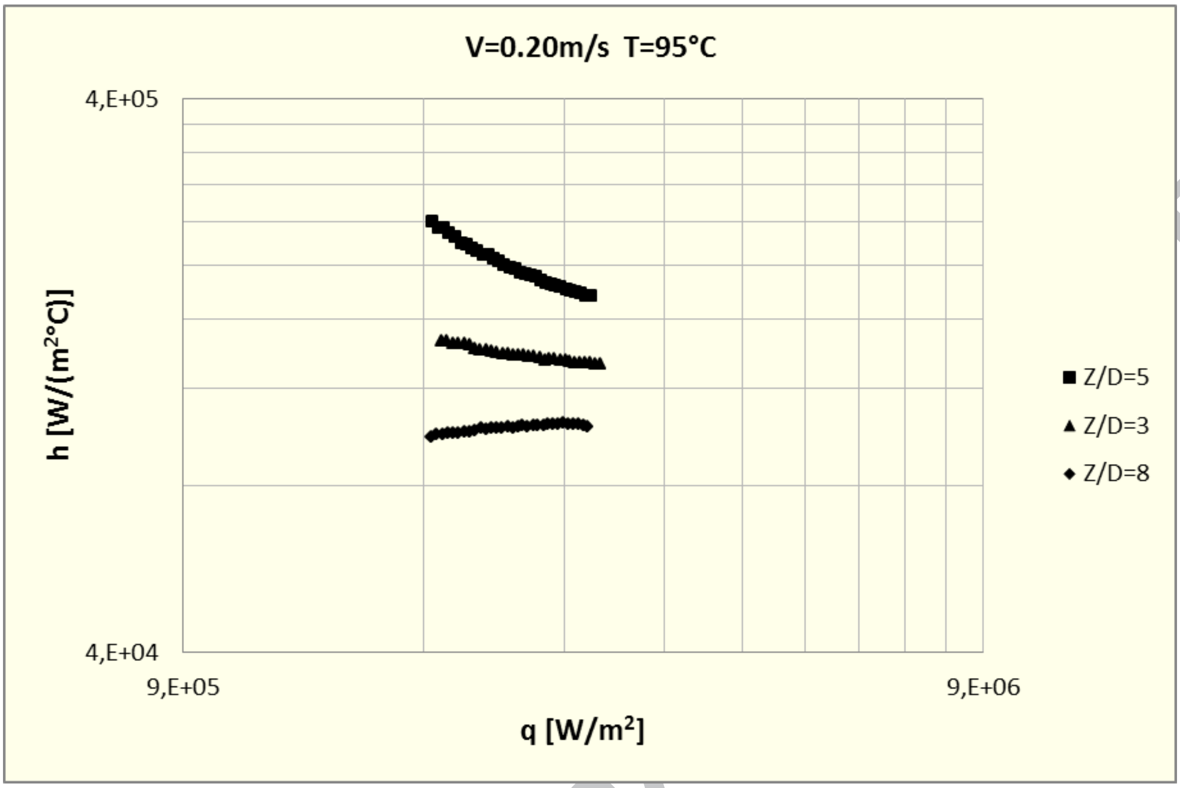


Fig.18: Trend of the heat transfer coefficient versus q'', at v=0.2 m/s, T_{H20}=95°C, at different Z/D.

