The analysis of the mechanic features for the pipes and compensators from the hydraulic systems

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Abstract— This article shows the results of such a study, which has been triggered by a technological incident: the wrenching from the compensator and its axial shifting of a large diameter air pipe in a wastewater treatment plant, event that occurred after many years of normal operating. The phenomenon that occurred can be justified only as a result of action of some axial forces, taking into account that axial shifting of rectilinear pipe sectors is not impeded by any special blocking device, but only by the friction of the pipe on its supports (axially) and by the resistance torque generated by its own weight (in the supports of the perpendicular sectors). The friction forces on supports correspond to the total weight of the studied pipe (length and number of flanged couplings) and to the friction factor between pipes and supports. The axial forces that act upon bends correspond to the action's direction is sensibly depending of friction factor between mains and supports. *The vertical force on the supports V* corresponds to the quasi-even distribution of the total weight of the free shifting sector, on the n_r supports that exist on its route. Deformation occurring in the console pipe sector allows an axial shifting of the perpendicular sector, corresponding to the arrow in its free end *f*.

Keywords—friction force, deformation arrow, inertia momentum, pressure force, temperature, weight force.

I. Introduction

The behaviour in time of fluid pipes is determined by the numerous quantifiable factors took into account by corresponding values of a quasi-normal operating in specific conditions of locations. During operation may occurred situations which, due to unusual combinations of factors, lead to serious loss of balance which impacts on the structures' stability and eventually leads to serious or less serious incidents. Assessing such events and identifying their causes bring useful information which can optimize models and criteria used in design process and furthermore, may lead to enhancing of safety and efficiency of respective systems, [1, 7].

The study is also very valuable because all plant's personnel was believed that the accident has been caused by the relatively high temperature of transported air in the pipe and hence by large temperature variations. Subsequently these theories have been proven to be false, [10, 11].

The most probable causes have been identified through a study of the structure's mechanical behaviour, under the complex stresses that impacted it at the existing conditions of the accident and considering also the way of how these have evaluated. The proof of identified causes is made by the fact that, in the system's evolution, all corresponding circumstances have occurred only in the moment when the incident took place, [2].

The physical and mechanical characteristics of the air pipe are provided by the topographic measurements and the current pipe route, constructive details of materials, $d_e \times s$, the compensators' features, flanges and bolts, the mounting temperature (t_m = 15°C) and certain specific properties of the fixings which support the pipe, [4].

The conditions are represented by the parameters of operational ratings for the air plant: flow Q' = (9.500 ÷ 21.000) Nm^3/h per blower; 1 ÷ 3 active blowers; pressure p = (0,6 ÷ 0,8) bar; fluid's operating temperature t = (50 ÷ 70) °C; maximal variation of pipe's temperature Δt < 100° C. The climatic features of geographical area are: minimal recorded temperature –30,6° C; maximal recorded temperature: +40,0° C; annual average number of snowy days: 134 d/yr; average thickness of snow layer: 310 mm - in December and January; main winds blowing on North-West direction (22,8 %) and on East direction (14,5 %); NW wind monthly average velocity: (4,9 ÷ 6,7) m/s; 1/10 frequency maximal wind velocity: 27 m/s, [5].

II. Problem Form

Circumstances in which the incident occurred: last week of year 2009; frosty weather and sleet; plant operating on auto mode, by night.

Incident consisted of a detachment of the air pipe from the *C*2 compensator which is located in the aeration tank area (Fig. 1) and the shifting of the upstream pipe sector on approximately 60 cm distance, with a subsequent deformation of the perpendicular pipe sector (*12 – 14*), (fig. 2).

After replacing the pipe on position, with hydraulic jacks, the coupling with the compensator has been mounted with shaped seals.

Fig. 3 presents the compensator on the air pipe Dn 1200 for the air transport at aeration basin 16 C1 from the cleaning station Dancu of Iasi.

Stability of the pipe sector has been provided by blocking the axial motion with a metallic shaft supported against the secondary clarifier's structure present nearby the pipe, section *D*.

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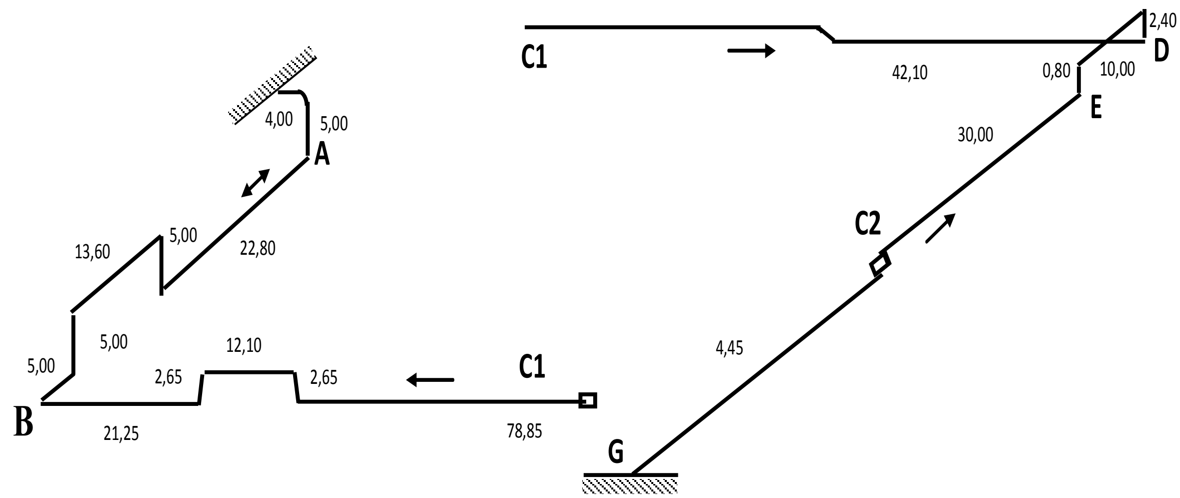

Fig. 1. Diagram of relevant pipe sectors study.

