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ANALYSIS OF STEADY AND TRANSIENT REGIMES IN OIL TRANSMISSION PLANTS

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Pri projektovanju, izgradnji i eksploataciji naftnih postrojenja sprovode se kompleksne analize stacionarnih i prelaznih radnih režima u cilju postizanja bezbednih, funkcionalnih i ekonomičnih tehničkih rešenja njihovih vitalnih elemenata. U radu je za jedno takvo postrojenje prikazana analiza prelaznih režima ispada pumpnih agregata sa elektronapajanja za naftovodno postrojenje. Projektovano je rešenje zaštite postrojenja primenom membranskih vetrenika neposredno na potisu pumpnih agregata. Pre analize prelaznih, urađeni su proračuni stacionarnih režima uz adekvatno preračunavanje radnih karakteristika pumpnih agregata, snimljenih na mernim štandovima sa vodom kao ispitnim fluidom, na uslove konkretnog industrijskog postrojenja..

Ključne reči: *prelazni režimi u naftnim postrojenjima, radne karakteristike pumpi, vetrenik*

In order to achieve safe, functional and economical technical solutions for oil-transfer plants, comprehensive analyses of steady and transient operating regimes are implemented within their design, development and exploitation stages. This paper presents a case study of analysis of transient regimes of pumping units power failure in an oil transmission plant. Design solution of protective measures is based on implementation of membrane pressure vessels located directly at the pumping units discharge. Before addressing transient regimes, due analysis and calculations of steady regimes is mandatory. Pump test characteristics obtained by using water as a test fluid are further converted onto plant prototype conditions via appropriate calculation procedures.

Key words: *steady and transient regimes; oil transmission plants; pump operating characteristics; pressure vessel*

1. Introduction

Water is the widest spread category of fluids which are subject to hydraulic transportation. This category comprises:

- potable water,
- waste water (industrial and municipal),
- sewer water (rain and fecal),
- fresh water (agricultural irrigation, flood water evacuation, underground water from Reni wells, etc.),
- industrial water (particularly treated water to meet requirements of operating processes of various industrial plants and factories, thermal power plants with steam turbines, etc.).

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Crude oil and oil derivates are second-most spread category of fluids which are subject to hydraulic transportation (hydraulic transportation by pumping). By rheological characteristics, oil derivates fall into the same group with water (so called Newtonian fluids). Anyhow, oil derivates are very different from water in regard of other physical properties (density, viscosity, vaporizing pressure, etc.). Density and viscosity of transported fluids have significant influence onto plant operating conditions.

2. Influence of density and viscosity of oil derivates onto plant operating conditions

Influence of density

Density of operating fluid, by itself (i.e. in the cases where viscosity of operating fluid is the same as for water, but density differs), does not impact hydraulic (head losses through pipeline system and pump), but energetic conditions only. That is because alteration of fluid density influences alteration of operating power of the pumping unit, i.e.

$$P_P = \frac{\rho \cdot g \cdot H_P \cdot Q_P}{\eta_P} \tag{1}$$

where: P_P is pump power, ρ is fluid density, g is gravitational constant, H_P is pump head, Q_P is pump discharge and η_P is pump efficiency.

2.1. Influence of viscosity

Viscosity of operating fluid significantly impacts:

- value of friction headlosses through pipeline system, and
- operating conditions of pumps (hydraulic losses within the limits of pumps, so imposing certain decrease of pump effective head, pump effective discharge and pump efficiency, with certain increase of pump operating power in comparison to the conditions of pump operating with water), ref. Figure 1.

These effects are intrinsic and natural because difference in viscosity of various kinds of crude oils and oil derivates vs. water are significant (kinematic viscosity being from several times to several orders of magnitude higher), as can be seen from Table 1.

Therefore, adequate understanding of operating characteristics of pipeline system and pumping units is of utmost importance when determining steady-state and transient operating conditions of plant, i.e. selection of pipeline materials and diameters, selection of pumping units, determination of head- and discharge- operating range of pumping units, pump regulating approach (adoption of regulating manner and equipment), analysis and adoption of appropriate manner of pipeline headlosses reduction (devices for heating of crude oil and oil derivates, causing some increase in exploitation costs vs. pipeline diameter or pipeline insulation increase, requiring some increase in investment costs), analysis and selection of measures (of constructive- or regime- nature) for protecting the plant in transient operating regimes, estimation of plant investment costs and exploitation costs, etc.