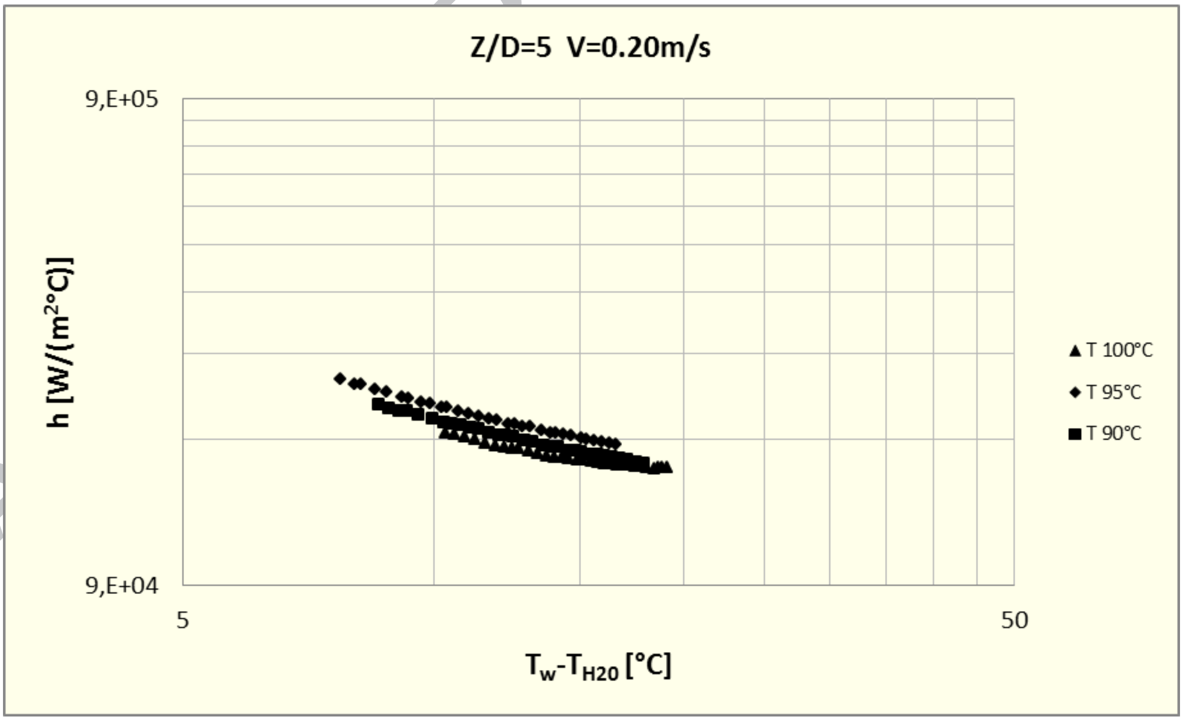


Fig.19: Trend of the heat transfer coefficient versus T_w-T_{H20} at Z/D=5, v=0.2 m/s, at different water temperature.

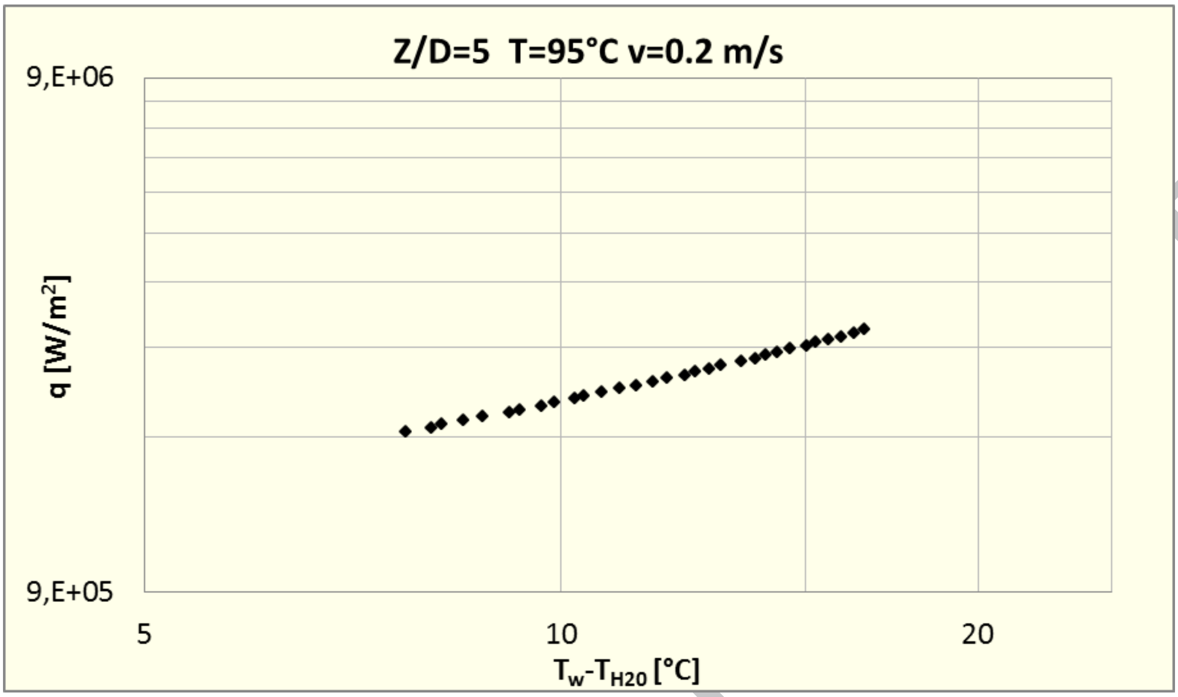


Fig.20: Trend of q'' versus T_w-T_{H20} at Z/D=5, v=0.2 m/s and T_{H20}=95°C (optimal conditions).

