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INFLUENCE OF JET-TO-MAIN STREAM TURNING ANGLE IN FLUID FLOW FROM CYLINDRICAL NOZZLE OF COLLECTOR-PIPELINE ON FLOW COEFFICIENT

For intake cylindrical nozzles with orthogonal lateral jet outlets, dependences of the flow coefficient μ on (1) Reynolds number Re_d , (2) jet-to-main stream turning angle β , which is measured relative to the direction of the main stream in a collector-pipeline, as well as (3) the ratio d/D of the diameter of the outlet hole of the nozzle to that of the collector-pipeline are obtained. The ratio d/D influences the value of the coefficient of flow more considerably than the jet-to-main stream turning angle does. The magnitude of flow coefficient varies most abruptly in the range of the magnitude of the ratio d/D from 0.35 to 0.40. For adjustment of non-uniformity of the fluid inflow into the pressure pipelines along their lengths, the nozzles of $0.35 \le d/D \le 0.40$ are the most suitable ones.

Keywords: intake cylindrical nozzle, lateral jet outlet, angle of jet inflow

1. Introduction

In many production processes, pipeline systems of variable flow rate pressure fluid flow are used. Streams with discrete inflow of fluid along the path take place in ventilation (exhaust ventilation systems); water drains (water drainage networks; sewerage purification plants), land reclamation (drainage systems), spillways; water supply (horizontal water intakes, beam water intakes, water supply treatment systems). The fluid inflow along the part in collector-pipeline (CP) through its apertures in walls is non-uniform. It increases with the approach to the mouth of a collector-pipeline. Exact hydraulic calculation of such pipe-

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lines can be done only by means of the theory of variable flow rate hydraulics [1]. Theoretical methods of calculation of collector-pipelines laid in fluid stream are being developed. It these methods, all the geometric parameters of a collector-pipeline and hydrodynamics characteristics of inner and outer fluid streams, as well as the jet-to-main stream angle relative to the direction of the main stream of the collector-pipeline are taken into account [2. 3]. The theoretical calculation dependences [2. 3] indicate that the non-uniformity of fluid inflow into the collector-pipeline can be reduced by means of adjustment of the angle β between the jet from outlet nozzle and the main stream of the pipe along the CP. To verify this idea, we have suggested application of intake cylindrical nozzle with lateral jet, the nozzle can rotate in the wall of CP about the longitudinal axis. Thus, the need to have the value of the flow coefficient μ of the intake cylindrical nozzle with orthogonal lateral outlet, the value being dependent on the angle β, has arose. In literature there is no functional dependence for determining its flow coefficient.

In the works [4.5], results of experimental investigations of a stream which flows out of a cylindrical pipe at an angle to its longitudinal axis are presented. However, the authors of the article [4] considered curling of the stream by two tangetional swirlers, which were located along the pipe. Besides, they introduced some air into the fluid stream at a butt end of the pipe. In the work [5], a fluid stream which flew from the pipe at the angle to its axis, the angle was varied from 60^o to 120^o, was investigated in cavitational and non-cavitational regimes. In the work [6], numerous investigations of the influence of spillihg angle of laminar free submerged jet of heated liguid fuel oil onto reservoir wall is reported. The curling of stream and injection of air into it in the work [4] as well as creation of cavitational regime in the work [5] have caused a change in hydraulic resistance, and as a conseguence, have led to a change in the value of the flow coefficient μ of the hole. In the work [6], the lateral jet flow is not considered.

We consider a nozzle in which the jet turns only by 90^o when flowing out of the nozzle and is not burdened by additional hydraulic resistances. Besides, the flow coefficient of this nozzle must take into fccount the counteraction of the main stream which flows in the collector-pipeline.

2. Aim of the work

The aim is to determine functional dependences for evaluation of the flow coefficient μ of the intake cylindrical nozzle with orthogonal lateral outlet as a function of the angle β to the direction of the main stream in a collector-pipeline, as well as a function of the ratio d/D of the diameter of the outlet of the nozzle to the diameter of the collector-pipeline.

