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ENTRANCE LOSS COEFFICIENTS IN PIPE HYDRAULIC SYSTEMS

The entrance loss coefficients have been investigated for four, most frequently occurring geometric shapes of inlets placed on the reservoir bottom. The minor (entrance, inlet) losses coefficients determined in the paper, having not been investigated experimentally so far, may be used, for example, to correctly dimension throttling pipes. Throttling pipelines are commonly used in sewage systems for limiting the volume flow rate of sewage channelled to sewage treatment plants from storage reservoirs, storm overflows or rainwater separators during torrential rainfall.

1. INTRODUCTION

The possibility to find correct values of entrance resistance coefficients of pipes in hydraulic systems given in available manuals, handbooks or standards and similar materials is only illusory. This is no surprise, since the coefficients are determined by specific shapes of a given minor loss and, in general, depend on the Reynolds number (*Re*). However, in practice, the Reynolds number is so high that the *K* coefficient can be constant. The paper presents the results of investigation of four specific shapes of pipe inlets typical of hydraulic (particularly sewage) systems, which however, have not been hydraulically investigated so far.

The sewage flow from a number of sewage facilities (storage reservoirs, storm overflows, separators) occurring during torrential rainfall must be limited prior to its introduction to the sewage treatment plant. To this end various types of flow regulators are used. However, most often this is realized by means of a pipeline working under pressure called a throttling pipe [1, 2]. The throttling pipe is an example of liquid flow regulator which makes it possible to empty a reservoir completely because it

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is placed on its bottom level. Minor losses on the inlet and outlet of a throttling pipe – apart from linear losses – influence the selection of the length and diameter of the pipe which should ensure the assumed distribution of volume flow rates on a storm overflow [3–5], or in a light liquid separator with inner by-pass channels [6].

The calculation of dimensions (diameter and length) of a throttling pipe is based on the phenomenon of liquid flow between two open reservoirs and connected by means of a pipeline operating under elevated pressure. Its correct dimensioning requires the knowledge of the linear (f) and minor (K) loss coefficients because they appear in the Darcy–Weisbach formula which is used to calculate energy losses in closed conduits. Therefore, the energy loss head $\Delta h_{l,d}$ occurring in liquid flow with the mean velocity u through the pipeline of the diameter d and length l amounts to

$$\Delta h_{l,d} = \left(f\frac{l}{d} + K + \alpha \right)\frac{u^2}{2g} \tag{1}$$

These are the entrance resistance of the coefficient K and outlet resistance of the coefficient α. It should be noted that the entire kinetic energy is lost on the outlet of the throttling pipe [6], thus, the outlet loss head is $\alpha u^2/2g$ (α is the kinetic energy correction factor).

As can be seen from Eq. (1), the knowledge of correct values of coefficients f, K and α is essential in the procedure of dimensioning of the throttling pipe. Thus, the friction factor f, which depends on relative roughness k/d (where k is a mean wall roughness head) and the Reynolds number and hence $f = f(k/d, Re)$, is assumed according to known formulae or from nomograms [8]. In the case of plastic sewage pipes of the wall smoothness comparable to glass the friction factor can be then calculated from the Blasius formula [7–11]:

$$f = \frac{0.3164}{\sqrt[4]{Re}} \wedge Re = \frac{ud}{\nu} = \frac{4q_V}{\pi d\nu} \tag{2}$$

where: ν is the kinematic viscosity (it was calculated using the formula making it dependent on the water temperature) and q_V is volume rate of flow.

As already mentioned, the entire kinetic energy is lost on the outlet of the throttling pipe, therefore the outlet loss head is $\alpha u^2/2g$. However, in many textbooks and publications [7, 10, 12] the outlet loss coefficient (α) is assumed equal to 1. It is nearly consistent with reality but only for a high turbulent flow. Whereas, Szewczyk [13] gave the following explicit dependence for the kinetic energy correction factor valid in the wide range of $Re \in [2.8\times10^3, 36\times10^6[$, for the flow in smooth hydraulic pipes

$$\alpha = 1 + 0.101\left(\frac{10}{\ln Re}\right)^6 - 0.107\left(\frac{10}{\ln Re}\right)^4 + 0.113\left(\frac{10}{\ln Re}\right)^2 \tag{3}$$

In order to illustrate the course of the dependence $\alpha(Re)$, a number of values of α calculated from Eq. (3) were given, that is: $\alpha(\{5, 10, 10^2, 10^3\}\times 10^3) = \{1.22, 1.15, 1.07, 1.04\}$.