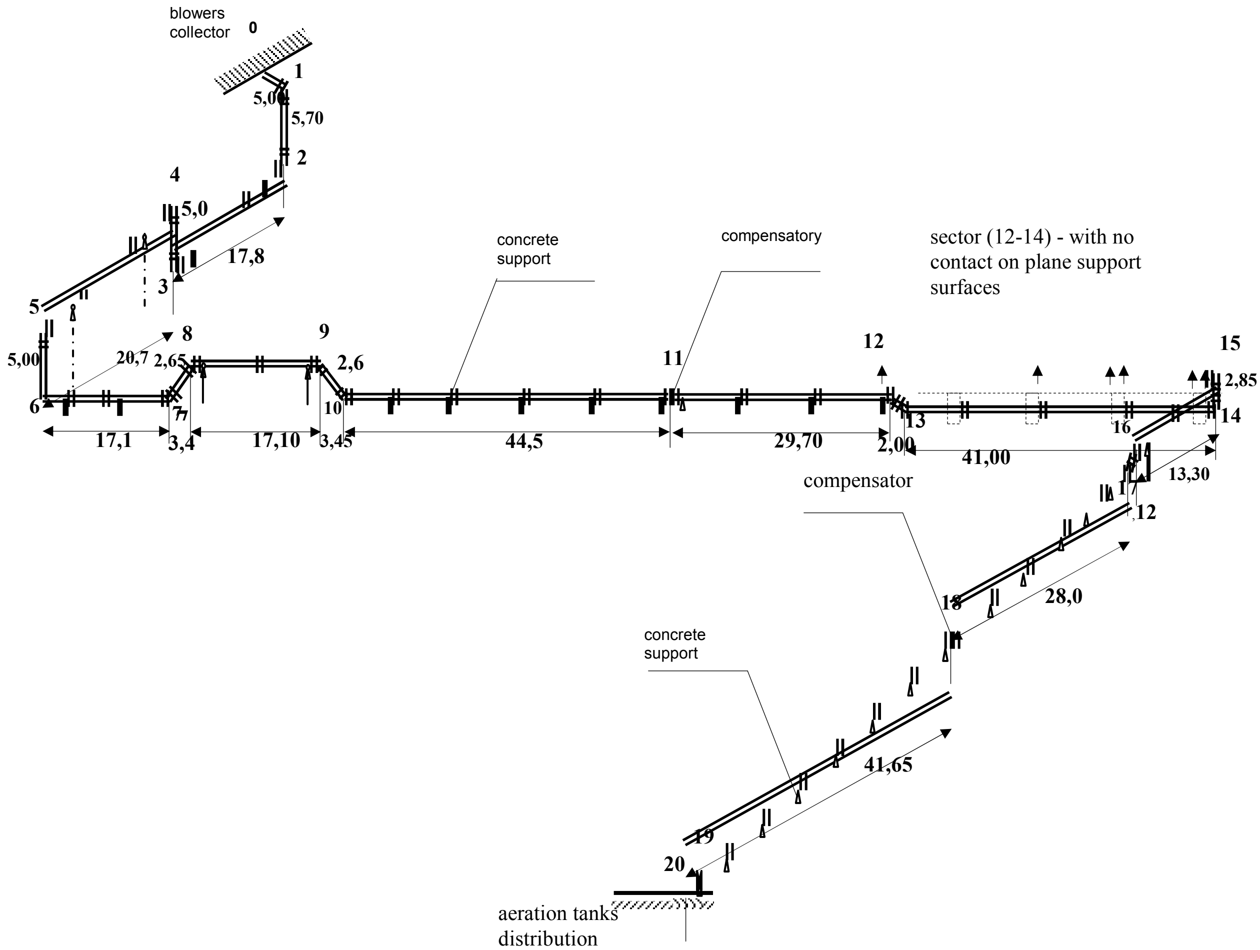

Fig. 2. Isometrical diagram of air main Dn 1200 mm

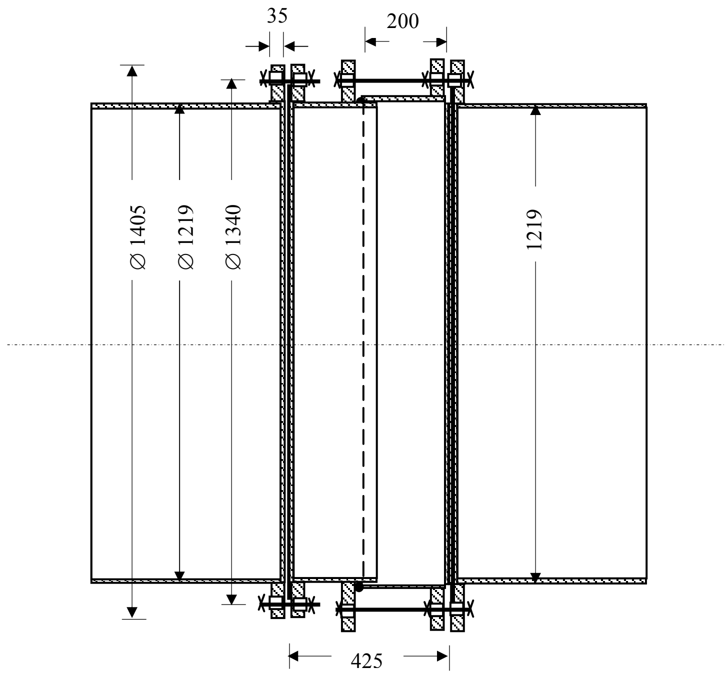

Fig. 3. Compensator on the air pipe Dn 1200 for the air transport at aeration basin 16 C1 from the cleaning station Dancu of Iasi.