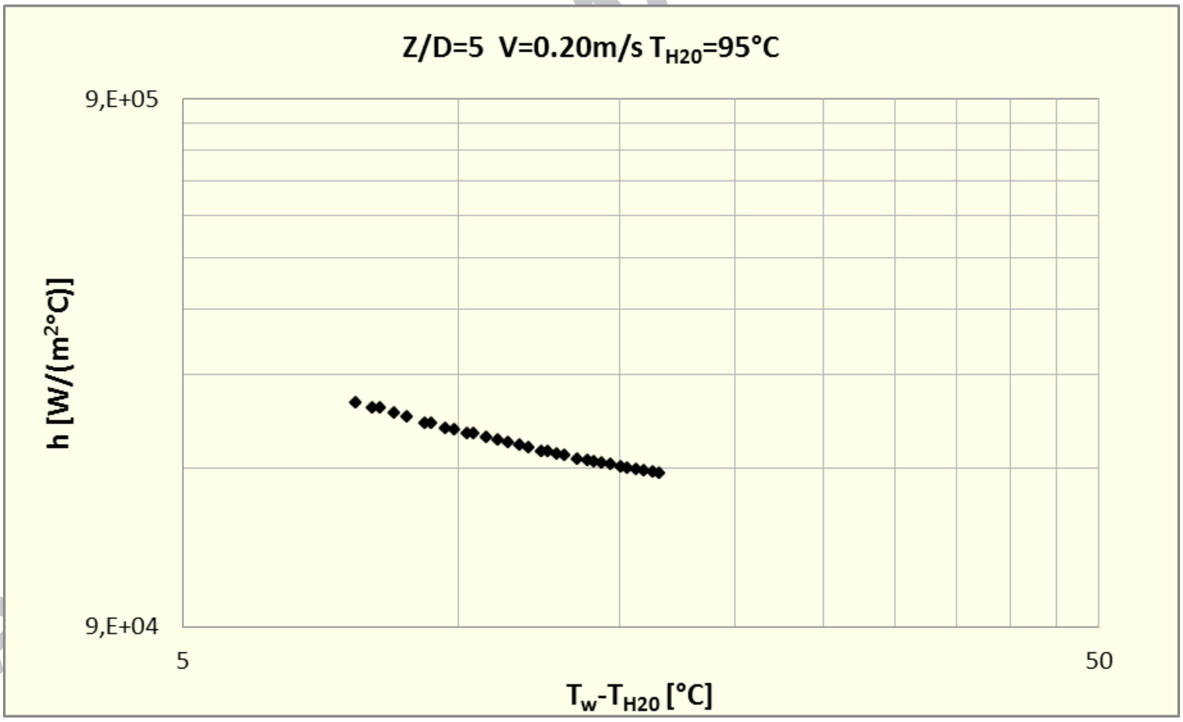


Fig.21: Trend of h versus T_w-T_{H20} at Z/D=5, v=0.2 m/s and T_{H20}=95°C (optimal conditions).