The selection of values for the inlet loss coefficient K encounters a number of difficulties, since literature data differ by as much as 100%. Typically, the Borda–Carnot formula was employed here [7, 10, 14, 15] and for the outflow from a large reservoir to the throttling pipe K is obtained as equal 0.5. According to the model testing on lateral overfalls with throttled outflow carried out by Kallwass [16, 17], values of the outlet loss coefficient amount to: $K = 0.25$, for the round-edged inlet, or $K = 0.35$ for the sharp-edged inlet. However, Imhoff and Imhoff [18] and the TGL standard [19] recommend to assume $K = 0.40$, if edges are technically sharp, but according to the DVWK-ATV-A 111 guidelines [1] give $K = 0.45$.

The reservoir wall containing the throttling pipe can be flat and perpendicular to the pipe axis, but it may also contain bevels – guides (usually at the angle of $\pi/4$ rad), and finally, the reservoir bottom may contain invert channels (typically of the depth of $d/2$). Furthermore, the influence of the height and width of a liquid jet in front of a pipe inlet have not been investigated.

2. THE IDEA OF ENTRANCE LOSSES MEASUREMENT, INVESTIGATION PROGRAM AND TESTING STATION

Figure 1 shows an idealized, axially symmetrical case of laminar and turbulent liquid flows through a straight pipe of the length more than l_e and diameter d connecting two quasi infinite liquid areas (areas of recirculation behind the pipe inlet were omitted for simplification).

The average flow velocity in the pipe amounts to u, whereas the shape of velocity distribution is formed at the entrance length l_e obtaining an unchangeable shape, characteristic of laminar or turbulent flow, only behind this length. Therefore, on the length l_e the energy grade line (EGL) drop is descending to become steady behind the entrance length. As can be seen, the minor loss is spread on the entire entrance length l_e, which in laminar flow amounts to approximately $0.06Red$ and $l_e \approx 40d^*$ in a turbulent flow. For practical reasons, energetic losses in the conduit are shown in the form of a simple sum of two components: minor losses and friction losses changing real curves EGL and HGL (the hydraulic grade line) into their apparent courses, which were marked with dotted lines in Fig. 1. As it can be seen in the figure, K cannot be well determined based on differential pressure $p_0 - p_i$ measurements in front of the inlet and in the section $]0, l_e[$, especially, when the value of K being measured is as

*Some authors give different values, e.g. according to Schlichting [20] $l_e \in [25, 40]d$ (in a turbulent flow).

small as in the investigated case. A method similar to the one, forced by many authors, can be used to determinethe minor losses coefficient of fittings offering a high resistance, e.g. a flow regulating valve. Then, two measurement sections are located close in front of and behind a fitting being investigated and their velocity heads are assumed identical, whereas friction losses are typically omitted. Such assumptions are not acceptable in the investigated case, where K is ca. 0.5 and f is ca. 0.02, and if the measurement is performed in a typical distance of $5d$ behind the inlet*, then friction losses constitute 0.2 of minor losses. It should be emphasized that the subject of the investigations was a flow far more complex than the one shown in Fig. 1. Namely, there is no axial symmetry of flow and recirculation areas are present, which were not accounted for in the figure.

