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Fluid Flow Control

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1. Introduction

The present chapter is dedicated to the general presentation of the control system structures for the flow control in the hydraulic systems that have as components centrifugal pumps. The chapter also contains modeling elements for the following hydraulic systems: pumps, pipes and the other hydraulic resistances associated to the pipe.

1.1 The structure of the fluid flow control systems

The fluid flow control systems can be classified depending on the type of pressure source, on the pipe system structure and on the control element. Depending on the pressure source type, the flow control systems can be equipped with centrifugal pumps or volumetric pumps. Concerning the pipe structure, the flow control systems can be used within the hydraulic systems with branches or without branches. The control element within the flow control systems can be the control valve or the assembly variable frequency drive – electric engine - centrifugal pumps.

Throughout the next part there will be taken into consideration only the flow control systems having within their structure centrifugal pumps and pipe without branches. Due to this situation, the chapter will contain only two flow control systems:

a. The control system having the control valve as a control element, figure 1;
b. The control system having the assembly variable frequency drive – electric engine – centrifugal pump as a control element,, figure 2.

The flow control system having as control element the control valve consists of:

- Process, made of centrifugal pump, pipe without brances, local hydraulic resistances;
- Flow transducer, made of a diaphragm as primary element and a differential pressure transducer;
- Feedback controller with proportional – integrator control algorithm;
- Control valve.

The operation of the control system is based on the controlled action of the control valve hydraulic resistance, so that the hydraulic energy introduced by the centrifugal pump ensures the fluid circulation, within the flow conditions imposed and at the pressure of vessel destination and recovers the pressure loss associated to the pipe, associated to the local hydraulic resistances and associated to the control valve. The study and the design of

this flow control system need the mathematical modeling of the centrifugal pump, of the pipe, of the local resistances, as well as of the control valve hydraulic resistance.
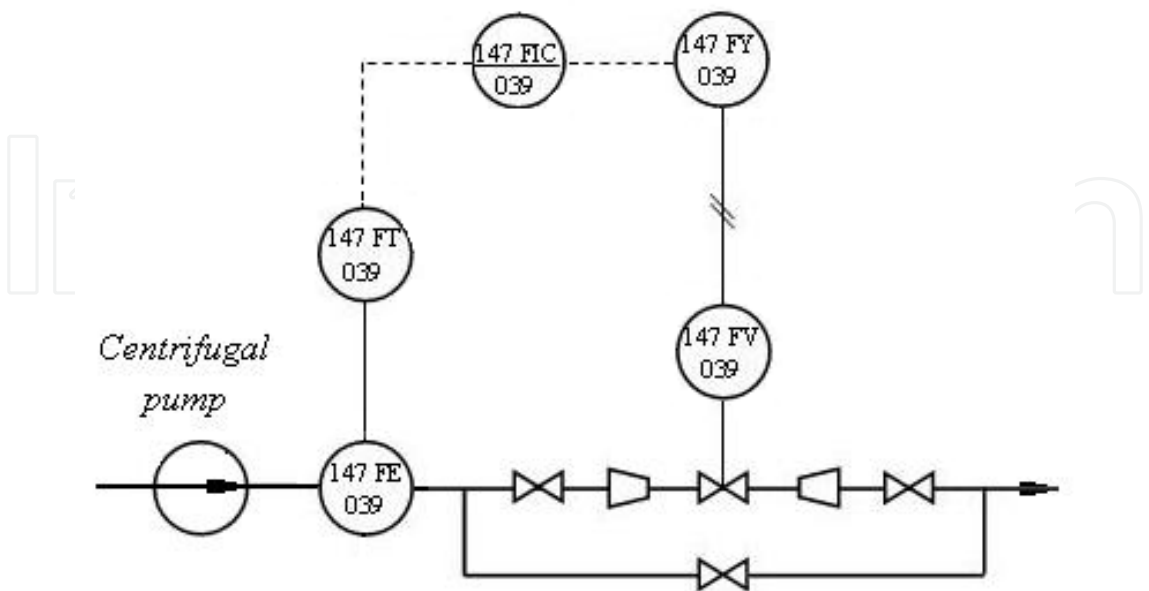

Fig. 1. The flow control system having as control element the control valve: FE – the sensitive element (diaphragm); FT – differential pressure transducer; FIC – flow controller; FY – electro-pneumatic convertor; FV – flow control valve.
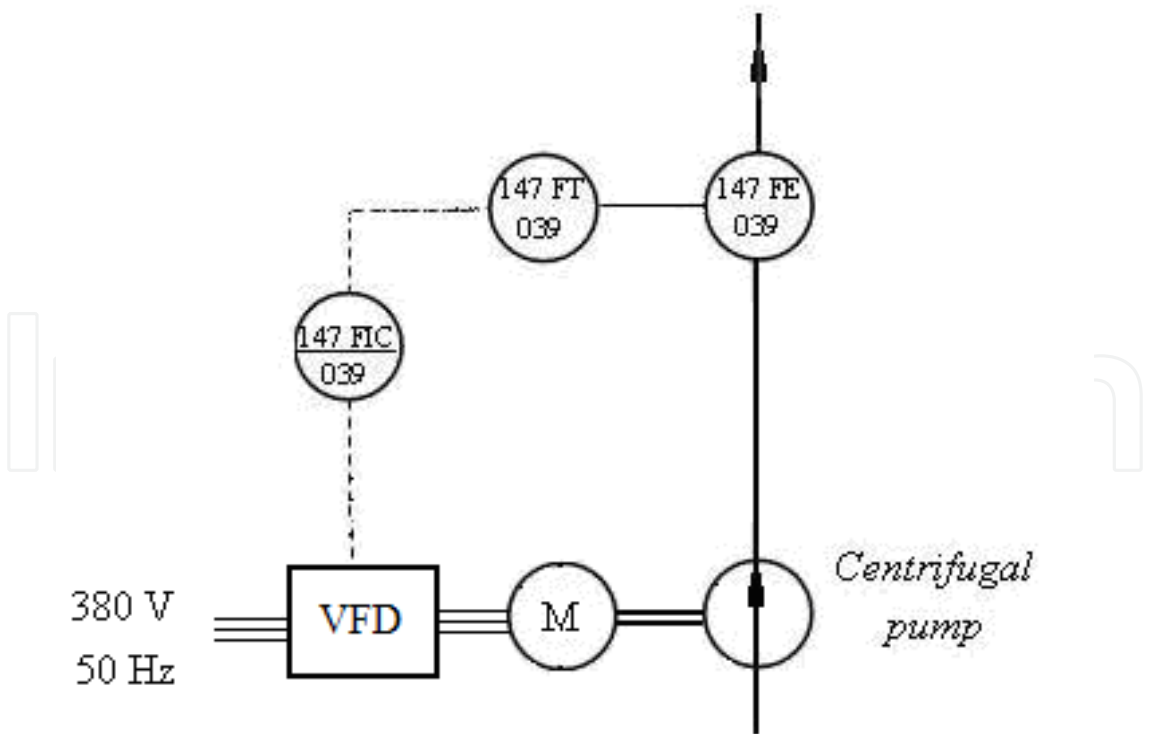

Fig. 2. The flow control system having as control element the assembly variable frequency drive – electric motor – centrifugal pump: M – electrical engine; VFD – variable frequency drive.

The flow control system having as control element the assembly - frequency static convertor – electric motor – centrifugal pump is made of:

- Process, made of a pipe without branches and local hydraulic resistances;
- Flow transducer, made of a primary element (diaphragm) and a differential pressure transducer;
- Feedback controller with proportional –integrator control algorithm;
- Control element made of variable frequency drive – electric motor – centrifugal pump.

The operation of this control flow system is based on the controlled actions of the centrifugal pump rotation, so that the hydraulic energy introduced ensures the fluid circulation, within the flow conditions imposed and the pressure in the destination vessel and recovers the pressure loss associated to the pipe and to the local hydraulic resistances. The study and the design of this control system needs the mathematical modeling of the centrifugal pump, of the pipe, of the local hydraulic resistances, as well as of the electric motor provided with frequency convertor.

1.2 The mathematical models of the fluid flow control elements

From the aspects presented so far there resulted the fact that the study and the design of the two types of control flow systems canot be done witout the mathematical modeling of all the subsystems that compose the control system. As a consequence, in the next part there will be presented: the mathematiocal model of the centrifugal pump, the mathematical model of the pipe, as well as of the local hydraulic resistance.

1.2.1 The simplified model of the centrifugal pumps

The operation parameters for the centrifugal pumps are: the volumetric flowrate, the ouput pumping pressure, the manometric aspiration pressure, the rotation speed, the hydraulic yield and the power consumption. The dependency between these variables is obtained experimentally. The obtained data is represented graphically, the diagrams obtained being named "the characteristics diagrams of the pump operation". In order to choose a centrifugal pumps to be used in a hydraulic system, there are applied the "characteristic diagrams at the constant rotation speed of pump".

The centrifugal pumps characteristics depend on the pumps constructive type. To exemplify, there has been chosen a pumps family used in refineries, figure 3 (Patrascioiu et al., 2009). The mathematical model of a centrifugal pump can be approximated by the relation

$$P_0 = a_0 + a_1 Q + a_2 Q^2 . \tag{1}$$

Nine types of pumps have been selected out of the types and characteristics presented in figure 3. For each of these there has been extracted data regarding the flow (the independent variable) and the outlet pressure (the output variabile). The data has been processed by using the polynomial regression (Patrascioiu, 2005), the results obtained being presented in table 1. Based on the numerical results, there have been graphically represented the characteristics calculated with the relation (1) for the following type of pumps: the pump 32-13, the pump 50-20 and the pump 150-26, figure 4.
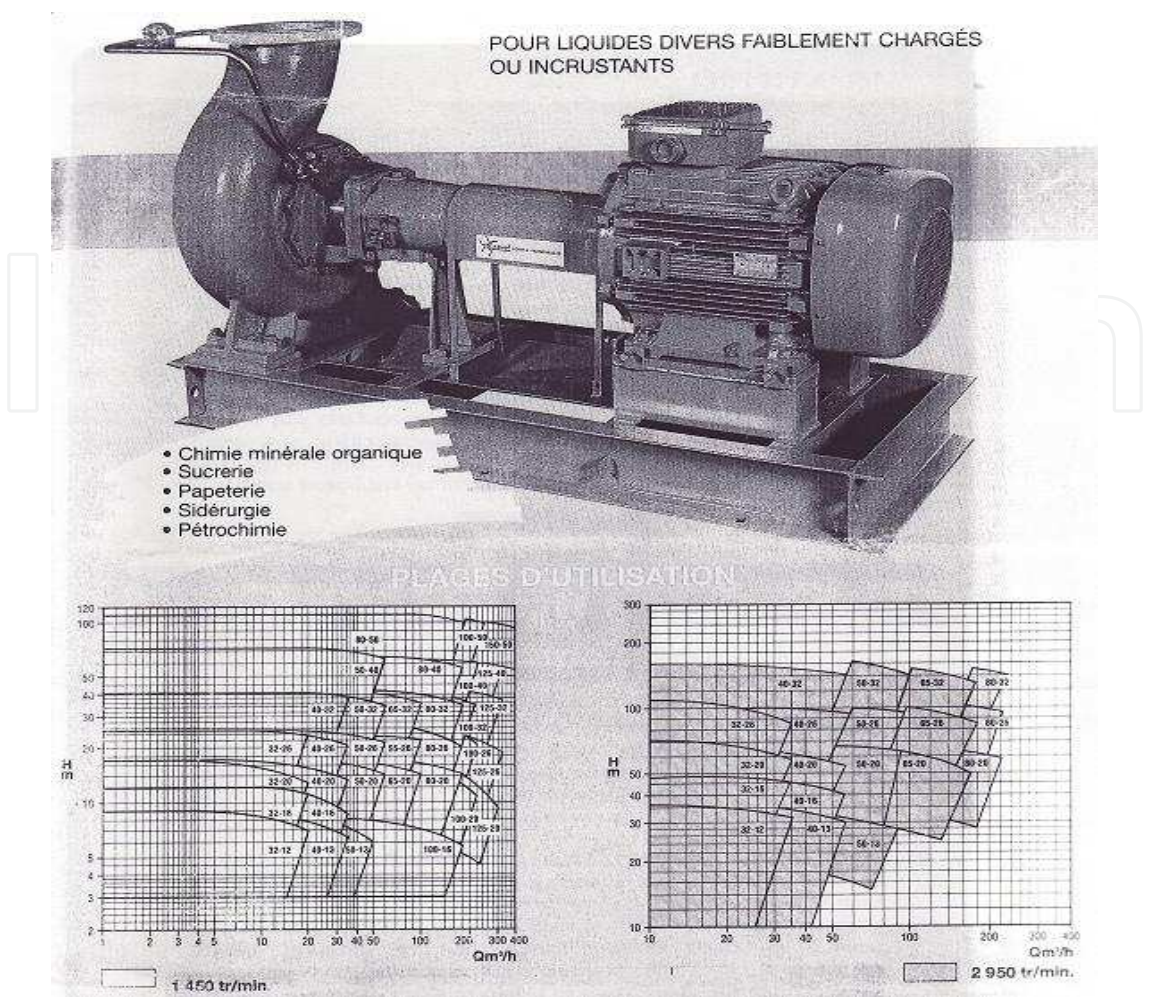
Fig. 3. The image and the characteristics of pumps used in a refinery