3. Techniques of carrying out the work

The experimental test bench (Fig. 1) enabled us to investigate water flow out of an outlet cylindrical nozzle under the action of the head which varied from 0 to 17 m. The working fluid was water. The heads which were less than 2.2 m were measured accurate to 0.5 mm by piezemeter whose division value was 1.0 mm. The heads which were greater than 2.2 m were measured by MTU pressure gauge whose accuracy class was 0.6 and whose division value was 0.02 kg / cm^2.

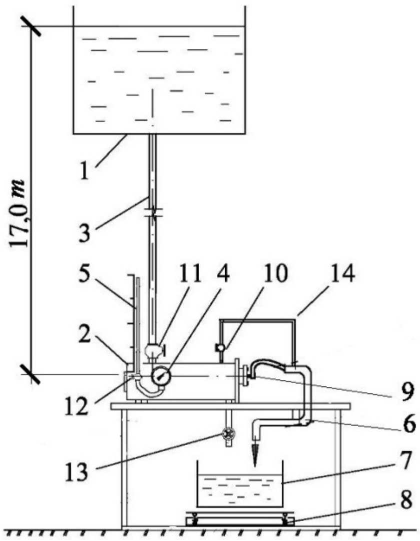

Fig. 1. Schematic diagram of experimental test bench: 1–reservoir; 2–head tank; 3–joining pipe; 4–pressure gauge; 5–piezemeter; 6–water–drain pipe; 7–water-receiving tank; 8–electronic weigher; 9–water–receiving tube with a nozzle; 10,11,12–faucets; 13–discharge pipe with a valve; 14–pipe for the exhaust of the pressure tank

Rys. 1. Schemat stanowiska badawczego: 1–górny zbiornik; 2–zbiornik ciśnieniowy; 3–połączenie rur; 4–manometr; 5–piezometr; 6–rura spustowa wody; 7–zbiornik wody odbiorczej; 8–elektroniczna waga; 9–rura z dyszą do badań; 10,11,12–zawory; 13–rura z zaworem do opróżnienia zbiornika ciśnieniowego; 14–rura do wypuszczania powietrza ze zbiornika ciśnieniowego

The investigation nozzles were installed at a butt end wall of the cylindrical tank 2 (see Fig. 1). The intake hole of the nozzle was placed in the head tank 2. To simulate the conditions of the work of the intake cylindrical nozzle in CP, the outlet end of the nozzle was introduced into the water-receiving tube, which was a part of CP in this case (Fig. 2).

As it was a single nozzle installed in the water-receiving tube that was investigated, a single end of the CP was closed by a pipe closer. The other end of the water-receiving tube was directed upward for filling it with water during the operation of the nozzle. Thus, water flow from the closed end to the other end of the water-receiving tube, which was connected with the water-draining pipe 6 (Fig. 1). The amount of water which flow through the outlet cylindrical tube was determining through volume method. The volume W of water which flow into the water-receiving tank 7 during the time t was determined through weight method with the help of electronic weather. The assembly of the nozzle attachment is given in Fig. 3.

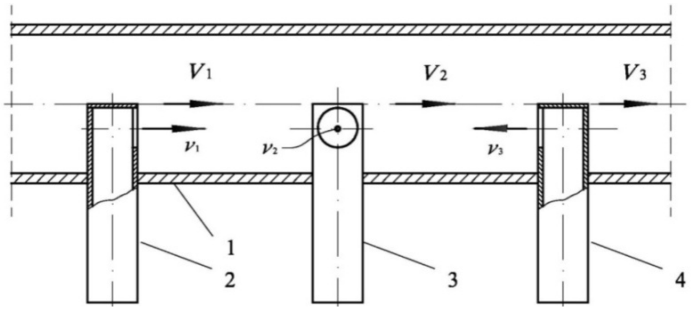

Fig. 2. Part of CP with nozzles installed with different outflow angles β: 1 – collector-pipeline; 2 – nozzle with $\beta = 0^o$; 3 – ditto $\beta = 90^o$; 4 – ditto $\beta = 180^o$; V_1 , V_2 , V_3 – velocities of streams in CP; v_1 , v_2 , v_3 – velocities of jet at outlets of nozzle

