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# Single phase pressure drop in minichannels

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## Abstract

This paper presents the results of experiential investigations of pressure drop in minichannels with use of water as the working fluid. The test section was made from stainless steel pipes with internal diameters of 0.55 mm, 0.64 mm and 1.10 mm respectively. A pressure drop was presented per a length unit as the function of Reynolds number both for laminar and turbulent flow. A comparison of the experimental friction factor with the results obtained from theoretical equations of Hagen-Poiseuille and Blasius are presented. The experiments were conducted in range of Reynolds number Re = 30 ÷ 6 400. Contradictory reports concerning a sooner transition to the turbulent flow, or the friction factor values which diverge from those occurring in conventional channels, were not confirmed here.

New experimental data presented in this paper constitute further contribution to the world's database of experimental results which constitute the basis for a verification of the existing theories or attempts to create new computational models.

**Keywords:** Minichannels; Experimental investigations; Friction factor; Laminar flow; Turbulent flow

## Nomenclature

| | | |
|---|---|---|
| $C^*$ | – | normalized friction coefficient Eq. (11), |
| $d$ | – | diameter, m |
| $L$ | – | length, m |
| $\dot{m}$ | – | mass flow rate, kg/s |
| $\Delta p$ | – | pressure drop, Pa |
| Po | – | Poiseuille's number Eq. (10), |
| Re | – | Reynolds number, |
| $w$ | – | mean velocity, m/s |

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$x$ – distance from inlet to pipe, m
$x^+$ – dimensionless length Eq. (1),

**Greek symbols**

$\lambda$ – friction factor,
$\mu$ – dynamic viscosity, Ns/m$^2$
$\nu$ – kinematic viscosity, m$^2$/s
$\rho$ – density, kg/m$^3$

**Subscripts**

$exp$ – experimental
$h$ – hydraulic
$th$ – theoretical
$w$ – inner

# 1 Introduction

Together with the technological advancement, heat exchangers of a size much smaller than that of exchangers generally known as conventional are becoming more and more common. Sample areas of application of this type of exchangers are as follows: car industry – air conditioning systems; electronic industry – cooling of elements generating heat, fuel cells; housing industry: air conditioners, heat pumps. The application of heat exchangers of that type is becoming the more common the more their sizes are reduced, and the manufacturing technology more available.

Heat exchangers in which channels of this type are used have their advantages. These include the following: the possibility to work with higher working medium pressures, a much greater contact zone of the medium with the channel wall in relation to the liquid volume unit, a substantially smaller mass of the applied refrigerant, experimentally confirmed higher heat transfer coefficients, a lower wear of materials, or a smaller weight of the whole system. In the case of a depressurization of the system, the mass of the medium which will get to the environment is very small, which is advantageous both to a decrease of gases which cause the greenhouse effect and degradation of the ozone layer. A drawback of compact heat exchangers is an increase of flow resistances together with a decrease of hydraulic diameter of the channel as a result of relative roughness increase. Another disadvantage is, with favorable conditions, a fluctuation of the flow cause by an abrupt increase of the size of bubbles.

In spite of the existence of numerous experimental and theoretical examinations, a certain number of the main hydrodynamic aspects have not received sufficient research. A short review of the present state of the knowledge is presented concerning a single-phase flow of liquids in minichannels, on the basis of papers published by several authors.

Pahlivan [14] made experimental investigations of the pressure drop of two-phase flow of water-air in channels with a circular section with an internal diameter of 3 mm, 1 mm and 0.8 mm. In the part concerning pressure drop of single-phase flow, he confirmed both for water and air a compliance of Poiseuille's conventional theory for laminar flow and Blasisus theory for turbulent flow with the experimental results for mini-channels.

Celata [3] examined the effect of a wall surface on the behavior of a liquid flowing inside channels with a circular section and internal diameters from 0.326 mm to 0.070 mm. The friction factor obtained could in every case be correctly foreseen by the dependencies valid for conventional channels.

The effect of air „dissolved" in water on, among others, a friction factor during a single-phase water flow was examined by Steinke M. E. [18]. Experimental investigations conducted in parallel channels with a hydraulic diameter of 0.207 mm yielded results which were coincidental with the results of computations of a classical theory.

Lelea [13] conducted experimental investigations and numerical computations of heat transfer and pressure drop of distilled water. The diameters of stainless steel pipes were applied as follows: 0.1, 0.3 and 0.5 mm. The experiments were conducted solely in the range of laminar flow ($\mathrm{Re} \leq 800$). The results of experimental investigations obtained matched well the results of theoretical dependencies from conventional channels.