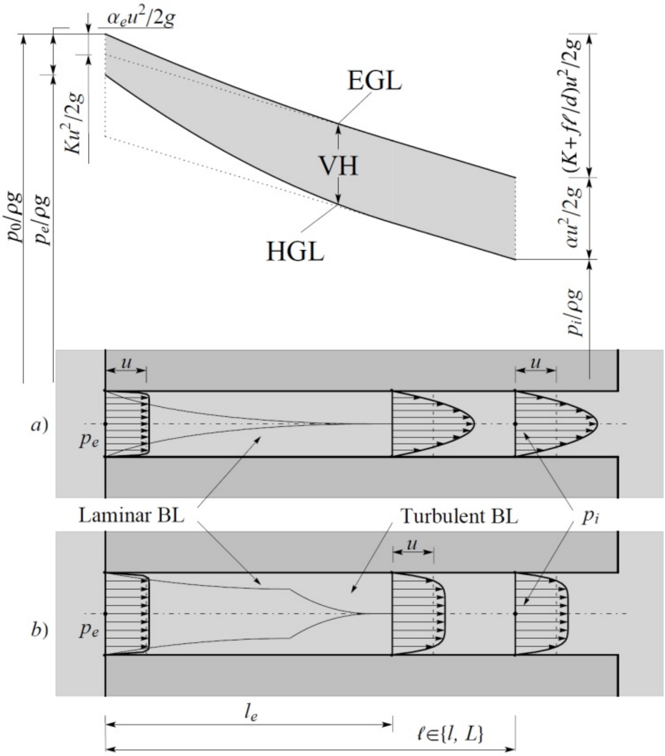


Fig. 1. Laminar (a) and turbulent (b) flows in the entrance of the tube,
EGL – the energy grad line, VH – the velocity head, HGL – the hydraulic grade line,
BL – the boundary layer, α – the kinetic energy correction factor ($\alpha_e \approx 1$),
l_e – the entrance length, p_k ($k \in \{i = 1, 2; 0, e\}$) – the liquid pressure

* Notice that additionally the measurement section may be located beyond a recirculation zone and in the section $]0, l_e[$ the kinetic energy correction factor α is unknown [21].

A similar explanation of the shape of the phenomenon velocity of distribution formation and interpretation of inlet losses are given by White [10] and Szewczyk [13].

The investigations aimed at determining the influence of geometrical shapes most frequently used in engineering practice forming reservoir sections adhering to the inlet of the pipe on its inlet resistance. The investigations covered:

- measurements related to the identification and calibration of model elements, which include: the inner diameter of polypropylene pipes used to produce the throttling pipe, the hydraulic characteristic of a triangular measuring overfall and physicochemical properties of circulating water used for the investigations,
- measurements related to the implementation of the fundamental paper objective, that is: determination of the piezometric pressure heads in hydrometric sections, measurements of water volume flow rates (by means of the overfall) for varying depths of reservoir filling and for different shapes of its inlet section adhering to the throttling pipe and variable heights (z_0) and widths (b) of the water jets, and finally, measurements of water temperature.

The inner diameter of polypropylene pipeline being investigated was determined by measuring the volume of water filling a pipe section of the known length, obtaining d = (71.0±0.1) mm. The hydraulic characteristic of the triangular measuring overfall (of the angle of π/2 rad) were determined by the volumetric method, establishing its flow coefficient at μ = 0.600. The physicochemical properties of water used in the investigations were verified at the beginning, in the middle and at the end of investigations, while the composition of circulating water during the test proved to be stable.

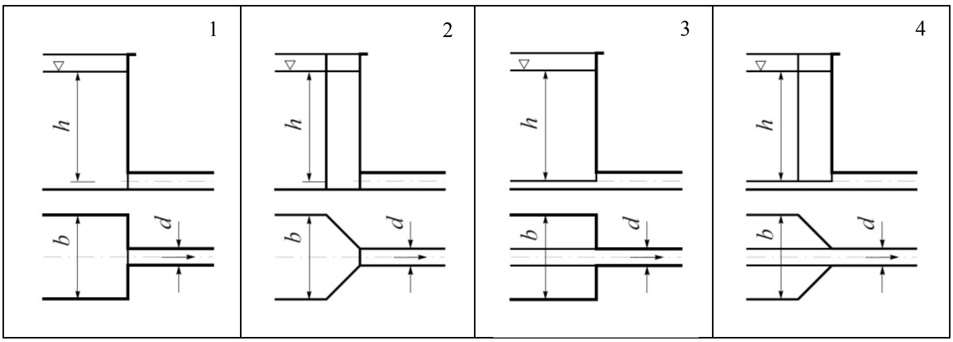


Fig. 2. The shapes of pipe inlets (1–4): vertical cross-sections and horizontal projections ($b/d \in$ {1.25, 2.5, 5, 10}, $h/d \in$ {2.5, 5, 10, 19})

The testing program covered four basic options of geometrical shapes of reservoirs in the section adhering to pipe inlets that is (Fig. 2):

1) a flat wall perpendicular to the throttling pipe axis with sharp-edged inlet, without an invert channel on the reservoir bottom,

2) an inlet to the pipe in the form of guides produced from vertical, flat walls convergent at the angle of π/4 rad in the throttling pipe direction (sharp-edged), without an invert channel on the reservoir bottom;

3) a flat wall perpendicular to the pipe axis with a sharp-edged inlet and invert channel of the depth of $d/2$ in the reservoir bottom;

4) an inlet to the pipe in the form of guides produced from vertical, flat walls convergent at the angle of π/4 rad in the pipe direction (sharp-edged) and with an invert channel of the depth of $d/2$ in the reservoir bottom.