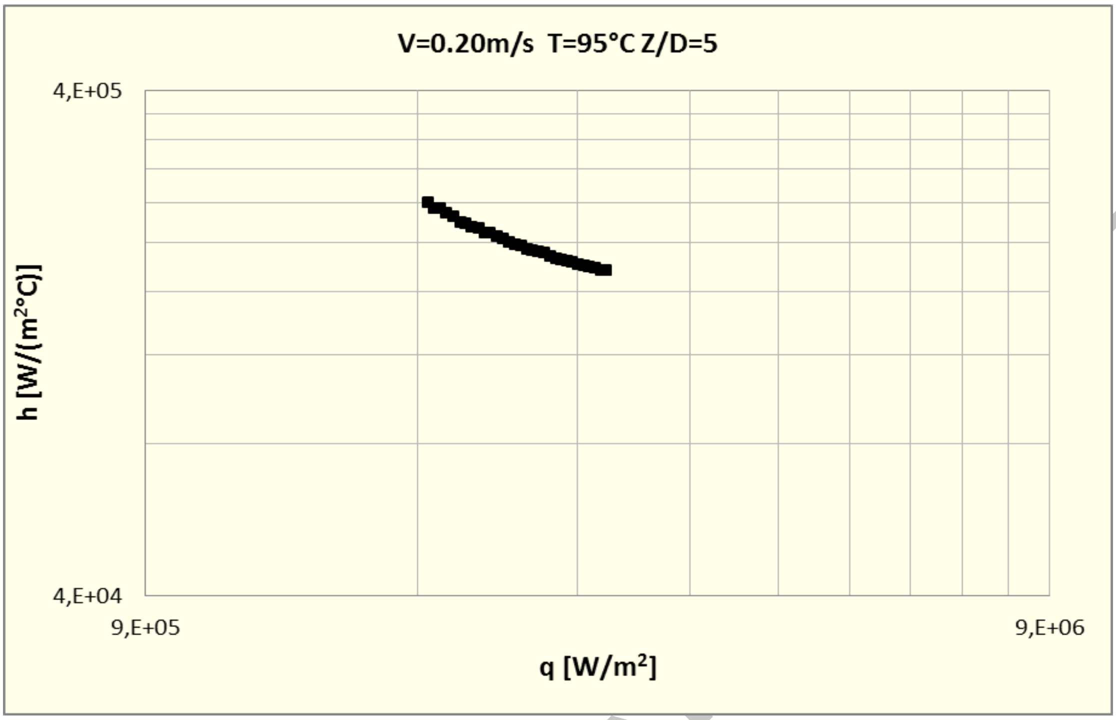


Fig.22: Trend of h versus q'' at Z/D=5, v=0.2 m/s and T_{H20}=95°C (optimal conditions).

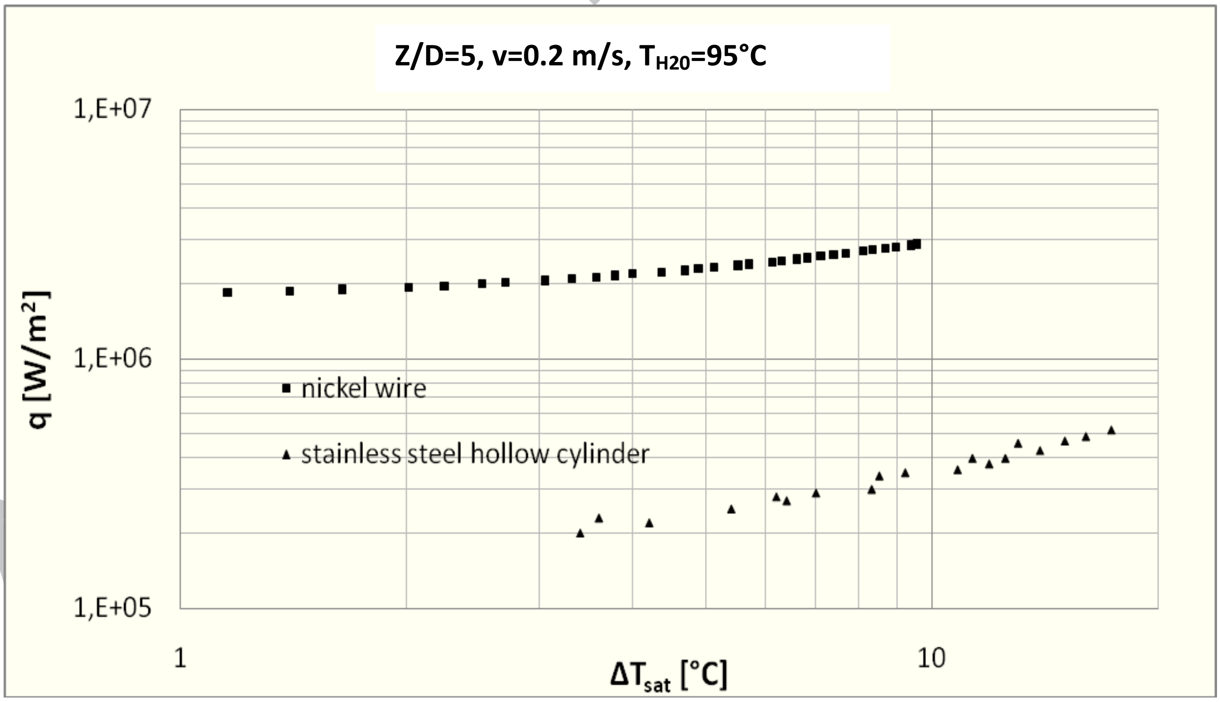


Fig.23: Comparison between the experimental results obtained with the nickel wire (0.25 mm. in diameter) and the ones obtained with the stainless steel hollow cylinder (3 mm. in diameter), in sub-cooled boiling conditions (Z/D =5, v=0.20 m/s and ΔT_{SUB}=5°C).

