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Thermal stresses in the heated wall of the minichannel

Abstract

This paper deals with thermal deformations of the heating foil in a heat exchanger with a rectangular minichannel. Flow boiling heat transfer was achieved to maximize the heat transfer coefficient. As confirmed by the experiments, the deformations resulting from the thermal expansion of the heating foil led to local narrowing of the channel, and consequently distorted flow and less effective heat transfer.

Keywords: thermal stress, distortion, minichannel, heat transfer, boiling.

1. Introduction

The aim of the study was to analyse thermal stresses and deformations accompanying flow boiling heat transfer in a rectangular minichannel of a heat exchanger. As flow boiling heat transfer was observed, it was vital to increase the efficiency of the heat transfer process. Thermal stresses were determined for a metal element of the heat exchanger where heat was generated by the flow of an electric current. The element considered in this paper was mounted between two fixed supports. However, with the variable distribution of temperature, different from that recorded during assembly, its displacements led to local narrowing of the minichannel and, accordingly, changes in the conditions of flow. This paper discusses thermal deformations obtained for three different settings of the heat flux supplied to the heated wall. The strains and the distributions of temperature were determined by means of the Ansys Workbench program, which employs numerical methods to solve the heat conduction and displacement equations.

The world's literature on the subject does not provide any information on thermal stresses in exchangers with channels rectangular in cross-section. Researchers concerned with thermal deformations of devices exposed to elevated temperatures have focused on many different problems. One study dealt with thermal stresses in a planar solid oxide fuel cell [1]. The aim of another was to analyze the influence of thermal stresses on the failure of a martensitic stainless steel (CA-15M) roll using numerical methods [2]. In Ref. [3], the researchers show the relationship between the geometry of the pressure vessel and the level of thermal stresses. Position [5] provides a model of a boiler start-up taking account of thermal stresses. In another study [6] thermal stresses were analyzed to optimize steam pipeline and T-pipe heating. Piasecka [7] discusses the influence of the microstructure of the enhanced heated surface on the efficiency of heat transfer during boiling for a refrigerant flowing in minichannels. Thermal deformations occurring on the minichannel surfaces may lead to changes in its geometry, which, in turn, has a considerable effect on the flow conditions and heat transfer in the minichannel, because of a very limited space width.

2. Experimental setup

The most important element of the setup was the test section with two parallel vertically-oriented rectangular minichannels, each 1.8 mm deep, 24 mm wide, and 360 mm long. The schematic diagram of the test section is shown in Fig. 1.

The heating element for FC-72 flowing in both minichannels (1) was a Haynes-230 alloy foil with a thickness of about 0.1 mm (2). The surface of the foil in contact with the fluid was enhanced. The two channels were observed through glass panes. One pair of panes (4) on one side enabled us to monitor changes in the foil temperature using infrared thermography and liquid crystal thermography, respectively. The data obtained from infrared thermography was used to calculate the local heat transfer coefficient. Before the measurements, the foil was coated with black paint (8) to achieve an emissivity of 0.83 [1]. Temperature was measured with an infrared camera in the central, axially symmetric part of the channel (approx. 10 mm × 350 mm). The sides of the channel were reinforced with glass panels to prevent the heating foil from deforming. The other side of the minichannels, where two-phase flow patterns occurred on the enhanced surface of the foil in contact with the fluid (3), was observed through the other pair of glass panes. Pressure converters and K-type thermocouples (7) were installed at the inlet and outlet of the minichannel.

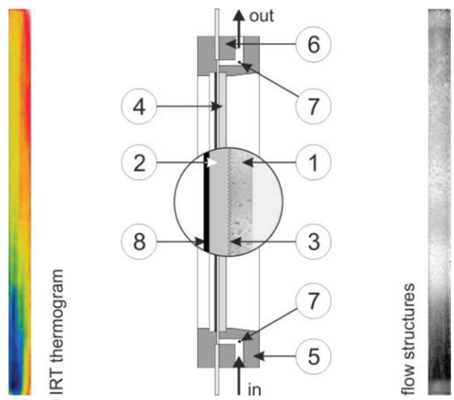

Fig. 1. The schematic diagram of the test section, 1 - minichannel, 2 - heating foil, 3 - enhanced surface of the foil, 4 - glass panes, 5 - channel body, 6 - front cover, 7 - thermocouple, 8 - black paint layer

The data and image acquisition system designed to collect measurement data was made up of a data acquisition station with a computer and appropriate software, an infrared camera, a digital camera, and a digital SLR camera. Another important system of the experimental setup was the supply and control system, which consisted of an inverter welder, a shunt, an ammeter and a voltmeter. Details of the experimental setup can be found in [7]. The minicavities on the enhanced foil surface in contact with the fluid flowing in the minichannel were produced by spark erosion using an arcograph. They were distributed unevenly and their dimensions varied. The craters were usually less than 1 μm deep.