Rys. 2. Część rurociągu-zbieracza z dyszami zamontowanymi pod różnymi kątami odpływu: 1 – rurociąg-zbieracz; 2 - dysza z kątami odpływu β równymi 0^o ; 3 - to samo, 90^o ; 4 - to samo, 180^o ; V_1 , V_2 , V_3 - prędkości strumieni w rurociągu-zbieraczu; v_1 , v_2 , v_3 - prędkości strumienia na wylocie z dysz 2, 3, 4

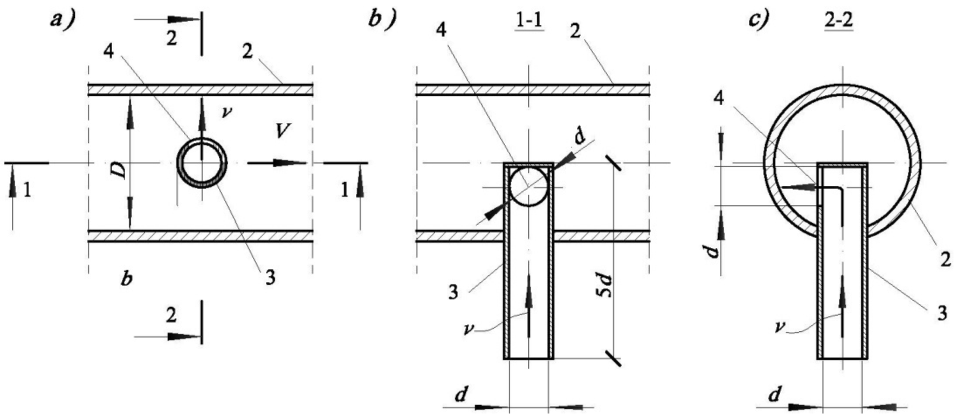

Fig. 3. Schematic-diagram of attachment of inner-nozzle with orthogonal lateral jet outlet: a – longitudinal section of receiving tube; b - section 1-1; c – section 2-2; 1 - pipe closer; 2 – receiving tube; 3 – nozzle; 4 – outlet hole

Rys. 3. Schematyczny diagram mocowania wejściowej dyszy z prostopadłym bocznym wylotem strumienia: a – przekrój podłużny doprowadzającego przewodu; b - przekrój 1-1; c – przekrój 2-2; 1 - korek; 2 – doprowadzający przewód; 3 – dysza wejściowa; 4 – otwór wylotowy

The intake cylindrical nozzle is mode with a lateral outlet at 90^o to its longitudinal axis (Fig. 4).

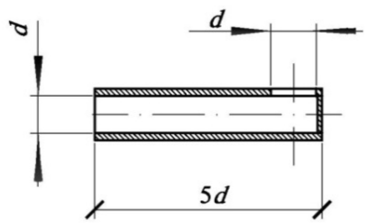

Fig. 4. Intake cylindrical nozzle with orthogonal lateral outlet

Rys. 4. Wejściowa dysza cylindryczna z prostopadłym bocznym wylotem

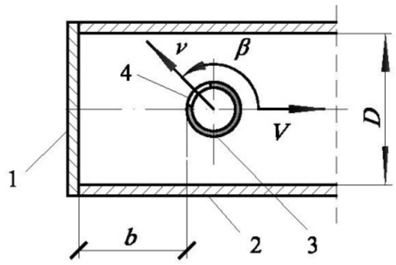

Fig. 5. Schematic diagram of reference system for angle β: 1 – pipe closer; 2 – receiving tube; 3 – intake nozzle; 4 – intake hole; V –velocity of streams in water-receiving; v – velocity of outflowing jet

Rys. 5. Schemat obliczenia kąta β odprowadzania strumienia wody: 1 - korek; 2 – doprowadzający przewód; 3 – dysza wejściowa; 4 – otwór wylotowy; V – prędkość doprowadzającego strumienia wody; v – prędkość strumienia wypływającego z dyszy wejściowej

Geometrical characteristics of the investigated intake nozzles are given in Table 1.