Anyhow, it should be taken into consideration that obtaining of operating characteristics of roto-dynamic pumps at test stands is usually done with water as test fluid, because it is neither usual nor rational to test pumps over wide range of various fluids from the group of crude oils and oil derivates (various extra light, light, intermediate, heavy and extra heavy fuels) for each investigated or newly-developed pump model. Therefore, in practice of pump engineering, some appropriate methods for correction or adaptation of results obtained by measurements with one standard test fluid are applied. By that way, pump characteristics of head vs. discharge $H_P=H_P(Q_P)$ and efficiency vs. discharge $\eta_P=\eta_P(Q_P)$, obtained for water as test fluid are being appropriately altered for influence of density and viscosity of fluids other than water.

Table 1. Kinematic viscosity of water and some of oil derivates and crude oils

Fluid		t_{fluid} [°C]	ν [mm^2/s]	*Fluid*		t_{fluid} [°C]	ν [mm^2/s]
Water		15	1.154	Crude oils	"Kutubu"	20	2.1
		20	1.002		"West Texas", intermediate	20	4.9
		38	0.697		"Arabian"	20	10.7
		50	0.547		"Tia Juana", light	38	8.8
Oil derivates	Eurodiesel	15	3.27		"Tia Juana", heavy	38	88.6
	Biodiesel	20	6.5÷9		"Escalante"	38	307
	Diesel D1	20	1÷6.5		"Boscan"	38	11233
	Diesel D2	20	1.8÷9		"Brent"	50	2.86
	Diesel D3	20	5÷25		"Bonny"	50	2.9

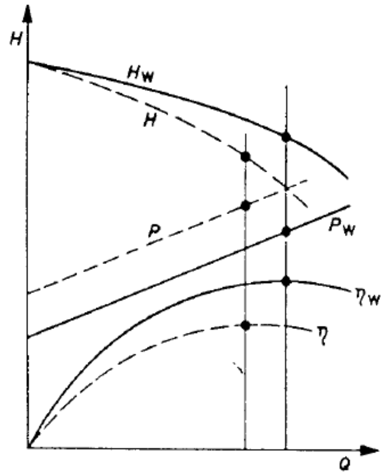

Figure 1. Alteration of pump operating curves – head vs. discharge, efficiency vs. discharge and power vs. discharge (shifting of particular operating points) due to viscosity effects of pumped fluid

Over decades, three methodologies for determination of viscosity-correction factors for recalculation of pump characteristics emerged as standard approaches in worldwide pump engineering practice. Two of them had been developed in the USA, i.e. by Hydraulic Institute of New York (1955.) [1], [2], [3] and by A. J. Stepanoff [4]. The third one had been developed in the USSR by B. M. Pevzner (1958.) [5], but its application is restricted to radial pumps of specific speeds within the range of $22 \le n_Q \le 38$ ($80 \le n_S \le 140$). In any case, all these methods lead to more or less the same conclusions in their comparative applications. In this paper, the authors refer to the nomogram which relates to the first of the mentioned methodologies (ref. Figure 2), and present results obtained by its application (ref. Item 3 hereafter).

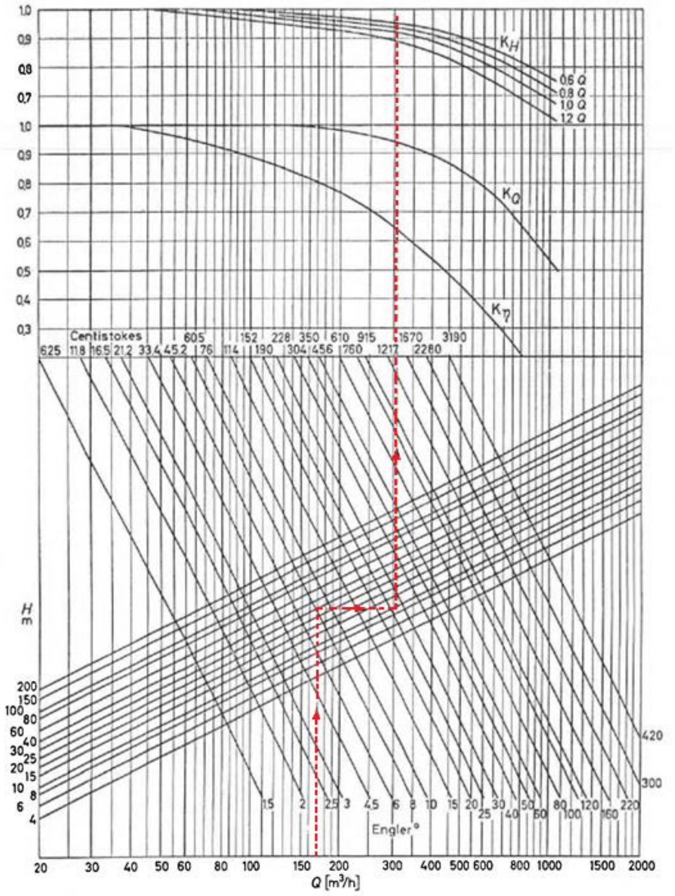

Figure 2. Nomogram for determination of correction coefficients for pump head, pump discharge and pump efficiency due to viscosity effects of pumped fluid