Figure 2 shows investigated geometrical shapes for pipe inlets. Each of them includes four suboptions of the reservoir width (b) on the outflow to the pipe, that is: $b/d \in \{1.25, 2.5, 5, 10\}$. Whereas, each suboption included four measurement runs of the relative height of reservoir filling: $h/d \in \{2.5, 5, 10, 19\}$. Each run contained at least 10 measurements of the volume flow rates and simultaneous reading out of piezometric pressure heads (z_i) and water temperature. The diagram of the testing station used to carry out model tests in question is shown in Fig. 3.

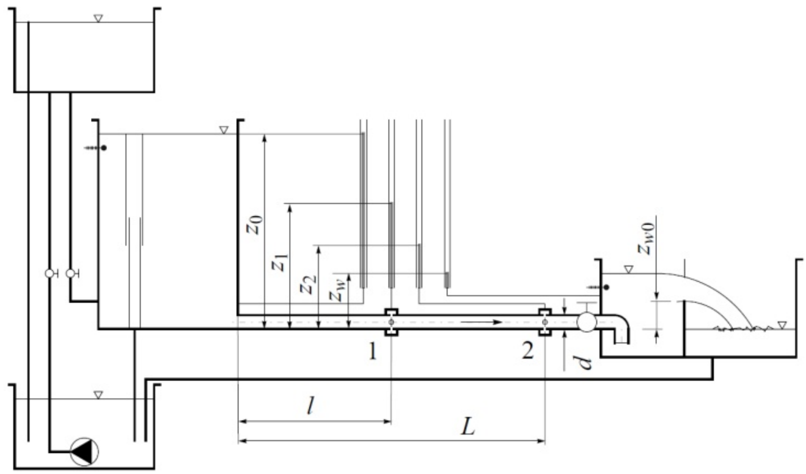


Fig. 3. The testing station (description in the text)

The testing section was produced from connection bell pipes of the mounting lengths of 1.0 m. Two hydrometric sections were placed on the testing pipeline: the first was located in the distance of $l = 60d$, while the other in the distance of $L = 110d$ behind the pipeline inlet. Thus, the turbulent flow was fully developed in the hydrometric sections, which occurs when water flows the distance of approximately $40d$, counting from the pipe inlet. The straight section of the testing pipeline behind the second hydrometric section was of the length of ca. $20d$, which was followed by a ball valve, and then, by the reservoir with a measuring overfall (triangular).

The adjustment of the valve, as well as of other valves shown in Fig. 3, made it possible to set required parameters (h/d, q_V). The remaining values of quantities necessary to determine the loss coefficients (f, K) and Reynolds number (Re) were read out from piezometers (z_i, $i \in \{0, 1, 2, w\}$), and water temperature was measured in two locations shown in Fig. 3. Water was supplied to the piezometers by means of circuit pressure delivery chambers to which pressure signal was provided through 4 piezometric holes (∅3 mm), drilled precisely and symmetrically in the pipe circumference. The obtained volume flow rates were in the range]1.45, 10.3[dm^3/s, while the water flow velocities were in the range]0.37, 2.6[m/s, the temperatures of water were in the range]7.0, 13.1[°C thus, the flow was always turbulent, since $Re \in$]19, 155[$\times 10^3$.