Coleman [4] conducted investigations concerning two-phase flow R134a in channels with the diameter of 0.83 mm. Initial results of a measurement for single-phase flow in range $\mathrm{Re} = 800 \div 21000$ yielded a compliance with Churchill's correlation with the average absolute error of 3.53%.

Kaminaga [7] presented results of a measurement of the friction factor during a single-phase flow of water in range $\mathrm{Re} = 60 \div 3200$. Both departing from laminar flow and the friction factor values both during laminar and turbulent flow were compliant with the theoretical calculations.

Hwang [6] made an investigation of pressure drop in mini-pipes made from stainless steel. In a review of the literature, he provides a tabular list of studies from which it is evident that the friction factor is larger that it results from theory (5 studies); compliant with the theory (4 studies) or smaller (4 studies). The results of his experiments conducted on pipes with internal diameters of 0.244, 0.430 and 0.792 mm were compliant with the forecasts of the classical theories, while the transition flow started with Re slightly smaller than 2000.

Hetsroni [5] in his survey article presents a list of results of experimental investigations of the single-phase flow through smooth and rough channels with various shapes of cross sections. Hydraulic diameters were from 4.010 mm to 0.003 mm. The boundaries of a transition from laminar to turbulent flow occurred from Re = 300 to Re = 3800, while the value of Poiseuille's number exceeded the theoretical values by even as much as 37%.

Wang [19] examined the friction coefficient during the flow of water and lubricating oil through channels with a circular and rectangular cross sections with hydraulic diameters $d_h = 2.01 \div 0.198$ mm. The experimental values obtained of the friction factor for oil in rectangular channels were 10 - 30 % lower, while for water slightly lower from those obtained in the theoretical way.

Agostini [2] made research into the heat transfer coefficient and pressure drop of liquid R134a flowing in rectangular channels with hydraulic diameters of 1.17 and 0.77 mm. The results of investigation obtained into the flow pressure drop were substantially higher (up to 50%) than those obtained from Shah-London's equation for laminar flow and from Blasius equation for turbulent flow, especially for the channel with greater dimensions. In paper [1], Agostini B. presents the result for flow R134a through eleven parallel channels with a rectangular section ($d_h = 2.01$ mm). As regards the laminar single-phase flow, a good agreement was obtained with Shah and London's correlation, and as regards turbulent flow, with Colebrook's equation.

The paper by Shuai [16] includes results of the measurements of the friction coefficient for single-phase water flowing through channels with a rectangular section, corresponding to hydraulic diameters of 2.67 and 0.8 mm. The values obtained for the larger channel matched the values obtained from Hagen-Poiseuille equation and Blasius equation. The results for the channel with the smaller diameter exceed the theoretical values by even as much as 100%.

The short review of the present state of the knowledge was presented concerning a single-phase flow of liquids in mini-channels, i.e. channels with a hydraulic diameter smaller than 3 mm, serves to confirm the fact that there are divergences in the assessment of its parameters. A problem of a vital importance is an empirical verification whether procedures can be applied, and to what extent, concerning hydrodynamic calculations, which are known and well-checked for a single-phase flow in conventional channels. This applies especially to such issues as a criterion of transition from a laminar to turbulent flow, the possibility of application of Hagen-Poiseuille formula (for a laminar flow,) and of Blasius' formula (for a turbulent flow) for the calculation of the friction factor.

Two reasons was an inspiration to undertake experimental investigations concerning the abovementioned subject matter. The first one will serve to dispel the abovementioned doubts and to obtain an explicit view on the hydrodynamic properties of a single-phase flow of liquid (water) in minipipes. The second one concerns testing of an experimental set-up prepared for the testing of a two-phase flow in minichannels. In the light of the published works by the other authors of experiments of a two-phase flow in minichannels, a resistance of flow and the heat exchange coefficients are different from those from conventional channels. Execution of an experiment with a single-phase flows facilitates an evaluation of the error of the test methodology planned in the scope of the two-phase flow; it can also be stated owing to it that the divergences obtained in this case of the two-phase flow testing are the result of new phenomena in mini-channels, and not an error of the measuring method.

## 2 Test section

The experimental set-up consists of a particulate filter (1), a magnetic mini gear pump with a by-pass (2), and adjustment valve, a mass Coriolis flow-meter (3), a test section (4, 5, 6) connected to a data acquisition system (7, 8) and an outlet tank (9). Fig. 1 shows the experimental set-up.