3. Experimental data

Thermal stresses in the heating foil were calculated using data recorded for the heat transfer in the main parts of the test section. From a series of experiments it was found that when the desired pressure and flow rate were reached, there was a gradual increase in the electric power supplied to the heating foil followed by an increase in the heat flux transferred to the fluid in the minichannel. This led to the onset of nucleate boiling and the heat transfer enhancement. The results correspond to steady-state regimes.

A one-dimensional model was used to calculate the local heat transfer coefficient [8]. The model took into account the heat flow direction, which was perpendicular to the direction of the flow of the fluid in the minichannel, see Fig. 2. Thus, the heat transferred to the fluid (see Fig. 2) was equal to the heat generated by the heating foil. The local values of the heat transfer coefficient $\alpha(y)$ in the area between the heating foil and the fluid in the minichannel were calculated from the Robin boundary condition (third-kind boundary condition) on the basis of the measured distribution of temperature on the heating foil and the known local fluid temperature or local saturation temperature. The calculations were as follows:

- for the subcooled boiling region

$$\alpha(y) = q_{w\_IRT} / \left[T_F(y) - T_f(y)\right], \tag{1}$$

- for the saturated boiling region

$$\alpha(y) = q_{w\_IRT} / \left[T_F(y) - T_{sat}(y)\right], \tag{2}$$

where

$$q_{w\_IRT} = q_w - q_{wl}, \tag{3}$$

q_{w_IRT} - heat transferred to the fluid in the minichannel, $T_F(y)$ - local foil temperature measured by infrared thermography, $T_f(y)$ - local bulk liquid temperature determined on the basis of the linear distribution of temperature along the length of the channel from the inlet ($T_{f,in}$) to the outlet ($T_{f,out}$), $T_{sat}(y)$ - local liquid saturation temperature determined on the basis of the linear distribution of pressure along the length of the channel from the inlet (p_{in}) to the outlet (p_{out}), q_w - density of the heat flux supplied to the heated foil, q_{wl} - heat lost to the surroundings in the minichannel, y - distance from the minichannel inlet.

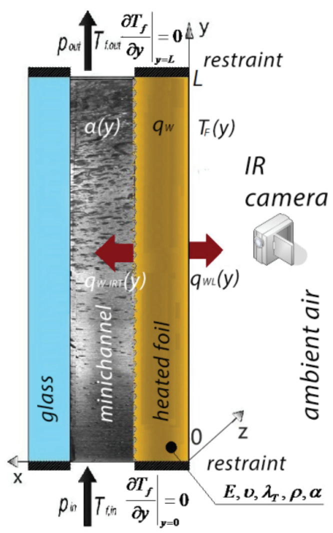

Fig. 2. The schematic diagram of the main elements of the central part of the test section with the minichannel; with the diagram including the boundary conditions and the input data for the Ansys Workbench program

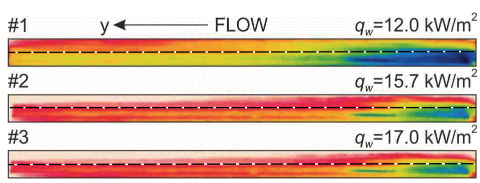

Fig. 3. Infrared thermograms obtained for the heated foil at a mass flux of 145 kg/(m^{2}s), an inlet pressure of 160 kPa, and inlet liquid subcooling of 45 K; settings for three different densities of the heat flux supplied to the heated foil : #1) 12.0 kW/m^2 , #2) 15.7 kW/m^2, #3) 17.0 kW/m^2

4. Results

The equation used to determine the effects of temperature changes on the stress in the heating foil is Hooke's law and its general form can be written as:

$$\sigma_{ij} = 2\mu\varepsilon_{ij} + (\lambda e - \beta\tau)\delta_{ij}, \tag{4}$$

e - the dilation $e = \varepsilon_{kk} = \varepsilon_{xx} + \varepsilon_{yy} + \varepsilon_{zz}$, λ and μ - the Lame constants, β - the thermoelastic constant, σ_{ij} - the stress value, τ - the temperature distribution, δ_{ij} – the Kronecker delta, υ - Poisson's ratio.

The relationships between the elastic constants and the thermoelastic constant can be written as:

$$\mu = \frac{E}{2(1+\upsilon)} \quad \lambda = \frac{2\upsilon\mu}{1-2\upsilon} \quad \beta = \alpha(3\lambda + 2\mu), \tag{5}$$

α - thermal expansion coefficient, E- Young's Modulus.

