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Investigation of Optimum Heat Flux Profile Based on the Boiling Safety Factor

Assit.Prof. Dr. Akram W. Ezzat*
Mechanical Eng. Dept.
University of Baghdad / Iraq
Email: akramwahbi@yahoo.ie

Dr. Sajida Lafta Ghashim
Mechanical Eng. Dept.
University of Baghdad / Iraq
Email: sajda_lafta@yahoo.com

ABSTRACT

An experimental study is conducted to investigate the effect of heat flux distribution on the boiling safety factor of its cooling channel. The water is allowed to flow in a horizontal circular pipe whose outlet surface is subjected to different heat flux profiles. Four types of heat flux distribution profiles are used during experiments: (constant distribution profile, type a, triangle distribution profile with its maximum in channel center, type b, triangle distribution profile with its maximum in the channel inlet, type c, and triangle distribution profile with its maximum in the channel outlet, type d). The study is conducted using heat sources of (1000 and 2665W), water flow rates of (5, 7 and 9 lit/min). The water temperature at cooling channel inlet is kept constant at (25°C). Copper test section of (0.6 m) length (0.025m) inner diameter is used during the experiments. The electrical heater used for water heating is wrapped around the copper pipe covering (50 cm) of its length. Calibrated thermocouples are distributed along pipe surface at distances (0.1, 0.2, 0.3, 0.4 and 0.5 m) from pipe inlet to measure pipe surface temperature. The results shows that the heat source with heat flux profile of type (c) is the most reliable one from thermo-hydraulic safety point of view for both types of heat sources, as it ensures a maximum boiling safety factor (K) of (1.6, 1.7, 2) at water flow rates of (5, 7 and 9 lit/min) respectively based on maximum heat capacity of (2665 w), while the heat source with heat flux profile of type (d) which posses minimum boiling safety factors of (1, 1.2, 1.3) at water flow rates of (5, 7 and 9 lit/min) respectively based on same heat capacity value is the worst one from same point of view.

Keyword: boiling safety factor; heat flux profile; parameters of heat transfer; horizontal pipe, water flow rate.

دراسة شكل الفيض الحراري الأمثل المبني على أساس معامل الامان للغليان

د. ساجدة لفته غشيم
قسم الهندسة الميكانيكية
جامعة بغداد

أ.م.د. اكرم وهبي عزت
قسم الهندسة الميكانيكية
جامعة بغداد

الخلاصة

تم اجراء دراسة عملية لتأثير شكل الفيض الحراري على معامل الأمان للغليان لقناة التبريد الخاص بها. تم أمرار الماء خلال انبوب افقي تحت تاثير اشكال الفيض الحراري المختلفة. تم اختيار اربع أنواع لتوزيع الفيض الحراري في التجارب: (شكل a منتظم ، شكل b مثلث يكون تمركز القمة في منتصف القناة ، شكل c مثلث يكون تمركز القمة في بداية القناة و شكل d مثلث يكون تمركز القمة في نهاية القناة). تم أعتماد قيم قدرة حرارية (1000، 2665) واط، كمية الماء المتدفق (5, 9,7) لتر/ دقيقة. تم تثبيت درجة حرارة الماء عند دخول القناة في 25 درجة مئوية. تم أستخدام أنبوب نحاسي بأبعاد 0.6 م طول و 0.025 م قطر أثناء التجارب. تم لف المسخن الكهربائي على السطح الخارجي للأنبوب النحاسي أبتدا من مسافة (5سم) الى مسافة (55سم). تم قياس درجة حرارة السطح على طول الانبوب في خمس مناطق على بعد (0.5 , 0.4 , 0.3 , 0.2, 0.1) م من مدخل الانبوب. بينت الدراسة ان شكل توزيع الفيض الحراري نوع c هي الأكثر أعتمادية من وجهة نظر السلامة ذات العلاقة بالخصائص الثرموهيدروليكية لنوعي المصدر الحراري لكونها تحقق أعلى قيم لمعامل السلامة للغليان على طول القناة (1.71.6، 2) عند معدلات جريان (5,7 , 9) لتر/ دقيقة بالتتابع، بينما النوع d والذي يمتلك أوطا قيم لمعامل السلامة للغليان على طول القناة (1,1.2,1.3) عند معدلات جريان (5,7 , 9) لتر/ دقيقة بالتتابع ولنفس نوعي المصدر هي الأسوء من نفس وجهة النظر.
الكلمات الرئيسية: معامل السلامة للغليان، شكل الفيض الحراري، معاملات انتقال الحرارة، انبوب افقي ، معدل التدفق .

1. INTRODUCTION

Cooling systems are utilized nowadays in different ways. The major initiators of their usage are increased power, reduction of pressure or coolant flow rate which gives rise to poor heat transfer process. Likely improvement occurs for the methods of coolant transfer, heat flux profile and other areas. A large number of studies have been directed to investigate nucleate boiling, locating peak of heat flux and safety factor**.** Safety factor is an expression depict the ability of a system beyond the real capacity. The term of safety factor using in different application, beginning from construction, electrical, mechanics and ending with industrial applications. The focus during this research will be on boiling safety factor as a part of the main study.

Breen, 1964, studied experimentally boiling heat flux by using thermocouples measured temperature of boiling surface in nucleate and transition boiling. Thermocouples were calibrated by using boiling point of Acetone, isopropanol and Freon-113. Thermocouples of 0.025 cm diameter measured temperature inside and outside surfaces of the boiling tubes in nucleate and transition boiling. The results showed that, the amount of the heat flux in relation to the high heat flux may be achieved from the nucleate boiling fluctuations. The initiation of transition boiling is defined by a broken rise in wall temperature and a large rise in temperature fluctuation capacity. Also, thermocouples fixed on surface the best method to prediction and control on peak heat flux in nucleat boiling.

