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EXPERIMENTAL INVESTIGATION OF THE CRITICAL HEAT FLUX AND POST-DRYOUT TEMPERATURE REGIME OF HELICAL COILS

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Abstract—The results of an experimental investigation of the temperature regime of a helical coil with the upflow and downflow of a steam–water mixture are presented. The experiments were carried out in the following range of operational parameters: pressure, $P = 9.81$–17.6 MPa; mass velocity, $\rho w = 500$–1500 kg m^{-2} s^{-1}; heat fluxes, $q = 100$–1500 kW m^{-2}. At $x > 0.2$, the critical heat fluxes in the coil are much higher than in a vertical straight tube, while the temperature regime of the coil tube in the post-dryout region is less severe. The wall temperature jump in the post-dryout region changes around the tube perimeter and depends on the flow direction.

NOMENCLATURE

- D helix diameter measured from tube axis to tube axis [m, mm]
- d inside tube diameter [m, mm]
- h enthalpy [J kg^{-1}]
- P pressure [Pa, MPa]
- q heat flux [W m^{-2}, kW m^{-2}, MW m^{-2}]
- T temperature [K, °C]
- x steam quality.

Greek symbol
- ρw mass velocity [kg m^{-2} s^{-1}].

Subscripts
- c coil
- cr critical
- in inlet
- max maximum
- s saturation
- st straight
- w wall.

AS EARLIER, when analysing the reliability of operation of steam-generating systems, a considerable amount of attention is being given to the phenomenon of burnout heat transfer in boiling. Although an extensive investigation of this phenomenon has been carried out, one, unfortunately, cannot say that the solution of the problem is nearing completion. New aspects of the phenomenon are revealed, new questions are raised in developing advanced types of apparatus. Thus, coil elements have found wide use in a number of steam-generator designs. Examples are supercharged steam generators, special types of steam generators applied in technological processes of metallurgical, oil and other industries. Since steam generators are developed in the direction of increased heat loadings at high exit steam qualities, the elucidation of the conditions for the occurrence of burnout in coils is of great importance.

Past investigations of critical heat fluxes in coils [1–8] have been carried out in a relatively limited range of operational parameters and geometric dimensions, with the most detailed of these being concerned with the liquids that have low heat of evaporation (Freons). Based on these studies, it is possible to distinguish some specific features of the boiling crisis in coils.

In the majority of experiments, the crisis first originated on the internal side of the coiled tube at moderate and high flow velocities and then spread around the remaining portion of the perimeter. At $x > 0$, the critical heat fluxes in coils are higher than in straight tubes. In the case of subcooled liquid flow, the critical loads in helical coils are lower than in straight tubes.

The heat transfer in the post-dryout region of coils is better than in straight tubes [5, 9, 10], the wall temperature jump is smaller and the change of temperature along the tube length is smoother. The temperature rise, in this case, on the internal side of a coiled tube is much higher than on the external side. It should be noted that the specific features of coils are mainly due to the centrifugal effect and to the secondary flow circulation.

The data available are insufficient for the development of new steam-generator designs. There is no information on the effect of flow direction on the burnout heat transfer and temperature regimes of coil turns. The specific features of boiling crisis occurrence in helical coils with vertical and horizontal axes have still not been clarified. There are no investigations of the effect of heat release around the perimeter and along the length of coiled tubes on the critical parameters. For these reasons it is essential that further experimental and theoretical investigations should be carried out. Experimental studies will aid in accumulation of corresponding experimental facts and numerical data, while analytical investigations will make it possible to correlate them and to develop the engineering calculation methods.

The present paper reports the results of experimental investigation of critical heat fluxes and the temperature regime of a coiled steam-generating channel in the following range of operational parameters: pressure,

$P = 9.81$–17.6 MPa; mass velocity, $\rho w = 500$–1500 kg m^{-2} s^{-1}; heat fluxes, $q = 100$–1500 kW m^{-2}.

The experiments were carried out on a thermal-hydraulic rig. The rig represents a closed circulation loop. The heat agent circulation is accomplished by plunger pumps. From a charging tank the heat agent enters the loop, successively passing preheaters, a test section and refrigerators, and then enters a discharge tank. In order to reduce the flow oscillations in the loop, a receiver, partially filled up with argon, is mounted ahead of the preheaters. The flow stability was ensured by sustaining the required pressure difference at the throttle valve. Chemically desalted water was used as a heat agent.