From the equation of equilibrium of forces $\sigma_{ji,j} + F_i = 0 \quad (i,j = 1,2,3)$ and component of deformations, we can determine the displacement equation using index notation:

$$\mu\nabla^2 u_i + (\lambda + \mu)u_{k,ki} - \beta\tau_{,i} + F_i = 0 \qquad (i = 1,2,3), \tag{6}$$

$$\nabla^2 = \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}, \tag{7}$$

F_i - the unit forces, $\sigma_{ji,j}$ - the stress divergence, μ_i - the displacement, ∇^2 - the Laplacian.

The heat conduction and displacement equations were solved numerically using Ansys Workbench software. By appropriately combining the modules containing mathematical description of physical phenomena and introducing boundary conditions and the experimental data, we obtained the distributions of temperature and the displacement of the heated wall along the channel symmetry axis. The numerical analysis was conducted using a calculation grid of the wall cross-sectional area divided into 43 750 elements and the material constants provided in Table 1. The boundary conditions for the problem are shown in Fig. 2. The results refer to one direction coinciding with the direction of flow. The graphical representation of the simulation results obtained by means of Ansys Workbench are shown in Figures 4, 5 and 6. The influence of heat transfer processes in minichannel on the distortion along the wall for three different densities of the heat flux supplied to the heated foil is presented in Fig. 6. Analysis of heat transfer processes in the minichannel was discussed in [7], [8]. The difference in heated wall temperature along the minichannel length was shown in Figures 5 and 6. It causes changes in the shape of the minichannel up to 40% in relation to the nominal length.

Tab. 1. Hastelloy material constants used in the calculations

Density, ρ	8 970 kg/m^3	Young's Modulus, E	2.11*10^{11} Pa
Poisson's Ratio, v	0.32	Coefficient of Thermal Expansion, α	1.17* 10^{-5} 1/K
Thermal Conductivity, λ_T	8.9 W/(m K)		

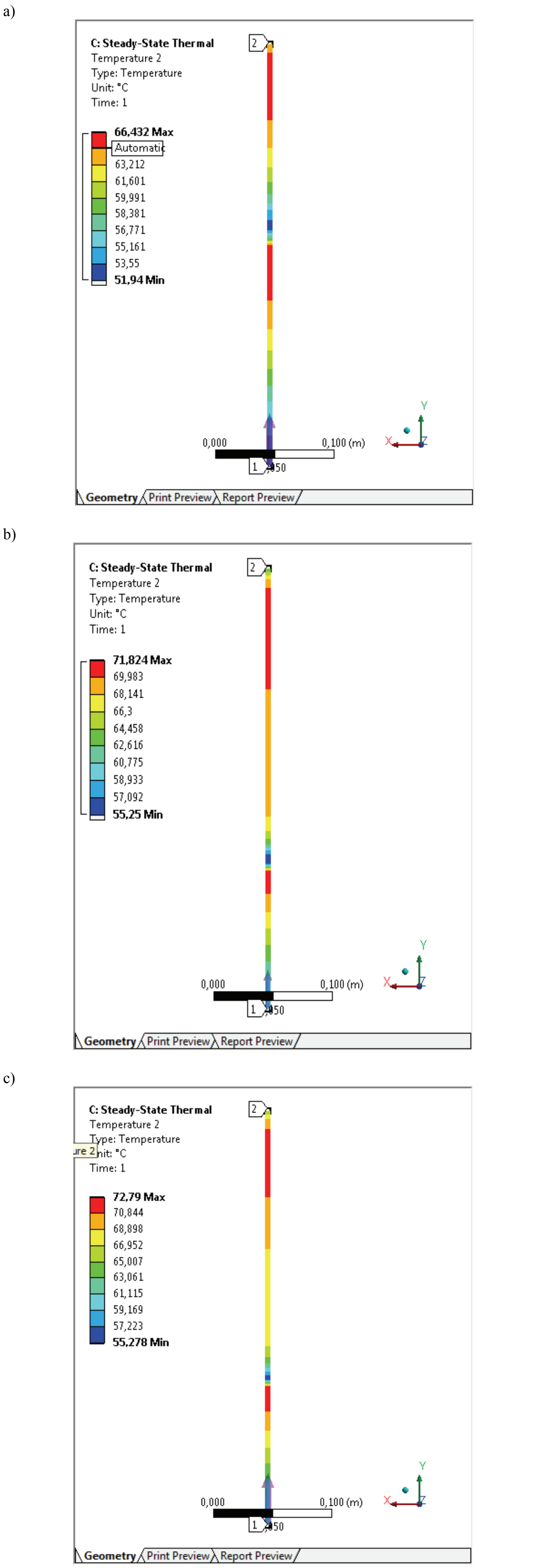

Fig. 4. Distribution of temperature along the heated wall at an increase in the heat flux supplied to the wall; the main experimental parameters as in Fig. 3; data for three heat flux densities: a) 12.0 kW/m^2 , b) 15.7 kW/m^2, c) 17.0 kW/m^2

