SPECIFICATION OF ACTUATORS INTENDED TO USE FOR BENCHMARK DEFINITION

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Abstract. This paper serves some fundamental issues more or less important for benchmark definition in the field on industrial applied actuator diagnosis. Significant part of this paper is based on valuable tutorial [1]. Paper is primary intended as working background suitable for all the DAMADICS[*)] project members joining the actuator diagnosis challenge. This paper sets the *actuator knowledge start conditions* equal for all challenge participants. Some physical phenomena important for process understanding are briefly described. The paper is divided roughly into two parts; one that brings more general acuator description and second that provides more detailed information concerning the actuators chosen for benchmark in the polish sugar factory Lublin S.A.

Keywords: actuators, pneumatic servomotors, control valves, positioners, fault detection, fault isolation

1. INTRODUCTION[*]

Control tasks for the technological processes may be generally defined in the terms of acting on the energy and mass flows. Actuators (final control elements) are applied for real acting on that flow. Faults or malfunctions of final control elements (e.g. control valves, servomotors, positioners) are appearing relatively often in the industrial practice. The actuators are installed mainly in harsh environment: high temperature, pressure, humidity, pollution, chemical solvents, aggressive media, vibration, etc. This influenced on the final control element predicted lifetime. The malfunction or failures cause long-term process disturbs or even sometimes forces the installation shut down. Moreover, final control element faults may influence the final product quality. This is the source of potential reasonable economic losses. For fault prevention or prediction, the real time diagnostics of final control elements may be applied. Continuously or periodically performed diagnosis of actuators cuts the maintenance costs. The introduction of remote real time diagnostic of actuators may bring down the periodical inspection costs by factor 2. In such cases, the inspections and repairing of the actuators are undertaken only if necessary.

2. DEFINITIONS

Not to being confused later let us set for our purpose the following primary definitions:

Actuator or *final control element* is a physical device, structure or assembly of devices acting on controlled process. Taking into account a benchmark definition we will further understand the actuator as a set consisting of:

- control valve
- spring-and-diaphragm pneumatic servo-motor
- positioner

Control valve is the mean used to prevent, allow and/or limit the flow of fluids through control systems. Changing the state of the control valve is accomplished by a servomotor. In industrial practice this element is sometimes called as actuator. According to above given definition

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we will understand actuator in more broad sense.

A spring-and-diaphragm pneumatic servomotor can be defined as a compressible (air) fluid powered device in which the fluid acts upon the flexible diaphragm, to provide linear motion of the servomotor stem.

Positioner is a device applied to eliminate the control-valve-stem miss-positions produced by the external or internal sources such as friction, pressure unbalance, hydrodynamic forces *etc*.

Hysteresis – property of an element evidenced by the dependence of the value of the output, for a given excursion of the input, upon the history of prior excursions and the direction of the current traverse.

Linearity – the closeness to which a curve approximates a straight line.

Dead band – the range through which an input signal may be varied, upon reversal of direction, without initiating an observable change in output signal.

Resolution – the least interval between two adjacent discrete details that can be distinguished one from the other.

The dead band and resolution are measured by making small moves about the operating point. In contrast, the hysteresis and linearity are continuously varying errors, which only accumulate measurably if the control valve is stroked over most of its span.