Nukiyama, 1966, illustrated experimentally the effect of heat transferred from a metal surface to water under boiling process. The experiments were conducted using thin metal. The wire is inserted in the water with electric current through the wire. The results showed the temperature difference between the surface and the water increases with increased heat transferred from a metal surface to water. But after the temperature difference reached to certain limit, the value of heat transfer decreases with increased in temperature difference

Ezzat and Taki, 1988, studied the thermal and hydraulic design of IRT-5000 reactor theoretically in steady state condition. The study reconnoitered the range of boiling safety factors that predict nucleate boiling upon the expected changes of reactor operating parameters initiated by excess reactivity insertion due to the oscillation of the control rods inside the reactor core. Also they calculated value of the maximum wall temperature during normal operating taking into account hot channel factors, hot spot factors and all uncertainties, which is proved to be lower than the boiling temperature. The maximum heat flux is estimated to be lower than the critical heat flux for forced convection.

Nariai and Inasaka, 1994, investigated experimentally critical heat flux CHF under circumferentially non-uniform heating conditions by using direct current heating of a stainless steel pipe. The results showed the critical heat flux in a smooth pipe without internal twisted tape under non-uniform heating conditions was slightly higher than that under uniform heating conditions because of the lower average qualities at the tube outlet under non-uniform heating condition than those under uniform heating condition. The higher critical heat flux under the non-uniform heating conditions was explained by the alternating development and disruption of the bubble boundary layer.

Divavin, et al., 1996, studied numerically and experimentally effect of high heat flux on one side heating in nuclear reactor at sub cooled flow boiling. A numerical analysis was made on circular channel under non-uniform heat flux on one side at 5-30 MW /m^2. The results showed, under non uniform heat flux all ways of heat transfer of sub cooled flow boiling sitting under stable condition relation to intensive peripheral temperature gradient

Moon, et al., 2002, illustrated experimental water flow in vertical annulus under non-uniformly heated condition at low flow and a large range of pressure conditions. The experiments were conducted using three types of flow (slow, normal and fast) transient in nuclear power plant. The inner rod having heated length of 1843 mm was uniformly divided into ten steps to simulate a symmetric chopped cosine axial heat flux profile. The results represented flow pattern of two-phase.

Yapıcı and Albayrak, 2004, investigated numerically laminar and forced convection flow in a circular pipe under effect of heat flux for two cases uniform and non-uniform. The conservation equations were used to simulate numerical results solved by using package Fluent (version 4.5.0) and Heating 7. The effect of heat flux on temperature and thermal stress distributions have been presented for different mean flow velocities.

Moon, et al., 2006, studied experimental critical heat flux of water flow in vertical annuli under uniform and divided cosine axial heat flux distributions with low range of flow and a high value of pressure conditions. The influence of distribution axial heat flux was large at low-pressure conditions compared to large pressure.

Salama, 2011, investigated numerically a steady state, three-dimensional flow in IAEA MTR type of reactor. CFD code Fluent (version 6.2.16) was used to solve numerical solution. The governing conservation laws and turbulence models k-w were used for the solution.

Al-Malikya, 2013, studied numerically liquid flow in a horizontal circular pipe under non uniform heat flux for six cases. FLUENT version (6.3) and GAMBIT were used to solve continuity, momentum and energy equations. Laminar flow is ensured in a pipe which has L/D=40 .The results showed that Nusselt number for each case of non-uniform heat flux increased with increasing Reynolds number .Correlation equations of Nusselt number for all cases were obtained.

Alimoradi and Shams, 2017, developed a numerical model for study wall temperature and volume fraction in sub- cooled flow boiling. Euler–Euler model with finite volume method was used to simulate water vapor flow in a vertical pipe. The experiments were conducted using range of pressure of 1.5 Mpa, heat flux 0.8 MW/m^2, mass flux 900 kg/m.s, and inlet sub-cooled temperature 47°K . The results showed that wall temperature increases by increasing the pressure value and heat flux, while it decreases by increasing the mass flux. Pressure decrease increased also volume fraction while wall temperature decreased by decreasing pressure. The comparison between numerical and experimental results showed good agreement.

The present work investigates experimentally the effects of heat flux profile on boiling safety factor and find the optimum profile that ensures maximum boiling safety at the position of maximum surface temperature. This optimum profile will ensure heat exchange process far from any bubble formation or nucleate boiling inside the pipe.

2. EXPERIMENTAL WORK

A schematic diagram of the test rig is shown in **Fig. 1**. The test section is a horizontal circular pipe manufactured from copper. The length of test section is (0.6 m) with inside and outside diameters of (0.025 m) and (0.028 m), respectively as shown in **Fig. 2**. The outer surface of pipe was heated electrically using an electrical heater. The heating element covers (50 cm) of the pipe length, as shown in **Fig. 3**. The inlet temperature of the water is kept constant during the experiment. Different types of heat flux distribution profiles are used in the experiment at different values of heat flux rates, as shown in **Fig. 4**. The heat flux distribution profiles are classified as given below:

- Constant heat flux distribution profile, type a.
- Triangle heat flux distribution profile with its maximum in channel center, type b.
- Triangle heat flux distribution profile with its maximum in the channel inlet, type c.
- Triangle heat flux distribution profile with its maximum in the channel outlet, type d.

These profiles could be ensured by adopting proper spacing of the coil wire along its length. Heater wire of (2mm) diameter, (3m) length and (5.6Ω/m) specific resistance is connected to an electrical power of (3000 W) capacity. The heater wire is electrically insulated by ceramic beads and (2 cm) layer of fiberglass which covered it to ensure a reliable thermal insulation. The heater is supplied with AC-current from 220V voltage regulator. The circuit is connected to digital voltage regulator to control the current according to the desired heat flux. Clamp meter is used to measure the current passing through the heater. The temperature of the pipe outside wall is measured using thermocouples type K (chromium - aluminum) installed in seven positions along pipe length. The first thermocouple is located at pipe entrance while the seventh is located at pipe exit. The other five thermocouples are distributed at equally spaced distances (10, 20, 30, 40 and 50 cm) along the heated wall. The end of thermocouple wires are connected with the digital thermometer. The water is supplied to the test section from (50 lit) water supply tank. Proper flow meter is used to read water flow rate in the test section. Double pipe heat exchanger is used to control heat removed from the test section.