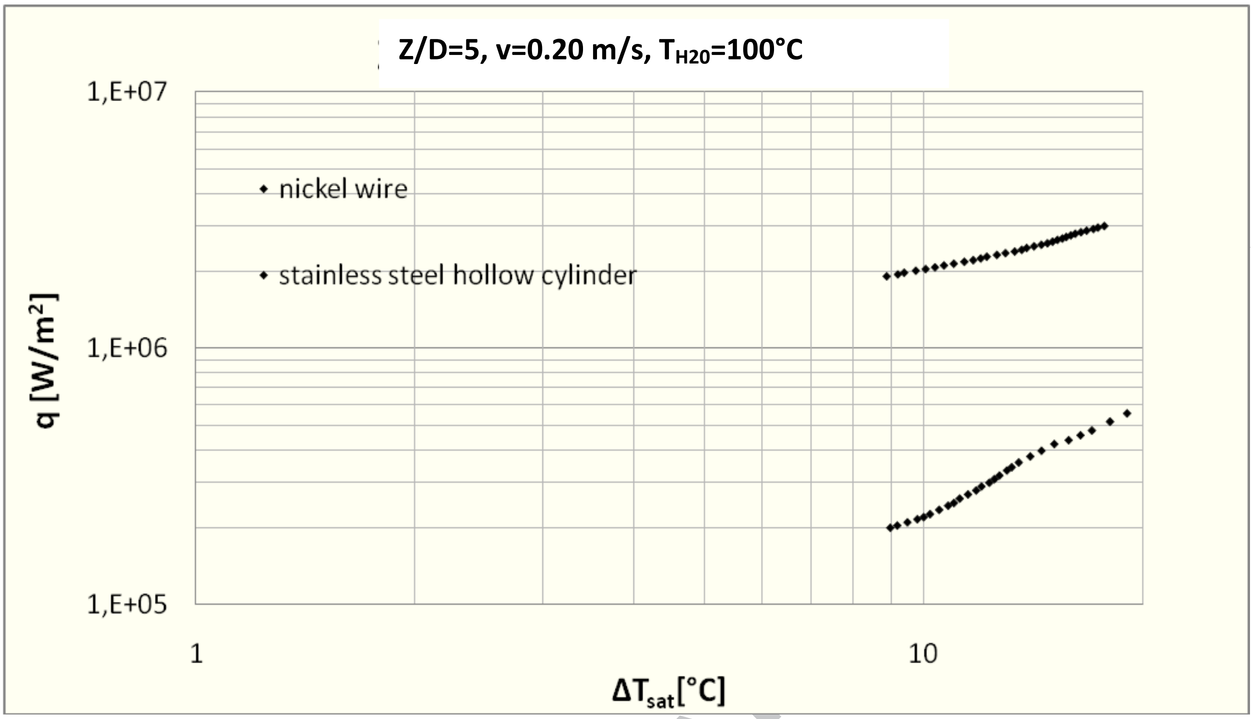


Fig.24: Comparison between the experimental results obtained with the nickel wire (0.25 mm. in diameter) and the ones obtained with the stainless steel hollow cylinder (3 mm. in diameter), in saturated boiling conditions (Z/D =5, v=0.74 m/s in saturated boiling).

Highlights

We investigated the heat transfer from an upward impinging slot jet of distilled water to a nickel wire, 0.25 mm. in diameter;

We set up a proper experimental apparatus to study the influence of the jet exit velocity, the wire placement and the thermal regime;

We measured the wire temperature starting from electric resistance measurements, because the nickel has good thermometric properties;

We found out the optimal conditions to enhance the heat transfer;

We compared these results with the previous ones with cylinders, instead of wires, which were one order of magnitude bigger than the wire;

The nickel wire could dissipate one order of magnitude higher heat flux per unit surface with respect to the flux dissipated by the stainless steel cylinders.