Pump type	Model coefficients (1)			Standard deviation [bar]
	a_0	a_1	a_2	
32-13	6.614213E+00	4.727929E-02	-3.938231E-02	6.621437E-02
32-16	8.851658E+00	7.501579E-01	-1.081358E-01	3.631092E-01
50-20	1.418321E+01	-2.746576E-02	-1.080953E-03	2.887783E-01
150-26	2.037071E+01	-7.850898E-03	-3.007388E-05	4.614905E-01
32-12	9.351883E+00	-1.262669E-01	3.939355E-04	1.890947E-01
50-13	8.227606E+00	-4.412075E-02	-1.679803E-05	1.578524E-01
40-16	1.258403E+01	-9.795947E-02	7.796262E-05	5.434285E-01
32-20	1.781198E+01	-2.755295E-01	1.361482E-03	3.952989E-01
40-20	1.742569E+01	-1.339487E-01	2.504382E-04	2.826216E-01

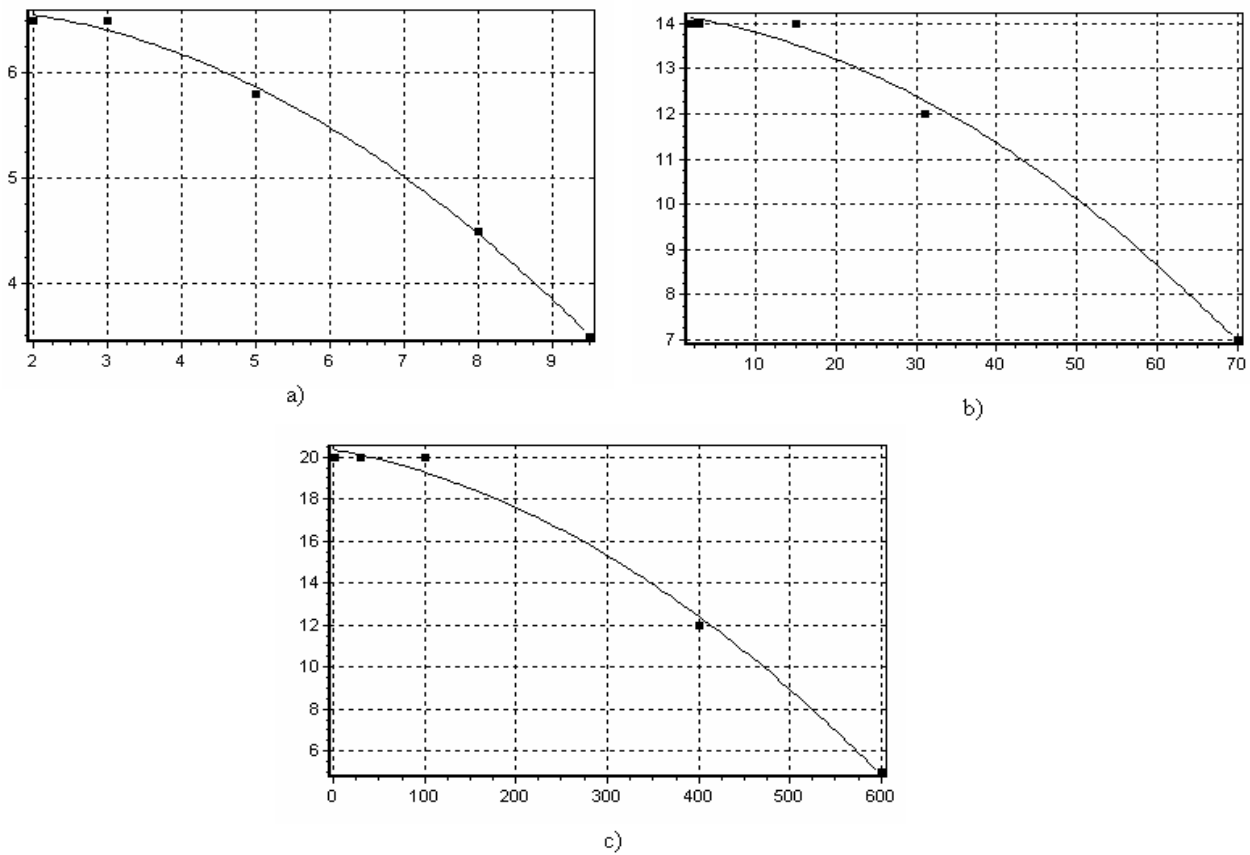Table 1. The mathematical model coefficients of the pump family

Fig. 4. The calculated characteristics of the pumps: a) pump 32-13; b) pump 50-20; c) pump 150-26.

1.2.2 The model of the pipe flow

The pipe represents a resistance of the hydraulic system. The mathematical model of the pressure lost by friction, for a straight pipe, with a circular section, is expressed by

$$\Delta P_{pipe} = \lambda \frac{8L}{\pi^2 D^5} Q^2 \quad \left[\frac{N}{m^2} \right] \tag{2}$$

where: λ – is the friction coefficient; L – the pipe length, in meters; D – the pipe diameter, in meters; Q – the fluid volumetric flow, in m³/s.

The application of the relation (2) implies determining the friction coefficient λ. The λ value depends mainly on the flowing regime, characterized by the Reynolds number Re, by the rugosity ε/D and the diameter D of the pipe. The Reynolds number is expressed by the relation

$$\mathrm{Re} = \frac{Dw}{v}, \tag{3}$$

where w represents the fluid linear velocity; v - the fluid's dynamic viscosity.

For the calculus of the friction coefficient λ there are used the relations (Soare 1979):

$$\lambda = \begin{cases} \lambda = \dfrac{64}{\text{Re}}, \quad \text{Re} < 2300 \\[2mm] \dfrac{1}{\sqrt{\lambda}} = 1.74 - 2\lg\left(2\dfrac{\varepsilon}{D} + \dfrac{18.7}{\text{Re}\sqrt{\lambda}}\right), \quad 2300 < \text{Re} < 3000 \\[2mm] \dfrac{1}{\sqrt{\lambda}} = -2\lg\left(\dfrac{D}{3.7\varepsilon} + \dfrac{2.51}{\text{Re}\sqrt{\lambda}}\right), \quad \text{Re} > 3000 \end{cases} \tag{4}$$

The relations group (4) contains two nonlinear equations, their solution being the friction coeffiecient λ (Patrascioiu et al., 2009). Then, for the intermediate flow regime, $2300 < \text{Re} < 3000$, is available this relation

$$\frac{1}{\sqrt{\lambda}} = 1.74 - 2\lg\left(\frac{2\varepsilon}{D} + \frac{18.7}{\text{Re}\sqrt{\lambda}}\right).$$

This relation is brought to the expression of the nonlinear equation

$$f(\lambda) = 1.74 - 2\lg\left(\frac{2\varepsilon}{D} + \frac{18.7}{\text{Re}\sqrt{\lambda}}\right) - \frac{1}{\sqrt{\lambda}} = 0. \tag{5}$$

For the turbulent flowing regime, $\text{Re} > 3000$, the relation

$$\frac{1}{\sqrt{\lambda}} = -2\lg\left(\frac{\varepsilon}{3,7D} + \frac{2.51}{\text{Re}\sqrt{\lambda}}\right)$$

represents a nonlinear equation

$$f(\lambda) = -2\lg\left(\frac{\varepsilon}{3.7D} + \frac{2.51}{\text{Re}\sqrt{\lambda}}\right) - \frac{1}{\sqrt{\lambda}} = 0. \tag{6}$$

All nonlinear equations have been solved by using the numerical aghorithms (Patrascioiu 2005). The mathematical model of the pressure loss was simulated for the hydraulic system presented in table 2.

Variable	Measure unit	Value
Pipe		
Diameter	m	0.05
Length	m	20
Rugosity	-	0.03
Max flow rate	m³/h	
Fluid		
Viscosity	m² s⁻¹	0.92e-6
Density	Kg m⁻³	476

Table 2. The geometrical characteristics of the pipe and the physical properties of the fluid

The variation of the friction factor used within model (2) was calculated with the relations (4), the result being illustrated in figure 5. The increase of the fluid flow, its rate and the Reynolds factor respectively, leads to the decrease of the pipe-fluid friction factor.
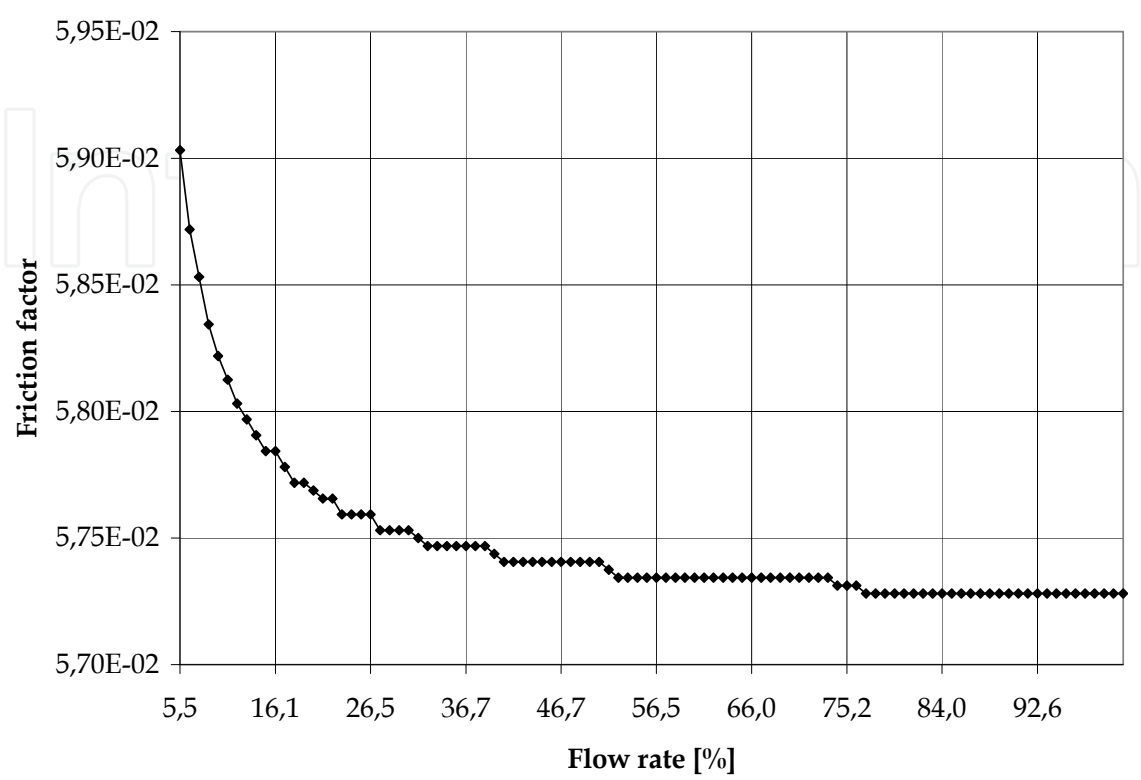

Fig. 5. The pipe drop pressure versus the fluid flow rate

In figure 6 there is presented the variation of the pressure drop in the pipe depending on the fluid flow. Due to the theoretical principles, the pressure drop on the pipe has a parabolic variation in relation to the fluid flow, although the friction factor decreases depending on the fluid rate.