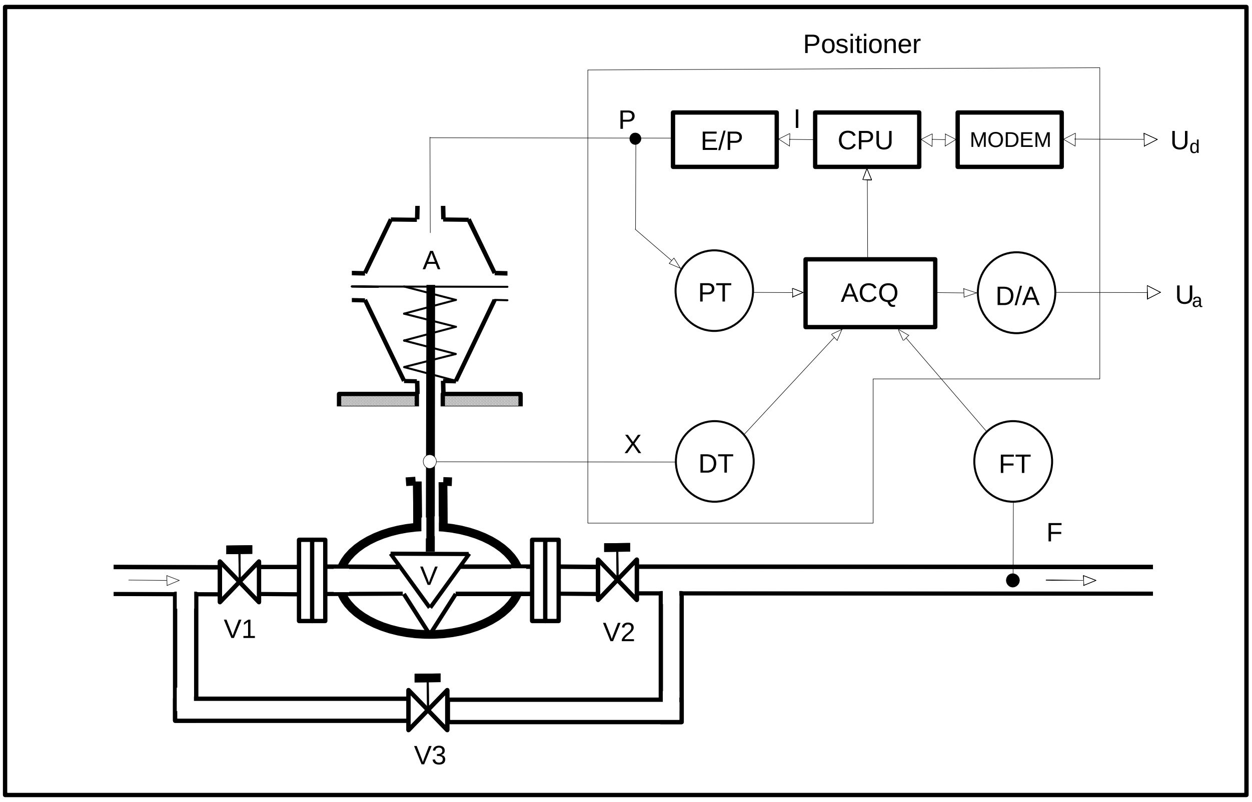


Fig.1. An example of the control valve-pneumatic servo-motor, positioner assembly.
Notations:

A	- pneumatic servo-motor	U_a	- analogue communication link
V	- control valve,	E/P	- electro-pneumatic transducer
V1,V2,V3	- hand-driven valves	DT	- displacement transducer
CPU	- positioner central processing unit	PT	- pressure transducer
ACQ	- data acquisition unit,	FT	- volume flow rate transducer
MODEM	- system for digital communication	I	- control current of E/P transducer
D/A	- digital-to-analogue converter	P	- output pressure of the E/P transducer
U_d	- digital communication link	F	- volume flow rate signal

3. CONTROL VALVE

3.1. Valve sizing

The importance of the correct sizing of control valves cannot be overemphasised. The most expensive, best performance controller is of little value if the control valve cannot correct the flow properly to maintain a desired set point. Oversized valves provide poor control and can lead to system instability, excessive wear and cycling of internal trim parts. Undersized valves generally cannot pass the required flows and thus starve the process.

The control valve sizing may be defined as follows: the flow rate of the process fluid is mathematically converted to an equivalent flow rate of a reference fluid. Then, a value size is selected which is known by test to be capable to flowing the equivalent quantity of the reference fluid at the process pressure conditions specified. For liquid flow the reference is pure water. For gas and vapour flows, the reference fluid is air at standard conditions of temperature and pressure.

Using the fluid mechanics theory, a basis control valve sizing equation can be derived from (1).

$$Q = C_v \sqrt{\frac{\Delta p}{\rho}} \qquad (1)$$

where:

Q	- flow rate	[gal/min]
C_v	- liquid sizing coefficient	
Δp	- differential pressure across valve	[p/si]
ρ	- liquid specific gravity	[p/ci]

When using SI units the basic control valve sizing equation for liquid is expressed in form (2)

$$Q = 100 \; K_v \sqrt{\frac{\Delta p}{\rho}} \qquad (2)$$

where:

Q	- flow rate	[m^3/h]
K_v	- liquid sizing coefficient	
Δp	- differential pressure across the valve	
ρ	- liquid specific gravity in upstream	[kg/ m^3]

Easy one can see from (1) and (2) the C_v and K_v values are equivalent. Taking into account that the equations (1) and (2) are sized in different physical units one can obtain:

$$K_v = k_{vc} \, C_v \qquad (3)$$

where:
k_{vc} – scaling factor
1 gallon = 3.78541 * 10^{-3} m^3
1 lb = 0.45359237 kg

3.2. Cavitation

At the *vena contracta* (smallest cross sectional area of the flow stream) the fluid pressure is minimal while the velocity is maximal.

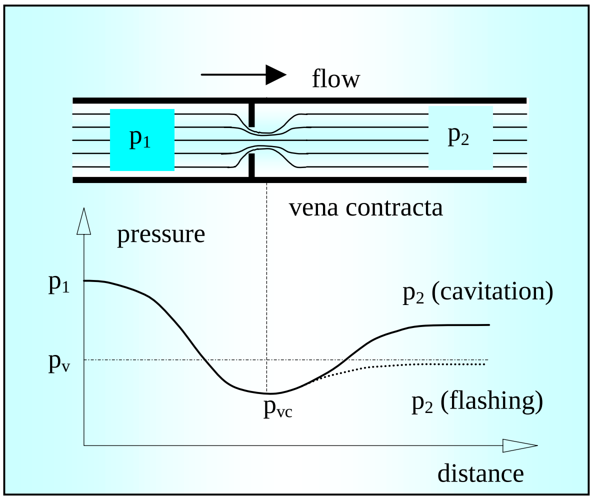


Fig. 2. Illustration of vena contracta effect.
p_1 – fluid inlet pressure
p_2 – fluid outlet pressure
p_v – fluid vapour pressure
p_{vc} – vena contracta pressure

If the pressure at the vena contracta falls below the vapour pressure of the liquid, vapour cavities forms in the flow stream.