3. THEORETICAL APPROACH

The power supplied to the wall is calculated as follow;

$$Po = I.Vo \tag{1}$$

Where:
Po = electrical power
I= current flow
Vo= voltage

Heat transferred by convection to the water is calculated by:

$$Q_{conv} = Po - Q_{loss} \tag{2}$$

Where Q_{loss} is the total heat losses by conduction and radiation and becouse the heat loss by radiation is very small , therefore can be neglected. Then the heat loss can be found from the following relation:

$$Q_{loss} = \frac{T_{center} - T_{ambient}}{\frac{1}{h_i A_i} + \frac{\ln\frac{r_2}{r_1}}{2\pi k_{pipe} L} + \frac{\ln\frac{r_3}{r_2}}{2\pi k_{in.1} L} + \frac{\ln\frac{r_3}{r_2}}{2\pi k_{in.2} L} + \frac{1}{h_o A_o}} \tag{4}$$

Heat flux along the length of pipe is calculated by using the following relation:

$$q'' = \frac{Q_{conv}}{A_s} \tag{5}$$

Where: $A_s = \pi \times D_i \times L$

The mean velocity of water defined as follows:

$$u = \emptyset_{water}/A \tag{6}$$

The Reynolds number according to the pipe diameter and the velocity of fluid at the inlet defined as:

$$Re = \frac{\rho \, u_{in} \, D_i}{\mu} \tag{7}$$

The bulk temperature profile over the length of pipe could be represented by:

$$T_{bulk}(x) = T_{in} + \frac{q'' \pi \, D_i x}{\dot{m} cp} \tag{8}$$

The local and average heat transfer coefficient is defined as follows:

$$h_x = \frac{q''}{(T_{wall(x)} - T_{bulk(x)})} \tag{9}$$

$$h = \frac{1}{x} \int_0^x h_x \, dx \tag{10}$$

The local and average Nusselt number is calculated using following equation:

$$Nu_x = \frac{h_x\, Di}{k} \tag{11}$$

$$Nu = \frac{1}{x}\int_0^x Nu_x\, dx \tag{12}$$

3.1 Heat flux calculation
The heat flux profiles used in the experiments are estimated as follows:

A. Constant heat flux distribution profile
$$q''_{ave} = \frac{Q_{conv}}{A_s} \tag{13}$$

B. Triangle heat flux distribution profile with its maximum in channel center.
$$q''_{max} = \frac{Q_{conv}}{\frac{1}{2}A_s} \tag{14}$$

$$q''_x = q''_{max}\left(\frac{2x}{L}\right) \qquad \text{when} \qquad 0 \leq x \leq \frac{L}{2} \tag{15}$$

$$q''_x = q''_{max}\left(2 - \frac{2x}{L}\right) \qquad \text{when} \qquad \frac{L}{2} \leq x \leq L \tag{16}$$

C. Triangle heat flux distribution profile with its maximum in the channel inlet.
$$q''_{max} = \frac{Q_{conv}}{\frac{1}{2}A_s} \tag{17}$$

$$q''_x = q''_{max}\left(1 - \frac{x}{L}\right) \qquad \text{when} \qquad 0 \leq x \leq L \tag{18}$$

D. Triangle heat flux distribution profile with its maximum in the channel outlet.
$$q''_{max} = \frac{Q_{conv}}{\frac{1}{2}A_s} \tag{19}$$

$$q''_x = q''_{max}\left(\frac{x}{L}\right) \qquad \text{when} \qquad 0 \leq x \leq L \tag{20}$$

3.2 Sub-cooled boiling temperature
In nucleate boiling, bubbles are formed by the extension, of entrapped gas or vapor at small cavities in the wall. When topical boiling conditions are observed, the primary mechanism of heat transfer is thought to be the intense agitation at the heat-transfer wall, which formed the high heat-transfer rates observed in boiling. Sub-cooled nucleate boiling is ensured when the heat source clad wall temperature reaches to the boiling temperature at that position. The following correlation is used for boiling temperature estimation based on local heat flux and water pressure, **Ezzat and Taki, 1988.**

$$T_B = T_{sat.} + 2.03\left(q''\right)^{0.35}P^{-0.23} \tag{21}$$

Preclude sub-cooled nucleate boiling at wall surface, by make the hot spot on the pipe surface have maximum temperature lower than boiling temperature. The value of boiling safety factor (K) can be defined , **Ezzat and Taki, 1988** :