3. RESULTS OF INVESTIGATIONS

3.1. THE OUTPUT EQUATIONS FOR FRICTION FACTOR AND MINOR LOSS COEFFICIENT

By writing generalized Bernoulli's equation in relation to the comparative level located on the inlet chamber bottom, set for the first section lying on the water surface in the chamber and for the second cross-section of the water jet flowing through the pipe, the following two equations can be yielded in the form of

$$z_0 = z_i + \alpha\left(\frac{4q_V}{\pi d^2}\right)^2 \frac{1}{2g} + f\frac{\Lambda}{d}\left(\frac{4q_V}{\pi d^2}\right)^2 \frac{1}{2g} + K\left(\frac{4q_V}{\pi d^2}\right)^2 \frac{1}{2g}, \qquad (4)$$

$$\{i, \Lambda\} \in \{\{1, l\}, \{2, L\}\}$$

which gives the following dependences with regards to f and K

$$f = \frac{\pi^2 g d^5 \left(\dfrac{z_1}{L-l} - \dfrac{z_2}{L-l}\right)}{8q_V^2} \qquad (5)$$

$$K = \frac{\pi^2 g d^4 \left(z_0 - \dfrac{z_1}{1-\dfrac{l}{L}} + \dfrac{z_2}{\dfrac{L}{l}-1}\right)}{8q_V^2} - \alpha \qquad (6)$$

The fact that the flow in Sections 1 and 2 is fully developed was accounted for, thus α is identical there. It follows from Eqs. (5) and (6) that owing to the applied compensatory measurement method the influence of the minor losses on the measured value of the friction factor and the influence of the friction losses on the measured value of the minor loss coefficient can be eliminated. It can be seen from Eq. (6) that a sufficiently accurate knowledge of the kinetic energy correction factor is very important in the measurement procedure of the coefficient *K*. It should be remembered that the minor loss coefficient has the numerical value of the order of 0.5 in relation to $\alpha > 1.04$, and thus, the assumption of $\alpha = 1$ involves the error of the order of 10% measurement results of the factor *K*.

In the experiment, it can be acknowledged that the pipe is hydraulically smooth, because the empirically obtained dependence of $f(Re)$ diverges insignificantly from the Blasius curve (2). This is illustrated in Fig. 4, which shows the results of the determination of the friction factor *f* based on thet measurements using Eq. (5). The volume flow rate q_V, as in Eqs. (5) and (6), was determined based on equation specifying the liquid flow through the measuring triangular overfall (Fig. 3)

$$q_V = \frac{8}{15}\mu\sqrt{2g(z_w - z_{w0})^5} \tag{7}$$

In Equation (7), $\mu = 0.60$ was taken as the value obtained as a result of the overfall calibration on the testing station presented in the Fig. 3.

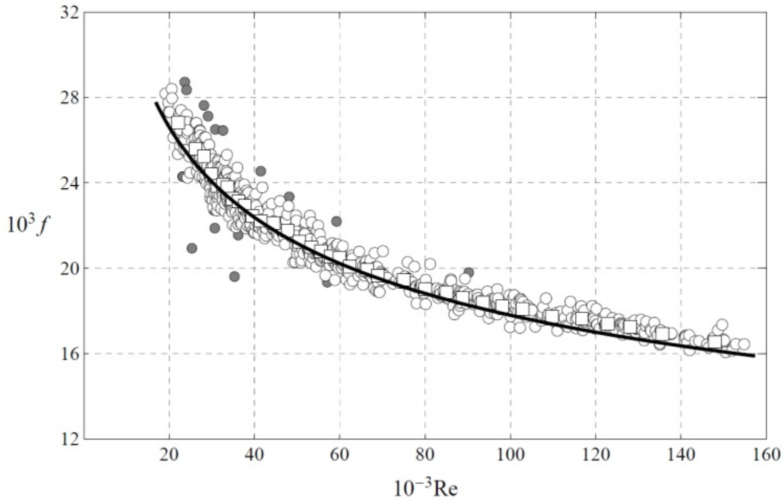


Fig. 4. The dependence $f(Re)$ obtained empirically by the authors (the solid line is the Blasius curve)

In Figure 4, the results of all measurements (the total of 719) are given. They are marked with circles, therewith, the grey circles indicate results recognized as inaccurate – divergent by more than 5% from the mean value (there were 18). The square centers visible in Fig. 4 contain coordinates of mean values of *Re* and *f* calculated in 35 subsets of measurement results, which the entire usable set of measurement results was divided into (701 points). The averaging of results makes sense, since there is a relatively large dispersion of results, which have normal distribution in the subsets. The subsets contained 20 results each (only the last one contained 21 results).

The solid line in Fig. 4 is the Blasius curve and, as can be seen, it lies close to the path set out by the squares. Therefore, the pipe being investigated was deemed hydraulically smooth and the Eq. (3) was employed to determine the kinetic energy correction factor.