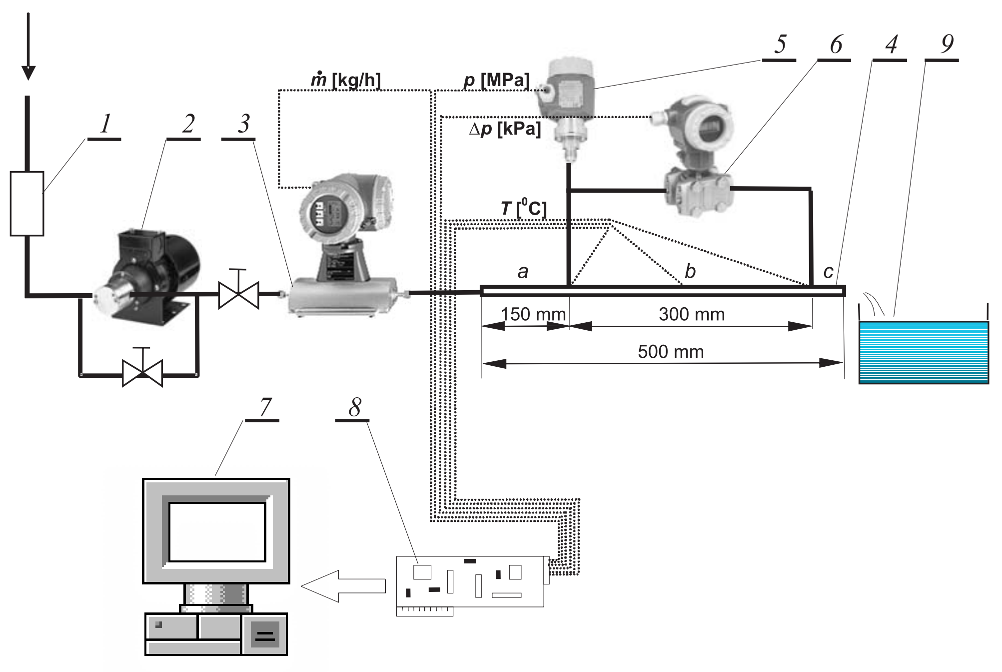


Figure 1. Experimental set-up: 1 – filter, 2 – pump with valves, 3 - flow meter, 4 – test section, 5 – pressure transducer, 6 – difference pressure transducer, 7 – measurement card, 8 – computer, 9 – tank: a, b, c – zones of the test section.

Water from the tap flew through the particulate filter. Its purpose was to prevent both the flow meter and the test section from being damaged. Water without particulates flew by the by-pass or, if a higher pressure was required on the input to the test section, through a magnetic pump (D Series Magnetically

Coupled Gear Pump manufactured by Tuthill Corporation) to the flow-meter. Coriolis mass flow-meter (Promass 80A manufactured by Endress+Hauser) was used with the measuring range of $0 \div 20$ kg/h. Measuring accuracy of this device is ±0.15 % of the measured value. With the flow intensity of 20 kg/h, this gives a measuring error of ±0.03 kg/h. Further, the water with the already known flow rate flew to the test section.

The test section was composed of exchangeable stainless steel pipes. The whole pipe length was 500 mm, while its internal diameters (according to the manufacturer) were 0.55 mm, 0.64 mm and 1.10 mm respectively. In accordance with the classification proposed by Kandliakar [8–11], channels with these diameters can be considered to be mini-channels. On the pipes, in the distances of 150 mm from the front and 50 mm from the end of the mini-channel, small cuts were made with a milling cutter. The cuts were made in compliance with the remarks concerning execution of experiments in channels with diameters smaller than those conventional ones (Kandlikar [10]), and therefore in such a manner so as to make it possible to receive the pressure impulse while not interfering the flow inside the mini-channel. In this way, the whole 500 mm mini-channel was divided into 3 sections. The first 150 mm section "a" stabilized the flow, the second – insulated – 300 mm section "b" constituted the measurement section, and the third 50 mm section was the outlet section – "c". Water, while leaving the mini-pipe, flew into an open tank, and so its pressure corresponded to the surrounding atmospheric pressure.