According to personnel's' statements the phenomenon started with air leaks on the compensator's o-shaped seal. These leaks slowly developed, while pipe sector *C*1 - *D* was getting deformed and while pipe sector *C*2 - *D* was axially shifting, (Fig. 2). The acute phase (detachment from compensator and shifting) occurred by night when the blower plant was on auto mode. The air leaks being heavier and heavier, the air flow need grow on and on till the second blower automatically has been started up by the system. The two blowers, parallel operating, led to a serious rise of air pressure which eventually triggered the start up of the third blower. All these conditions led to the acute phase above described.

Given the extent of pipe's shifting (approximately 60 cm) and the circumstances of the event (frost, sleet, snow) this cannot be directly declared to be caused by temperature variation because, even at Δt = 100 ^{o}C, the main's length variation (on the aeration tank) would never exceed ΔL = 10 cm.

The phenomenon can only be explained as a result of action of axial stresses, in the conditions when axial shifting of rectilinear pipes is not impeded by any stopping device but only by the friction of the main against its supports (on axial direction) and by the resisting torque developed (on the supports of perpendicular pipe sectors) by the pipes' own weights, [3].

So that the phenomenon to be explained it has to be stated that, because the presence of the two compensators, the air pipe in question can be illustrated in terms of mechanical behaviour as is shown in Fig. 4.

Therefore the next items can be identified:

- Three pipe sectors that operate quasi-independently such as three beams (having annular sections) and having different spatial shapes and different connecting points: ABC1, C1DC2 and C2G;
- Three pipe sectors relevant for the calculation of length variation induced by temperature variation: AB, BD and DG.

Considering the air main configuration, the axial forces occurring in bends are balancing themselves two by two, their result being null on each of the system's pipe sectors, [7].

Presence of compensators *C*1 and *C*2 may lead to axial unbalanced forces that might induce shifting of rectilinear pipe sectors; this may occur in (Fig. 1 and 2):

- Bend *6* (pipe sector *B*, Fig. 2), action towards the exterior, on direction *11 - 6*;
- Bend *14* (pipe sector *D*, Fig. 2), action towards the exterior, on direction *11 - 14*;
- Bend *15* – force transferred through the adjacent vertical pipe sector *14,* on which acts towards the exterior, on direction *18 - 14.*

When system is operated balance of axial horizontal forces is provided by (Fig. 4):

- Friction force between pipe and pipe sector's supports on which direction it occurs.
- Horizontal reactions in supports of the perpendicular pipe sector, on which it acts.
- Resilience of consoled pipe sectors, between the applying point and first active support.

The consoled pipe sector *C*1 - *D*, which limits axial shifting of sector *C*2 - *D*, will be deformed by bending stresses, according to its length and its resilience modulus of the pipe's transverse section, depending on the force that act upon the free end. This deformation of the consoled sector will allow an axial shifting of the perpendicular pipe sector, which will be equal to the arrow of the free end *f*.

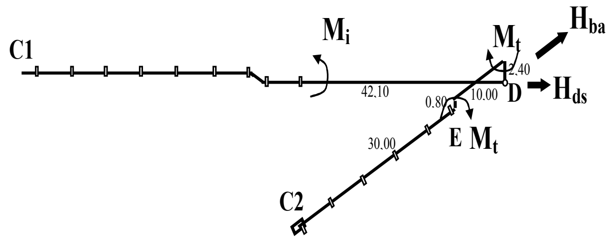

Fig. 4. Stresses that might induce axial shifting of air main's C2 - D sector.

Considering the items above the study of the main's mechanical stability included the following:

- Calculation of main loads that act upon studied system: weight forces G, pressure forces P, hydrodynamic forces in bends F_x and friction forces in supports Q_f.
- Calculation of axial horizontal forces in unbalanced bends.
- Calculation of vertical and horizontal forces that occur in supports.
- Checking the stability to rolling of the pipe, on its support.
- Calculation of unitary efforts and calculation of sectors' deformation in console.
- Assessing axial shifting of perpendicular pipe sector.
- Checking distance between supports.
- Calculation of maximal variation of rectilinear sectors' lengths when temperature varies during operation.
- Establishing the number of compensators needed to prevent the main's loss of tightness and their type.

Calculations have been made for each pipe sector separately considering the specific features of supports and the actual position of the pipe compared to these, [8, 9]. It has to be stressed that the study accorded special interest to deteriorated sector C1DC2, but also checking the existing situation on sector ABC1.

On pipe sector ABC1 the air main is layer on supports made of low grade reinforced concrete, which is roller-supports mounted on poles, in high areas where walkways are over-crossed.

On pipe sector C1DC2, the air main is supported as it follows:

- On low grade concrete supports between sectors 11 and 12 (Fig. 1);
- On metallic surfaces that cover manholes of the secondary clarifier between sectors 13 and 14;
- On the metallic supports of the aeration tank's trestle bridge.

The console sector which is the weak link that impacts on the system's stability is located in the pipe sector laid on the metallic manholes of the clarifier, area in which some 30 m of pipe is literally suspended (with no contact with supports). Study has been developed on Math Cad and Math Lab software of author, with classic methods of the field.

III. Experimental Results

The even distributed head of main's weight [kN/m] during operating period corresponds to the pipe's diameter and to its wall thickness, respectively to the air density (kg/m^3), depending on pressure, Table I.

TABLE I
THE VARIATION OF AIR DENSITY, AIR WEIGHT AND PIPE WEIGHT

air density	1,0	1,1	1,2	1,3	1,4
air weight	0,011	0,012	0,013	0,014	0,015
Pipe weight	3,059	3,060	3,061	3,062	3,063

In calculation the average value is adopted G_c = 3,061 kN/m. Weight of a flanged coupling is G_f = 2,326 kN/buc. Pressure force in bend (A = 1,128 m^2) varies depending on air pressure in main, between P = 56,379 kN, at pressure p = 0,5 bar and P = 90,207 kN, at p = 0,8 bar, (Table II).