3. Case study

3.1. Short technical description of the plant

The investigated industrial facility serves for tanking and storing oil derivates, i.e. several kinds of industrial fuels and is located in Serbia. The plant comprises three parts:

- fuel manipulation zone (tanking area and fuel pumping station, with tanking reservoirs, pipework, pumping units, auxiliary equipment and devices – armature valves, measuring, regulating and signaling devices),
- fuel transmission pipeline (two main reaches, the first one being 1 × DN150 and approx. 5500 m long, whereas the second one being 2 × DN150 and approx. 1500 m long), and
- fuel storing zone (reservoirs for various kinds of industrial fuels, with appurtenant pipework and auxiliary equipment and devices – armature valves, measuring and signaling devices).

Even though the plant pipeline system represent complex branched pipe reaches (there are many separate branch-lines within the manipulation and the storing zones), it is to be regarded as a simple linear system in hydraulic regard, because at one moment just one of the pumping units is active and performs filling of just one of the storage reservoirs. Plant operating modes comprise:

- filling mode (for the purpose of fuel storing, by its hydraulic transportation from manipulation zone towards storing zone), at rated discharge of 27.5 l/s, and
- outflowing mode (for the purpose of consuming the stored fuels, by gravitational streaming from storing zone towards manipulation zone), at rated discharge of 26 l/s.

Geodetic head of the plant equals approx. 210 m and insignificantly varies regarding operating levels within the reservoirs. Total headloss coefficient (to include friction and local resistance elements) of the pipeline system equals K = 147246 m/(m³/s)² (i.e. total headloss at discharge of 26 l/s equals 99.54 m).

Table 2. Pump operating curves (for water and for fuel), tabulated

Kind of fluid	*Quantity*	*Numeric values*								
Pump operation with water (*w*), obtained at test stand	Q_w [l/s]	5	7	10	15	20	25	27.5	30	35
	H_w [m]	464	467	480	465	450	424	406	385	334
	η_w [%]	30	34	38	50	59	65	66	65	57
	$P_{P,w}$ [kW]	118	122	130	142	154	166	173	178	190
Pump operation with fuel (*v*), recalculated as per correction coefficients	Q_v [l/s]	4.7	6.6	9.4	14.1	18.8	23.5	25.85	28.2	32.9
	H_v [m]	450	458	466	446	423	394	373.5	343	297
	η_v [%]	19.2	21.8	24.3	32	37.8	41.6	42	41.6	36.5
	$P_{P,v}$ [kW]	92.8	114.2	148.5	161.9	173.3	183.3	189.4	195.6	220.5