In accordance with the equation (Steinke [17])

$$x^{+} = \frac{x}{d_h \cdot \mathrm{Re}} \tag{1}$$

where $x^{+}$ is a non-dimensional length, $x$ is a length measured from the channel inlet; the flow can be considered to be fully developed when non-dimensional length $x^{+} = 0.05$ ($x^{+}= 0.055$ – according to Celata [3]). Accepting $\mathrm{Re} = 2000$ and the hydraulic diameters applied during the examinations, i.e. $d_h = 0.55$, 0.64 and 1.10 mm the required length of the hydraulic development phase is 55 mm, 64 mm and 110 mm respectively. A section of 150 mm used on the stand is sufficient to state that in the measuring section with deal with a fully developed flow.

Through the $T$-junction, which was fixed at the first orifice, it is possible to measure the pressure at the input to the measuring section and one of impulses necessary for the measurement of the pressure drop. The second impulse, through the $T$-junction, is received from the next orifice. For the pressure measurement, a piezoelectric sensor with a transducer (Cerabar M PMP41 manufactured by Endress+Hauser) was applied. This sensor has a measuring range of $0 \div 1$ MPa, and its measurement error does not exceed 0.2% of the measurement range. This gives a pressure measuring error of $\pm 2$ kPa.

A pressure drop on the pipe length was measured with a piezo-resistive pressure difference sensor with a transducer (Deltabar S PMD75 manufactured by Endress+Hauser). The factory measuring range of the device is $0 \div 500$ kPa. The measuring accuracy for this device is 0.075 % of the measuring range. In the measuring range of $0 \div 500$, the measuring error is $\pm 0.375$ kPa.

The flow-meter, the pressure sensor and the pressure difference sensor were individually calibrated by the manufacturer.

The temperature of the liquid flowing inside a mini-pipe on the length of the measuring section was measured with three K type thermocouples with a thickness of 0.2 mm. The thermocouples were individually calibrated in the range of $10 \div 30^{\circ}$C with an accuracy of $\pm 0.1^{\circ}$C, and were soldered on the length of 300 mm of the test section; right after the beginning, in the middle of the length, and right before the end. The whole was separated from the environment with a 10 mm thick silicone insulation. The flow process was considered to be adiabatic as water had a temperature slightly higher than the ambient temperature, which was the result of the work of the pump. As a consequence a constant temperature on the whole length of the test section was kept. Thermal and physical properties of the liquid were read with the use of [15].

Data from the mass flow-meter, pressure on the input of the measuring section, pressure drop on the length of the test section of the mini-pipe and temperature, were all registered by the data acquisition system made up of 16 bit, 1 MHz DaqBoard 3005 measuring card working with a PC.

The measurement procedure comprised the following stages. The mass flow rate was set with the aid of a valve located before the flow-meter. Depending on the required mass flow rate, and so on the pressure of the input to the measuring section, water flew out under the influence of the pressure in the laboratory in the water system, or was removed by the applied gear pump. Next, the researchers waited the required time, i.e. until the readings of pressure difference sensor settled. After this time, depending on the pipe applied and the current flow rate, in settled conditions, the recording of measurement started. The measurement lasted ca. $10 \div 20$ s. In this time, data from the measuring devices were registered every 0.5 s. The average from the individual results gave quantities corresponding to a given measurement, which was further used in the calculation procedure.

## 3 Data reduction

Reynolds number and friction coefficient $\lambda$, were determined for each case. Reynolds number was calculated from its definition equation with the following form:

$$\mathrm{Re} = \frac{w \cdot d_w}{\nu} \tag{2}$$

where $\nu$ is the kinematic viscosity. The measured mass flow rate $\dot{m}$ can be written as follows:

$$\dot{m} = \rho \cdot w \cdot \frac{\pi d_w^2}{4}, \tag{3}$$

where $\rho$ – water density, $w$ – its average flow velocity.

Once the average water velocity has been determined from Eq. (3) and put in Eq. (2), the following is obtained:

$$\mathrm{Re} = \frac{4 \cdot \dot{m}}{\pi \cdot \mu \cdot d_w}, \tag{4}$$

where $\mu$ is the dynamic viscosity.

From Darcy-Weisbach equation:

$$\frac{\Delta p}{L} = \lambda \cdot \frac{w^2 \cdot \rho}{2 \cdot d_w} \tag{5}$$

the friction factor $\lambda$ is determined:

$$\lambda = \Delta p \frac{d_w}{L} \frac{2}{\rho \cdot w^2} \tag{6}$$

while the value $\left(\frac{\Delta p}{L}\right)$ determines the water pressure drop referred to a length unit of the mini-channel. Equation (6), considering relation (3), obtains the following form:

$$\lambda = 0.125\pi^2 \Delta p \frac{d_w^5}{L} \frac{\rho}{\dot{m}^2} \tag{7}$$

Theoretical value of the friction coefficient accepted in calculations for conventional channels was determined for laminar flow from the following formula:

$$\lambda = \frac{64}{\mathrm{Re}} \tag{8}$$

where $A$ – Poiseuille's constant ($A = 64$ for channels with a circular section).