The heating of the working elements of electrical heaters and also of the test section was accomplished by passing a low-voltage alternating current along the tube. Power variation was achieved with the aid of thyristor-type single-phase voltage regulators connected in series with single-phase step-down transformers.

The temperature of water (upstream and downstream of the preheaters and at the test section inlet and outlet) was measured by chromel–copel immersion thermocouples. Potentiometers were used as secondary instrumentation. The pressure at different points in the loop was controlled by standard 24.5 MPa manometers of 0.6 accuracy class. The coolant flow rate in the loop was measured by a disk flowmeter orifice with recording on a recorder of 1.6 accuracy class. The electrical power supplied to the test section was determined on the lower side of the step-down transformers with the aid of current transformers, ammeters and voltmeters of the electrodynamic system of 0.5 accuracy class. The power of electric heaters was computed on the high side of transformers with the aid of ammeters and voltmeters of the same system.

The test section (Fig. 1) was made from 10 mm diameter stainless steel tubing in the form of a coil having a vertical axis. It has six coils 136 mm in diameter measured from tube axis to tube axis. Four middle coils were heated by alternating current and had three current leads connected in the 'zero-phase-zero' circuit. The pitch of the coil was 55 mm and the length of the heated portion of the test section was 1709 mm. The coils were formed by cold bending of tubing filled up with sand. The wall thickness at the inside of the coil was 0.1 mm larger, and at the outside 0.1 mm smaller, than on the top and bottom of the coiled tube. As a result, the heat load at the inside surface of the coil was about 10% higher, and at the outside surface 10% lower, than at the top and bottom surfaces. Twelve thermocouples were attached to each, internal and external, side of the inlet and outlet heated coils which were connected to the

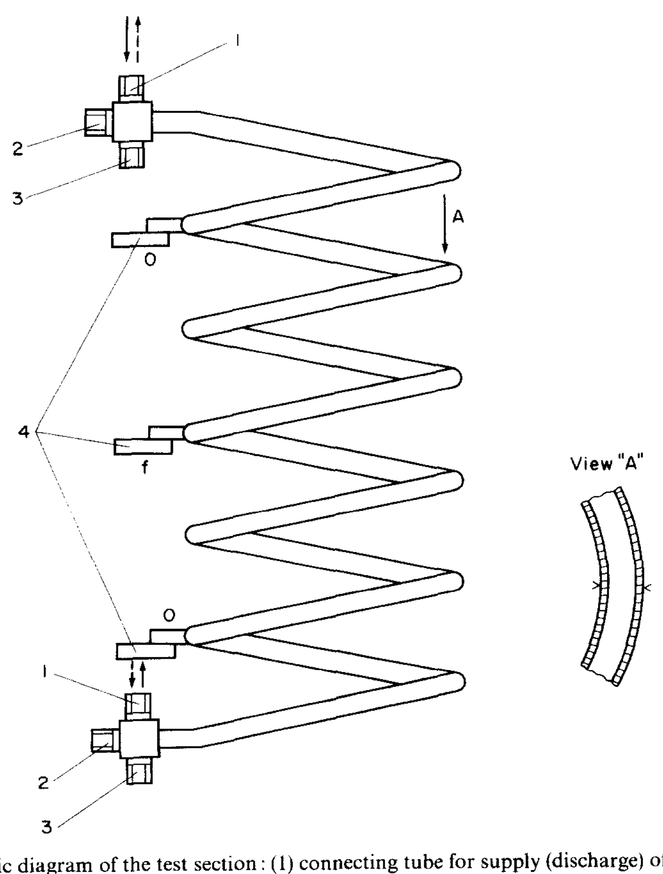


FIG. 1. Schematic diagram of the test section: (1) connecting tube for supply (discharge) of working liquid; (2) connecting tube for insertion of thermocouple; (3) pressure tap; (4) current lead.

potentiometers. Some of the thermocouples were connected to single-point potentiometers provided to protect the test section from burnout. The thermocouples were made from chromel–copel wire 0.2 or 0.5 mm in diameter. They were attached to the outside surface of the tube by electric-spark or argon-arc welding. The distances from the start of heating to the sections, where the thermocouples are located, are listed in Table 1.

The boiling crisis over the test section was achieved by two techniques. In the first technique, it originated at the constant values of heat flux q, exit pressure P and mass velocity ρw due to a smooth increase of enthalpy at the inlet, h_{in}. In the second, it occurred at the constant values of P, h_{in} and ρw due to an increase of the heat load over the test section. The results on q_{cr} obtained by the two procedures were the same.