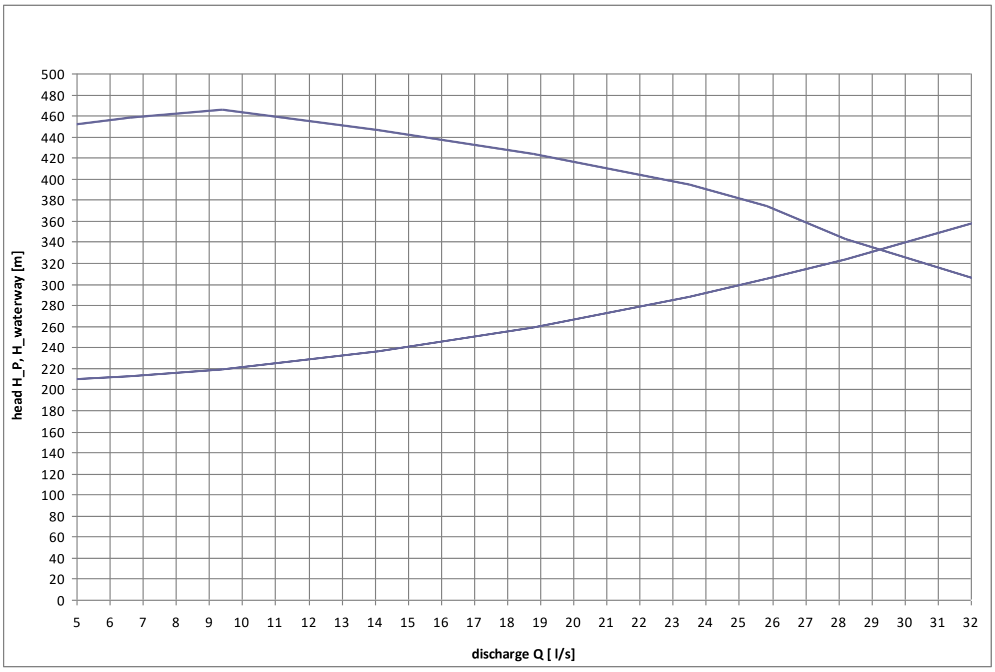

Figure 3. Pump operating curve (head vs. discharge) for nonregulated pump conditions and pipeline headloss curve

The methodology for recalculation of pump characteristics due to fluid viscosity effects (industrial fuels), as described in Item 2 hereabove, had been applied for analyses of steady-state and transient behaviour of the plant and adoption of plant protective measures (ref. Table 2 and Figure 3). It is to be underlined that the pumping units are additionally equipped with frequency regulators for the purpose of fine tuning of operating discharge.

3.2. Results of the analyses

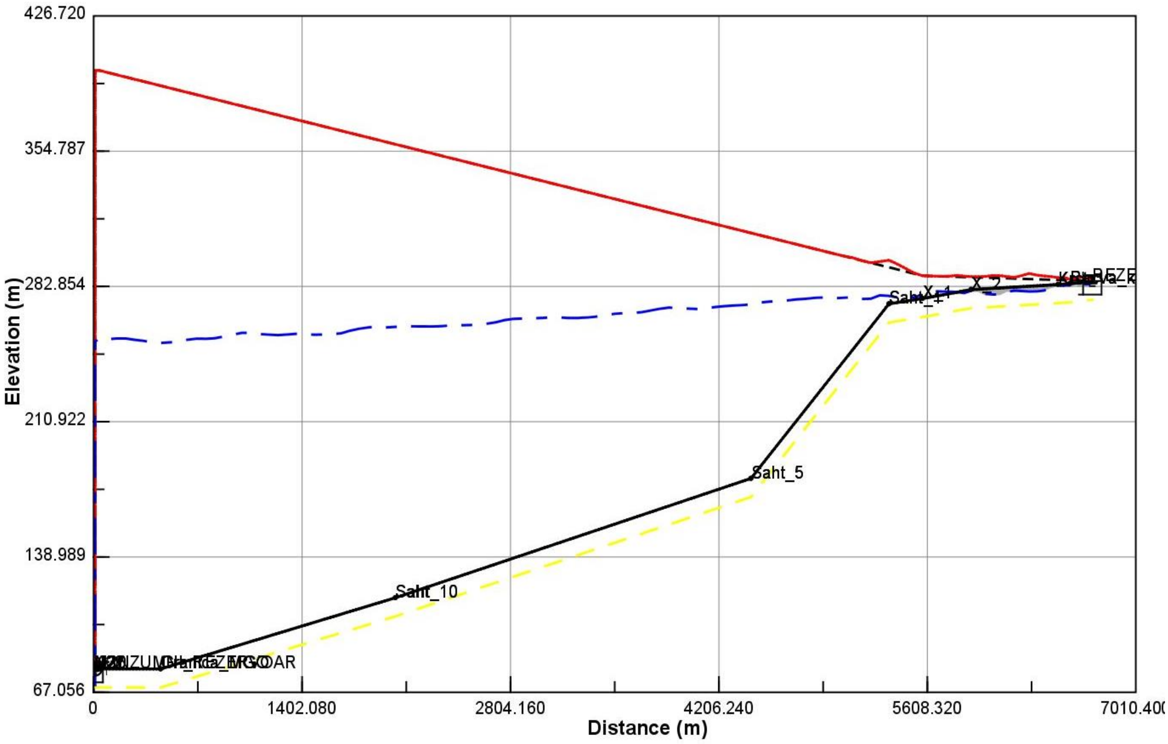

Figure 4. Pumping mode, input power failure.
Head envelopes along the pipeline profile