Cavitation results if fluid outlet pressure p_2 recovers to a pressure above the vapour pressure of the liquid. As the vapour cavities collapse, **noise is generated and damage can occur**. Cavitation damage produces a rough, pitted, cinder-like surface.

If the fluid outlet pressure p_2 remains below the vapour pressure of the liquid, flashing effect occurs. **Flashing damage resembles erosion** and is distinguished by the [MPa] smooth polished appearance of the eroded surface.

3.3. Choked flow

Both flashing and cavitation limit the flow of the liquid through the valve. During flashing and cavitation bubbles begin to form in the flow stream when the pressure drops below the vapour pressure of the liquid. The bubble formation at the vena contracta restricts the amount of liquid that can be forced through the valve. A condition develops where the flow chokes. Flow is no longer increasing with decreases in downstream pressure.

A plot of the flow rate versus the square root of pressure drop Δp across the valve is a straight line (1) whose slope is equal to the valve sizing coefficient C_v. The equation imply that there is no limit to flow as long as Δp across the valve increases. If the cavitation or flashing occurs this is not longer true. With upstream pressure constant, there is a limit to the flow increase that can occur as a result of decreasing downstream pressure.

As the valve pressure drop is increased beyond the point of bubble formation, the choked flow condition is reached. At this point any further increase in Δp does not increase the flow. This differential pressure is called the *allowable Δp.* The K_m factor defined in (4) specify the pressure recovery in downstream. Greater K_m factor denotes lower pressure recovery .

$$K_m = \frac{\Delta p_{allow}}{(p_1 - p_{vc})} \qquad (4)$$

where:

Δp – fluid pressure drop across the valve

p_1 – upstream fluid pressure

p_{vc} – vena contracta pressure

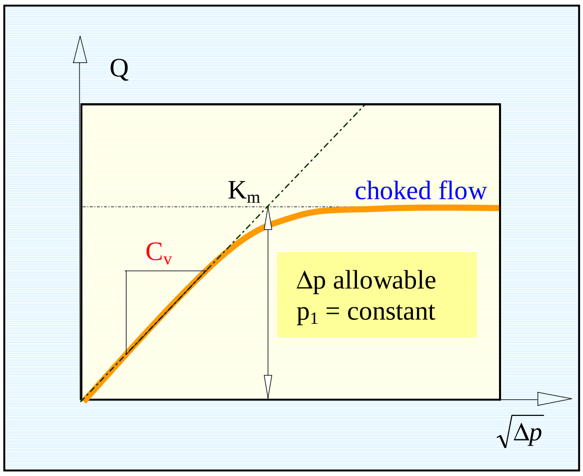


Fig. 3. Flow rate Q versus square root of pressure drop across valve. Figure get graphic interpretation of liquid sizing coefficient C_v, allowable Δp and K_m, factor.

The allowable Δp for any application must be known in order to accurately predict the flow rate. **Choked flow can result in severe damage to the valve**, and needed flow requirements may not be reached, that starving the process. The Δp at the choked flow condition is a function of the flow geometry of the control valve. The experimentally determined coefficient used to define the point of choked flow condition for any value is called K_m . To determine the allowable pressure drop that is effective in producing flow the equation (5) may be used.

$$\Delta p_{allow} = K_m (p_1 - r_c p_v) \qquad (5)$$

where:

Δp_{allow} – maximum allowable fluid pressure drop across the valve

K_m – valve recovery coefficient

p_1 – upstream fluid pressure

r_c – critical pressure ratio of the liquid

p_v – upstream fluid vapour pressure

The valve recovery coefficient K_m and critical pressure ratio of the liquid r_c can be determined from the tables and curves given by valve manufacturer.

In most globe-style valves, minor cavitation occurs at a slightly lower pressure differential than that predicted by the equation (5). In high recovery valves such as ball or butterfly valves, significant cavitation can occur at pressure drops below that which produces choked flow. So while *allowable Δp* and K_m factor are useful in predicting choked flow capacity, the point where cavitation-related problems begin may be described by a dimensionless ratio called the cavitation index K_c . This sizing limit may be expressed as a percentage of the *allowable Δp* and depends on the valve style and service conditions. For properly applied cavitation control trim this percent may be as high as 100%. On the other hand for some high recovery valves in may be as low as 50%. This data is supplied by the valve manufactures for specific hardware. For example the equation for typical V-notch ball, high-recovery valves is:

$$\Delta p_c = K_c (p_1 - p_v) \qquad (6)$$

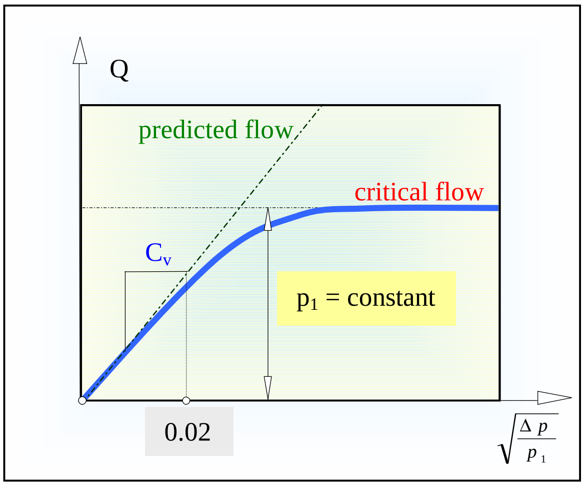
The equation (6) is invalid if the downstream pressure is equal to or less than the vapour pressure, because in this case, flashing rather than cavitation occurs. When the actual pressure differential Δp exceeds Δp_c anti-cavitation trim should be considered. The equation (6) could be used anytime outlet pressure is greater than the vapour pressure of the liquid. The typical plots of K_m and K_c decline versus valve opening in ratio 1:3. For typical V-notch ball, high recovery, rotary valves K_m is considerably higher than K_c . Addition of anti-cavitation trim to the same high recovery V-notch valve tends to increase of K_m in the upper two-thirds of the travel range. In other words choked flow and incipient cavitation occur at substantially higher pressure drops than was the case without the anti-cavitation accessory.

3.4. Valve sizing for gas and steam

The measured gas or steam flow rate shows good agreement with theoretical curve (1) at low pressure drops but a significant deviation occurs at pressure drop ratios greater than approximately 0.02 (see Fig. 4). This is because the equation assumes an incompressible fluid. When the pressure drop ratio exceed approximately 0.02 the gas behaviour is no longer like an incompressible fluid.

The serious limitation on the modified equation involves critical flow. The control valve can be treated as a simple restriction in the pipeline. As the flow passes through valve the flow streams contracts. As shown on Fig. 2 in a short distance downstream of the physical flow restriction *vena contracta* area occurs. With the steady gas flow through the valve, the stream velocity in the point of vena contracta is the highest. As the differential pressure across the valve increases, the flow increases and increases the velocity of gas in the vena contracta area. At some value of Δp however the gas reaches the sonic velocity at vena contracta. The gas doesn't travel any faster than this limiting velocity, and choked flow condition known as **critical flow** occurs.

If the pressure drop ratio is greater than the value required to produce choked flow, the modified C_v equation is no more applicable for determining the flow.



Thus the different sizing equations must be used (please refer to table 1).

Fig. 4. Flow rate Q versus square root of pressure ratio drop across valve. Figure get graphic interpretation of gas sizing coefficient C_v and critical flow.

3.5. Valve noise

Control valves have been long recognised as a major noise sources.

Mechanical vibration noise is a result of pressure fluctuations within the valve body and fluid impingement upon the moveable or flexible parts. The most prevalent source of noise resulting from mechanical vibration is the lateral movement of the valve plug relative to the guiding surfaces. Sound produced by this type of vibration lies in the region below 1500Hz as is often recognised as a metallic rattling. Mechanical vibration **may cause physical damage** of the valve plug and associated guiding surfaces.
The second source of the mechanical vibration noise is the valve components resonance. Resonant vibration produces a sound that is a single-pitched tone ranging from 3000 to 7000Hz. This type of vibration produces high levels of mechanical stress and **may produce fatigue failure** of vibrating part. Valve components susceptible to natural frequency vibration including contoured valve plugs with hollow skirts and flexible valve members such as the metal seat ring of a ball valve. Noise resulting from the mechanical vibration may be used as a signal warning that conditions exist which could produce valve failure.

Table 1. Valve sizing equations

Flow conditions	Liquid	Gas	Steam
Subsonic flow $p_2 > \frac{p_1}{2}$	$K_v = \frac{Q}{100}\sqrt{\frac{\rho_1}{\Delta p}}$	$K_v = \frac{Q_n}{5042}\sqrt{\frac{\rho_N \cdot T_1}{\Delta p \cdot p_2}}$	$K_v = \frac{G}{100}\sqrt{\frac{V_2}{\Delta p}}$
$\Delta p_2 < \frac{p_1}{2}$ *Subsonic flow*		$K_v = \frac{Q_n}{5042}\sqrt{\frac{T_1}{\rho_N \cdot \Delta p \cdot p_2}}$	
Ultrasonic flow $p_2 < \frac{p_1}{2}$	$K_v = \frac{G}{100}\sqrt{\frac{1}{\rho_1 \cdot \Delta p}}$	$K_v = \frac{Q_n}{2521 \cdot p_1}\sqrt{\rho_N \cdot T_1}$	$K_v = \frac{G}{100}\sqrt{\frac{2 \cdot V_2}{p_1}}$
$\Delta p > \frac{p_1}{2}$		$K_v = \frac{Q_n}{2521 \cdot p_1}\sqrt{\frac{T_1}{\rho_N}}$	

K_v [m^3/h] - flow coefficient
Q [m^3/h] - flow
Q_N [Nm^3/h] - gas volume flow under normal conditions (0°C, 760 mm Hg)
G [kg/h] - mass flow
p_1 [MPa] - valve inlet fluid pressure
p_2 [MPa] - valve outlet fluid pressure
Δp [Mpa] - differential pressure across the valve
ρ_1 [kg/ m^3] - specific fluid gravity in upstream flow
ρ_N [kg/ m^3] - specific fluid gravity in normal conditions
T_1 [K] - fluid temperature on the valve inlet
V_2 [m^3/kg] - specific steam volume for {p_2, T_1}
V [m^3/kg] - specific steam volume for {p_1/2, T_1}