Usually, the crisis occurred at the exit from the test section. When the crisis occurred at high values of q, the load on the test section was, as a rule, switched off. The critical section was taken to be the last section of the segment which had the temperature typical of nucleate boiling.

The main experiments were preceded by experiments in which the errors of the measurement of temperature of, and heat losses from, the heated elements of the rig were determined, the calibration of the flowmeters and drop-meters was made. These experiments allowed a reliable calculation of the main parameters. The processing of the experimental data and other procedural aspects are described in detail in ref. [11].

Figure 2 presents the dependence of the critical heat flux on the steam quality. It is seen that with an increase in pressure and mass velocity, the value of x_{cr} decreases. In the studied region of heat fluxes and pressures at $\rho w = 500$ kg m^{-2} s^{-1}, the critical steam quality remains unchanged and close to $x = 1.0$. The same applies for $P = 9.81$ MPa and $\rho w = 1000$ kg m^{-2} s^{-1}, where $x_{cr} = 0.9$, and only at $q > 1.05$ MW m^{-2} a tendency is observed towards a decrease in x_{cr}. The direction of flow—upflow or downflow—practically does not influence the critical heat flux.

A comparison of the results of the present investigation with the data on q_{cr} for a coiled tube [2] and vertical tubes with upflow [12] is given in Fig. 3. It is seen that virtually within the whole range of the values studied, the x_{cr}'s for the coils are much higher than for straight tubes. In general, the data agree well with the results of previous investigations. However, it should be noted that a sharp change in the nature of the function $q_{cr} = q(x)$, observed in ref. [2] at $P =$ 20.0 MPa, can hardly be explained only by a small increase of P as compared with the data of the present investigation. This requires further verification.

The function $x_{cr}(\rho w)$ at different heat fluxes is given in Fig. 4. It might be expected that with a decrease in the mass velocity, the influence of the coil diameter D_c on the critical steam quality decreases. This conclusion can be drawn taking into account the fact that the results of the present investigation and of ref. [2],

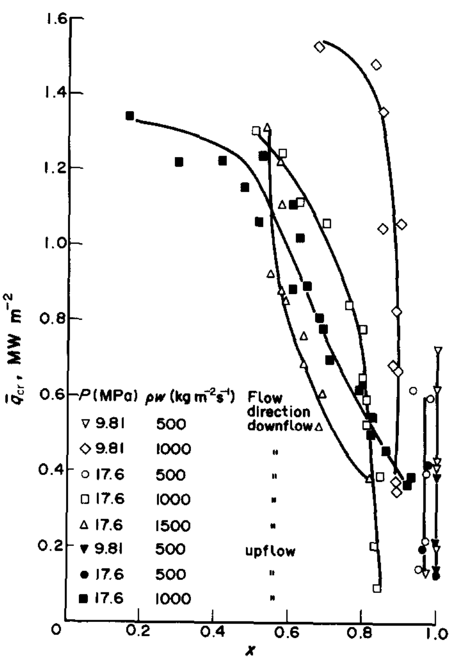
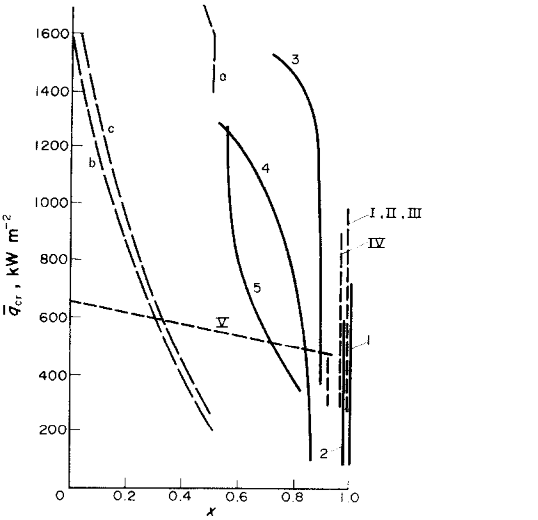


FIG. 2. The effect of steam quality on critical heat fluxes in a helically coiled steam-generating tube.

Table 1.