$$K = \frac{T_B - T_{in}}{T_{wall} - T_{in}} \tag{22}$$

K factor is used as a safety criterion by nuclear regulatory body for research reactors. This factor is adopted a value of 1.6.
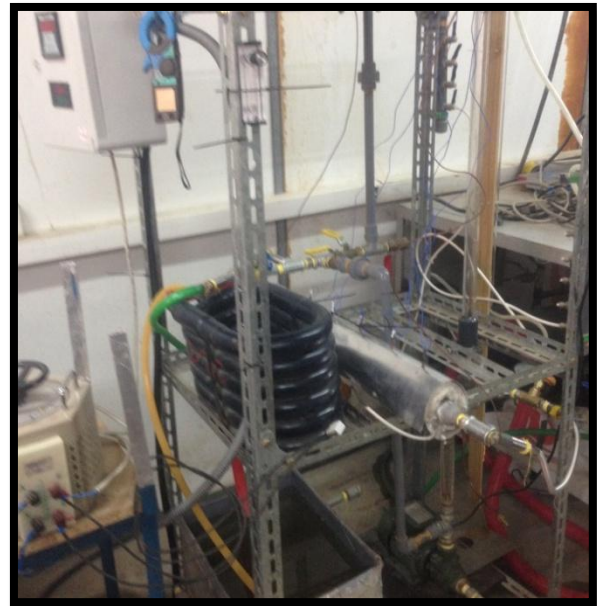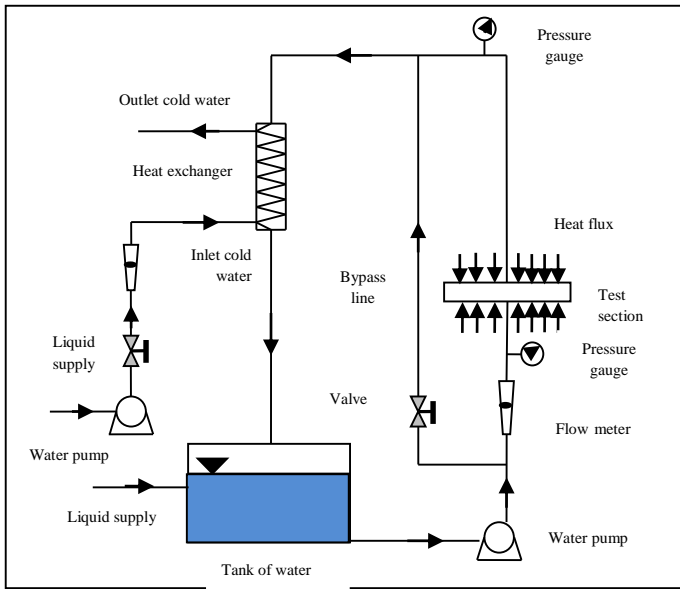
(a)   (b)
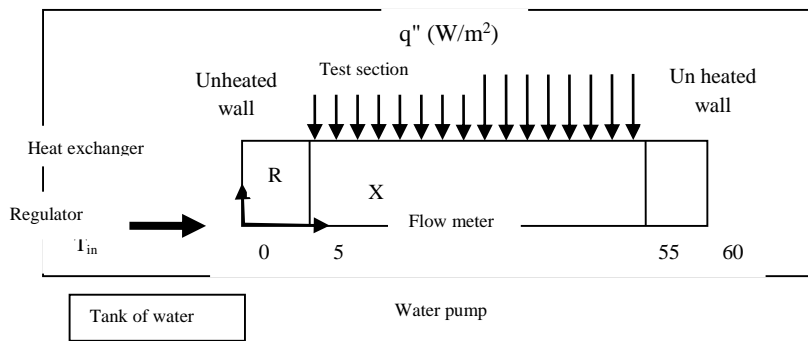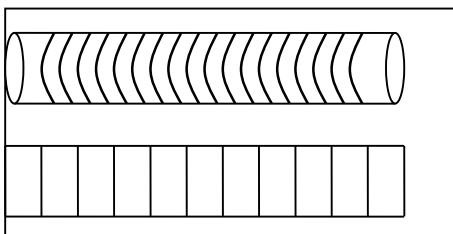
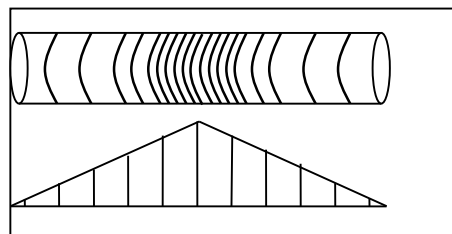**Figure 1.** Schematic diagram of the test rig.

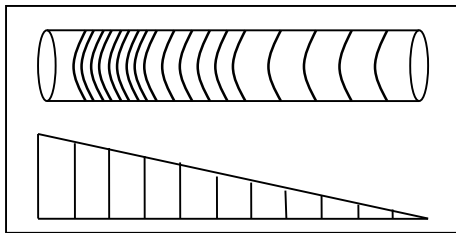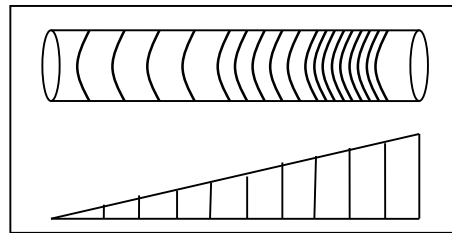**Figure 2.** Schematic diagram of the heat source and its boundary conditions.

Type (a) heat flux distribution profile   Type (b) heat flux distribution profile

Type (c) heat flux distribution profile   Type (d) heat flux distribution profile

Figure 3. Types of heat flux distribution profiles.

4. RESULTS AND DISCUSSION
Table 1. Shows the correlation of average Nusselt number (Nu) versus Reynolds number (Re) that covers the data of experimental results (Re = 4968 to 8672), **Dittus and Boelter, 1930:**

$$Nu = c(Re)^n \qquad (23)$$

Where c and n are empirical constants.

Table 1. Correlations of the average Nusselt numbers.

| Heat flux shape	Po (W)	Q (lit/min)	Correlation equation
Constant heat flux	1000	5	$Nu = 0.389\,(Re)^{0.5323}$
		7	
		9	
	2665	5	$Nu = 0.258\,(Re)^{0.631}$
		7	
		9	
Increasing heat flux	1000	5	$Nu = 0.044(Re)^{0.7301}$
		7	
		9	
	2665	5	$Nu = 2.256(Re)^{0.338}$
		7	
		9	
Decreasing heat flux	1000	5	$Nu = 0.798(Re)^{0.418}$
		7	
		9	
	2665	5	$Nu = 3.200(Re)^{0.319}$
		7	
		9	
Concentrated heat flux	1000	5	$Nu = 0.286(Re)^{0.528}$
		7	
		9	
	2665	5	$Nu = 2.82(Re)^{0.325}$
		7	
		9	

Fig. 4 shows experimental results of wall temperature variation along dimensionless length of pipe at value of heating capacity of (1000 and 2665 W), water flow rates of (5, 7 and 9 lit/min) and different heat flux profiles. At uniform heat flux, type a, the wall temperature increases versus normalized length and reaches its maximum value at the end of the pipe. The wall temperature is proportional to heat flux value subjected on the water channel and inversely proportional to the water flow rate. It could be noticed also that the figure related to type (a) and (d) occupies same trend. The figure related to type (b) reaches to its maximum value at certain position within second half of the channel. This position normally depends on the heat source capacity and water flow rate in addition to the heat source profile. The case of type (c) shows that wall temperature decreases versus the normalized distance.

Fig.5 shows experimental results related to bulk temperature variation along dimensionless length of pipe for heating capacity of (1000 and 2665 W), water flow rates of (5, 7 and 9 lit/min) and different heat flux profiles. It is noticed from the figures that the bulk temperature reaches to its maximum value at exit of pipe and that the differences between these temperatures at the end of the channel related to the four cases are almost negligible which complies with physical justifications of water heating up.