TABLE II
THE VARIATION OF PRESSURE FORCE *P* DEPENDING ON AIR PRESSURE *P*

air pressure p [bar]	0,5	0,6	0,7	0,8
pressure force P [kN]	56,379	67,655	78,931	90,207

The hydro dynamical force in the 90^o bend varies very little compared to air density and flow in the main, depending, as the pressure force does, on the operating pressure, (Table III), [6].

TABLE III
THE PRESSURE *P* VARIATION DEPENDING ON AIR FLOW *Q*

air flow [m^3/s]	Pressure [bar]		
	0,6	0,7	0,8
5	67,686	78,962	90,238
10	67,779	79,055	90,331
15	67,934	79,210	90,486
20	68,152	79,427	90,703

The analysis is realized at the air pipe Dn 1200 mm from the cleaning station Dancu of Iasi. The air pipe is made on iron OL 1020; the outside diameter is D_e = 1,219 m; the wall thickness is δ = 0,0104 m. The pipe's weight is G_c = 312 daN/m and the iron's specific weight is γ_o = 7800 daN/m^3, (Fig. 3).

The longitudinal structure of the air pipe Dn 1200 from the cleaning station Dancu of Iasi is presented on three directions proper of the right (*X, Y, Z*) system in the following figures: for *Z* axle Fig. 5; for *Y* axle Fig. 6; for *X* axle Fig. 7.

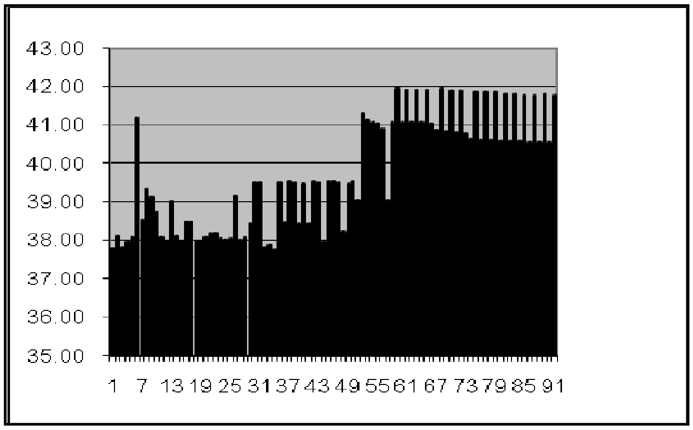

Fig. 5. The longitudinal structure of the air pipe Dn 1200 from the cleaning station Dancu of Iasi on Z direction of (*X, Y, Z*) system

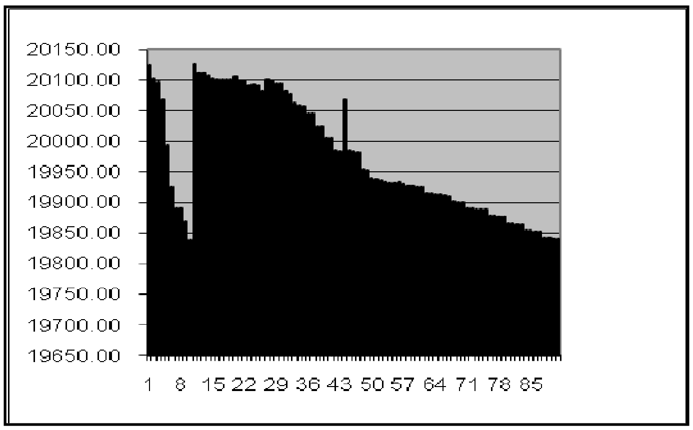

Fig. 6. The longitudinal structure of the air pipe Dn 1200 from the cleaning station Dancu of Iasi on Y direction of (X, Y, Z) system

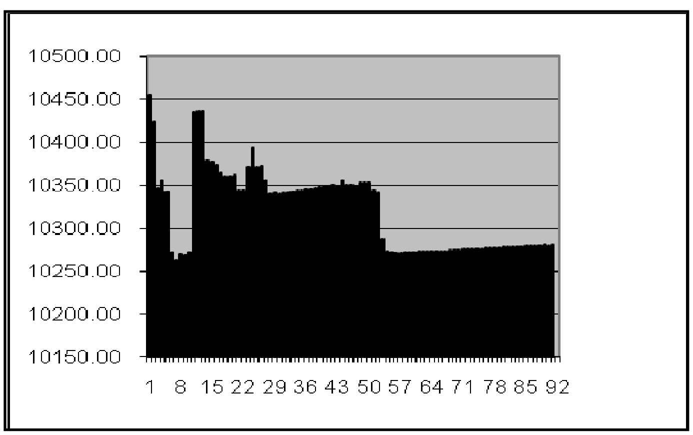

Fig. 7. The longitudinal structure of the air pipe Dn 1200 from the cleaning station Dancu of Iasi on Z direction of (*X, Y, Z*) system

The mechanically and geometrical characteristic for the flange splice are following:

- Flange: material OL37.2k, STAS 500/2 – 80; bursting resistance $\sigma_r \geq 360$ n/mm^2; diameter d = 1219 mm; d_1 = 1405 mm; d_2 = 1340 mm; n x d_3 = 32 x 33; outside diameter d_e = 1223 mm; b = 40 mm; e = 5 mm; flange weight = 101 kg/ piece.
- Bolt: material M 30 x 110 STAS 920 – 69; bolt weight = 0,824 kg/ piece.
- Screw: material B – M 30 STAS 922 – 75; screw weight = 0,216 kg/ piece.
- Armatures: material 6 – 1200/5 STAS 1733 – 79/M; inside diameter d_i = 1220 mm; outside diameter d_e = 1310 mm; armatures weight = 1,8 kg/ piece.

The splice's weight through flange is G_f = 237,08 daN/m.