3.2. RESULTS OF MEASUREMENT OF THE ENTRANCE LOSS COEFFICIENT *K*

The results of measurements conducted for each determined sub-option $b/d \in \{1.25, 2.5, 5, 10\}$ and all measurement runs $h/d \in \{2.5, 5, 10, 19\}$, in each geometric option of the reservoir (that is, for each of four pipe inlet shapes – Figure 2) were properly prepared prior to their marking on the diagrams. Namely, raw results (719) were sorted, while results whose values deviated by more than ±3×(Standard Deviation) from the mean value for a given set were rejected as incredible (41).

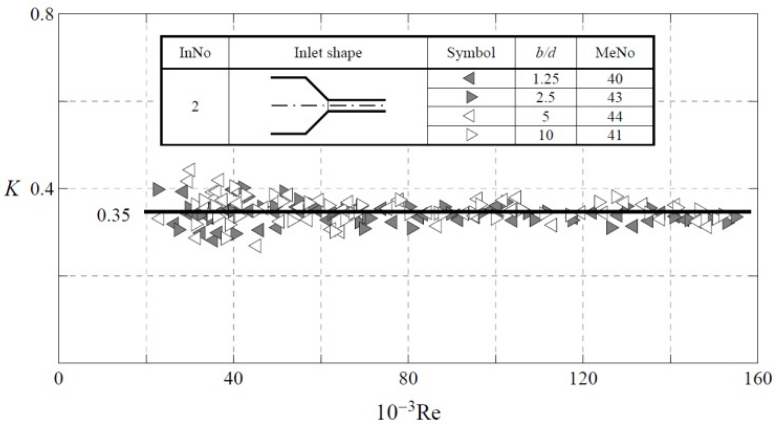


Fig. 5. Results of measurement of the entrance loss coefficient *K* (an example for the pipe inlet No. 2) for $h/d \in \{2.5, 5, 10, 19\}$ and each $b/d \in \{1.25, 2.5, 5, 10\}$ (InNo is the pipe inlet number, while MeNo denotes the number of measurement results)

Thus, results for four measurement runs (h/d) were presented for the inlet option No. 2, in Fig. 5. Each suboption (for specified b/d) was marked with a different square subsequent to the rejection of incredible values. The horizontal line is located at the height of the mean value from all 4 measurement runs (the total of MeNo = 168 results). All measurement runs were marked in Fig. 5 by means of layers (a layer of symbols denote a single run). Analyzing the distribution of symbols on the diagram plane, the dependence of *K* on *Re* [14, 15, 22] and the relation of *K* with values b/d

and h/d could not be found. All symbols are mixed and do not form separate paths, which would mark out 4 different curves. Furthermore, many symbols are not visible, since previous layers were covered with subsequent layers of symbols belonging to subsequent measurement runs. The mean value of the set produced from the sum of measured values of K in all four measurement runs were taken credible in the situation.

The set of K values assigned in such manner to a single pipe inlet shape and sorted by ascending Reynolds numbers was presented in Fig. 6. The figure shows a considerable dispersion of results of single measurements around the mean value of the entire 168-element set. However, the confidence interval is relatively narrow ±0.01 in relation to the mean 0.35. This can be explained by the fact that the value set is numerous and has normal distribution. Thus, the uncertainty of the K value, and thus the confidence interval were calculated based on the mean square error of the arithmetic mean with the probability of 95%.

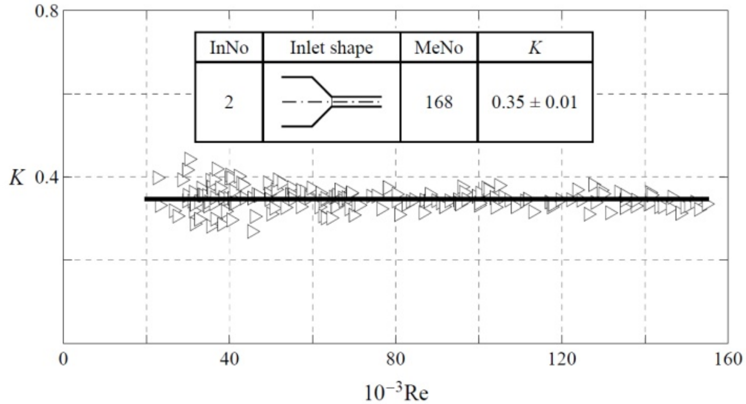


Fig. 6. Complete results of measurements
of the coefficient K for the pipe inlet No. 2

Subsequent results of measurements assigned to other investigated inlet shapes (Nos. 1, 3 and 4 cf. Fig. 2) are given in Table 1.