Aerodynamic noise is generated by the turbulence associated with gas flow control. Because of the relative flow velocities, high-intensity noise levels resulting from turbulent flow are more common to valves handling gases or steam than to those controlling liquids. Aerodynamic noise is the major source of stresses or shear forces that are the property of turbulent flow. Aerodynamic noise can be classified as a non-periodic or random noise with the predominant frequencies occurring between 1000 and 8000Hz.
The main sources of turbulence phenomena in transmission lines are usually restrictions in the flow path, rapid expansions or decelerations of high-velocity gas and directional changes in the fluid stream. As the gas flows through control valve it loses some of its energy in the form of heat, noise and vibration. The loss of energy is evident in the pressure drop from inlet to outlet. Pressure is an indicator of the potential energy and fluid and velocity is the kinetic energy indicator. Because the pressure decreases while the velocity usually increases across the valve, the pressure drop is the main source of noise generation.

Hydrodynamic noise is generated by the liquid flow. The cavitation is the major source of this noise. The noise is caused by the implosion of vapour bubbles formed in the cavitation process. As the vapour bubbles move through the valve and encounter a pressure above the vapour pressure, they collapse resulting in severe damages to any adjacent valve or pipeline surface. Hydrodynamic noise sounds like gravel flowing through a pipe. Intensive cavitation can cause noise levels as high as 115dB and drastically **shorten the operating life** of installation. The noise levels for non-cavitating or flashing liquids are comparably quite low.

4. PNEUMATIC DIAPHRAGM SERVO-MOTORS

Pneumatic operated spring-and-diaphragm servomotors (Fig. 1) are the most popular acting devices in industrial applications. As it was mention above a spring-and-diaphragm pneumatic servomotor is a compressible (air) fluid powered device in which the fluid acts upon the flexible diaphragm and spring to provide linear motion of the servomotor stem. The pneumatic servomotor is intended to have several distinct purposes:

- moving the control valve closure member (disk, ball or plug) to the desired position
- holding the valve closure member in the desired position
- seating the valve closure member with sufficient force to provide the desired shutoff specification
- providing the failure mode in the event of system failure
- providing the required stroking speed

Fail modes define the servomotor in the event of loss of input signal or supply pressure. During such a failure, the servomotor's spring will move the control valve closure member to the position associated with relaxed (fully extended) spring. Fail-safe valve positioning plays a major role in which type of actuator is to be selected.

A *fail open* configuration uses the spring to open the valve in the event of a system failure.

A *fail closed* configuration uses spring to seat the value during system failure.

All three servomotors chosen for DAMADICS benchmark purposes are pneumatic linear movement direct action ones. All are of *fail open* mode type too.

4.1. Static properties

Both: static an dynamic forces must be taking into account when sizing and analysing behaviour of pneumatic servo-motor.

To move the valve closure member between two positions the following static force F_s should be considered:

$$F_S = F_u + F_{sh} + F_f + F_a \qquad (7)$$

where:

F_u - the force necessary to overcome the static unbalance of the valve plug

F_{sh} - the force necessary to provide sufficient seat load to attain the desired shutoff

F_f - the force necessary to overcome the packing friction

F_a - the force necessary to overcome the additional forces depending upon valve and/or servomotor design.

Static unbalance refers to the effect of process pressure on a specific valve plug when *the valve is seated*. Depending of the direction of flow and design of control valve, static unbalance may oppose or assist servomotor force. During end phase of shutoff operation this force is developing rapidly and from other hand when starting opening seated valve plug this force is decreasing. Unbalance effect is mostly influenced by pressure drop across the valve during stem travel This is known as balanced valve design employing the methods of equalising process pressure above and below the valve plug. These designs minimise the net fluid flow forces that create static unbalance and allow the usage of a smaller servomotor.

Seat load is the simply product of the port circumference and the lineal force value recommended by the valve manufacturer to attain the desired shutoff.

Packing friction (see Fig. 5) opposes valve steam movement in any direction. Friction is strongly dependent upon packing material applied. For example the graphite packing can produce friction in order of 5 to 10 times greater compared to those produced by TFE packing. Packing friction is a main source of the hysteresis in the assembly consisting control valve – servomotor. Packing friction may cause also a *stick-slip* effects resulting in pour stem position controlling quality performance.

Additional static forces depend upon particular valve design. The additional friction may be produced for example by the piston rings

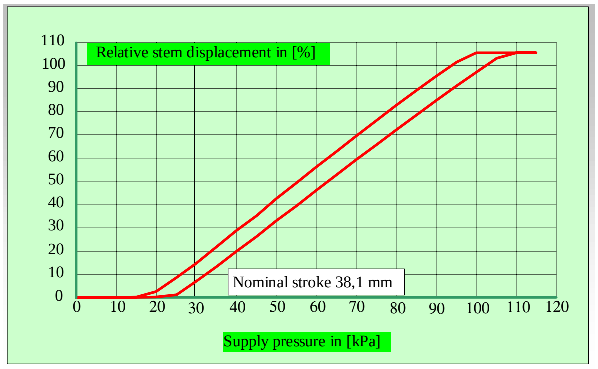


Fig.5. An example of experimental relative stem displacement versus servomotor supply pressure of reverse-acting pneumatic-diaphragm servomotor. Hysteresis loop results mainly from the packing friction. Please note that hysteresis is approximately 10% wide.