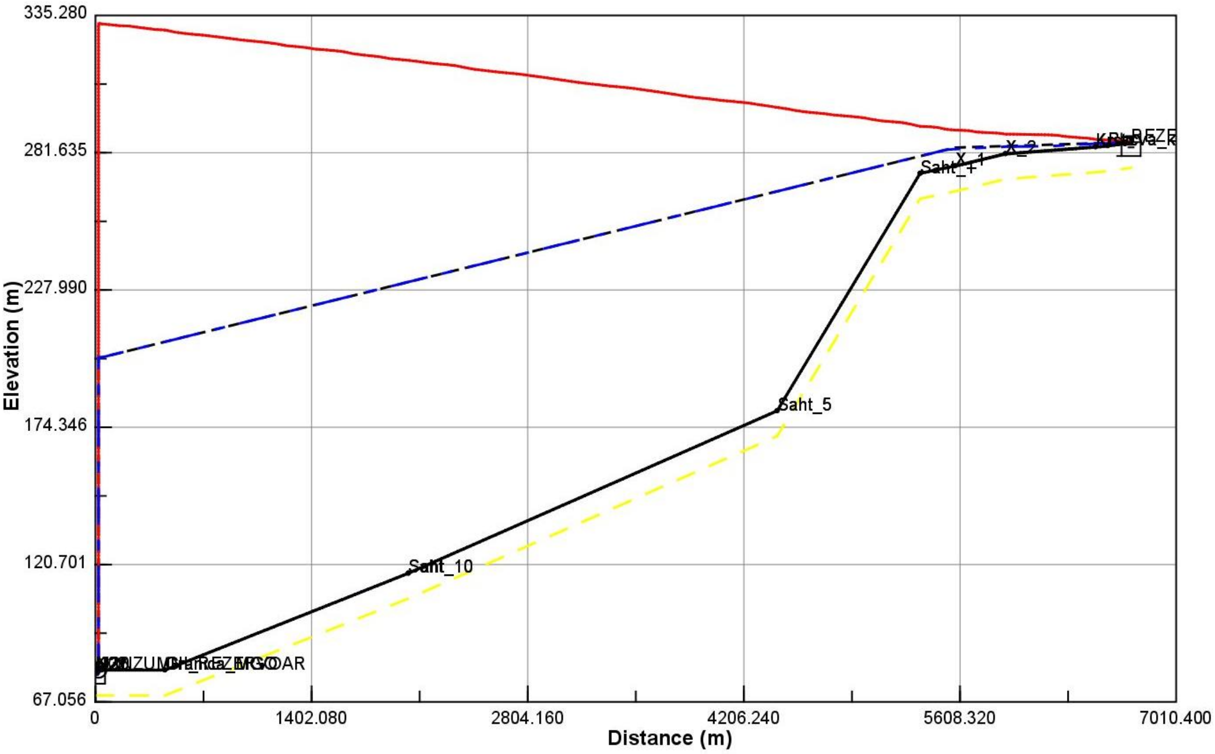

Figure 5. Gravitational flow mode, regulating valve closing.
Head envelopes along the pipeline profile