1.2.3 The model of the hydraulic resistors

The local load losses, at the turbulent flow of a fluid by a restriction in the hydraulic system that modifies the fluid rate as size or direction are expressed either in terms of kinetic energy by relations under the form:

$$h_{hr} = \zeta \cdot \frac{w^2}{2g} \ , [\text{m}] \ , \tag{7}$$

or

$$\Delta p_{hr} = \zeta \cdot \frac{\rho w^2}{2} \ , [\text{N}/\text{m}^2] \ , \tag{8}$$

or in terms of linear load loss through an equivalent pipe of the length l_{hr} that determines the same hydraulic resistance as the considered local resistance

$$l_{hr} = \frac{\zeta}{\lambda} \cdot D .$$ (9)

The values of the local load loss coeffiecient ζ are usually obtained experimentally, in an analytical way, being estimated only in the case of turbulent flowing of a newtonian liquid. In table 3 there are presented the values of equivalent lengths (in metres) for different types of local resistances.
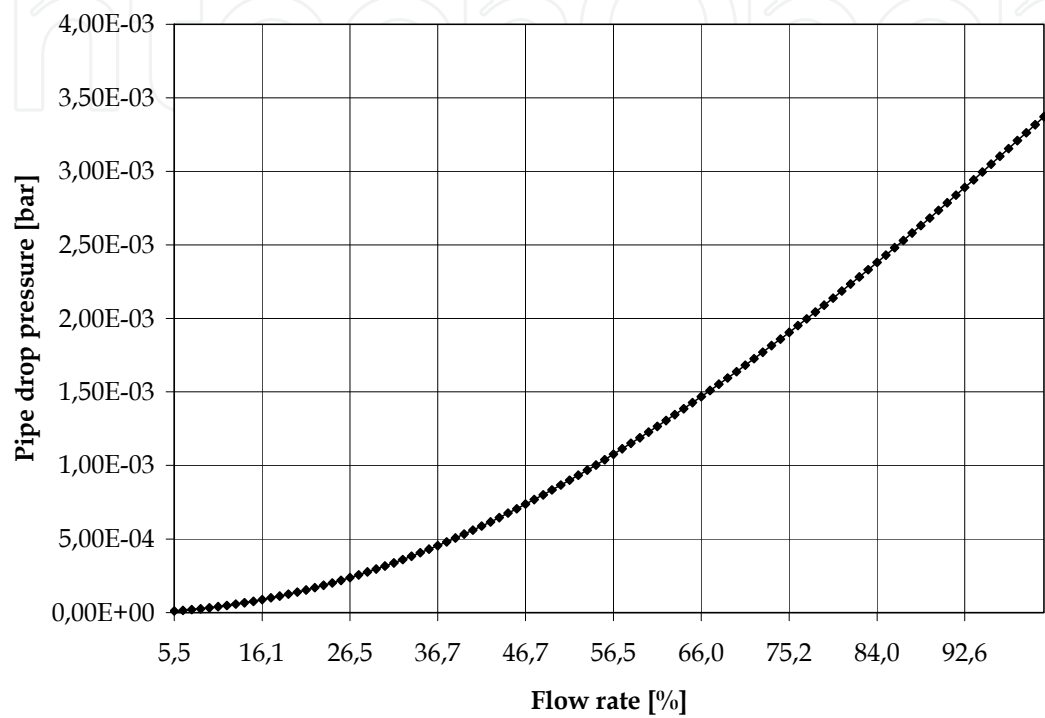

Fig. 6. The pipe static characteristic

The pipe nominal diameter [mm] Local resistance	50	100	150	200	300	400	500
T-square	4.5	9.0	14.5	20.0	34.0	37.0	63.0
Crossover tee	5.0	11.5	17.5	26.0	47.0	74.0	100.0
Quarter bend: $\alpha = 90°$; $Re/R = 8$	1.0	1.7	2.5	3.2	5.0	7.0	9.0
Quarter bend: $\alpha = 90°$; $Re/R = 6$	1.5	2.5	4.0	5.0	7.5	11.0	44.0
Cast curve	3.2	7.5	12.5	18.0	30.0	44.0	55.0
Slide valve	0.6	1.5	2.0	3.0	5.0	7.5	10.0
Tap valve	0.6	-	1.2	1.8	-	-	-
Flat compensator of expansion shape	4.0	9.5	14.5	20.0	33.0	48.0	64.0
Choppy compensator of expansion	5.0	12.0	18.5	26.0	42.0	61.0	82.0
Safety valve	3.6	7.5	12.5	18.0	130.0	-	-
Valve with normal pass	13.0	31.0	50.0	73.0	130.0	200.0	270.0
Valve with bend pass	10.0	20.0	32.0	45.0	77.0	115.0	150.0

Table 3. The equivalent lengths (metres of pipe) of some local resistances

2. Flow transducers

The flow transducer is included within the structure of the automatic system of flow control. The design of the flow control systems includes the stage of choosing the transducer type and its sizing. From the author's experience, in the domain of the chemical engineering there are especially used flow transducers based on the fluid strangling. From these, the most representative ones are the flow transducers with a diaphragm (for pipes with a circular section) and the flow transducers with a spout with a long radius (for the rectangular flowing sections).

2.1 The flow transducers with a diaphragm

The decrease section method is governed by national standards (STAS 7347/1-83, 7347/2-83, 7347/3-83). Within the flow transducer, the primary element is the diaphragm, classified as follows:

- with pressure plugs an angle;
- with pressure plugs at D and $D/2$;
- with pressure plugs in flange.

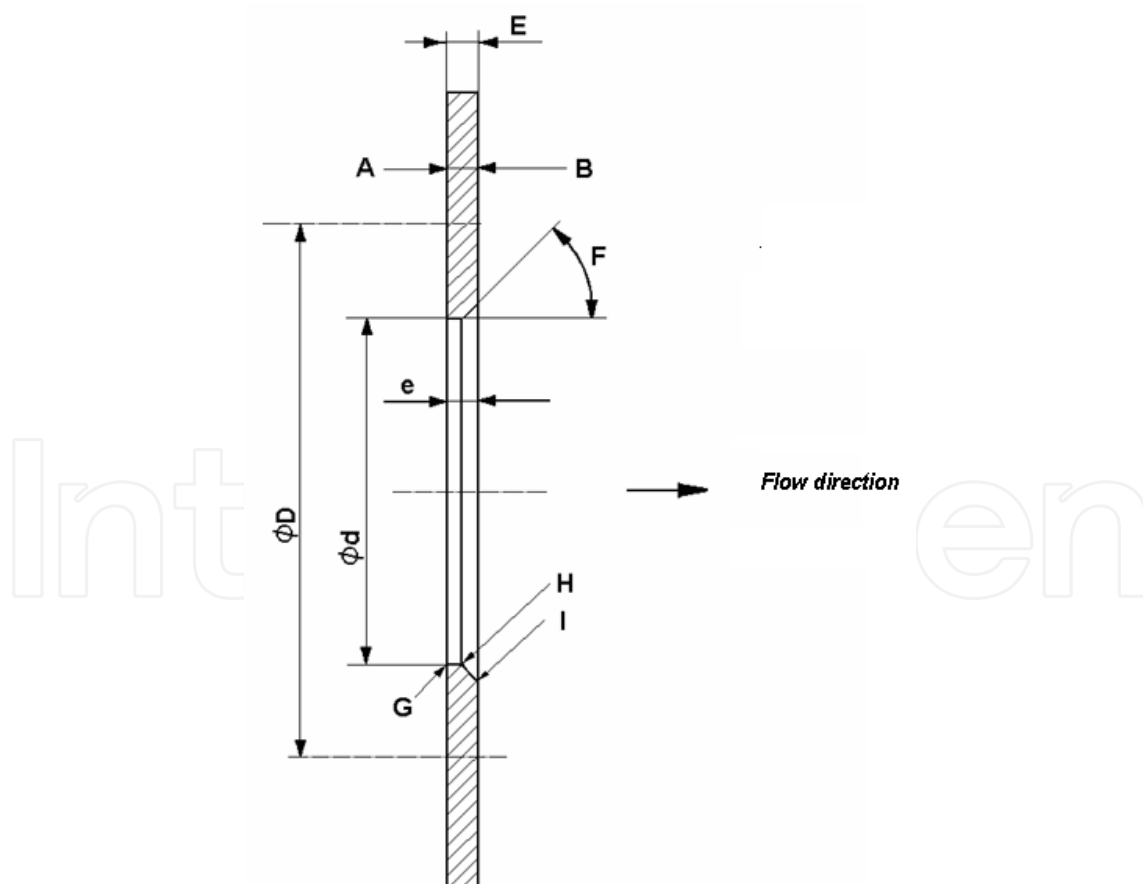Constructive elements specific to diaphragms are presented in figure 7.

Fig. 7. The normal diaphragm construction: A – upstream face; B – downstream face; E – plate thickness; G – upstream edge; H,I – downstream edges; e – hole thickness.

The domain of the diaphragm is restricted to circular section pipes, the diameter of the pipe and of the diaphragm being also restricted, table 4.

Characteristic	*Plugs at flange*	Plugs at D and $D/2$	Plugs in angle
d [mm]	≥ 12.5	≥ 12.5	≥ 12.5
D [mm]	$50 \leq D \leq 760$	$50 \leq D \leq 760$	$50 \leq D \leq 1000$
β	$0.2 \leq \beta \leq 0.75$	$0.2 \leq \beta \leq 0.75$	$0.23 \leq \beta \leq 0.80$
Re_D	$\geq 1260\beta^2 D \leq 10^8$	$\geq 1260\beta^2 D \leq 10^8$	$5000 \leq \mathrm{Re}_D \leq 10^8$ $0.23 \leq \beta \leq 0.45$ $10000 \leq \mathrm{Re}_D \leq 10^8$ $0.45 \leq \beta \leq 0.77$ $20000 \leq \mathrm{Re}_D \leq 10^8$ $0.77 \leq \beta \leq 0.80$

Table 4. Diaphragms use domain

The mass flow, Q_m, is calculated with the relation

$$Q_m = CE\varepsilon \frac{\pi}{4} d^2 \sqrt{2\,\Delta p\,\rho_1} \; . \; [\mathrm{kg/s}] \tag{10}$$

The significance of the variables is:

- C discharge coeffiecient,

$$C = \frac{\alpha}{E} \; ; \tag{11}$$

- d diameter of the primary hole   [m];
- D pipe diameter   [m];
- β diameter ratio,

$$\beta = \frac{d}{D} \; ; \tag{12}$$

- E closing rate coeffiecient,

$$E = \frac{1}{\sqrt{1 - \beta^4}} \tag{13}$$

- Δp differential pressure   [Pa];
- ε expansion coefficient
- ρ_1 fluid density upstream the diaphragm.   [kg/m³]

The discharge coefficient is given by the Stoltz equation

$$C = 0.5959 + 0.0312\,\beta^{2.1} - 0.1840\,\beta^8 + 0.0029\,\beta^{2.5}\left[\frac{10^6}{\mathrm{Re}_D}\right]^{0.75}$$

$$+\,0.0900\,L_1\,\beta^4\left(1-\beta^4\right)^{-1} - 0.0337\,L_2'\,\beta^3 .\qquad(14)$$

The volumetric flow, Q_v, is calculated with the classic relation

$$Q_v = \frac{Q_m}{\rho}. \quad [\text{m}^3/\text{s}]\qquad(15)$$

Within Stoltz equation, L_1, L_2 variables, respectively, are defined as follows:

- L_1 is the ratio between the distance of the upstream pressure plug measured from the diaphragm upstream face and the pipe diameter

$$L_1 = l_1 / D ;\qquad(16)$$

- L_2 is the ratio between the distance of the downstream pressure plug, measured by the diaphragm downstream face and the pipe diameter

$$L_2' = l_2' / D .\qquad(17)$$

The particular calculus relations for L_1 și L_2' are presented in table 5. The expansion coefficient ε is calculated irrespective of the pressure plug type, with the empirical relation

$$\varepsilon = 1 - \left(0.41 + 0.35\beta^4\right)\frac{\Delta p}{\chi\,p_1}\qquad(18)$$

this relation being applicable for $\frac{p_2}{p_1} \geq 0.75$.