4.2 Bench set

Bench set is expressed as a pressure range through which the servomotor will begin and end its stroke when *disconnected* from the control valve. For example, a typical bench set range for a direct-acting servomotor is equal to 20 – 100 kPa. In other words the servo-motor begin its stroke when 20 kPa loading pressure is applied to the diaphragm and reach its maximum rated travel with loading pressure of 100 kPa when detached from the valve under static conditions.

The purpose of identifying bench set is to provide a consistent and standard means with which manufacturers and users can describe and verify servomotor performance off-line. Additionally it provides a simple method of quantifying what forces will be used for initial force F_i and spring compression F_c over the required travel and what forces will remain available for satisfying control valve force requirements.

Most servomotors used for sliding-stem valves have spring adjusters which allow manipulation of spring compression. Therefore by altering spring compression, bench set specifications may by fine tuned to specific applications.

Two bench sets are used for servo-motors used for DAMADICS benchmark: standard 20..100kPa, and 40..200 kPa. For details please refer to table 2.

Two distinct diaphragm servomotor designs are known:

- direct acting servo-motor and
- reverse acting servo-motor

Direct acting servomotor is defined as a device whose travel member (stem) is moving in direction that oppose both: the spring and valve static unbalance forces, when increasing loading pressure.

Reverse acting servo-motor is defined as a device whose travel member (stem) is moving in direction that oppose the spring force and is unidirectional with valve static unbalance force, when increasing loading pressure.

4.3 Dynamic properties

Dynamic properties of pneumatic diaphragm servomotor are strongly dependent on stem movement direction (see Fig. 6, 7). This phenomena can be explained when taking into consideration the energy transformation process.

Let us consider direct acting servomotor. Let the Δp input signal increase. The force ΔF generated oppose the dynamic and static forces induced mainly by:

- accelerated mass load
- hydrodynamic forces induced by flow rate changes
- hydrostatic unbalanced forces
- packing friction force and
- diaphragm and spring compression forces

Let now Δp input signal decrease. Diaphragm and spring decompress. This forces act now in the same direction as ΔF induced by decreased pressure input. The energy stored in diaphragm and servomotor spring containers will be loosed acting as additional input forcing the stem movement. This can explain the asymmetry of step response characteristics.

The third order well damped system give the rough approximation of dynamic servomotor behaviour (pressure is the input while stem displacement is output). The dominant time constant is a function of servomotor chamber mechanical dimensions and air inflow rate limitation. Typical values are ranging from 1 to 50s.

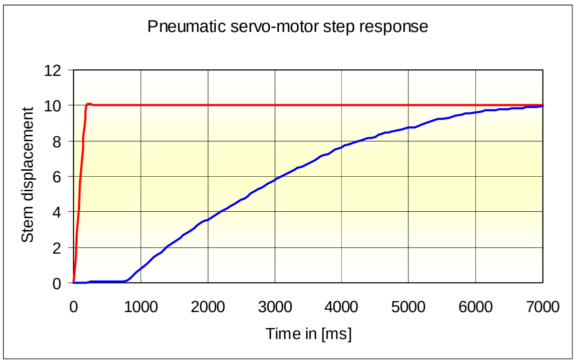


Fig. 6. An example of positive step response of direct acting pneumatic servomotor. The pressure step value ranges beyond the servomotor bench set (20..100kPa). This explains relatively large dead time.

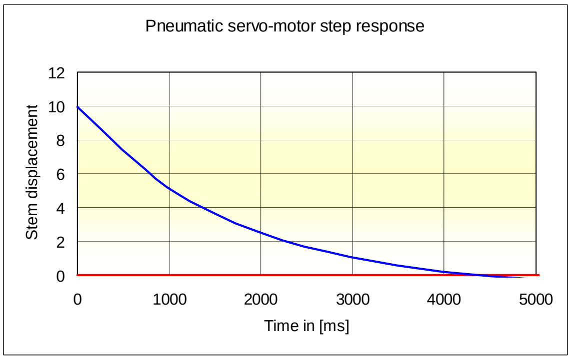


Fig. 7. An example of negative step response of direct acting pneumatic servo-motor. The response time is significantly shorter compared to observed in Fig. 6.

5. CONTROL VALVE POSITIONER

Positioners are applied to eliminate valve-stem miss-positions produced mainly by changing load, friction forces, supply pressure and flow deviations, system non-linearities e.t.c.

In other instances they are used to allow the implementation of system design technique such as for example split ranging.

To provide good control most systems are designed to be linear. Non-linearities when left uncompensated have the potential to degrade system performance.