The axial pressure force $P(p)$ at direction's upheaval is calculated with following mathematical relation:

$$P(p) = A(\delta).p.10^4 = \frac{\pi}{4}.(d_e - 2.\delta)^2.p.10^4;$$
$$A(\delta) = \frac{\pi}{4}.(d_e - 2.\delta)^2 = 1{,}128 \ \ m^2. \tag{1}$$

The hydro dynamic force in blend can be calculated for 30°, 45° and 90°.The air speed in pipe v varies for flow Q = (5 ÷ 25) m^3/s between values v = (4,434 ÷ 22,171) m/s:

$$v(Q) = \frac{Q}{A(\delta)} = \frac{Q}{\frac{\pi}{4}.(d_e - 2.\delta)^2}. \tag{2}$$

The hydro dynamic force in blend has two compounds:

- Horizontal compound F_x has the form:

$$F_x(Q,\rho,p,\alpha) = 10^{-1}.Q.\left(v(Q).\rho + 10^5.\frac{p}{v(Q)}\right).$$
$$.(1-\cos\alpha), \ \ [\mathrm{daN}]. \tag{3}$$

- Upright compound F_y has the form:

$$F_y(Q,\rho,p,\alpha) = 10^{-1}.Q.$$
$$.\left(v(Q).\rho + 10^5.\frac{p}{v(Q)}\right).\sin\alpha, \ \ [\mathrm{daN}]. \tag{4}$$

The 90° bend has the two equal compounds: $F_x = F_y$.

The work conditions are following: flow Q = (1,2 ÷ 22) m^3/s, pressures p = {0,6; 0,7; 0,8} bar. It reckon with the air density ρ = {1,1; 1,4} kg/m^3.

The hydrodynamic force varies in the 90° bend depending on flow Q, pressures p and air density ρ = 1,1 kg/m^3, (Fig. 8).

For an air density ρ_{aer} = 1,4 kg/m^3, when an average 10 m^3/s flow is transported, hydrodynamic force takes values between F_x = 67,779 kN, at p = 0,6 bar and F_x = 90,331 kN, at pressure p = 0,8 bar, (Fig. 9):

1. *The friction forces on supports*, (Fig. 10) correspond to the total weight of the studied pipe (length and number of flanged couplings) and to the friction factor between pipes and supports. For friction factors having values μ = (0,1 ÷ 0,4) the

friction force that impedes the axial shifting takes different values from a sector to another as it follows Table IV.

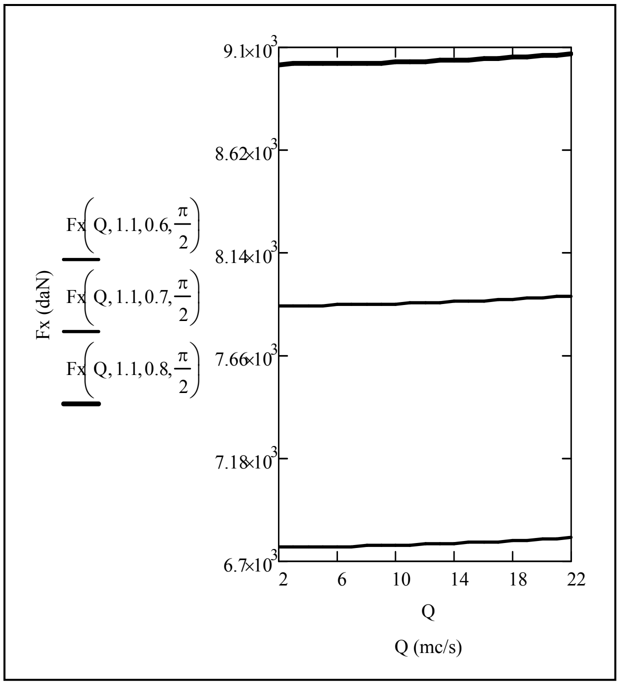

Fig. 8. Horizontal hydraulic force F_x depending on air flow Q for bend at 90° for an air density ρ_{aer} = 1,1 kg/m^3, from air pipe Dn 1200 mm at the cleaning station Dancu of Iasi.

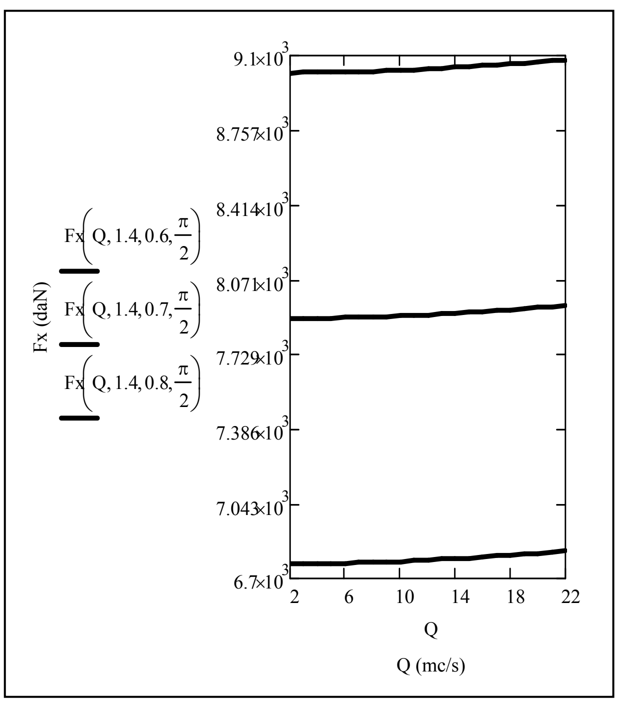

Fig. 9. Horizontal hydraulic force F_x depending on air flow Q for bend at 90° for an air density ρ_{aer} = 1,4 kg/m^3, from the air transport pipe Dn 1200 mm at the cleaning station Dancu of Iasi.