The pressure plugs type	Calculus relations	Observations
Pressure plugs in angle	$L_1 = L_2' = 0$	-
Plugs at D and D/2	$L_1 = 1$ $L_2' = 0.47$	As $L_1 \leq 0.4333$, $\beta^4\left(1-\beta^4\right)^{-1} = 0.039$
Plugs at flange	$L_1 = L_2' = 25.4 / D$	For the pipes with the diameter $D \leq 58.62$ mm, $L_1 \leq 0.4333$, respectively $\beta^4\left(1-\beta^4\right)^{-1} = 0.039$

Table 5. Calculus relations for L_1 and L_2'

2.3 Case studies: The flow and diaphragm calculus for a flow metering system

In this paragraph there will be presented the calculus algorithm of the flow associated to a flow metering system and the diaphragm calculus algorithm used for the design of the metering system.The two algorithms are accompanied by industrial applications examples.

2.3.1 Algorithm for the calculus of the flow through diaphragm

The fluid flow that passes through a metering system having as primary element the diaphragm or the spout cannot be determined directly by evaluating the relation (10) due to the dependency of the discharge coefficient in ratio with the fluid rate, $C = f(v)$.Based on the relations presented in the previously mentioned standard, there has been elaborated a calculus algorithm of the fluid flow that passes through a metering system having the diaphragm as a sensitive element. Starting from the relation (10) there is constructed the nonlinear equation

$$g(Q_m) = 0 \, , \tag{19}$$

where the function $g(Q_m)$ has the expression

$$g(Q_m) = Q_m - CE \frac{\pi}{4} d^2 \sqrt{2 \, \rho \, \Delta P} \, . \tag{20}$$

As the factors E, ε, d, ΔP and ρ do not depend on Q_m, the relation (20) can be expressed under the form:

$$g(Q_m) = Q_m - KC \, , \tag{21}$$

where

$$K = E \, \varepsilon \, \frac{\pi \, d^2}{4} \sqrt{2 \, \rho \, \Delta P} \, . \tag{22}$$

Solving the equation (21) is possible, using the successive bisection algorithm combined with an algorithm for searching the interval where the equation solution is located (Patrascioiu 2005) .Based on the algorithm presented, there was achieved a flow calculus program for a given metering system.

2.3.2 Industrial application concerning the flow calculus when diaphragm is used

A flow metering system is considered, having the following characteristics:

- Pipe diameter = 50 mm
- Diaphragm diameter = 35 mm
- Measuring domain of the differential pressure transducer = 2500 mmH₂O
- Fluid density = 797 kg/m³
- Fluid viscosity = 3.76×10^{-6} m²/s

The request is the determination of the flow valve corresponding to the maximum measured differential pressure.

The results of the calculus program contain the values of the discharge coefficient *C,* the closing-up rate coeffiecient *E,* the diameters ratio β, as well as the mass flow value Q_m, calculated as a solution of the equation (21). A view of the file containing the results of the calculus program is presented in list 1.

_____

List 1

The results of the calculus program for flow through diaphragm
_____

Metering system constructive data	
Pipe diameter (m)	5.0000000000E-02
Diaphragm diameter (m)	3.5000000000E-02
Diaphragm differential pressure (N/m2)	2.4525000000E+04
Fluid characteristics	
Density (kg/mc)	797.000
Viscosity (m2/s*1e-6)	3.7600000000E-06
Auxiliary parameters calculus	
Beta	7.0000000000E-01
E	1.1471541425E+00
C1	6.2390515175E-01
K	6.9005483903E+00
kod	1
Flow rate (kg/s)	4.4015625000E+00
Flow ratet (m3/s)	1.9881587202E+01

_____

2.3.3 The algorithm for the diaphragm diameter calculus

The diaphragm calculus algorithm is derived from the calculus relation for the mass flow (10). Starting from this relation, the next nonlinear equation is drawn

$$g(d) = 0,\qquad(23)$$

where the function $g(d)$ has the expresion

$$g(d) = Q_m - C\,E\frac{\pi}{4}d^2\sqrt{2\,\rho\,\Delta P}\ .\qquad(24)$$

Since the factors *ε, ΔP* şi *ρ* do not depend on the diaphragm diameter *d,* the relation (24) can be written in the form:

$$g(d) = Q_m - K\,C\,d^2\ ,\qquad(25)$$

where

$$K = \varepsilon\,\pi\,\frac{\sqrt{2\,\rho\,\Delta P}}{4}\ .\qquad(26)$$

The diaphragm diameter calculus algorithm was transposed into a calculus program.

2.3.4 Industrial application concerning the diaphragm diameter calculus

A flow metering system is considered, having the characteristics:

- Pipe diameter = 50 mm
- Measuring domain of the differential pressure transducer = 1000 mm H_2O
- Fliud density = 940 kg/m^3
- Fluid viscisity = 10.6×10^{-6} m²/s
- Fluid flow rate = 1.1 kg/s.

The request is the determination of the diaphragm diameter corresponding to the flow rate and to the known elements of the flow measuring system. The numerical results of the calculus program are presented in list 2.

_____
                                                                                                           List 2

The results of the diaphragm diameter calculus program

_____

The constructive data of the metering system
Pipe diameter   (m)   5.0000000000E-02
Diphragm differential pressure (N/m2)   2.4525000000E+04

Fluid characteristics
Density (kg/mc)   797.000
Viscosity   (m2/s*1e-6)   4.6700000000E-06
Flow rate   (kg/s)   1.5000000000E+00

Diameter   (m)   2.2250000000E-02
_____

2.4 Flow transducers with tips

A particular case is represented by the air flow metering at the pipe furnaces. Since the pipe furnaces or the steam heaters are provided with air circuits having a rectangular section, there are no conditions for the diaphragm flow transducers to be used. For this pipe section type, there are not provided any calculus prescriptions. In this case, there has been analysed the adaptation of the long radius tip in the conditions. A cross-section through the sensitive element is presented in figure 8.

As the long radius tip is characterized by a continous and smooth variation of the throttling element, there is justified the hypothesis according to which the pressure drop for this sensitive element is due to the effective decrease of the following section.

The calculus relations for the design of the sensitive element are derived from the relation (10), written in the form

$$Q_m = a\, A_0 \sqrt{2\Delta P \rho}\ , [\text{kg/s}] \tag{27}$$

where α represents the flow coefficient; A_0 – the maximum flowing area; ΔP – the drop pressure between the upstream and downstream plugs of the sensitive element.
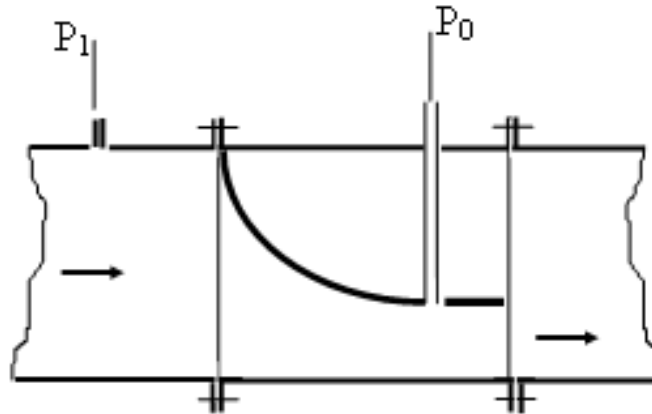

Fig. 8. The geometry of the long radius tip

By combining the relation (27) with the relation (15) there is obtained

$$Q_v = \alpha A_0 \sqrt{\frac{2\Delta P}{\rho}} \cdot [\text{m}^3/\text{s}] \qquad (28)$$

The design of the sensitive element presented in figure 8 means determining the value of area A_0. In terms of the hypotheses enumerated at the beginning of the section, the calculus algorithm for the flow transducer dimensioning consist of the following calculus elements:

- Bernoulli equation

$$P_1 + \frac{\rho_1 w_1^2}{2} = P_0 + \frac{\rho_0 w_0^2}{2} \qquad (29)$$

- The mass conservation equation

$$\rho_1 w_1 A_1 = \rho_0 w_0 A_0. \qquad (30)$$

Since the density variation is nonsignificant for the difference in 100 mm CA, $\rho_0 = \rho_1$ is considered, that leads to

$$P_1 - P_0 = \Delta P = \frac{\rho_1 w_o^2}{2}\left(1 - \frac{w_1^2}{w_o^2}\right). \qquad (31)$$

By combining the relations (28), (29), (30) and (31) there is obtained the expression used at the design of the flow transducer based on the long radius tip

$$A_0 = A_1 \sqrt{\frac{1}{1 + \dfrac{2\Delta P}{\rho}\left(\dfrac{A_1}{Q_0}\right)^2}}. \qquad (32)$$

The standards in the domain of the Venturi type flow transducers specify an ellipse spring for the diaphragm profile, described by the equation

$$\frac{x^2}{a^2} + \frac{y^2}{b^2} = 1 \,, \tag{33}$$

where a and b are the demi-axis of the ellipse, figure 9.

The ellipse quotes, the pairs of coordinates points (x, y) can be calculated from the relation (33), where the variable x has discrete values in the domain $[0, a]$.
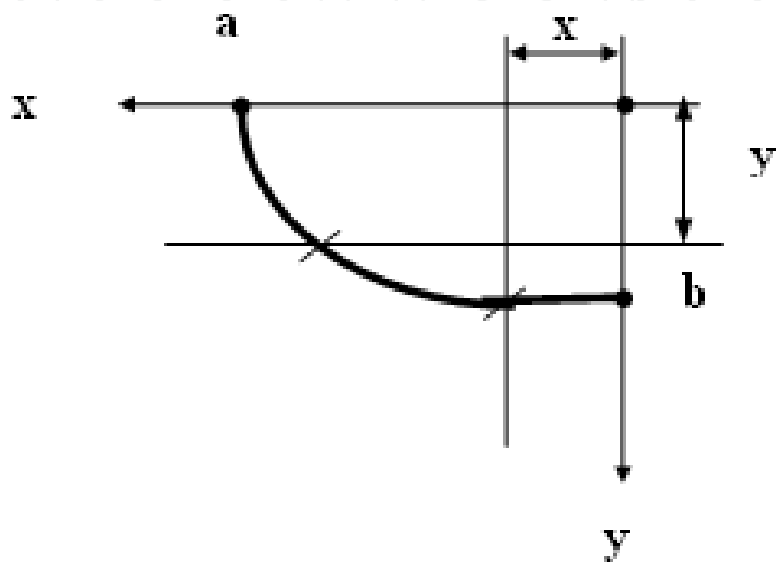

Fig. 9. Cross-section through long radius tips

2.5 Case study: The design of the long radius tip for an air flow metering system

There is considered a steam furnace within a catalytic cracking unit (CO Boyler). The initial design data is presented in table 6. The request is the dimensioning of the sensitive element of the air flow transducer.