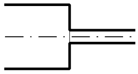
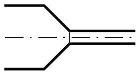
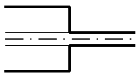
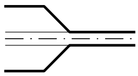
It follows from the cumulative juxtaposition of the results of investigation (Table 1) that the invert channel (on the reservoir bottom) and the guides (vertical flat converging toward the pipe) have the influence on the value of the entrance loss coefficient K, that is:

- the presence of guides decreases the value of K by 16–19%,
- the presence of invert channel increases the value of K by 22–24%,
- the lowest value of $K = 0.35$ exists when the guides are located in the tank without invert channel on its bottom,

• the highest value of $K = 0.55$ exists when the tank has not a guides but the invert channel is located on its bottom.

Table 1

The cumulative juxtaposition of results of measurement of the entrance loss coefficient K of all investigated pipe inlets

InNo	Inlet shape	K	MeNo	$10^{-3}Re$
1		0.43 ± 0.01	176	]20, 155[
2		0.35 ± 0.01	168	]22, 155[
3		0.55 ± 0.01	164	]20, 135[
4		0.46 ± 0.01	170	]19, 135[

It was found that an invert channel of the depth $d/2$, located on the tank bottom, increases the values of the coefficient K, whereas guides, produced from vertical, flat walls converging at the angle $\pi/4$ rad towards the pipe, decrease the values of K in relation to inlets without such elements. This can be explained by the fact that a liquid inlet to the pipe with an invert channel is hindered with respect to an inlet without an invert channel. Namely, the bottom half of a pipe flow section is only supplied by liquid from an invert channel, while the top half draws liquid from a quasi quarter-space. This results in an increase of the energetic loss in the pipe inlet. Whereas, the guides ease the pipe inlet, thus reducing recirculation areas constituting the source of relation to a inlet without and invert channel, where the space from which liquid flows to a energetic losses. Hence, losses in a supply system containing guides are lower in comparison to a system without guides.

4. CONCLUSIONS

The subject matter literature does not provide the influence of a geometrical shape of a reservoir section adhering directly to a pipe on its entrance loss, which the pipe is placed on the reservoir bottom. The paper presents results of investigation of the entrance losses coefficients for four, most frequently used in practice, geometrical sections of pipe inlets: a sharp-edged inlet from a wall perpendicular to the pipe axis, an inlet with guides produced from two vertical wall deviated from the pipe axis at the angle of π/4 rad, therewith, both cases may include (or not) invert channels of the depth of *d*/2. Sufficiently accurate results were obtainrd owing to the application of the measurement compensatory method (the influence of the friction losses on the measured value of the minor loss coefficient was eliminated) and the consideration of the kinetic-energy correction factor.

It was found that the invert channel of the depth of *d*/2 located on the reservoir bottom produces the increase of the value of the coefficient *K* by 22–24%, whereas the guides built from flat, vertical walls divergent at the angle of π/4 rad towards the throttling pipe decrease the value of *K* by 16–19% with respect to the inlets without such elements. The values of the minor loss coefficients, that were measured empirically, can be applied in the process of dimensioning of many hydraulic systems.

ACKNOWLEDGEMENTS

All calculations, figures and tables were carried out by means of *Mathematica* 8 by S. Wolfram, where own algorithms were prepared and ready-made calculation packets were applied.

SYMBOLS

b – width of a reservoir
d – diameter (inner) of the pipe
f – Darcy's friction factor
g – acceleration due to gravity
h – reservoir filling
Δh_{ld} – energy loss head (pipeline of the diameter d and length l)
k – pipeline roughness (mean height of wall roughness)
K – minor loss coefficient
L l – length of a pipeline section
Re – Reynolds number
q_V – volume flow rate
u – liquid average velocity
z_i – piezometric pressure head in the ith hydrometric section
Δz – pressure loss head
α – kinetic energy correction factor
μ – discharge coefficient

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