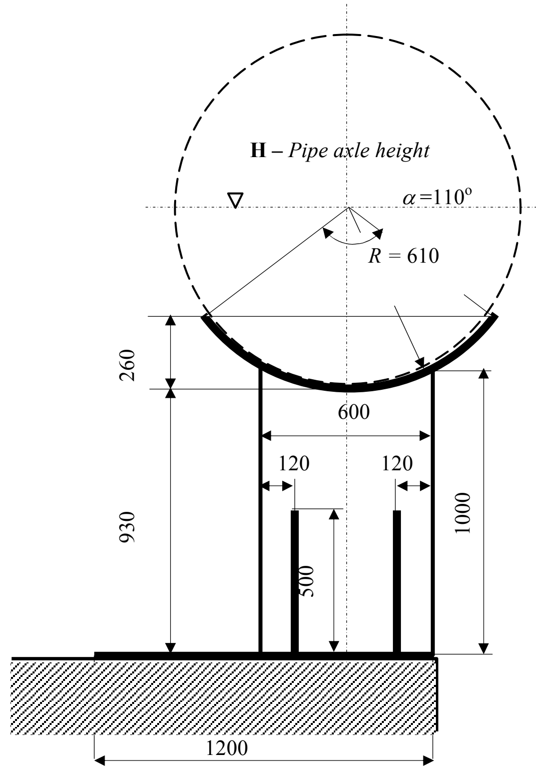

Fig. 10. Metallic support of the compensator from the Dn 1200 mm air transport pipe at the aeration basin 16 C1, the cleaning station Dancu of Iasi

2. *Horizontal axial forces in vertical bends D* (14) *and* 6*:* the axial forces that act upon bends correspond to the action's direction is sensibly depending of friction factor between mains and supports. Assuming the following values ρ_{aer} = 1,4 kg/m^3, Q = 10 m^3/s, α = 90° and μ = (0,1 ÷ 0,4), axial forces in bends 14 and *6* take the values shown in Table V.

When monitoring the variation of the active axial force in bends which depends on the air pressure within the pipe, for different values of the friction factor, we may observe the following:

- in the case of bend 6, when friction factor drops down to μ = 0,2 , the force will have values which will not exceed (10 ÷ 15) kN, even if air pressure boosts to (0,9 ÷ 1) bar; in the exceptional situation when friction factor drops down to μ = 0,15, the values are not exceeded at air pressures of (0,7 ÷ 0,8) bar, [4].
- In the case of bend *D*, active axial force will take values sensibly superior compared to those above soon as μ < 0,25 (on direction *C*1 - *D*) and even if μ = 0,30 (on direction *C*2 - *D*), and this at lower pressures within the main (0,65 ÷ 0,85) bar, on direction *C*1 - *D*, respectively (0,5 ÷ 0,8) bar, on direction *C*2 - *D*).

3. *The vertical force on the supports V* corresponds to the quasi-even distribution of the total weight of the free shifting sector, on the n_r supports that exist on its route, (Table VI):

$$V(L,n_f,n_r)=\frac{L\cdot G_c+n_f\cdot G_f}{n_r} \quad (5)$$

4. *Horizontal axis-normal force in the supports H*: generated by the active axial force applied by the perpendicular sector, the horizontal force that acts upon supports, in a direction normal to axis, results from, Table VII:

$$H(p,\mu,L,n_f,n_r)=\frac{F_{a1}(p,\mu,L,n_f)}{n_r} \quad (6)$$

5. *Pipe roll-over stability on supports c* is provided by over-unitary values for the safety coefficient, (Table VIII):

TABLE IV
THE FRICTION FORCE ON SUPPORTS F_R AND FLOW Q_F VARIATION DEPENDING ON FRICTION FACTORS μ

μ	***Sector 14 - 18*** L = 45,35 m; n_{fl} = 11; n_r=5		***Sector 11 - 14*** L = 72,70 m; n_f = 10; n_r = 5		***Sector 1 - 11*** L = 139,9 m; n_{fl} = 30; n_r = 14	
	Q_f	F_r	Q_f	F_r	Q_f	F_r
0,1	16,44	3,288	24,58	4,916	49,80	3,557
0,2	32,88	6,576	49,16	9,832	99,60	7,114
0,3	49,32	9,864	73,74	14,748	149,40	10,672
0,4	65,76	13,152	98,32	19,663	199,21	14,229

TABLE V
THE AXIAL FORCE VARIATION DEPENDING ON PRESSURE P AND FRICTION FACTORS μ

	Sector 18 - 14			***Sector 11 - 14***			***Sector 11 - 6***		
	L = 86,35 m; n_{fl} = 11; n_r = 5			L = 118,05 m; n_{fl} = 10; n_r = 5			L = 139,9 m; n_{fl} = 30; n_r = 14		
p / μ	0,6	0,7	0,8	0,6	0,7	0,8	0,6	0,7	0,8
0,1	38,789	50,65	61,341	29,318	40,594	51,87	17,978	29,254	40,530
0,2	9,799	21,075	32,351	-9,142	2,133	13,41	-31,82	-20,55	-9,271
0,3	-19,19	-7,915	3,361	-47,60	-36,33	-25,05	-81,62	-70,35	-59,07
0,4	-48,18	-36,91	-25,63	-86,06	-74,79	-63,51	-131,4	-120,1	-108,9

TABLE. VI
VERTICAL FORCE ON SUPPORT V VARIATION

Pipe sector	length L [m]	flange silver n_f	Support	vertical force on support $V(L,n_f,n_r)$ [kN]
1 – C1	139,90	30	*D*	35,572
C1 – C2	118,50	21	*C*1	37,415

$$c(p,\mu,L,n_f,n_r)=\frac{V\cdot a}{H(p,\mu,L,n_f,n_r)\cdot b} \quad (7)$$

As it can be observed, the roll-over stability of mains is assured for pressures of (0,5 ÷ 0,8) bar, and this even when the friction factor drops down to μ = 0,1.

In the existing situation, when the pipe sector between sector *12* and bend *14* do not have any support area, this pipe operates such as an annular beam having diameters d_e, d_i and length L, being embedded at one end *12* and stressed at the other end by a concentrated force which corresponds to the active axial force F_{a1}, applied to the perpendicular sector (*18 - 14*).