Fig. 6 shows experimental results of the boiling safety factor K variation along dimensionless length of pipe for heating capacity of (1000 and 2665 W), water flow rates of (5, 7 and 9 lit/min) and different heat flux profiles. It is obvious, this figures related to the four heat flux profiles have the inverse trend of those related to the wall temperature figures shown in **Fig. 4**.

Fig. 7 shows experimental results of the Boiling safety factor K variation with Re number at maximum wall temperature for different profile and values of heat fluxes. The results illustrates the idea of relating the coolant flow rate with boiling safety factor at highest wall temperature. As shown from figures, when the mass flow rate increases, the boiling safety factor increased. This explains that when the flow rate is higher, the fluid inside the pipe has less time to suck in thermal energy which leaves it cooler. Therefore this low temperature of the coolant will cool the pipe faster than when its temperature is higher. In other words, the coolant helps in the prevention of equating the boiling temperature and the saturation temperature, leaving the boiling safety factor away from unity

Figs. 8 and 9 show minimum boiling safety factor K variation at maximum wall temperature positions. The figures cover the experimental values of both water flow rates and heat capacities. The figures prove that K values are proportional to water mass flow rate and inversely proportional to heating capacity**.** However all the figures illustrates that boiling safety factor at maximum surface temperature position is property of the heat flux profile and that its maximum value is related to heat flux profile type (c) while its minimum value is related to heat flux profile type (d).
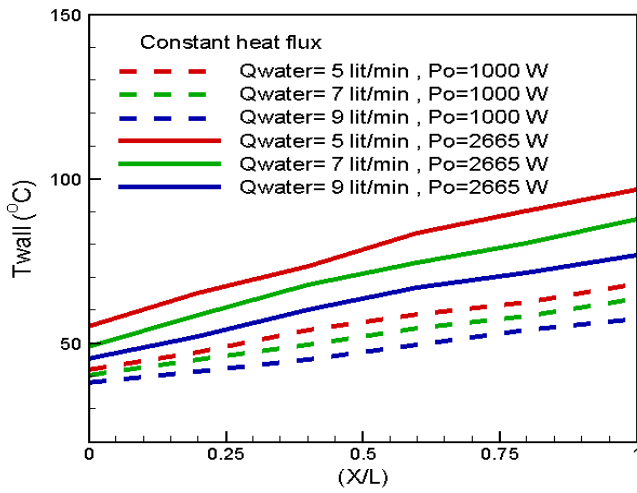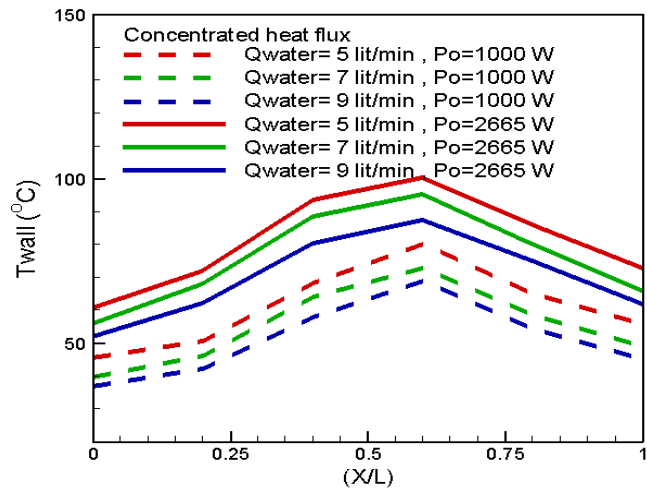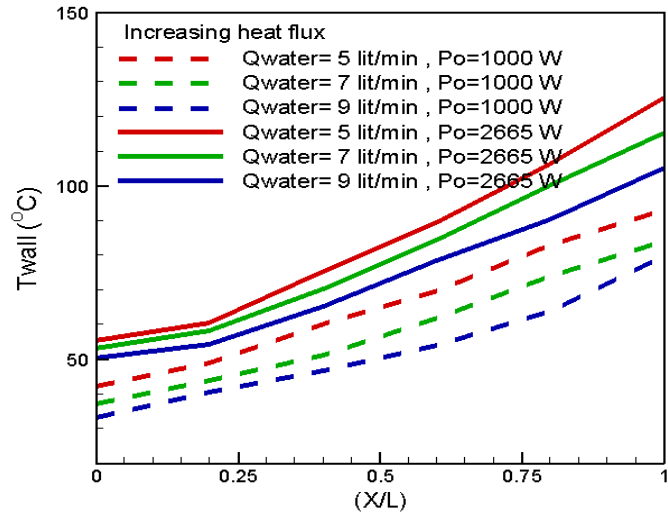
Type (a)   Type (b)
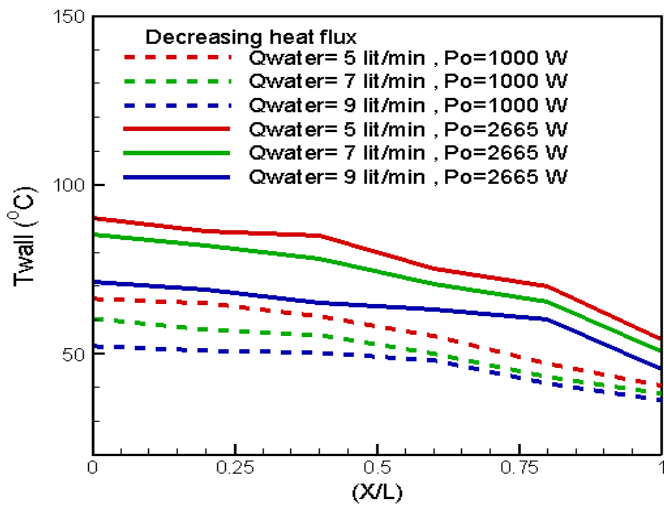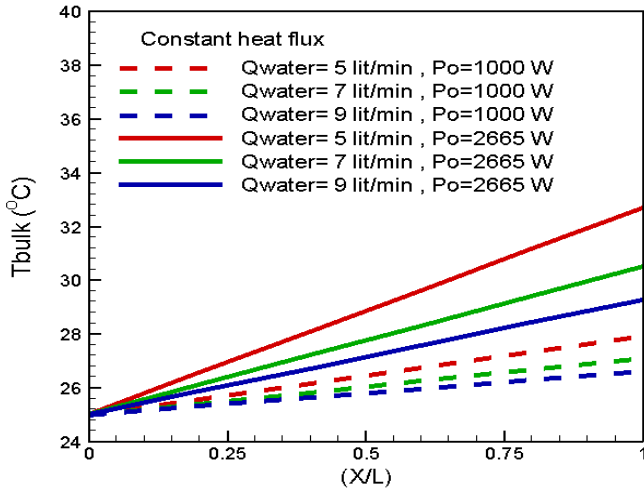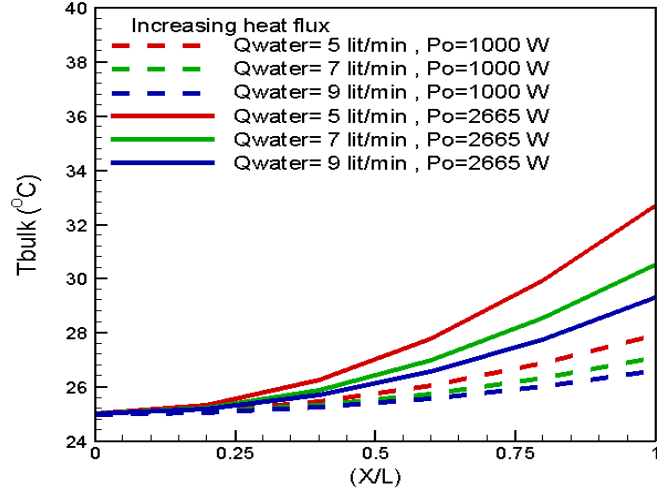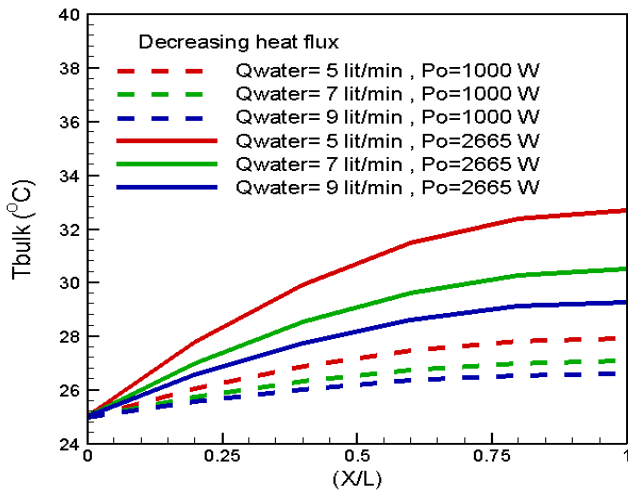
Type (c)   Type (d)