For turbulent flow, the theoretical value of the friction coefficient was determined from Blasius formula:

$$\lambda = 0.316\mathrm{Re}^{-0.25} \tag{9}$$

In an analysis of the results of experimental tests, Poiseuille's number defined with product was applied [5, 17]:

$$\mathrm{Po} = \lambda \cdot \mathrm{Re} \tag{10}$$

as well as a normalized friction coefficient described with equation [5, 12, 17]:

$$C* = \frac{(\lambda \cdot \mathrm{Re})_{\mathrm{exp}}}{(\lambda \cdot \mathrm{Re})_{th}}. \tag{11}$$

# 4 Results and discussion

## 4.1 Resistance of laminar flow

Figure 2 presents the results of experimental investigation of flow resistance depending of Reynolds number for water flow in three circular minichannels with internal diameter $d_w$ = 0.55, 0.64 and 1.10 mm. The investigations were made in the range of Reynolds number Re = 30 ÷ 6400. The flow resistance was expressed with the aid of value $\left(\frac{\Delta p}{L}\right)$, i.e. a water pressure drop referred to the unit of the length of the measuring section [kPa/m]. An increase of Re, reflecting an increase of the water flow intensity, resulted in an increased flow resistance. For the mini-channel diameter $d_w = 0.55$ mm, the flow resistance value was four times greater than for a mini-channel with a diameter of 1.10 mm, with the same flow intensity. It is to be noted that on the transition from the region of laminar flow into turbulent flow, i.e. the so-called transition area, the nature of the flow diagram $(\Delta p/L) = f(\mathrm{Re})$ is subject to change. Similar trends were observed in paper [3].

Figures 3÷5 present diagrams of an experimental dependence of frictional factor $\lambda$ vs. Reynolds number, with the emphasis on the laminar flow range, for pipe mini-channels with internal diameters of 0.55 mm (Fig. 3), 0.64 mm (Fig. 4) and 1.10 mm (Fig. 5) respectively.

For all the cases of water flow in minichannels, similarly as in the case of flow in conventional channels, a linear drop of the value of frictional factor $\lambda$ has been observed in the laminar flow. All the results of experimental investigations were in a deviation being not greater than ±10% (±5% for diameter $d_w$= 0,55 mm), as referred to the theoretical values of coefficient $\lambda_{th}$, described with Eq. (8). The continuous line in Figs. 3÷5 is a dependence $\lambda_{th} = f(\mathrm{Re})$.

There is a clear-cut boundary of a transition from the laminar flow into turbulent flow (a transitory area) described with Reynolds number equal to 2000. In this transition area, the nature of dependence $\lambda = f(\mathrm{Re})$ is subject to change, while once this threshold has been crossed, there occurs a slight increment in

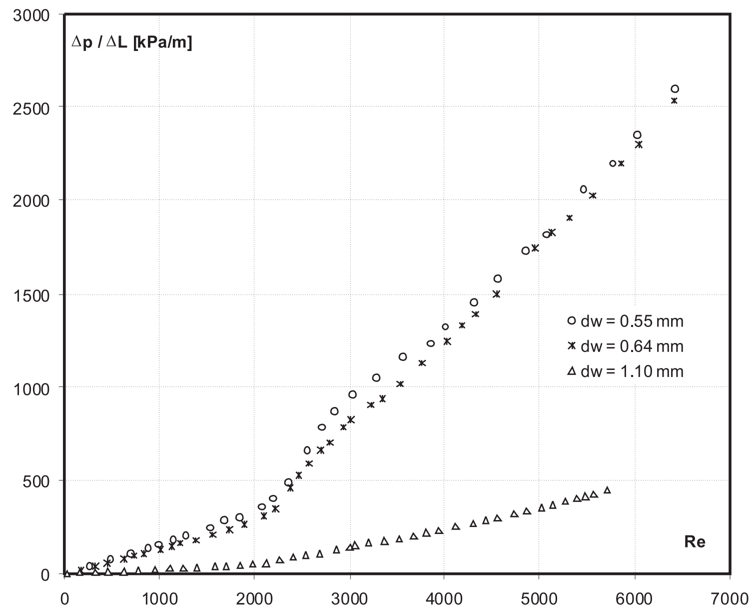


Figure 2. Results of experimental investigations of dependencies of water flow resistance from Re number in pipe minichannels with internal diameters: 0.55, 0.64 and 1.10 mm.