TABLE VII
HORIZONTAL AXIS-NORMAL FORCE IN SUPPORTS VARIATION

Sect.	***Sector 11-13***: L = 86,35 m; n_f = 11; n_r = 4				***Sector 15-18:*** L=118,05m; n_f=10; n_r=5				***Sector 2-5***: L = 139,90 m; n_f=30; n_r=4			
p / μ	0,5	0,6	0,7	0,8	0,5	0,6	0,7	0,8	0,5	0,6	0,7	0,8
0,1	6,847	9,666	12,485	15,304	3,584	5,839	8,094	10,349	1,645	4,464	7,283	10,1
0,2	-	2,419	5,238	8,057	-	-	0,402	2,657	-	-	-	-
0,3	-	-	-	0,809	-	-	-	-	-	-	-	-

TABLE VIII
THE UNITARY VALUES FOR THE SAFETY COEFFICIENT VARIATION

Sect.	**Sector 11 - 13**: L = 86,35 m; n_f=11; n_r = 4; V = 35,57 a = 0,4 m; b = 0,46 m				**Sector 15 - 18:** L = 118,05m; n_f = 10; n_r = 5; V = 37,42; a = 0,9 m; b = 1,24 m				**Sector 2 - 5**: L = 139,90 m; n_f = 30; n_r = 4; V = 35,57; a = 0,4 m; b = 0,46 m			
p / μ	0,5	0,6	0,7	0,8	0,5	0,6	0,7	0,8	0,5	0,6	0,7	0,8
0,1	4,521	3,203	2,480	2,023	7,579	4,652	3,356	2,624	18,82	6,936	4,251	3,065
0,2	-	12,80	5,911	3,842	-	-	67,59	10,22	-	-	-	-
0,3	-	-	-	38,26	-	-	-	-	-	-	-	-

The inertia momentum I_x and the resilience modulus of the annular section W_x having $d = d_i/d_e$ = 0,983, being:

$$I_x = \pi \cdot \frac{d_e^4}{64} \cdot \left(1 - d^4\right) = 7{,}177 \cdot 10^5 \quad [\text{cm}^4], \tag{8}$$

Respectively:

$$W_x = \pi \cdot \frac{d_e^3}{32} \cdot \left(1 - d^4\right) = 1{,}177 \cdot 10^4 \quad [\text{cm}^3], \tag{9}$$

Corresponding to the axial load with which acts the perpendicular pipe sector $C2$ - D, within the embedding $C1$, will generate a bending momentum M_i:

$$M_i = 10^{-1} \cdot F_{a1} \cdot L, \ [\text{MN.cm}], \tag{10}$$

Which will next generate the unitary effort σ_x, [6]:

$$\sigma_x = 10^2 . \frac{M_i}{W_x}, \ [\text{N/mm}^2], \tag{11}$$

Maximal arrow f in D sector is calculated with following form: [7]:

$$f = \frac{10^9 \cdot F_{a1} \cdot L^3}{3 \cdot E \cdot I_x}, \ [\text{cm}] \tag{12}$$

The deformed angle φ has the mathematical relation:

$$\varphi = \tan^1 \left[\frac{10^7 \cdot F_{a1} \cdot L^2}{2 \cdot E \cdot I_x} \right], \ [\text{rad}]. \tag{13}$$

Variation of the arrow function of the axial active load on sector *12 - 14* of the main (Fig. 11), correlated with its dependence on air pressure and support friction factor, leads to the conclusion that when the axial shifting on direction $C2$ - D is not impeded, section D will sensibly shift as soon as the air pressure exceeds 0,8 bar, even in normal conditions of support friction. Subsequently the phenomenon occurs at lesser and lesser pressures even when the friction factor takes values below μ = 0,25.

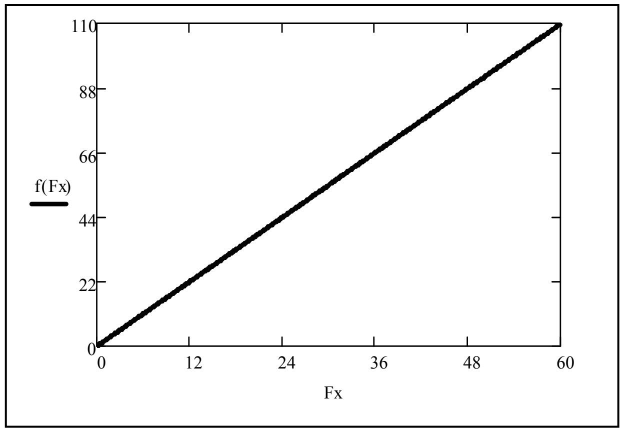

Fig. 11. Arrow variation f depending on horizontal hydraulic force F_x on sector *12 - 14* of the main.

If accept an elastic modulus E = 2,02.10[7] N/cm[2], for sector $C1$ - D, having a length L = 43 m, the main characteristics of the bending stress occurring in section $C1$, under the active axial forces in range from 0 to 60 kN, are shown in Table IX.

The inertia momentum and the resilience modulus of the annular section having $d = d_i/d_e$ = 0,983, being the same as in the previous case (I_x = 7.177.10^5 cm^5, respectively W_x = 1,177.10^4 cm^3), corresponding to the axial load with which the perpendicular sector $C1$ - D acts, in the embedding E will occur a bending momentum M_{iE}, which generates:

- The unitary effort σ_{x17} [N/mm^2],
- The maximal arrow in D, f_D [cm],
- The deformation angle φ_E [rad],

Which, accepting an elastic modulus E = 2,02. 10^7 N/cm^2 and a length for sector E - D, L = 15 m, for active axial forces in the range of 0 to 60 kN, takes the values shown in the Table X.