Figure 4. Wall temperature variation versus cooling channel length at different heat flux profiles, water flow rate and thermal loads.
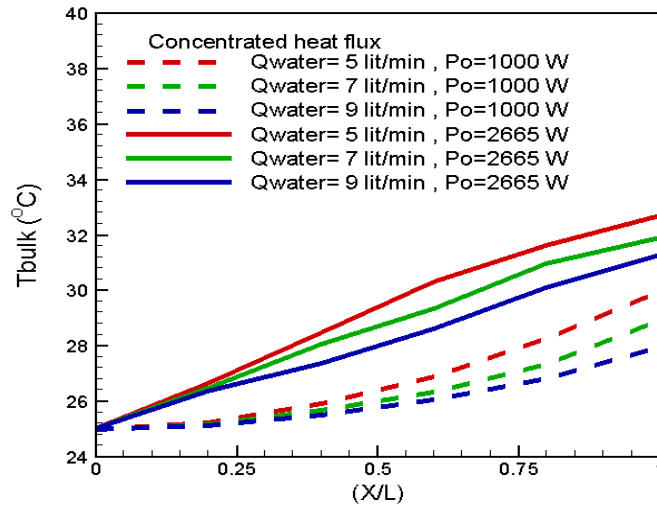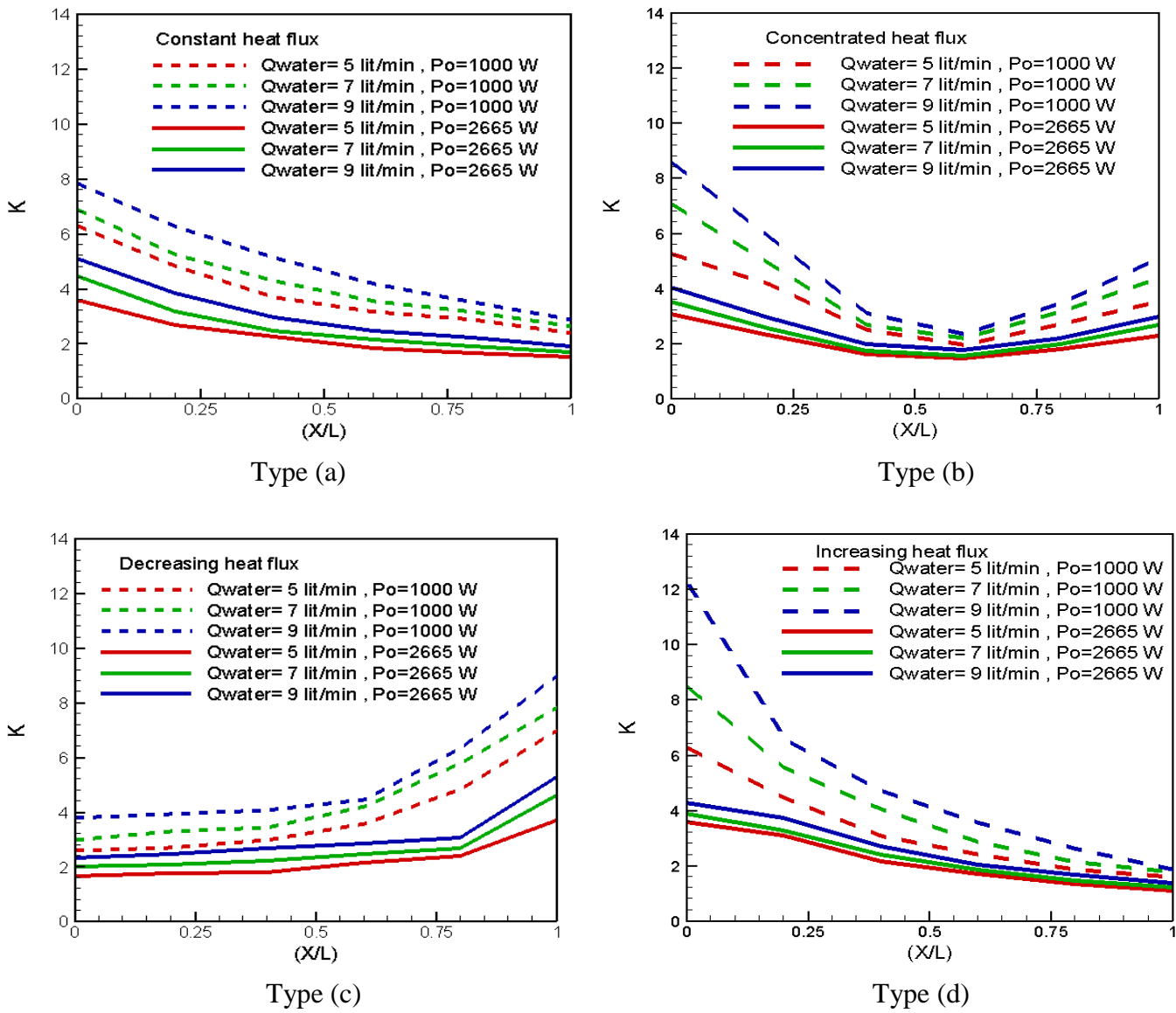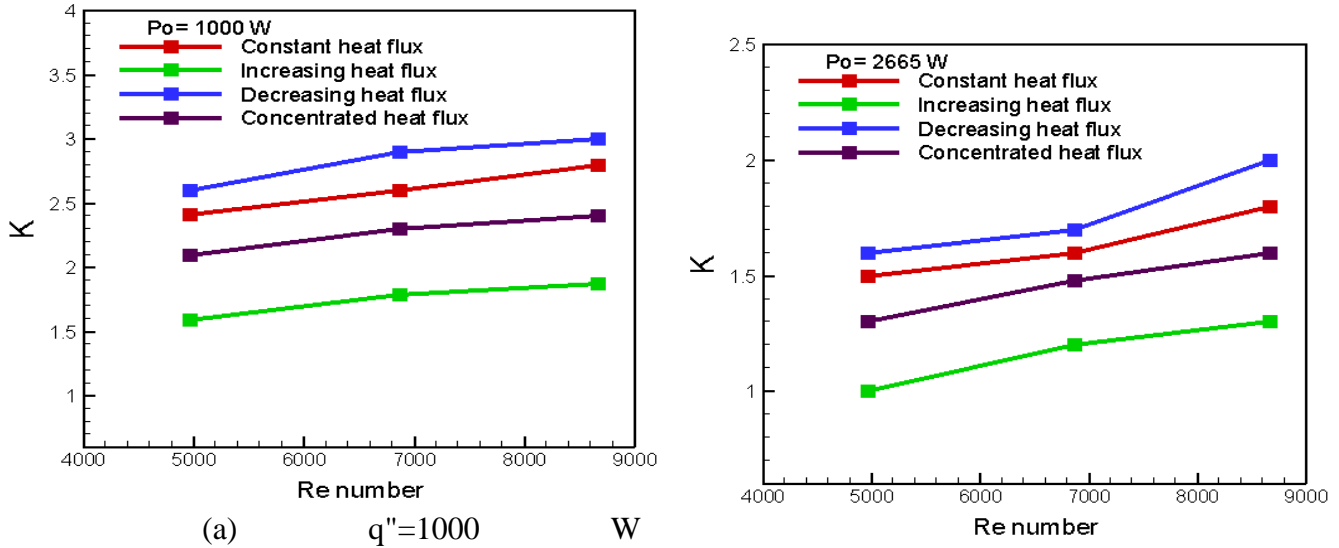
Figure 5. Bulk temperature variation versus cooling channel length at different heat flux profiles, water flow rate and thermal loads.