The common cause of non-linear actuator response is friction and backlash. There are many potential sources of friction within the valve and servomotor assembly. Friction is expected from the normal operation of components such as: stem packing, guide bushings and seals. Accumulated dirt and debris on the valve stem, accumulations from cooking fluids or solidified residue may increase the amount of friction present.

There are two basic friction symptoms: dead band and hysteresis.

Dead band is the condition where is no change in output observed for a given change in input signal. The dead band measure is the ratio of the input signal span through which the input may be varied without producing an output change to the whole input span. When refer to Fig.5, one can easily estimate the 10% servomotor dead band. When uncompensated result in actuator cycling (limit cycle conditions met). To illustrate limit cycling assume a closed loop system with PI controller (what is the case in benchmark) Because of significant friction portion present increasing controller output may not produce servomotor stem position. This causes an increasing control error so the controller output continues to increase producing more and more actuator thrust. When induced force finally exceed static friction the servomotor suddenly start moving the valve into new position (the kinetic friction will decrease). However the moving valve member may overshoot or undershoot the desired stem position. This creates control error that dynamically amplified by controller cause next cycle. The limit cycle phenomenon **is extremely limiting the actuator lifetime** because of extensive wear of packing, sliders and bushing. Limit cycle can be identified as a low frequency component of control error by constant controller set point values.

In this case the positioners may be applied with the aim to compensate the dead band effect and eliminate the limit cycle phenomena. The effect of applying positioner to the servomotor with considerably broad dead band (see Fig. 5) was shown on Fig. 8.

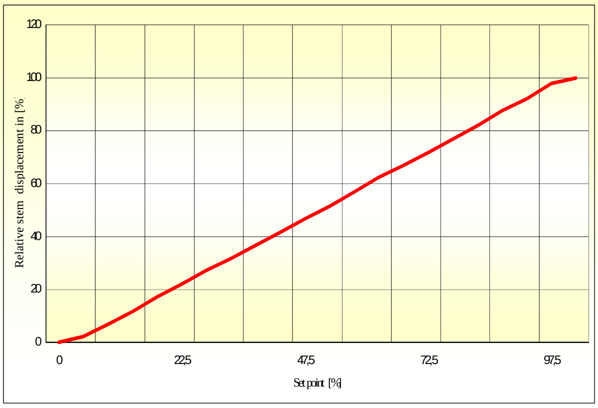


Fig.8. An example of experimental relative stem displacement versus control signal when positioner applied to servomotor with static characteristics shown on Fig. 5. Significant hysteresis and dead band reduction is to be observed. Static linearity improvement is to be noted.

The *dead band measure* is the maximum relative separation in horizontal direction between measured indications of the measured variable when the input is first increased from one end of the scale to the other and then decreased.

The *hysteresis* in opposite to dead band is the maximum relative separation in vertical direction between measured indications of the measured variable when the input is first increased from one end of the scale to the other and then decreased.

The positioners intend to use for benchmark purposes are all the same type. The positioners belongs to the group of microprocessor based smart devices. However the communication facilities are limited only for two current (4..20 mA) signalling loops. One loop provides set point value, where second signal the actual valve stem position. The positioner coarse parameter values are given in table 4 while the simplified positioner internal structure is shown on Fig. 9. The parameters will be tuned after positioner mounting and process running. The up-dated parameter values will be available in the next release of this document.

6. THE SET OF FAULTS

The total of 19 faults $\{f_1 .. f_{19}\}$ are distinguished [5] in the assembly consisting of: control valve, pneumatic servomotor and positioner. The faults are classified into four following groups:

- Control valve faults $\{f_1 .. f_7\}$
- Pneumatic servo-motor faults $\{f_8 .. f_{11}\}$
- Positioner faults $\{f_{12} .. f_{14}\}$
- General faults/external faults $\{f_{15} .. f_{19}\}$

Control valve faults

f_1 - valve clogging
f_2 - valve or valve seat sedimentation
f_3 - valve or valve seat erosion
f_4 - increased of valve or bushing friction
f_5 - external leakage (bushing , covers, terminals)
f_6 - internal leakage (valve tightness)
f_7 - medium evaporation or critical flow

Pneumatic servo-motor faults

f_8 - twisted servo-motor's piston rod
f_9 - servo-motor's housing or terminals tightness
f_{10} - servo-motor's diaphragm perforation
f_{11} - servo-motor's spring fault

Positioner faults

f_{12} - electro-pneumatic transducer fault (E/P)
f_{13} - rod displacement sensor fault (DT)
f_{14} - pressure sensor fault (PT)

General faults/external faults

f_{15} - positioner supply pressure drop
f_{16} - increase of pressure on valve inlet or pressure drop on valve output
f_{17} - pressure drop on valve inlet or increase of pressure on valve output
f_{18} - fully or partly opened bypass valves
f_{19} - flow rate sensor fault (FT)