Table 1. Nozzles investigated

Tabela 1. Badane dysze

Diameters, *mm*		Ratios of diameters	Relative distance from nozzle to butt end pipe closer of water-receiving,
D	*d*	*d/D*	*b/D* (fig. 3,*a*)
10	4	0.400	1.1
22	8	0.364	1.55
22	10	0.455	1.36
45	13	0.289	0.49
45	15,6	0.347	0.24

4. Mathematical experimental data processing

The volumetric flow rate we calculated according to the formula:

$$Q = W/t \tag{1}$$

where, W is the volume of water which inflow into the water-receiving tank during the time t.

The flow coefficient μ of the nozzle we calculated according to the formula:

$$Q = \mu\omega\sqrt{2gH} \tag{2}$$

where, Q is the flow rate of water through the hole of the nozzle, m^3 / s;

ω is the area of the effective cross-section, m;

g is the gravity acceleration, $g = 9{,}81\, m/s^2$;

H is the head at the outlet of the outlet hole of the nozzle, m.

The value of Reynolds number we calculated for d of the outlet hole of the nozzle according to the formula:

$$Re = \frac{vd}{\nu} \tag{3}$$

where, v is the speed of fluid outflow through the outlet hole of the outlet nozzle, m / s;

d is the diameter of the outlet hole of the outlet nozzle, m;

ν is the kinematic viscosity of water depending on its temperature, m^2 / s

The relative change of the flow coefficient μ we determined according to the formula:

$$\psi = \frac{\mu_i - \mu_{0^o}}{\mu_{0^o}} 100\% \tag{4}$$

where, μ_{0^o}, μ_{i^o} are the flow coefficients of the nozzle for $\beta = 0^o$ and $\beta = i$ when i^o = 45^o; 90^o; 135^o; 180^o. 45^o; 90^o; 135^o; 180^o.

5. Results of experimental investigation

On the basis of experimental data processing, the dependence of the flow coefficient μ of the outlet nozzle with lateral jet flow on a previously set value of the Reynolds number Re_d is obtained (Fig. 6).

The dependence $\mu = f(Re)$ is obtained for all the investigated nozzles. For the nozzle of d = 4 mm (d / D =0.400), the independence of flow coefficient on Reynolds number (sell-similarity with respect to Reynolds number Re) was obtained at $Re = 2 \cdot 10^4$ for β = 0^o, at $Re = 3.5 \cdot 10^4$ for β = 135^o - (see. Fig. 6). For the nozzles of d / D = 0.289; 0.347, 0.364; 0.455 the self-similarity of the coefficient μ with respect to Reynolds number Re is observer at 80000; 90000; 65000; 65000, respectively. For the plotting of the dehendences $\mu = f(\beta)$, $\mu = f(d/D)$, the values of μ were taken from segments of sell-similarity of the graphs $\mu = f(Re)$ for each of the nozzles individually, i.e. for Re = 80000; 90000; 65000; 35000; 65000 for d/D = 0.289; 0.347, 0.364; 0.400, 0.455 respectively.

The dependence of the flow coefficient μ on the angle β are given in Fig. 7; ditto on the ratio d/D in Fig. 8.

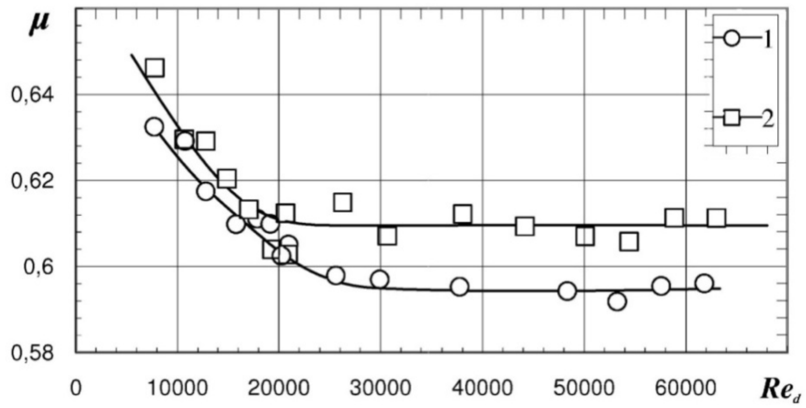

Fig. 6. Dependence $\mu = f(Re)$ for nozzle of $d = 4$ mm and $D = 10$ mm for anqles β : $0^o - (1)$; $135^o - (2)$;