Specification	Value
The pipetubes section profile upstream the sensitive element	1094 mm x 1094 mm
Total length of the part that contains the sensitive element	1100 mm
Maximum air flow	75000 m³ₙ/h
Air pressure upstream the sensitive element	100 mm CA relativ
Maximum pressure drop on the sensitive element	100 mm CA
Air temperature	20° C

Table 6. Steam furnace design data

Solution. The problem solution has been obtained by passing through the following stages:

1.   The air flow calculus in conditions of flowing through the sensitive element.

2. Determination of the area and of the minimum flowing limit.
3. The calculus of the coordinates of the sensitive element component ellipse.

Stage 1. The air flow calculus in the conditions of flowing through the sensitive element:

- air density in the conditions of flowing into the sensitive element

$$\rho_N = \frac{M}{RT_N} P_N = \frac{28.8}{8314 \times 273} \times 13590 \times 9.81 \times 0.76 = 1.286 \quad \left[kg / m_N^3 \right];$$

- air density in the conditions of flowing into the sensitive element (P_0, T_0)

$$\rho_0 = \frac{M}{RT_0} P_0 = \frac{28.8}{8314 \times 293} \times 13590 \times 9.81 \times 0.767 = 1.209 \quad \left[kg / m^3 \right];$$

- volumetric flow in the conditions (P_0, T_0)

$$Q_0 = \frac{\rho_N}{\rho_0} Q_N = \frac{1.286}{1.209} \times 75000 = 79777 \quad \left[m^3/h \right];$$

$$Q_0 = \frac{79777}{3600} = 22.1602 \quad \left[m^3/s \right].$$

Stage 2. The determination of the area and the minimum flow limit value:

- pipetubes area calculus

$$A_1 = 1.094 \times 1.094 = 1.1968 \quad \left[m^2 \right];$$

- calculus of pressure drop on the sensitive element

$$\Delta P = 1000 \times 9.81 \times 0.1 = 981 \quad \left[N / m^2 \right];$$

- calculus of the minimum section A_0

$$A_0 = 1.1968 \sqrt{\frac{1}{1 + \frac{2 \times 981}{1.209} \times \left(\frac{1.1968}{22.1602} \right)^2}} = 0.4998 \quad \left[m^2 \right];$$

- calculus of the minimum flow limit, figure 9

$$S = \frac{0.4998}{1.094} = 0.456 \quad [m].$$

Based on the result obtained, there is adopted S = 460 mm.

Stage 3. The calculus for the coordinates component ellipse of the sensitive element is based on the relation (33). From figure 10 and from the data presented in table 9 there result the following dimensions:

-    the big demi-axi of the ellipse

$$a = 1100 - 300 - 50 = 750 \quad [mm];$$

-    the small demi-axis of the ellipse

$$b = 1094 - 460 = 634 \quad [mm].$$

By using the values of the ellipse, the relation (33) becomes

$$\frac{x^2}{0.75^2} + \frac{y^2}{0.634^2} = 1.$$

In table 7 there are presented the values of the points that define the profile of the ellipse spring.
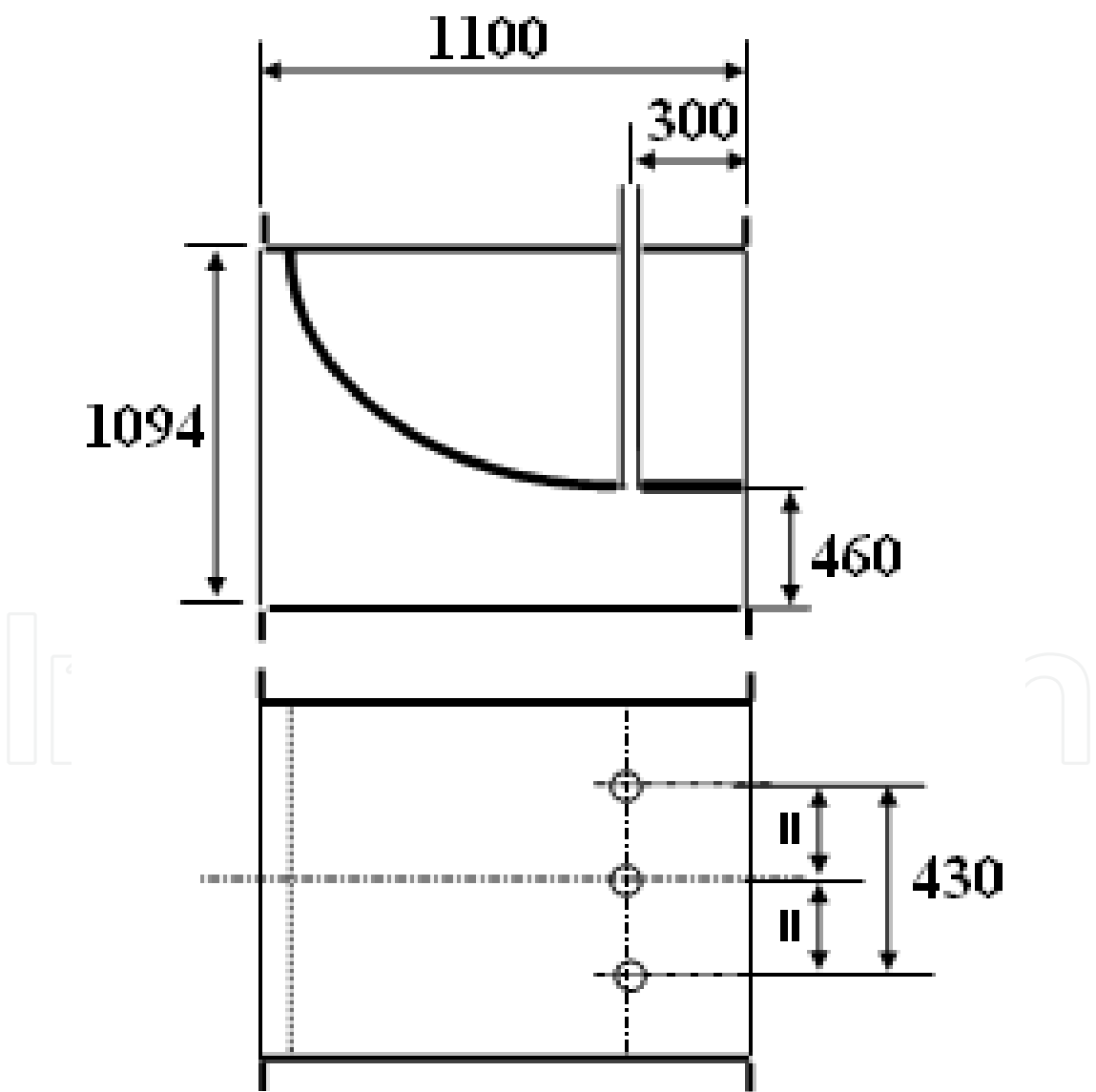

Fig. 10. The sensitive element basic dimensions

x [mm]	y [mm]	1094-y [mm]	(1094-y) [mm]
0	634	460	460
50	632	461	460
100	628	465	465
150	621	472	470
200	611	482	480
250	598	496	495
300	581	513	515
350	560	533	535
400	536	558	560
450	507	587	590
500	472	621	620
550	431	663	665
600	380	714	715
650	316	778	780
700	227	866	865
750	0	1094	1094

Table 7. The values of the ellipse spring profile

3. Fluid flow control systems based on control valves

3.1 The structure of the control valves

The control valves are the most widely-spread control elements within the chemical, oil industry etc. For these cases, the execution element is considered a monovariable system, the input quantity being the command u of the controller, the output quantity being identified with the execution quantity m, associated to the process. Taking into consideration the fact that the command signal u is an electrical signal in the range $[4…20]$ mA, the drive of the control element needs a signal convertor provided with a power amplifier, a servomotor and a control organ specific to the process. In figure 11 there is presented the structure of an actuator of control valve – type. This is made of an electrical – pneumatic convertor, a pneumatic servomotor with a membrane and a control valve body with one chair.

The electro – pneumatic convertor changes the electrical signal u, an information bearer, into a power pneumatic signal p_c. The typical structure of this subsystem is presented in figure 12 (Marinoiu et. al. 1999). This contains an electromagnet (1), a permanent magnet (3), a pressure – displacement sensor (2), a power amplifier (4) and a reaction pneumatic system (5). A lever system ensures the transmission of information and of the negative reaction into the system.

The other two subsystems, the servomotor and the control valve body, are interdependent, their connection being of both physical and informational nature. In the drawing of the figure 13 there is presented a servomotor with a diaphragm, a servomotor that ensures a normally closed state of the control valve. The contact element between the two subsystems