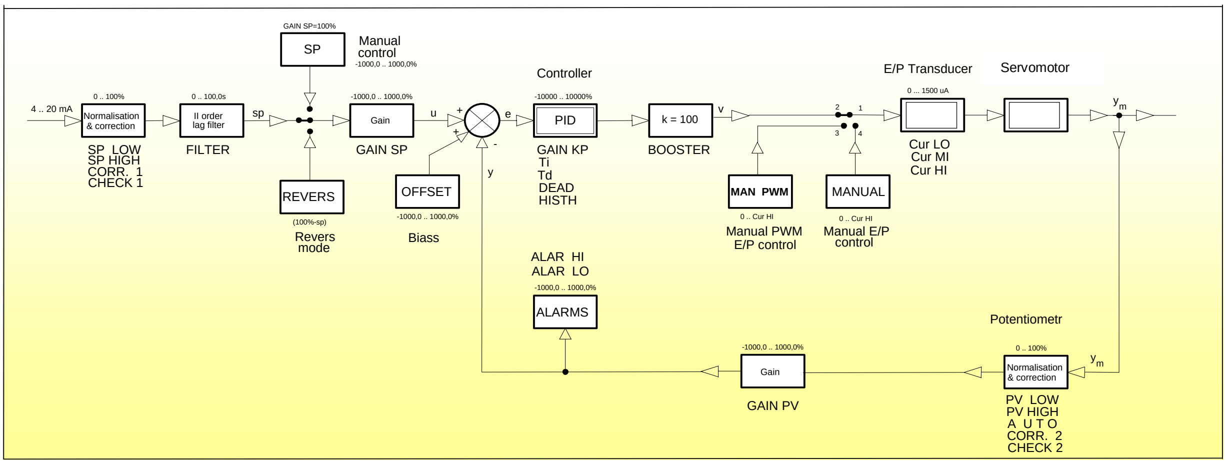


Fig.9. Simplified block structure of smart positioner A 785.

Table 2. Specification of control valves and pneumatic servo-motors considered for benchmark purposes (to be verified)

Item	Parameter	Thin juice level control loop in the first stage of evaporation station **LC51_03**	Thick juice outflow control loop from the fiveth stage of evaporation station **FC57_03**	Water level control loop in the fourth boiler station **LC74_20**
1.	Servo-motor type	37-18	37-20571A	37-9
2.	Servo-motor manufacturer	POLNA S.A. PL	POLNA S.A. PL	POLNA S.A. PL
3.	Servo-motor stroke	38,1 mm	38,1 mm	38,1 mm
4.	Servo-motor diaphragm diameter	527 mm	527 mm	280 mm
5.	Effective diaphragm area	1290 cm^2	1290 cm^2	290 cm^2
6.	Maximal supply pressure	240kPa	240kPa	240kPa
5.	Supply pressure	140 kPa	140 kPa	240 kPa
6.	Control signal pressure nominal range (bench set)	20 ..100 kPa	20 ..100 kPa	40 ..200 kPa
7.	Fail mode	fail-open	fail-open	fail-open
8.	Reversal mode	no	no	no
9.	Hysteresis	±2%FS	±2%FS	±2%FS
10.	Linearity	±4%FS	±4%FS	±4%FS
11.	Ambient temperature	-30 .. 70°C	-30 .. 70°C	-30 .. 70°C
12.	Maximal relative humidity	98%	98%	98%
13.	Control valve nominal diameter	DN 200	DN 100	DN 50
14.	Flow range	0 .. 400 t/h	0 .. 80 t/h	0 .. 40 t/h
15.	Fluid	thin juice	thick juice	water

Table 3. Specification of typical controller settings in the control loops chosen for benchmark definition (to be verified).

Item	Control loop	Thin juice level control loop in the first stage of evaporation station	Thick juice outflow control loop from the fifth stage of evaporation station	Water level control loop in the fourth boiler station
1.	Loop descriptor	LC51_03	LC-57_03	LC74_20
2.	Controller type	P I	PI	PI
3.	k_p	1.5	0.5	5
4.	k_i (1000/T_i [s])	0.08	0.12	0.15

Table 4. Specification of the typical positioner settings of the actuators chosen for benchmark definition (to be verified).

Item	Control loop	Thin juice level control loop in the first stage of evaporation station	Thick juice outflow control loop from the fifth stage of evaporation station	Water level control loop in the fourth boiler station
1.	Loop descriptor	LC51_03	LC-57_03	LC74_20
2.	Positioner type	A785	A785	A785
3.	Positioner manufacturer	Controlmatica PL	Controlmatica PL	Controlmatica PL
4.	Positioner action	P	P	P
5.	k_p factor	40	40	40
6.	Set point gain factor	1	1	1
7.	Process value gain factor	1	1	1
8.	Set point filter	2 order lag	2 order lag	2 order lag
9.	Set point filter time constant	0.3 s	0.3 s	0.3 s
10.	Set point span (0.. 100%)	4..20 mA	4..20 mA	4..20 mA
11.	Process value (0..100%)	4..20 mA	4..20 mA	4..20 mA
12.	Positioner reverse mode	no	no	no
13.	Set point correction look up table	off	off	off
14.	Process value internal correction look up table	off	off	off
15.	Set point additional bias	0	0	0
16.	Speed limit	off	off	off
17.	Single/double action	single	single	single
18.	Supply presssure	140 kPa	140 kPa	240 kPa
19.	Transmitting distance	100 m	80 m	200 m

7. REFERENCES

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