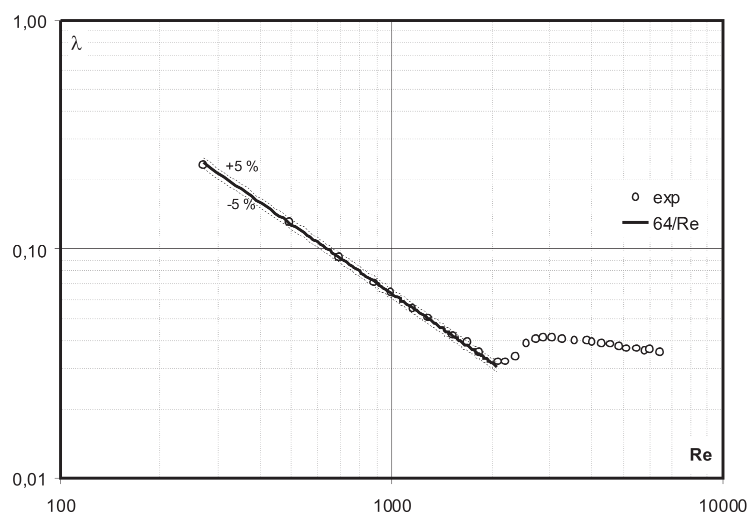


Figure 3. Results of experimental investigations dependence of frictional factor $\lambda$ vs. Reynolds number for water flow in laminar flow in a pipe minichannel with internal diameter $d_w = 0.55$ mm.

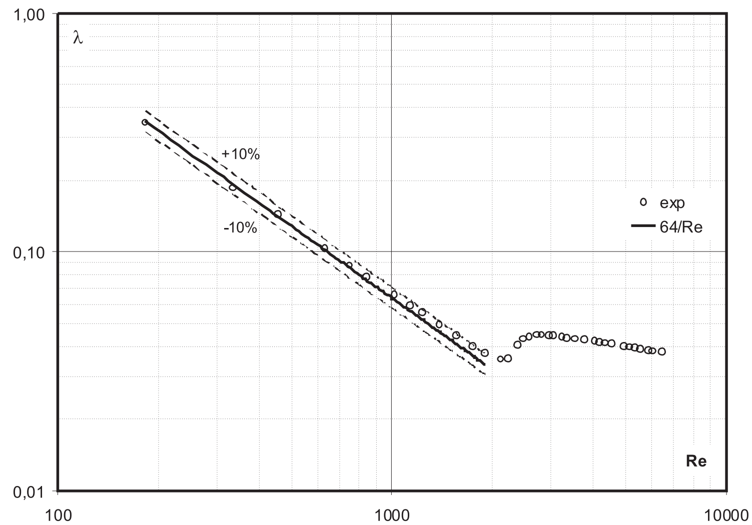


Figure 4. Results of experimental investigations dependence of frictional factor $\lambda$ vs. Reynolds number for water flow in laminar flow in a pipe minichannel with internal diameter $d_w = 0.64$ mm.

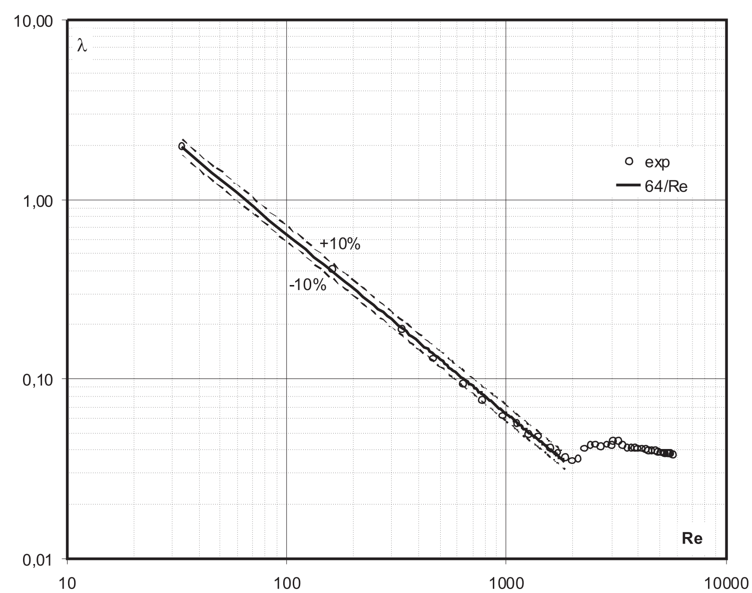


Figure 5. Results of experimental investigations dependence of frictional factor $\lambda$ vs. Reynolds number for water flow in laminar flow in a pipe minichannel with internal diameter $d_w = 1.10$ mm.