Figure 6. Boiling safety factor variation versus cooling channel length at different heat flux profiles, water flow rate and thermal loads.

(a)   q"=1000   W

(b) q"= 2665 W

Figure 7. Boiling safety factor variation at maximum wall temperature positions versus Re number for different heat flux profiles.
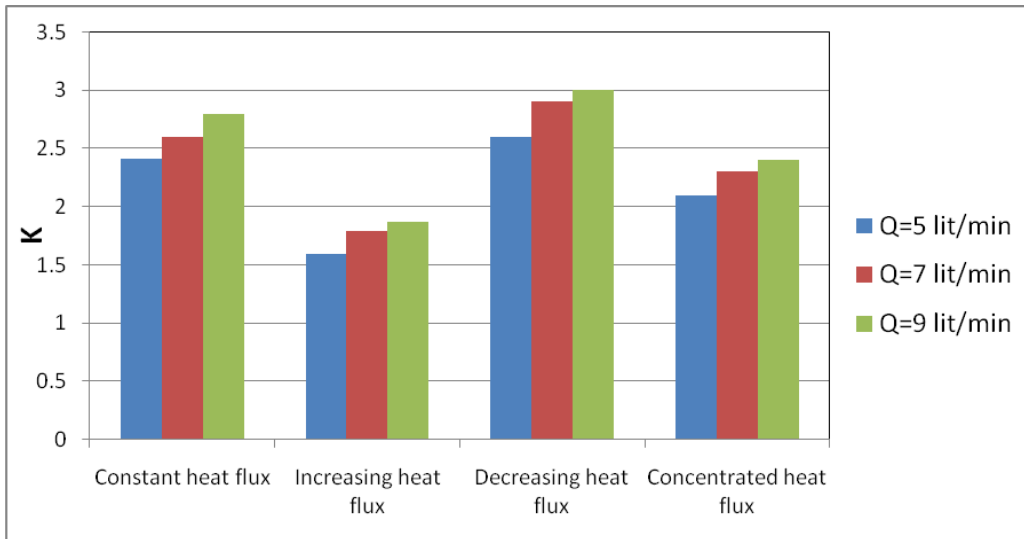
Figure 8. Minimum boiling safety factor variation at maximum wall temperature positions for different heat flux profiles at heating capacity of 1000 W.