Rys. 6. Zależności $\mu = f(Re)$ dla dyszy wejściowej o średnicy $d = 4$ mm oraz średnicy doprowadzającego przewodu $D = 10$ mm dla kątów β równych: $0^o - (1)$; $135^o - (2)$

In table 2, comparison of the effectiveness ψ of the adjustment of the flow coefficient μ of the nozzle for $\beta = 0^o$ and $\beta = 135^o$ for different values of the ratio of the diameter d of the intake hole of the nozzle to the inned diameter D of the receiving tube is given. As it is seen from Fig. 8, the ratio of the diameters d/D influences the coefficient μ more cousiderably than the value of the angle β does.

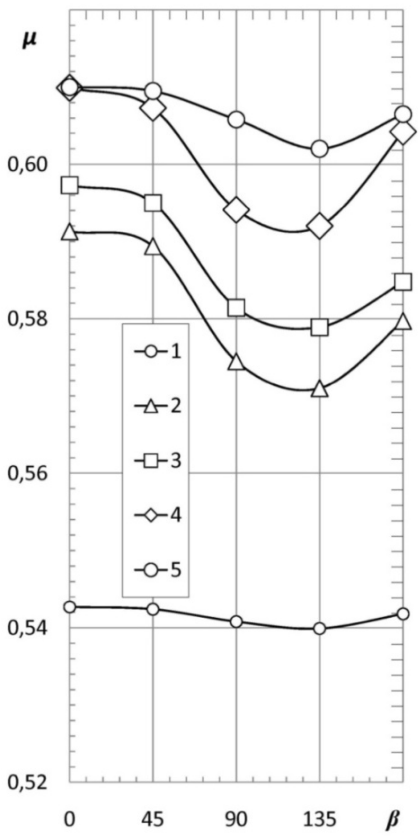

Fig. 7. Dependence of the flow coefficient $\mu = f(\beta)$ for different ratio of diameters d/D : $0.289-(1)$; $0.347-(2)$; $0.364-(3)$; $0.400-(4)$; $0.455-(5)$

Rys. 7. Zależności współczynnika przepływu dyszy wejściowej $\mu = f(\beta)$ dla różnych wartości stosunku średnic d/D : $0.289-(1)$; $0.347-(2)$; $0.364-(3)$; $0.400-(4)$; $0.455-(5)$

The greater the ratio of the diameters d/D is, the more cousiderably the value of the flow coefficient μ of the intake cylindrical nozzle with orthogonal lateral jet outflow increases for all the values of the angle β between the jet and the main stream (see Fig. 8). It is evident that with the increase in d/D, the extent of jet dilation decreases (from d to D). Thus the energy losses for dilation of the stream decrease. Accordingly, the increase in carrying capacity of the nozzle is observed, its flow coefficient μ increases.

For each of the investigated nozzle, i. e. for each value of d/D, it is established that the greatest value of the flow coefficient μ is at $\beta = 0^o$, and the least one at $\beta = 135^o$. At d/D = 0.289, the flow coefficient μ varies with the least rate (see Fig. 7; Fig. 8; Table 2): within the range 0.35 $\le d/D \le$ 0.40, it varies with the greatest rate. Thus, for adjustment of non-uniformity of fluid inflow along the path in pressure colltctor-pipelines, nozzles of 0.35 $\le d/D \le$ 0.40 are the most suitable for application.