is represented by the rod (3). This transmits the servomotor movement, expressed by the rod *h* displacement, towards the control valve body. The command pressure of the servomotor, p_C, represents the output variable of the electro – pneumatic convertor. The control valve body represents the most complex subsystem within the control valve. This will modify the servomotor race *h* and, accordingly, the valve plug position in ratio with the control chair. The change of the section and the change of the flowing conditions in the control valve body will lead to the corresponding change of the flow rate.
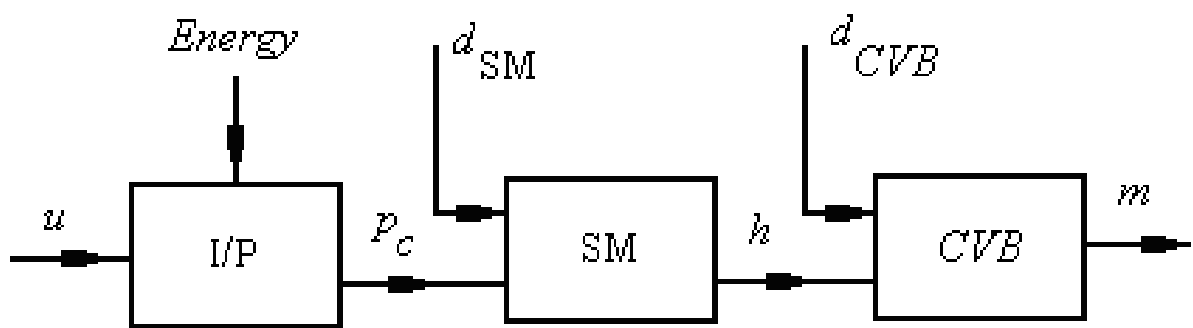
Fig. 11. The component elements of a control valve: *E/P* – electrical – pneumatic convertor; *SM* – servomotor; *CVB* – control valve body; *u* – electrical command signal; d_c – pneumatic command signal; *h* – servomotor valve travel; d_{SM} – disturbances associated to the servomotor; d_{CVB} – disturbances associated to the control valve body.
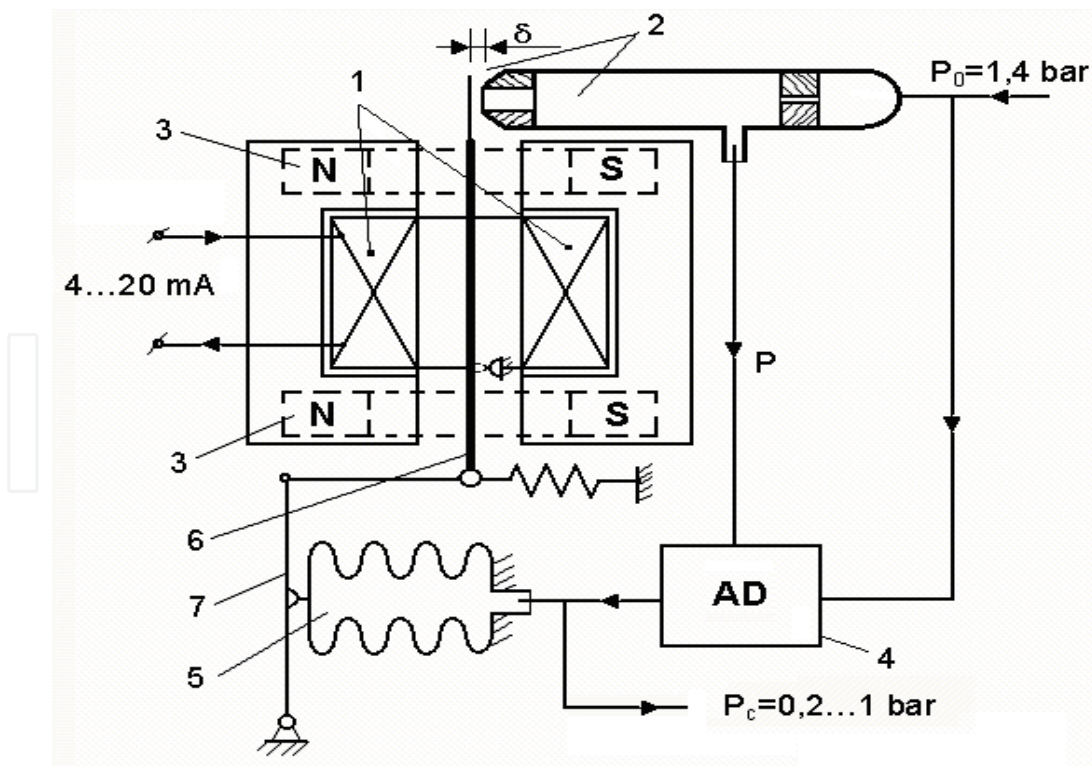
Fig. 12. The electro – pneumatic convertor: 1 – electromagnetic circuit; 2 – the pressure – displacement sensor; 3 – permanent magnets; 4 – power amplifier; 5 – the reaction bellows; 6 – articulate fitting; 7 - lever.
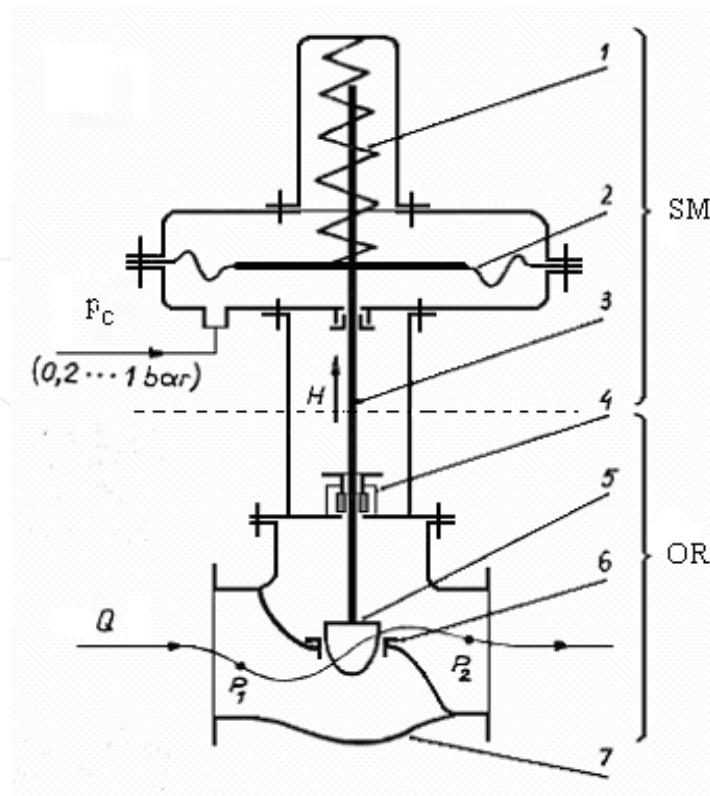
Fig. 13. The servomotor – control organ subsystem: SM – servomotor; CVB- control valve body; 1 - resort; 2 – rigid diaphragm; 3 - rod; 4 - sealing system; 5 - valve plug stem; 6 – chair; 7 - body.

3.2 The constructive control valve types

The usually classification criteria of the control valves are the following (Control Valve Handbook, Marinoiu et al. 1999):

a. The valve plug system:
    -   a profiled valve;
    -   a profiled skirt valve or a valve with multiple holes;
    -   a cage with V-windows;
    -   a cage with multiple holes;
    -   special valve plug systems;
    -   no valve plug systems;
b. The ways of the fluid circulation through the control organ:
    -   straight circulation;
    -   circulation at 90° (corner valves);
    -   divided circulation (valves with three ways).
c. Numbers of chairs:
    -   a chair;
    -   two chairs.
d. The constructive solution imposed by the nature, temperature and flowing conditions:
    -   normal;
    -   with a cooling lid with gills;

- with a sealing bladder;
- with an intermediate tube;
- with a heating mantle.

3.3 The control valves modelling

The modeling of the control valves represents a delicate problem because of the complexity design of the control valves, because of the hydraulic phenomena and the dependency between the elements of the control system: the process, the transducer, the controller and the control valve.

From the hydraulic point of view, the control valve represents an example of hydraulic variable resistor, caused by the change of the passing section. An overview of a control valve, together with the main associated values, is presented in figure 14. When the h movement of the valve plug modifies, there results a variation of the drop pressure ΔP_v and of the flow Q which passes through the valve.
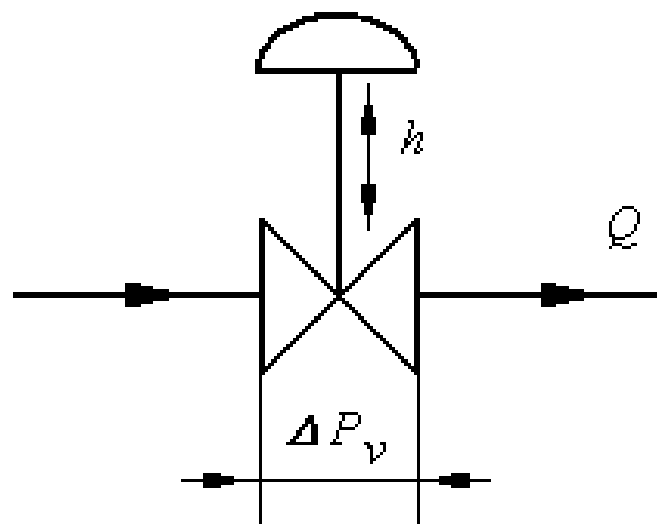
Fig. 14. Overview of a control valve: h – the movement of the valve plug's strangulation system; Q – the debit of the fluid; ΔP_v - the drop pressure on the control valve.

3.3.1 The inherent valve characteristic of the control valve body

The inherent valve characteristic of the control valve body represents a mathematical model of the control valve body that allows the determination, in standard conditions, of some inherent hydraulic characteristics of the control valve, irrespective of the hydraulic system where it will be assembled. A control valve can not always assure the same value of the flow Q for the same value of movement h, unless there is an invariable hydraulic system. This aspect is not convenient for modeling the control valve as an automation element, because it implies a different valve for every hydraulic system. A solution like this is not acceptable for the constructor, who should make a control valve for every given hydraulic system The inherent characteristic represents the dependency between the flow modulus of the control valve body and the control valve travel

$$K_v = f(h) \tag{34}$$

The flow modulus K_v represents a value that was especially introduced for the hydraulic characterization of the control valves, its expression being

$$K_v = \sqrt{2}\alpha A_v \quad [\text{m}^2], \tag{35}$$

where A_v – the flow section area of the control valve; α – the flow coefficient.

The way K_v value was introduced through relation (35) shows that it depends only on the inherent characteristics of the control valve body, which are expressed based on its opening, so based on the movement h of the valve plug. Keeping constant the drop pressure on the valve, there is eliminated the influence of the pipe over the flow through the control valve and the dependency between the flow and the valve travel is based only on the inherent valve geometry of the valve.

The inherent valve characteristics depend on the geometric construction of the valve control body. Geometrically, the valve control body can be: a valve plug with one chair, a valve plug with two chairs, a valve plug with three ways, a valve plug especially for corner valve etc. Consequently, the mathematical models of the inherent valve characteristics will be specific to every type of valve plug.

In the following part, there will be exemplified the mathematical models of the inherent valve characteristics for the valve control body with a plug valve with one chair. For this type of valve plug, there are used two mathematical models, named *linear characteristic* and *logarithmical characteristic,* models which are defined through the following relations:

- linear characteristic dependency

$$\frac{K_v}{K_{vs}} = \frac{K_{v0}}{K_{vs}} + \left(1 - \frac{K_{v0}}{K_{vs}}\right)\frac{h}{h_{100}} ; \tag{36}$$

- logarithmical characteristic dependency

$$\frac{K_v}{K_{vs}} = \frac{K_{v0}}{K_{vs}}\exp\left(\frac{h}{h_{100}}\ln\frac{K_{vs}}{K_{v0}}\right), \tag{37}$$

where h is the movement of the valve plug related with the chair; h_{100} – the maximum value of the plug's valve travel; K_{v0} – the value of K_v for $h = 0$; K_{vs} – the value of K_v at maximum valve travel h_{100}.

In figure 15 there are presented the graphical dependency for the two mathematical models of the inherent characteristics of the valve control body with a plug valve with one chair (Marinoiu et. al. 1999).

Observations. The value K_v, used within the mathematical model of the inherent valve characteristic and for the hydraulic measurement of the control valves, was introduced by Früh in 1957 (Marinoiu et al. 1999). Through the relation (35) he shows that the flow modulus K_v has an area dimension; out of practical reasons there has been agreed to be

attributed to K_v a physical meaning, which would lead to a more efficient functioning. This new meaning is based on the relation

$$K_v = \frac{Q}{\sqrt{\dfrac{\Delta P_r}{\rho}}} \quad [\text{m}^2], \tag{38}$$

and has the following interpretation:

K_v is numerically equal with a fluid of $\rho = 1$ kg/dm³ density which passes through the control valve when there takes place a pressure drop on it of $\Delta P_r = 1$ bar. The numerical values of K_v are expressed in m³/h .
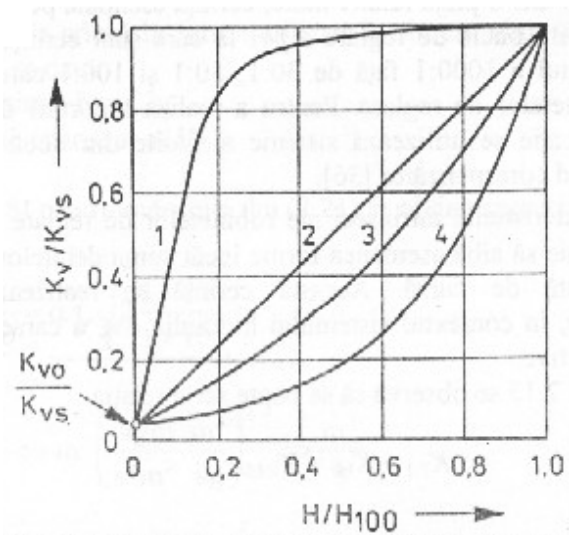

Fig. 15. Inherent valve characteristics types associated to the valve control body with a valve plug with one chair: 1 – fast opening; 2 – linear characteristic; 3 – equally modified percentage; 4 – logarithmical characteristic.