Number of thermocouples	Distance from the start of heating (mm) Upflow	Downflow	Remark
1;1[1]	1198	1231	Thermocouples 1–10 are located at the inside of the coil; thermocouples 1[1]–10[1], at the outside of the coil
2;2[1]	1252	1282	
3;3[1]	1306	1333	
4;4[1]	1360	1384	
5;5[1]	1414	1435	
6;6[1]	1465	1486	
7;7[1]	1516	1537	
8;8[1]	1567	1588	
9;9[1]	1618	1639	
10;10[1]	1669	1679	



Notation	1	2	3	4	5	I	II	III	IV	V	a	b	c
P, MPa	9.81	17.6	9.81	17.6	17.6	10	15	10	15	20	10	18	18
ρw, kg m^{-2}s^{-1}	500	500	1000	1000	1500	600	600	1000	1000	1000	1000	1000	1500
Test section	Coil d_{in} = 8 mm					Coil d_{in} = 10 mm					Vertical tube d_{in} = 8 mm		
Reference	Authors' data					2					12		

FIG. 3. Comparison between the results of the present investigation and the data of ref. [2] (coil) and ref. [12] (vertical straight tube).

obtained at $\rho w = 500$ kg m^{-2} s^{-1}, coincide, and at $\rho w = 1000$ kg m^{-2} s^{-1} are somewhat different.

Figure 5 presents the temperature regime of the coil in the post-dryout region at $P = 17.6$ MPa and $\rho w = 1000$ kg m^{-2} s^{-1}. As is seen from the figure, the temperature at the internal surface first increases sharply and only thereafter changes insignificantly as in straight tubes. The increase of the temperature at the external surface occurs at much higher steam quality and is much smaller in value. At the fluxes of 400 and

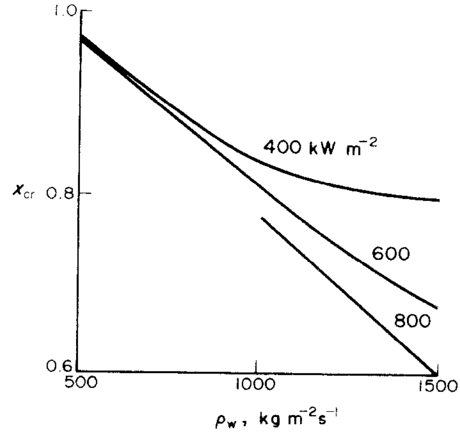


FIG. 4. The critical steam quality as a function of velocity at different heat fluxes, $P = 17.6$ MPa.

600 kW m^{-2}, the crisis at the external surface is preceded by a small increase of its temperature due to the spread of heat around the perimeter from the superheated internal surface. It follows from the figure that the critical steam quality at the external surface of the coil increases with the heat flux. A similar phenomenon was also observed at other values of P and ρw. This fact can be explained as follows: as a liquid droplet approaches the superheated surface, the liquid starts to vigorously evaporate from the side facing the wall. As a result, a reactive force appears which repels the droplets in the opposite direction. The higher the wall is superheated, the more vigorous is evaporation and the higher the reactive force. Consequently, after the crisis has originated at the internal surface of the coil, besides forces acting on the droplets, there appears the radial component directed toward the external surface of the coil. The larger the heat load, the higher radial momentum is acquired by the droplets. As a

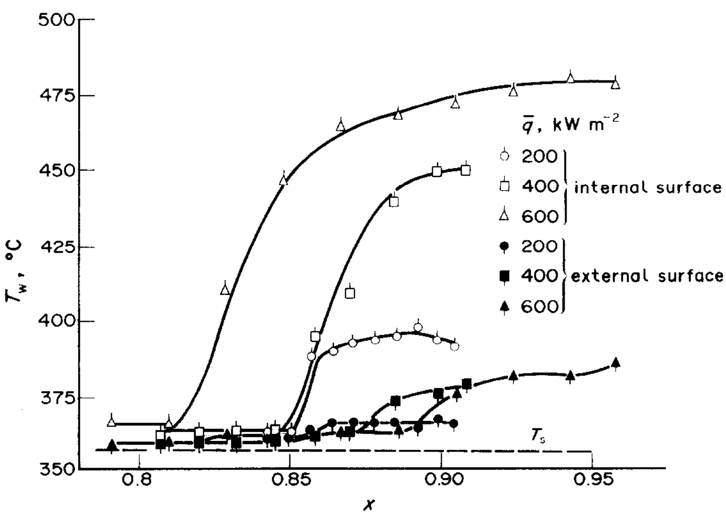


FIG. 5. Temperature regime of a helically coiled steam-generating tube at $P = 17.6$ MPa and $\rho w = 1000$ kg m^{-2} s^{-1}.

result, the wetting flux directed toward the external surface of the coil increases, which appears to be the cause of the burnout delay.