the value of coefficient $\lambda$, followed by its linear drop. The course of dependences $\lambda = f(\mathrm{Re})$ during a laminar flow of water in mini-channels is not contradictory to its comparable course for conventional channels. Figure 6 presents an experimental dependence of Poiseuille's number (Po) – equation (10) vs. Re number.

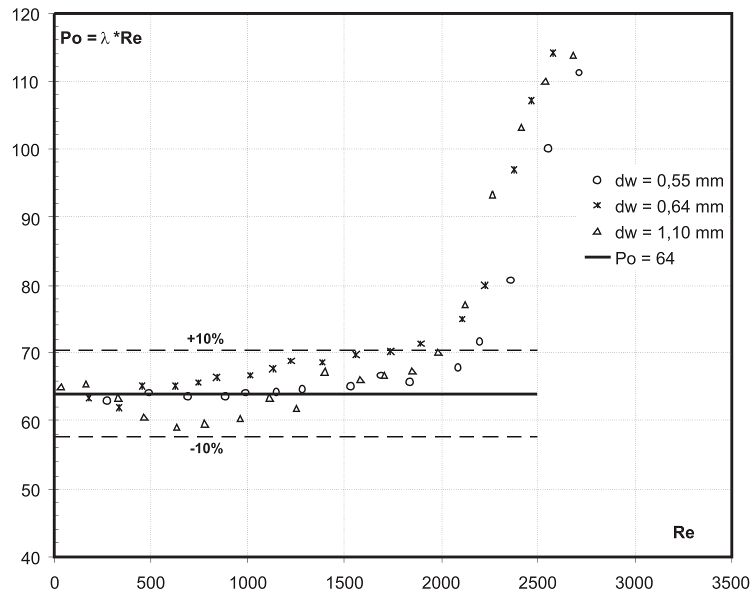


Figure 6. Experimental dependence of Poiseuille's number (Po) vs. Reynolds' number (Re).

Figure 7 presents dependence of normalized friction coefficient $C^*$ from Re number. This coefficient was defined with equation (11), and thus it is described with a quotient of experimental Poiseuille's number ($\mathrm{Po}_{exp}$) and its theoretical value $\mathrm{Po}_{th} = 64$. Experimental values of the normalized friction coefficient were in the range $0.9 < C^* < 1.1$, which should be considered as a very good compliance of the results of research into flow resistance in mini-channels for the range of laminar flow with Hagen-Poiseuille law, which hold true for conventional channels. It should be emphasized that the average values of experimental Poiseuille's number, for laminar ranges, were $\mathrm{Po}_{exp} = 64.51$ – for $d_w = 0.55$ mm. $\mathrm{Po}_{exp} = 66.96$ – for $d_w = 0.64$ mm oraz $\mathrm{Po}_{exp} = 63.98$ – for $d_w = 1.10$ mm respectively, which gives respective averages of the value of normalized frictional coefficient $C^* = 1.008$ $(d_w = 0.55 mm)$, $C^* = 1.046$ ($d_w = 0.64$ mm), $C^* = 0.9997$ ($d_w = 1.10$ mm).

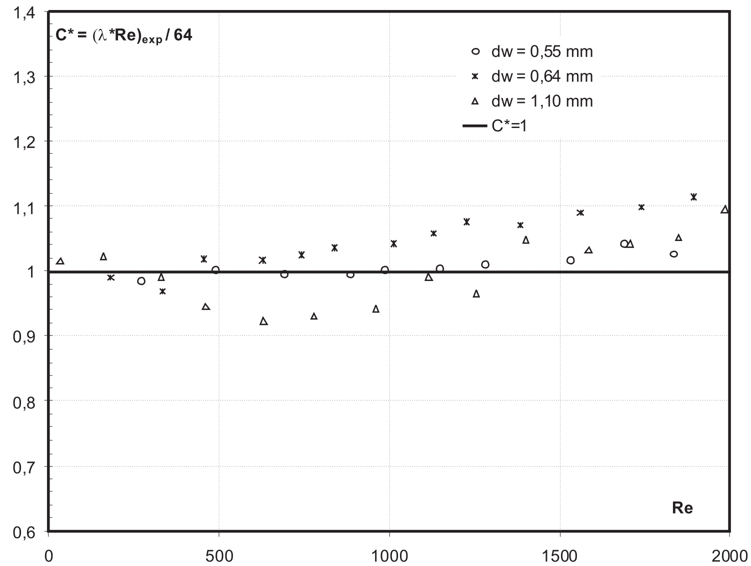


Figure 7. Results of experimental investigations of normalized frictional coefficient $C*$ vs. Reynolds number for water flow in pipe minichannels in laminar flow.