In the USA, by replacing the value K_v there is defined the value C_v as being the water flow expressed in gallon/min, which passing through the control valve produces a pressure drop of 1 psi. The transformation relations are the following:

$$\begin{cases} C_v = 1{,}156 K_v \\ K_v = 0{,}865 C_v \end{cases}; \tag{39}$$

3.3.2 The work characteristic of the control valve body

The work characteristic of the control valve represents the dependency between the flow Q and the valve travel of the h valve plug

$$Q = Q(h). \tag{40}$$

When defining the static work characteristic there is no longer available the restrictive condition concerning the constant pressure drop on the valve, as it was necessary for the

inherent valve characteristic, but the flow rate gets values based on the hydraulic system where it is placed, the size, the type and the opening of the valve control. From the point of view of the hydraulic system, the working characteristics can be associated to the following systems:

a. systems without branches;
b. hydraulic systems with branches;
c. **hydraulic systems with three ways valves.**

Due to the phenomena complexity, for the mathematical modeling of the working characteristic of the valve control body, there are introduced the following simplifying hypotheses:

a. there is taken into consideration only the case of the indispensable fluids in turbulent flowing behavior ;
b. there are modelled only the hydraulic systems without branches;
c. the loss of pressure on the pipe is considered a concentrated value.

The main scheme of a hydraulic system without ramifications is presented in figure 16. The system is characterized by the loss of pressure on the control valve ΔP_v, the loss of pressure on the pipe ΔP_p and the loss of pressure inside the source of pressure ΔP_{SI}.
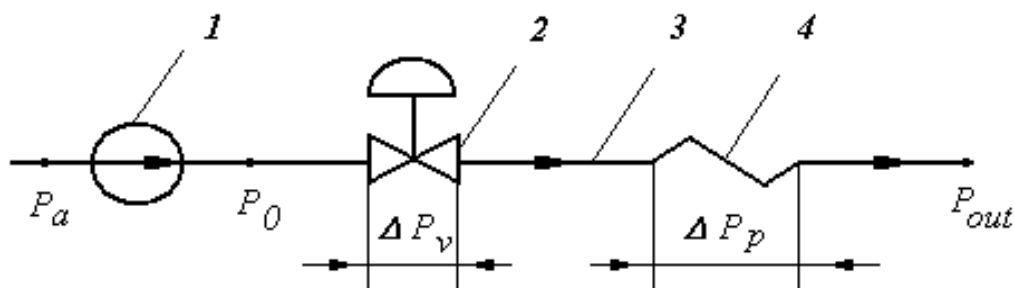

Fig. 16. Hydraulic system without branches: *1* - pump; *2* – control valve; *3* - pipe; *4* – the hydraulic resistance of the pipe.

For the modeling, the working characteristic of the valve control body, are defined by the following values:

- The flow rate that passes through the valve control

$$Q = K_V \sqrt{\frac{\Delta P_v}{\rho}} \quad [\text{m}^3/\text{h}]. \tag{41}$$

- The energy balance of the hydraulic system

$$P_o = P_{out} + \Delta P_v + \Delta P_p. \tag{42}$$

The connection of the control valve with the hydraulic system is very tight. To be able to determine the working characteristics of the control valve there have to be solved all the elements of modeling presented in this chapter: the centrifugal pipe characteristic, the inherent valve characteristic of the control valve and the pipe characteristic. The mathematical model of

the control valve working characteristic is defined by the block scheme presented in figure 17. The input variable is the valve travel h of the servomotor and implicit of the control valve and the value of exit is the flow rate Q which passes through the valve. Mathematically, the model of the control valve presented in figure 17 is a nonlinear equation

$$f(Q) = Q - K_v \sqrt{\frac{\Delta P_v}{\rho}} = 0 . \tag{43}$$

3.3.3 Solutions associated to the working characteristic of the control valve

The working characteristic of the control valve body, materialized by relation (40), can be determined in two ways:

a.   By introducing the simplifying hypothesis according to which there is considered that the flow modulus associated to the pipe does not modify, respectively ;
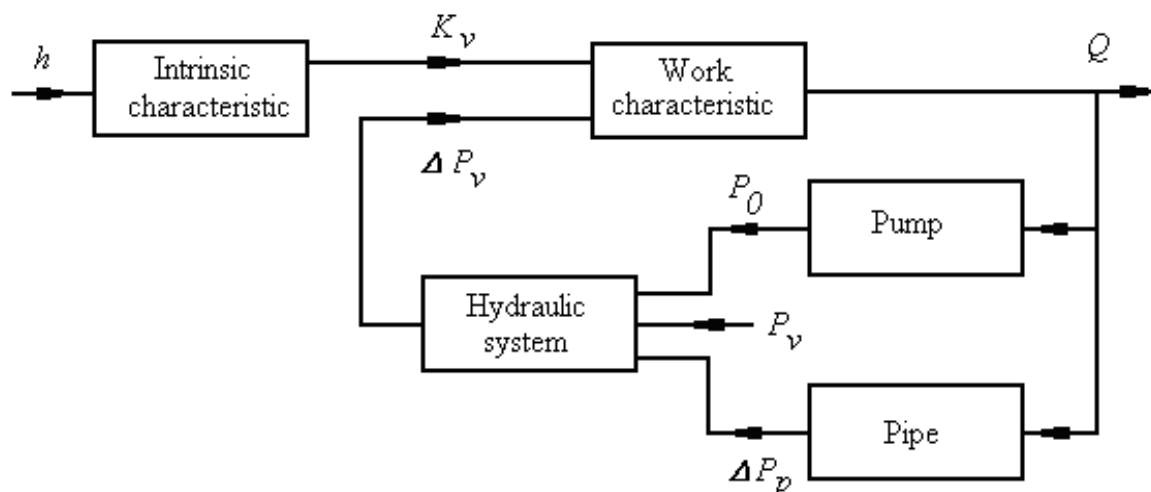b.   By resolving numerically the model presented in figure 17.

Fig. 17. The block scheme of the mathematical model of the body control valve.

The solution obtained by using the simplifying hypothesis represents the classical mode to solve the working characteristics of the control valve body (Früh, 2004). The solution has the form (Marinoiu et. al. 1999)

$$q = \frac{1}{\sqrt{1 + \Psi\left(\frac{1}{k_v^2} - 1\right)}} , \tag{44}$$

where $q = Q/Q_{100}$ represents the adimensional flow rate; $k_v = K_v/K_{v100}$ - the adimensional flow module; $\Psi = \Delta P_{v100}/\Delta P_{hs}$ - the ratio between the maximal control valve drop pressure and the maximal hydraulic system drop pressure.

In figure 18 there are presented the graphical solution of the working characteristics of the control valves for the valve plug type with inherent valve linear characteristic and inherent valve logarithmic characteristics.

The solution obtained by the numerical solving of the mathematical model presented in figure 17 has been recently obtained (Patrascioiu et al. 2009). Unfortunately, the mathematical model and the software program are totally dependent on the centrifugal pump, pipe and the control valve type (Patrascioiu 2005). In the following part there are presented an example of the hydraulic system model and the numerical solution obtained. The hydraulic system contains a centrifugal pump, a pipe and a control valve. The pump characteristic has been presented in figure 3 and the mathematical model of the 50-20 pump type is presented in table 1. The pipe of the hydraulic system has been presented in table 2 and the pipe mathematical model is expressed by the relation (2). The control valve of the hydraulic system is made by the Pre-Vent Company, figure 19, the characteristics being presented in table 8 (www.pre-vent.com).
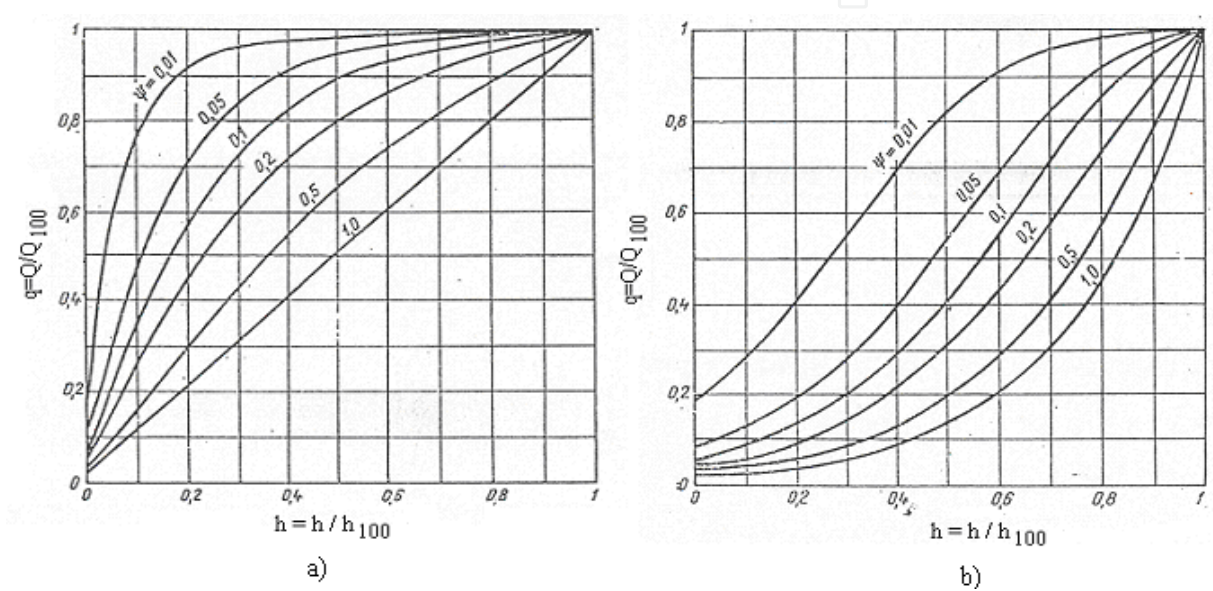

Fig. 18. The working characteristics of the control valves calculated (based on the simplifying hypothesis): a) valve plug with linear characteristic; b) valve plug with logarithmic characteristic.

The numerical results of the program are the inherent valve and the work characteristics. For theses characteristics, the independent variable is the adimensional valve travel of the control valve, $h/h_{100} \in [1...100] \%$.

Variable	Measure unit	Value
Inherent valve characteristic	Linear	
K_{vs}	m³ h⁻¹	25
K_{v0}	m³ h⁻¹	1

Table 8. The Control valve characteristics.

The inherent valve characteristic obtained by calculus confirms that the control valve belongs to the linear valve plug type. The working characteristic of the control valve is

almost linear, figure 20. The pipe drop pressure is very small, figure 5, and for this reason the influence of the control valve into hydraulic system will be very high, see figure 21. In this context, the inherent valve characteristic of the control valve body is approximately linear.
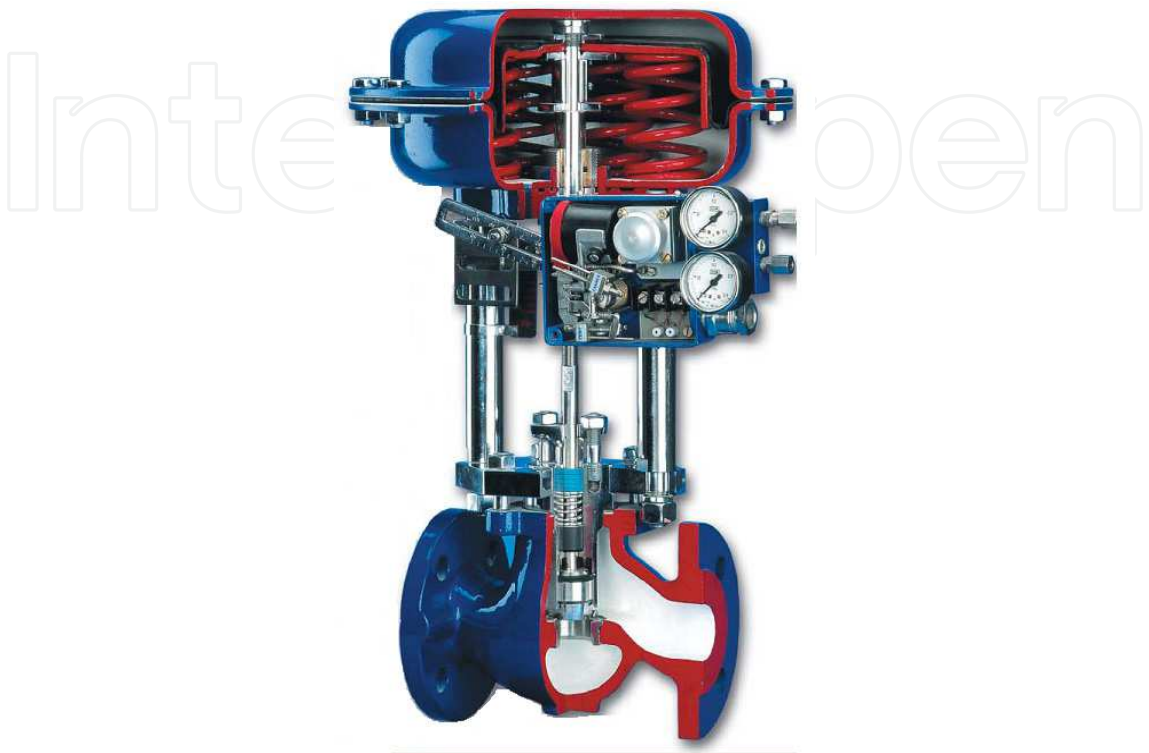