The dependence of the maximum wall temperature increment in the post-dryout region, $\Delta T_w^{max} = T_w - T_s$, on the heat flux, mass velocity and pressure are given in Fig. 6. With an increase of ρw and, to a lesser extent, of P, the temperature jump ΔT_w^{max} decreases. For comparative purposes, Fig. 6 contains the functions ΔT_w^{max} for a 6 mm diameter vertical tube [13, 14] which were obtained at a somewhat smaller pressure, 16.7 MPa, and larger velocities, 750 and 1250 kg m^{-2} s^{-1}. But even in this case the temperature jump turned out, under these conditions, to be larger than in a coil with $P = 17.6$ MPa, $\rho w = 500$ and 1000 kg m^{-2} s^{-1}, respectively.

Figure 7 shows the temperature regime of the internal tube surface for different directions of motion of the coolant. It follows from these plots that the flow direction influences the temperature regime mainly on the internal surface of the coiled tube, with the wall temperature in the post-dryout region in the case of upflow turning out to be lower than in the case of downflow. This fact seems to have been discovered for the first time. However, the data available are

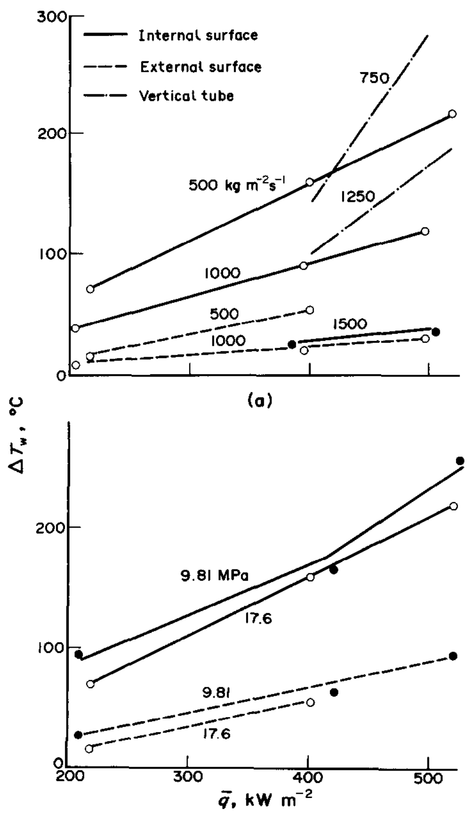


FIG. 6. The dependence of temperature rise, $\Delta T_w^{max} = T_w^{ex} - T_s$, in the post-dryout region on the heat load for helically coiled and vertical straight tubes: (a) $P = 17.6$ MPa; (b) $\rho w = 500$ kg m^{-2} s^{-1}.

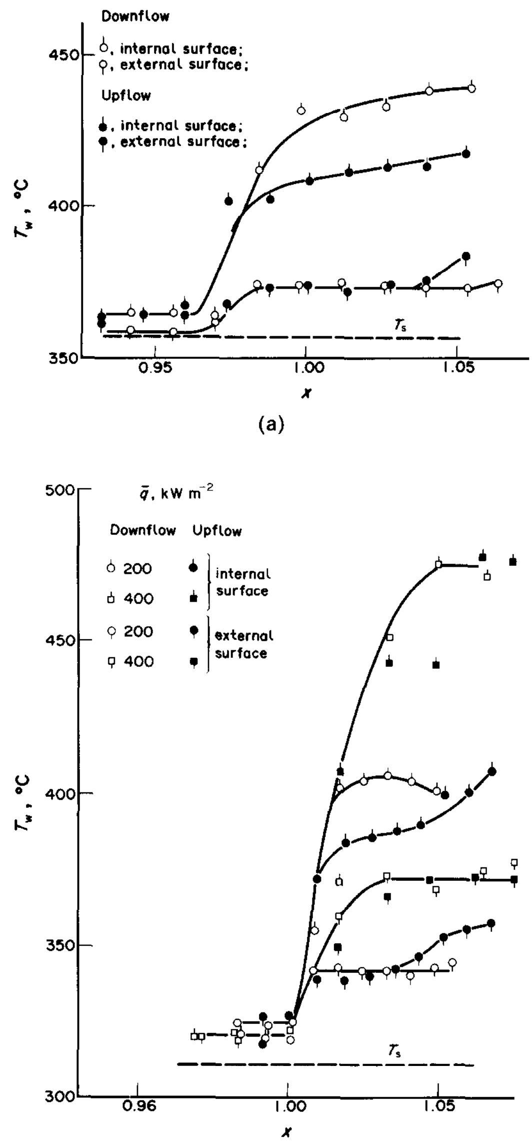


FIG. 7. The temperature regime of a helically coiled steam-generating tube in the post-dryout region as a function of the flow direction at $\rho w = 500$ kg m^{-2} s^{-1}: (a) $P = 17.6$ MPa; (b) $P = 9.81$ MPa.