TABLE IX

THE AXIAL FORCE F_{A1} , BENDING MOMENTUM M_{IC1} , UNITARY EFFORT σ_{XC1} , ARROW F_D , DEFORMED ANGLE φ_{C1} VARIATIONS BETWEEN $C1 - D$ BENDS

F_{a1}	kN	0	10	20	30	40	50	60
M_{iC1}	MN.cm	0	43	86	129	172	215	258
σ_{xC1}	N/mm^2	0	36,52	73,04	109,55	146,07	182,59	219,10
f_D	cm	0	18,3	36,6	54,8	73,1	91,4	109,7
φ_{C1}	Grd.	0	0,37	0,73	1,07	1,46	1,83	2,19

TABLE X

THE AXIAL FORCE F_{A1} , BENDING MOMENTUM M_{IE} , UNITARY EFFORT σ_{XE} , ARROW F_D , DEFORMED ANGLE φ_E VARIATIONS BETWEEN $E - D$ BENDS

F_{a1}	kN	0	10	20	30	40	50	60
M_{iE}	MN.cm	0	15	30	45	60	75	90
σ_{xE}	N/mm^2	0	12,74	25,48	38,22	50,96	63,69	76,43
f_D	cm	0	0,78	1,55	2,33	3,10	3,88	4,66
φ_E	grd.	0	0,04	0,09	0,13	0,18	0,22	0,27

Variation of arrow under the active axial load in node *14*, correlated to its dependence of the air pressure and the support friction factor leads to the conclusion that the axial shifting of section 14 on direction *12 - 14* will not reach intensities impossible to be undertaken by the compensation devices, even when air pressure exceeds (0,8 ÷ 1) bar, and the friction factor in supports drops down to $\mu = 0,15$.

IV. CONCLUSIONS

Deformation occurring in the console pipe sector allows an axial shifting of the perpendicular sector, corresponding to the arrow in its free end *f*.

In normal conditions the friction force that opposes to the axial load takes values Q_f = 65,76 kN - on sector *D*- *C*2; Q_f = 98,32 kN – on sector *C*1- *D* and Q_f = 199,21 kN – on sector 1 - *C*1.

Vertical force on supports corresponds to the quasi-even distribution of the total weight of the free shifting sector, on the n_r supports that exist on its route; it takes the values V = 35,572 KN on sector 1 - 11, respectively 37,415 KN on sector *C*1 – *C*2.

Axial shifting of a main is allowed only by bending deformation of the perpendicular sector and will correspond to the arrow generated in their free end, *f*.

Admissible distance between supports is conditioned by:

- Calculation distributed load, generated by the own operating pipe's weight, including the weight of flanges.
- The resilience modulus of the annular main section.

The admissible tension for wind and weight generated loads of the material of which the main is made (OLT 35 k, σ = 145,8 N/mm^2 [1]), is respected in the case of the air main having d_e = 121,9 cm, and d_i = 119,83 cm, even on sector *C*1 - *D* which is not supported on the plane supports existing nearby on the clarifier.

The necessary number of compensators matches the maximal temperature variation which occurs during operating ΔT, the length of straight main on which they are mounted L and the compensation capacity stated by the supplier ΔL, respectively the linear thermal dilatation factor for steel (α_d = $11,10^{-6}$ m/m. °C).

Keeping the same number of compensators on the two long sectors of the air main it results that at a maximal temperature variation ΔT = *100* °C, the main must undertake a length variation of ΔL = 9,5 ÷ 12 cm.

The phenomenon that occurred can be justified only as a result of action of some axial forces, taking into account that axial shifting of rectilinear pipe sectors is not impeded by any special blocking device, but only by the friction of the pipe on its supports (axially) and by the resistance torque generated by its own weight (in the supports of the perpendicular sectors) (Fig. 3).

Presence of compensators *C*1 and *C*2 may lead to axial unbalanced forces that might induce shifting of rectilinear pipe sectors; this may occur in (Fig. 2):

- Bend *6* (pipe sector *B*, Fig. 2), action towards the exterior, on direction *11-6*;
- Bend *14* (pipe sector *D*, Fig. 2), action towards the exterior, on direction *11-14*;
- Bend 15 – force transferred through the adjacent vertical pipe sector *D,* on which acts towards the exterior, on direction *C*2 - *D*.

When system is operated balance of axial horizontal forces is provided by (Fig. 5):

- Friction force between pipe and pipe sector's supports on which direction it occurs.
- Horizontal reactions in supports of the perpendicular pipe sector, on which it acts.
- Resilience of consoled pipe sectors, between the applying point and first active support which will bear deformation corresponding to their length and the resilience modulus of the pipe's transverse section, depending on the value of the axial force that acts on the free end.

The horizontal force that acts upon supports in a direction normal to axis is generated by the active axial force applied by the perpendicular sector and depends on the air pressure in the mains, their friction factor on supports. On relevant sectors for the study, considering the worst scenario (μ = 0,1), this force takes values which increase directly with the air pressure in the pipe: H = (6,847 ÷ 15,304) kN - on sector 11 - 14; H = (3,584 ÷ 10,349) kN - on sector 15-18; H = (1,645 ÷ 10,101) kN - on sector 2 - 5.

The pipe sector comprised between section *17* and bend *14* also do not have any supporting point. Therefore this sector works such as a annular beam of diameters D_e, D_i and a length L, embedded at one end *17*, and stressed at the other end by a concentrated active force corresponding to the active axial force F_{a1}, applied by the perpendicular sector (*12 - 14*). The inertia momentum and the resilience modulus of the annular section having $d = D_i/D_e$ = 0,983, being the same as in the previous case ($I_x = 7.177.10^5$ cm^5, respectively $W_x = 1,177.10^4$ cm^3), corresponding to the axial load with which the perpendicular sector (*12 - 14*) acts, in the embedding *17* will occur a bending momentum M_{i17}, which generates: the unitary effort σ_{x17} [N/mm^2], the maximal arrow - in *14* - f_{14} [cm] and the deformation angle φ_{17} [rad], which, accepting an elastic modulus $E = 2,02.10^7$ N/cm^2 and a length for sector *17 - 14*, L = 15 m, for active axial forces in the range of 0 to 60 kN.

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