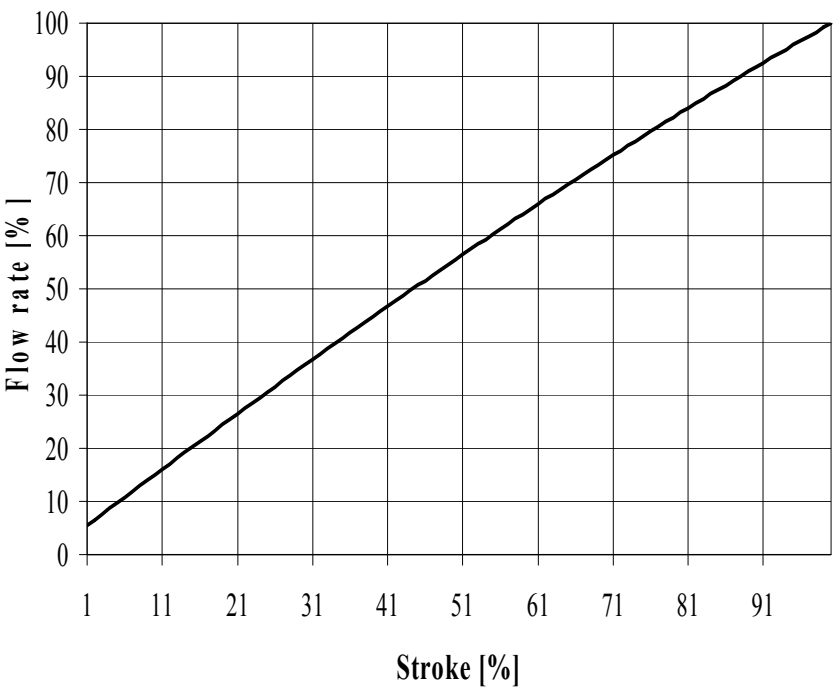Fig. 19. The control valve made by Pre-Vent Company.

Fig. 20. The work characteristic of the control valve from the studied hydraulic system.

The picture presented in figure 21 is similar to the output pressure of the pump. The conclusion resulting is that 99% of the hydraulic pump energy is lost into the control valve. For this reason, the choice of a control valve with linear characteristicis wrong, the energy being taken into consideration.
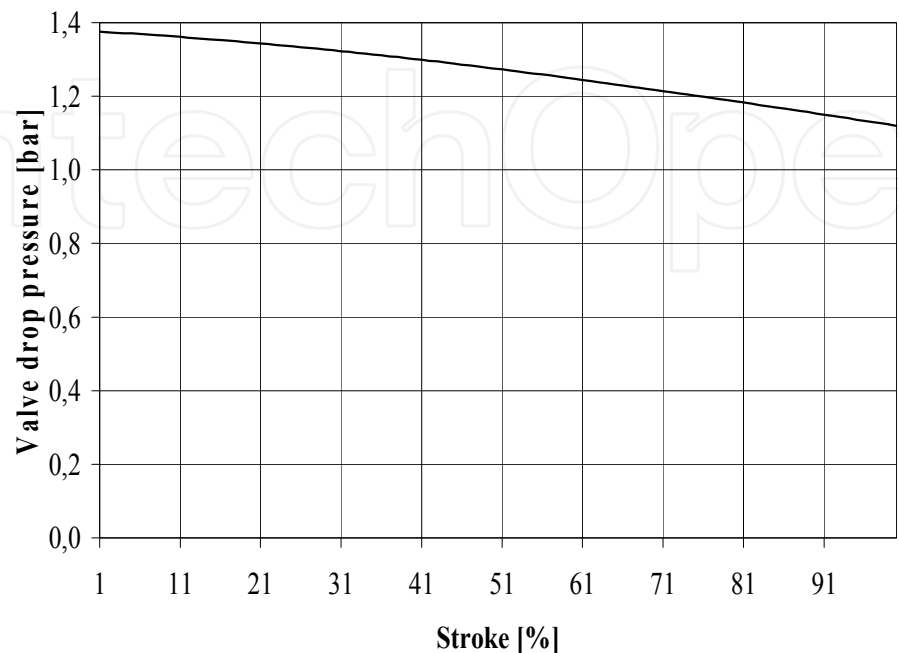

Fig. 21. The control valve drop pressure.

3.4 The control valve design and the selection criteria

The control valves are produced in series, in order to obtain a low price. For this goal, the control valve producers have realized the proper standards of the geometric and hydraulic properties. The control valves choice is a complex activity, composed of technical, financial and commercial elements. Mainly, the control valves choice represents the selection of a type or a subtype industrial data based on a control valve, depending of one or many selection criteria.

Technical criteria refer to the calculus of the technical parameters of the control valves. The financial elements include the investment value and the operation costs. The commercial elements describe the producers 'offers of various types of control valves.

The technical parameters of the control valves contain at least the flow module calculus of the control valve of the control system. The choice of the control valve involves the following elements: the constructive type of the control valve body, the standard flow module K_{vs} of the control valve manufactured by a control valve company and the nominal diameter D_n of the control valve.

3.4.1 The flow module calculus

The design relations of the control valves are divided in two categories: classical relations and modern relations. The classical relations have been introduced by Früh (Früh 2004). Theses relations are recommended for the design calculus of the control valves placed in the

hydraulic systems characterized by turbulent flow regime, hydraulic system characterized by without branches and for the calculus initialization of the other design algorithms. A short presentation of these relations is presented in table 9.

ΔP_v	Fluid type		
	Liquid	Gas	Overheated steam
$\Delta P_v < \dfrac{P_1}{2}$	$K_v = Q\sqrt{\dfrac{\rho}{\Delta P_v}}$	$K_v = \dfrac{Q_N}{514}\sqrt{\dfrac{\rho_N T_1}{P_2 \Delta P_v}}$	$K_v = \dfrac{Q_m}{31.6}\sqrt{\dfrac{v_2}{\Delta P_v}}$
$\Delta P_v > \dfrac{P_1}{2}$		$K_v = \dfrac{Q_N}{257}\sqrt{\dfrac{\rho_N T_1}{P_1}}$	$K_v = \dfrac{Q_m}{31.6}\sqrt{\dfrac{2v_2}{P_1}}$
Notation and measure units	Q – volumetric flow rate [m³/h]	Q_N – volumetric flow rate [m³$_N$/h]	Q_m – mass flow rate [kg/h]
	-	P_1 – upstream pressure [bar]; P_2 – downstream pressure [bar]	P_1 – upstream pressure [bar]
	ΔP_v - control valve drop pressure [bar]		
	ρ - density [kg/dm³]	ρ_N – normal density [kg/m³$_N$]	v_2 – specific volume of the steam [m³/kg]

Table 9. The classical design relations for control valve flow module.

The modern relations of the flow module are based on the ISA standards and are characterized by all flow regimes (laminar regime, crossing regime, turbulent regime, cavitational flow regime) and for a more hydraulic variety (ISA 1972, 1973). Also, the calculus relations are specific to the fluid type (incompressible fluid and compressible fluid).

3.4.2 The industrial control valves production

The control valves companies produces various types of standardized control valves. Each company has the proper types of control valves and each control valve type is produced a various but standardized category, defined by standardized flow module, nominal diameter and chair diameter. Each company presents their control valves offer for chemical and control engineering. In figure 22 there is presented an image of the BR-11 control valve type from the Pre-Vent Company. There are presented the standard flow module, the maximal valve travel and the offer of nominal diameter of the control valve.

3.4.3 The control valves choice criteria applied to the flow control system

The control valves choice represents an important problem of the control systems design. The control valves choice criteria are the following:

a. For each control system, there must be chosen an inherent valve characteristic of the control valve body (or a control valve type) so that all the components of the control system generate the lowest variation of the control system gain;
b. For each control system there must be chosen a working characteristic of the control valve body so that all the components of the control system to generate a linear characteristic.

The common flow control system has the structure presented in figure 1. Using the previous choice criterion, some recommendations can be made for the selection of the control valve of the flow control systems, table 10 (Marinoiu 1999). The flow control system can meet the stabilization control function or the tracking control function.

Kv_S (m³/h)	Stroke (mm)	Plug face (mm)	Valve nominal size DN												
			15	20	25	32	40	50	65	80	100	150	200	250
0,010														
0,016														
0,025														
0,040														
0,063														
0,10		6,35												
0,16														
0,25														
0,40														
0,63	20													
1,0														
1,6		9,52												
2,5		12,70												
4,0														
6,3		19,05												
10		20,64												
16		25,25												
25		31,72												
40		41,25												

Fig. 22. The BR-11 control valve type offer of the Pre-Vent Company

| Control system type	Flow transducer characteristic	Ψ	Disturbances	Inherent recommended characteristic
stabilization	$r \approx Q$	1	P_{r0}, P_v, T, v, ρ	logarithmic
	$r \approx Q^2$			linear only if $Q_{i1} \neq Q_{i2}$
	$r \approx Q$	< 1	$P_{r0}, P_v, R_1, R_2, T, v, \rho$	logarithmic
	$r \approx Q^2$			
tracking	$r \approx Q$	1	The disturbance variations are negligible as compared to the set point variations	linear
	Linearized by square root	≤ 0,3		linear
		≤ 0,3		logarithmic
	$r \approx Q^2$	1		logarithmic
	Non linearized	< 1		linear

Table 10. Recommendations for the choice of the control valves for the flow control systems.

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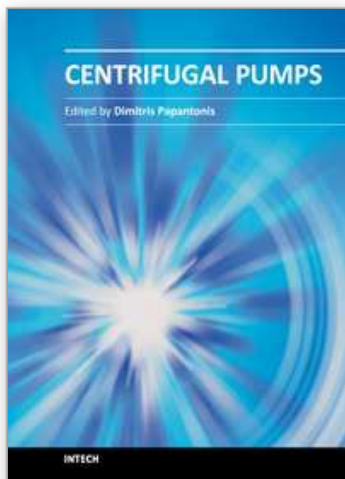
Centrifugal Pumps

Edited by Dr. Dimitris Papantonis

The structure of a hydraulic machine, as a centrifugal pump, is evolved principally to satisfy the requirements of the fluid flow. However taking into account the strong interaction between the pump and the pumping installation, the need to control the operation, the requirement to operate at best efficiency in order to save energy, the provision to improve the operation against cavitation and other more specific but very interesting and important topics, the object of a book on centrifugal pumps must cover a large field. The present book examines a number of these more specific topics, beyond the contents of a textbook, treating not only the pump's design and operation but also strategies to increase energy efficiency, the fluid flow control, the fault diagnosis.

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