insufficient to definitely state that in the case of an upflow the temperature regime of the internal surface of the coil in the post-dryout region is relatively more favourable than for a downflow. Further investigations in a wider range of parameters are required.

CONCLUSIONS

(1) The analysis of the experimental results on the boiling crisis in a coiled steam-generating tube has confirmed that in coiled tubes the critical steam quality is much higher than in straight vertical ducts. With an increase of pressure and mass flow velocity the critical steam quality in coils decreases.

(2) It has been confirmed that heat transfer in the post-dryout region in coils is better than in straight tubes. The temperature jump decreases with an increase of pressure, steam quality and mass flow velocity.

(3) It has been established that the flow direction practically does not influence the magnitude of the critical heat flux at $\rho w \geq 500$ kg m^{-2} s^{-1}.

It is possible to make a tentative conclusion that the temperature regime of the internal surface of a helical coil depends on the flow direction. In the case of upflow the heat transfer in the post-dryout region is higher than for a downflow.

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ETUDE EXPERIMENTALE DU FLUX THERMIQUE CRITIQUE ET DU REGIME POST-ASSECHEMENT DANS DES SERPENTINS

Résumé—On présente les résultats d'une expérimentation sur le régime thermique d'un serpentin avec un mélange eau–vapeur ascendant ou descendant. Les conditions expérimentales correspondent aux domaines des paramètres : pression, $P = 9{,}81$–$17{,}6$ MPa ; débit surfacique $\rho W = 500$–1500 kg m^{-2} s^{-1} ; densité du flux, $q = 100$–1500 kW m^{-2}. A $x > 0{,}2$, les flux thermiques critiques dans le serpentin sont plus élevés que dans un tube vertical droit, tandis que le régime de température d'un tube helicoïdal est moins sévère dans la région de post-assèchement. Le saut de température à la paroi dans cette région change sur la périphérie du tube et il dépend de la direction de l'écoulement.

EINE EXPERIMENTELLE UNTERSUCHUNG DER KRITISCHEN WÄRMESTROMDICHTE UND DES "POST-DRYOUT"-TEMPERATURVERHALTENS VON SPIRALROHREN

Zusammenfassung—Es wird über Ergebnisse einer experimentellen Untersuchung des Temperaturverhaltens eines Spiralrohres mit einem aufwärts- bzw. abwärts strömenden Dampf/Wasser-Gemisch berichtet. Die Versuche wurden in folgendem Parameterbereich durchgeführt: Druck, $P = 9{,}81$–$17{,}6$ MPa; Massenstromdichte, $\rho w = 500$–1500 kg m^{-2} s^{-1}; Wärmestromdichte, $q = 100$–1500 kW m^{-2}. Bei $x > 0{,}2$ sind die kritischen Wärmestromdichten im Spiralrohr viel höher als in einem vertikalen geraden Rohr, während die Temperaturänderung im "post-dryout"-Gebiet weniger stark ausgeprägt ist. Der Wandtemperatursprung im "post-dryout"-Gebiet ist über den Rohrumfang veränderlich und hängt von der Strömungsrichtung ab.

ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ КРИЗИСА ТЕПЛООТДАЧИ И ТЕМПЕРАТУРНОГО РЕЖИМА ВИНТОВЫХ ЗМЕЕВИКОВ

Аннотация—Приводятся результаты экспериментального исследования температурного режима винтового змеевика при подъемном и опускном течении пароводяной смеси. Опыты проводились в следующем диапазоне режимных параметров: давление $P = 9{,}81$–$17{,}6$ МПа, массовая скорость $\rho w = 500$–1500 кг м$^{-2}$ с$^{-1}$, тепловые потоки $q = 100$–1500 кВт м$^{-2}$. В области $x > 0{,}2$ критические тепловые потоки в змеевике значительно выше, чем в вертикальной прямой трубе, а температурный режим канала в закризисной области менее напряженный. Скачок температуры стенки в закризисной области изменяется по периметру трубы и зависит от направления течения.