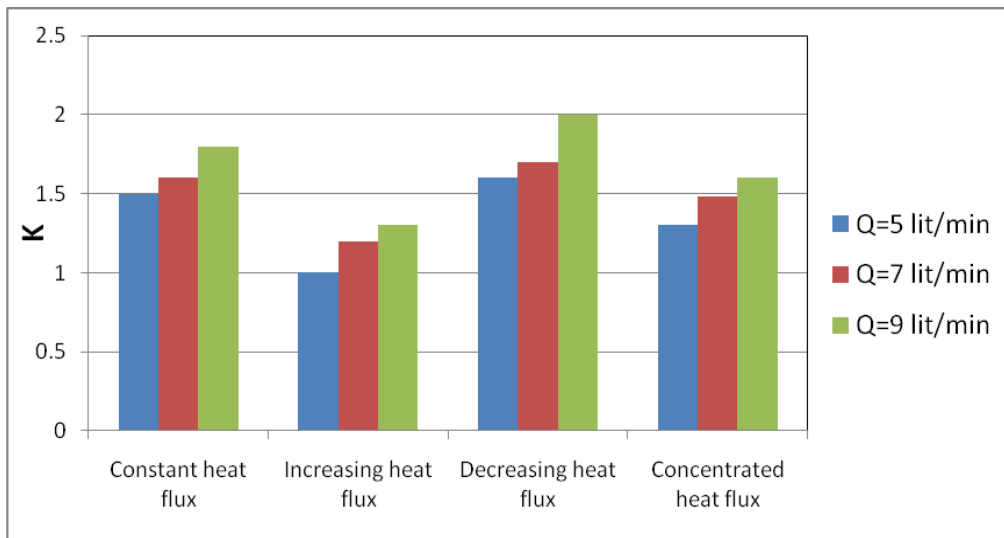**Figure 9.** Minimum boiling safety factor variation at maximum wall temperature positions for different heat flux profiles at heating capacity of 2665 W.

5. CONCLUSIONS

Heat transfer enhancement, avoidance of film boiling and its poor characteristic should be taken under consideration in any thermal design of power generation systems by adopting proper boiling safety factor, (K). Thermal design criteria for most nuclear research reactors adopts K values ranged (1-2) depending on its type, power and other design characteristics. Heat transfer parameters related to flowing water inside horizontal circular pipe subjected to different heat flux profile heat source are studied. The study covers the effect of different water flow rates and heating capacities. The following are concluded:

1. The boiling safety factor, K depends on both water mass flow rate and heating capacity as it is proportional to the first and inversely proportional to the second despite the heat flux distribution profile.

2. The heat flux distribution profile shows sensible effect on boiling safety factor, K as it has both direct and indirect effects on it. The direct effect is on the difference between coolant and surface temperatures at any point which is affected by the local value of the heat flux while the indirect effect is related to the accumulated heat removed by water from channel inlet to the position at which this factor is calculated.

3. The results shows that the heat source with heat flux profile of type (c) is the most reliable one from thermo-hydraulic safety point of view for both types of heat sources, as it ensures a maximum boiling safety factors of (1.6, 1.7, 2) at water flow rates of (5, 7 and 9 lit/min) respectively based on maximum heat capacity of (2665 w), while the heat source with heat flux profile of type (d) which possess minimum boiling safety factors of (1, 1.2, 1.3) at water flow rates of (5, 7 and 9 lit/min) respectively based on same heat capacity value is the worst one from same point of view.

4. It is clear also that the position of the maximum surface temperature depends on the heat flux profile in addition to the other design and operating condition parameters, for example channel dimensions, coolant velocity and heating capacity.

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7. NOMENCLATURE

Latin symbol

A	area of cross section, m^2
A_s	surface area, m^2
C_P	specific heat at constant pressure, J/kg.°C
D_i	tube inner diameter, m
h	average heat transfer coefficient, $W/m^2.°C$
h_x	local heat transfer coefficient , $W/m^2.°C$
K	boiling safety factor
k	thermal conductivity, W/m.°C
I	electrical current, Am
L	length of pipe
$\dot{m}$	mass flow rate, kg/s
Nu	average Nuesslt number, $(Nu = hD_i/k)$
P	pressure, Pa
Po	electrical power, W
Q_{conv}	convection heat loss, W
Q_{loss}	total losses, W
Q_{cond}	conduction heat loss, W
q"	heat flux per unit area , W/m^2
$q^{"}_{ave}$	average heat flux, W/m^2
$q^{"}_{max}$	maximum heat flux, W/m^2
$q^{"}_{x}$	local heat flux, W/m^2
Re	reynolds Number
R,X	cylindrical coordinates
r	radius
T_{in}	inlet temperature, °C
T_{wall}	wall temperature, °C
T_{bulk}	bulk temperature, °C
T_{center}	temperature at the center of pipe, °C
$T_{ambient}$	temperature at the ambient, °C
T_B	boiling temperature, °C
$T_{sat.}$	saturation temperature, °C
u_{in}	inlet velocity, m/s
Vo	voltage, Volt
X	axial distance along heat source, m

Greek Symbols

μ	Laminar viscosity, kg/m.s
ρ	Density, kg/m^3
$\emptyset_{water}$	Water flow rate, lit/min