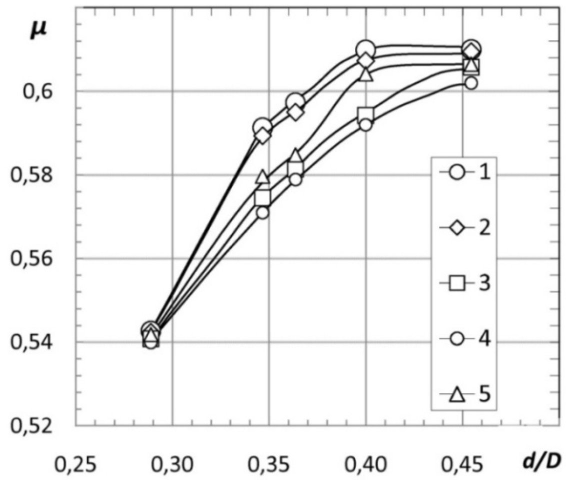

Fig. 8. Dependence of the flow coefficient $\mu = f(d/D)$ on the ratio of intake hole diameter of nozzle to CP diameter for different values of anqle β bettwen jet and main stream: $0^o - (1)$; $45^o - (2)$; $90^o - (3)$; $135^o - (4)$; $180^o - (5)$

Fig. 8. Zależności współczynnika przepływu dyszy wejściowej $\mu = f(d/D)$ w stosunku do proporcji średnicy otworu wylotowego dyszy do średnicy rurociągu-zbieracza dla różnych wartości kąta β odprowadzania strumienia wody: $0^o - (1)$; $45^o - (2)$; $90^o - (3)$; $135^o - (4)$; $180^o - (5)$

Table 2. Comparison of effectiveness of adjustment of flow coefficient μ for different values of d/D and β

Tabela 2. Porównanie skuteczności regulacji współczynnika przepływu μ dla różnych wartości d/D i β

Ratios of diameters d/D	Flow coefficient μ of nozzle for values of angle β		Relative change of flow coefficient ψ, %
	0°	135°	
0.289	0.5427	0.5399	-0.52
0.347	0.5913	0.5710	-3.43
0.364	0.5973	0.5789	-3.08
0.400	0.6099	0.5920	-2.93
0.455	0.6100	0.6020	-1.31

6. Conclusions

Experimental dependences of the flow coefficient μ of a cylindrical nozzle with orthogonal lateral outflow of jet on (1) Reynolds number Re, (2) ratio of diameters d/D (diameter of nozzle outlet to that of CP), and (3) the angle β between the jet and the direction of the main stream in CP are obtained. The ratio d/D influences the value of the flow coefficient μ more considerably than the value of β does. The magnitude of the flow coefficient μ varies most steeply within the ratio of d/D from 0.35 to 0.40. For adjustment of non-uniformity of fluid inflow along the path in pressure collector-pipelines, nozzles of $0.35 \leq d/D \leq 0.40$ are the most suitable ones.

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WPŁYW ZMIANY KIERUNKU STRUMIENIA W PRZEPŁYWOWYCH CYLINDRYCZNYCH DYSZACH RUROCIĄGU NA WSPÓŁCZYNNIK PRZEPŁYWU

Dla wlotowych cylindrycznych dysz z prostopadłymi oraz bocznymi wylotami strumieniowymi, wyznaczamy współczynnik przepływu liczby Reynoldsa Re_d (1), zmiana kąta strumienia zwrotnym β, który jest mierzony w kierunku głównego strumienia w rurociągu (2) oraz iloraz średnicy otworu wylotowego dyszy do wnętrza rurociągu są uzyskane d/D. Iloraz d/D wpływa na wartość współczynnikę przepływu znaczniej niż zmiana kierunku strumienia. Wielkość współczynnika przepływu zmienia się w najwiekszym stopniu w zakresie wielkości od 0,35 do 0,40. Do regulacji nierównomierność dopływu cieczy do rur ciśnieniowych, wzdłuż ich długości, dysze $0.35 \leq d/D \leq 0.40$ są najbardziej odpowiednie.

Słowa kluczowe: cylindryczna dysza, boczny wylot dyszy, kąt wypływu strumienia

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