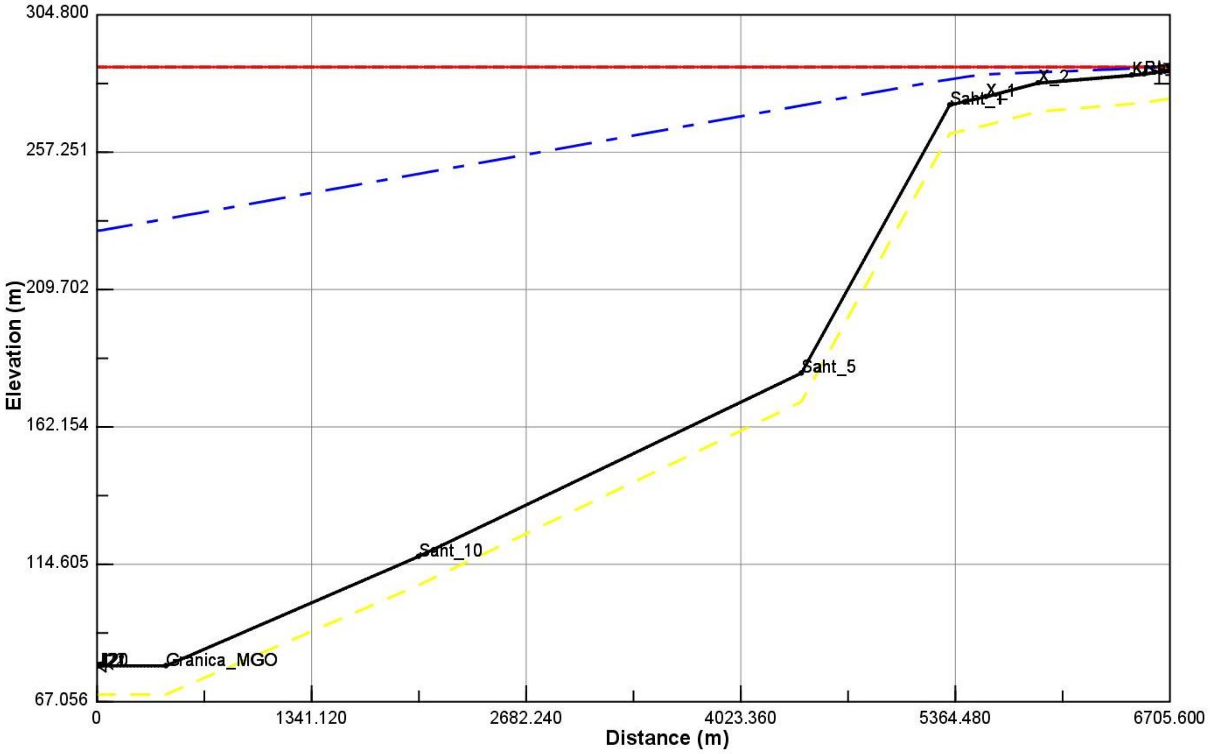

Figure 6. Gravitational flow mode, regulating valve opening.
Head envelopes along the pipeline profile

Note: Numerical simulations had been performed by Bentley HAMMER software, which uses steady-state friction model.

3.3. Discussion of results and comments

In regard of the plant's exposure to violent transient regimes, it is to be stressed out that the profile of pipeline system is not favourable because the last reach is low-sloped (close to horizontal) so that, without remedial measures, it would be endangered by minimal head envelopes that would emerge transient regimes. This issue had been duly addressed and investigated within the analyses of variant solutions and ultimate adoption of protective measures (i.e. technical characteristics and location(s) of surge-protective pressure vessels).

Adopted constructive-kind measure (applicable to the filling mode of plant operation) comprises installation of pressure vessel of the following characteristics:

– total number of vessels	1,
– location	within pumping station, at the discharge side,
– type	membrane, nitrogen-filled,
– total volume of the vessel	1.6 m^3,
– standpipe diameter	DN80 (3“),
– volume of membrane prime-filling	0.4 (±3%) m^3.

Such solution assures that pump input power failure during the filling mode of plant operation will:

- not induce envelopes of minimal heads that would jeopardize pipeline system, and
- induce envelopes of maximal heads that does not overpass steady-state piezometric line, at all.

Adopted regime-kind measures (applicable to the outflowing mode of plant operation) comprise:

- law and overall time of regulating valve closing – linear in 300 s, and
- law and overall time of regulating valve opening – linear in 300 s.

4. Conclusions

During design stages for the viscous fluids hydraulic transportation plants (crude oil and oil derivates) adequate estimation of viscosity effects onto pipeline hydraulic losses and pump operating characteristics is of crucial importance (total hydraulic losses, effective discharge and effective head and effective power of pumps, adoption of pump electromotor drive, plant exploitation costs, etc.).

In addition, within analyses of oil plants transient behaviour and adoption of protective measures, aeration valves and compressed-air pressure vessels should be avoided due to nature of such fluids. Instead, membrane type, compressed nitrogen pressure vessels are preferred (appropriate number of sets and arrangement along the pipeline), along with alteration of pipeline profile (if possible or practicable) at the reaches which may be endangered by appearance of unfavourable minimal pressures.

5. References

[1] *** KSB, *Handbook. Pumps*, Klein, Schanzlin & Becker Aktiengesellschaft, Frankenthal, Germany, 1968.

[2] *** KSB, *Centrifugal Pump Lexicon*, Klein, Schanzlin & Becker Aktiengesellschaft, Frankenthal, Germany, 1975.

[3] **Protic Z., Nedeljkovic M.,** *Pumps and ventilators*, University of Belgrade Faculty of Mechanical Engineering, Belgrade, Serbia, 1992.

[4] **Stepanoff, J. A.**, *Centrifugal and Axial Flow Pumps. Theory, Design and Application, 2nd edition*, Krieger publishing company, Malabar, Florida, 1957.

[5] **Певзнер, Б. М.**, *Судовые центробежные и осевые насосы*, Машиностроение, Ленинград, СССР, 1971.