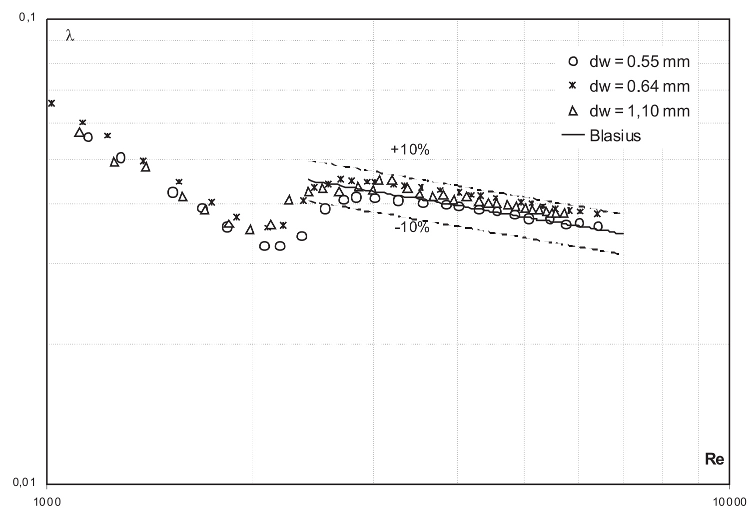


Figure 8. Results of experimental investigation of friction factor $\lambda$ vs. Reynolds number in the range of turbulent flow.

## 4.2 Resistance of turbulent flow

Figure 8 presents the results of experimental investigations of friction factor $\lambda$ vs. Reynolds number, made for water flow in pipe minichannels in the range of turbulent flow (the transition region and fully turbulent flow). Also, a diagram of dependence $\lambda_{th} = f(\mathrm{Re})$ is enclosed with a continuous line in Fig. 8, representing the frictional resistance coefficient described with Eq. (9). A linear course of this dependence is characteristic to the area of hydraulically smooth conventional channels, i.e. such channels where the thickness of the laminar boundary sublayer is greater than the height of roughness on the channel wall. The results of research for minichannels exhibit a consistency of this type of channels with conventional ones.

## 5 Conclusions

1. On the basis of the results of experimental investigations of a single-phase water flow in tubular mini-channels with internal diameters: 0.55 mm; 0.64 mm and 1.10 mm, made from stainless steel, the following conclusions have been made:

    - transition from laminar to turbulent movement occurred at a critical number Re = 2000;
    - compatibility (±10%) was obtained between the frictional resistance coefficient determined experimentally in the range of a laminar flow and calculated according to Hagen-Poiseuille formula;
    - compatibility (±10%) was obtained between the results of experimental investigations and the results of calculations from Blasius' formula applied to a turbulent water flow.

2. It was found, in the tested area of a single-phase water flow in minipipes, that the formula applied so far in the calculations of hydraulically smooth conventional tubular channels can be also used in minichannels. It is justifiable that these dependencies can be applied in a much wider range on stipulation that we deal with minichannels with a smooth surface.
3. The opinions held by other authors who indicated the need of a modification of the formula applied for conventional channels were not confirmed here.
4. No influence has been reported of the type of the material from which the minichannels are made on the flow resistance coefficient.
5. Some of the different opinions of other authors on the hydrodynamic assessment of the single-phase liquid flow in minichannels may be the result of the following:

    - roughness of the internal surface of minichannels being the result of a manufacturing process which is improper for minichannels;

- errors of the methodology of experimental tests which does not take into account the possibility of the occurrence of the so-called internal friction medium heating effect. Such an effect, for the conditions of the experiment conducted, is revealed with hydraulic diameters below 0.2 mm;
- inaccuracies in the measurement of the minichannels hydraulic diameter, which has a significant influence on the assessment of the flow resistance coefficient Eq. (7).

6. The tests results obtained confirmed the validity of the methodology and equipment applied for future tests, conducted during two-phase flows, which portends well for their correctness.

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