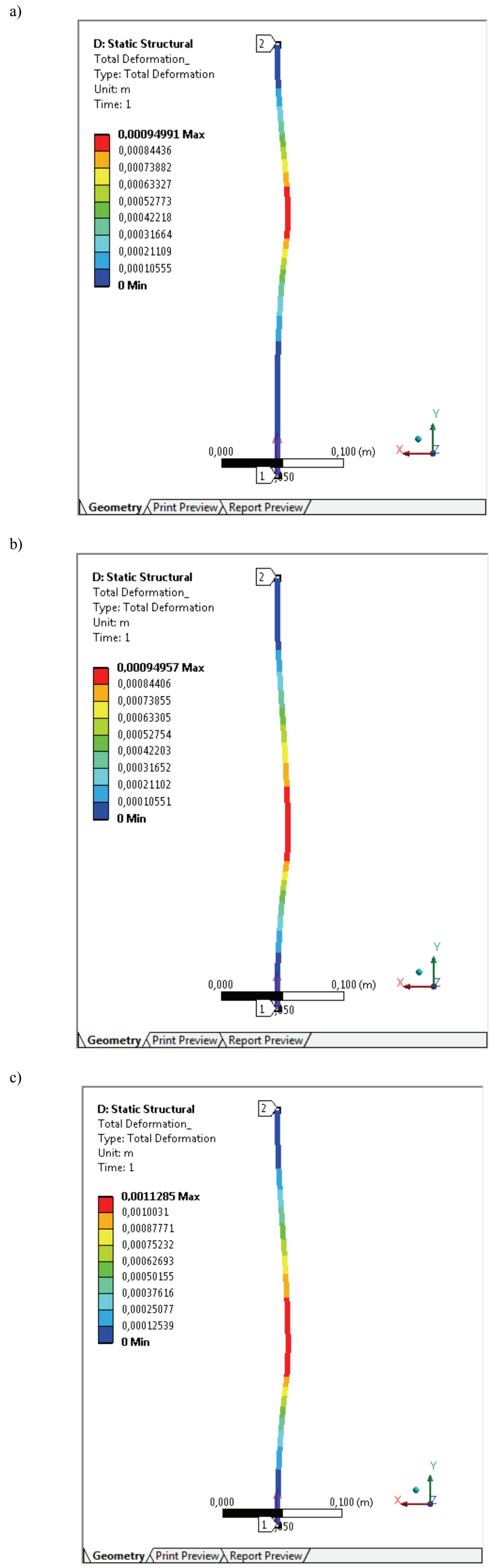

Fig. 5. Thermal distortions along the heated wall of the minichannel; the main experimental parameters as in Figs. 3 and 4

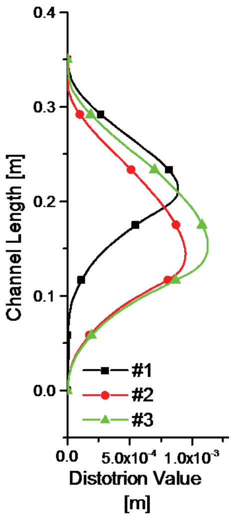

Fig. 6. Thermal distortions along the heated wall of the minichannel; the experimental parameters as in Figs. 3 and 4

5. Conclusions

The geometry of the minichannel may differ from that assumed at the design stage. The differences may result from dimensional inaccuracies relating to the technologies applied to produce the particular structural elements or from assembly errors. In the case of small-size channels these problems seem to be less frequent. It is necessary, however, to take into account deformations of the channel surfaces attributable to changes in the operating conditions. Thermoelastic deformations of the walls on both sides of the minichannel are one of the most important aspects to be taken into consideration in the modelling of the physical phenomena occurring in this minichannel with a refrigerant compressed to a higher temperature. As shown in Figs. 5 and 6, the difference in heated wall temperature along the minichannel length results in changes in its shape, and accordingly its local narrowing of up to 40% in relation to the nominal length. The deformations locally change the rate of the refrigerant flow in the channel and reduce the efficiency of heat transfer. From the results it is evident that the modelling of phenomena describing the mass and energy flow in minichannels requires taking into account also thermoelasticity.

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