© Strojniški vestnik 49(2003)3,150-160
ISSN 0039-2480
UDK 621.224.24:532.528
Izvirni znanstveni članek (1.01)
© Journal of Mechanical Engineering 49(2003)3,150-160
ISSN 0039-2480
UDC 621.224.24:532.528
Original scientific paper (1.01)

Obratovanje hidravličnega turbostroja med prehodnimi pojavi

The Behaviour of a Hydraulic Turbomachine during Transients

Anton Bergant

Prispevek obravnava obratovanje hidravličnega turbostroja med prehodnimi pojavi (vodni udar) v cevnih sistemih. Podane so osnove metode karakteristik in prehodnega kavitacijskega toka v ceveh. Hidravlični turbostroj je popisan kot robni pogoj v deltoidni mreži metode karakteristik. Prikazana sta dva industrijska primera: trenutna razbremenitev dveh 34 MW francisovih turbin in izklop centrifugalne črpalke. Rezultati izračunov in meritev na terenu se dobro ujemajo v obeh primerih.

(Ključne besede: stroji turbinski, udar tlačni, hidroelektrarne, sistemi črpalni)

This paper deals with the behaviour of a hydraulic turbomachine during transients (water hammer) in piping systems. A brief description of the method of characteristics and the fundamentals of transient cavitating pipe flow are given. The hydraulic turbomachine is treated as a boundary condition within the staggered grid of the method of characteristics. Case studies for a sudden load rejection of two 34-MW Francis turbines and a centrifugal pump rundown are presented. There is good agreement between the computational and the field-test results for both cases.

(Keywords: turbomachines, water hammer, hydroelectric power plant, pumping systems)

0 UVOD

V fazi izbire in načrtovanja hidravličnega sistema moramo izdelati analizo prehodnih pojavov, da zagotovimo varno obratovanje izbranega sistema. Glavni cilj tega prispevka je izluščitev vpliva hidravličnega turbostroja na odziv pretočnega sistema med prehodnimi režimi. Med prehodnimi režimi se lahko pojavita ekstremni vodni udar in prekinitev kapljevinskega stebra v sistemu. V pretočnem sistemu hidroelektrarne se pojavijo naslednji režimi: obremenitev turbine, zmanjšanje obremenitve in trenutna razbremenitev, pobeg turbine, zaprtje varnostnih zapiral ter kombinirano obratovanje turbine in zapirala. V črpalnih sistemih se vodni udar pojavi pri startu, izklopu črpalke ter pri odpiranju in zapiranju ventilov. Poleg tega lahko hidravlični turbostroj obratuje kot črpalka – turbina. V tem primeru je turbostroj v turbinskih in črpalnih področjih obratovanja. Vodni udar lahko povzroči motnje v obratovanju hidravličnega sistema in poškoduje elemente sistema (zlom cevovoda). Ekstremne tlačne utripe v pretočnem sistemu in vrtilno frekvenco hidravličnega stroja običajno krmilimo z ustreznimi obratovalnimi manevri (zapiranje in odpiranje

0 INTRODUCTION

Feasibility and design studies of hydraulic systems should include a water-hammer analysis in order to ensure safe operation of the system. The main objective of this paper is to identify the influence of hydraulic turbomachine on system behaviour during transient regimes. Transient regimes may cause excessive water hammer and possible column separation in the system. These include the turbine-load acceptance, load reduction or sudden load rejection, turbine runaway, shutoff valve closure, and a combined operation of the turbine and valve in hydroelectric power plants. In pumping systems, water hammer may be induced by the pump start-up, the pump rundown, and the opening and closing of valves. In addition, the hydraulic turbomachine can operate as a pump-turbine. In this case the turbomachine undergoes transient regimes in the turbine- and pump-operating modes. Water hammer may disturb the operation of the hydraulic system and damage the system components (pipe rupture). High-pressure fluctuations in the flow-passage system and the hydromachine speed are traditionally controlled by appropriate operational means (closing and opening wicket gates, preventing flow vodilnika turbine, preprečitev povratnega toka skozi črpalko). Vodni udar lahko blažimo tudi z vgradnjo zaščitne opreme proti vodnemu udaru (vztrajnik, tlačni varnostni ventil, izravnalnik, zračni kotel, prezračevalna cev, zračni ventil) in tehnično prerazporeditvijo cevnih elementov ([1] do [4]). Na izbiro metod za blažitev vodnega udara vplivajo obratovalni, varnostni in gospodarni kriteriji.

Vodni udar popišemo s sistemom hiperboličnih parcialnih diferencialnih enačb, kontinuitetne in gibalne enačbe. Obravnavani sistem enačb rešujemo z uporabo metode karakteristik. Hidravlični turbostroj popišemo kot robni pogoj v deltoidni mreži metode karakteristik. Podan je kratek oris prekinitve kapljevinskega stebra in neprekinjenega kavitacijskega toka. Kavitacijski tok se pojavi, ko se tlak kapljevine v cevi zniža na parni tlak kapljevine.

V sklepnem delu prispevka preverimo teoretični model z dvema primeroma iz industrije. Prvi obravnavani sistem, hidroelektrarna Toro II (Kostarika), ima vgrajeni dve 34 MW francisovi turbini (nazivni neto padec 376 m, pretok 10 m^3/s). Podani so rezultati trenutne razbremenitve obeh turbin s polne moči. Vodni udar blažimo z ustrezno nastavitvijo časa zapiranja vodilnika turbine. Drugi primer obravnava izklop centrifugalne črpalke v drenažnem črpalnem sistemu rudnika v Velenju. V obeh obravnavanih primerih se rezultati izračuna in meritev dobro ujemajo.

reverasal of the pump). Additional protective measures against unacceptable water hammer include installation of surge-control devices (flywheel, pressure-relief valve, surge tank, air-cushion surge chamber, aeration pipe, air valve) and redesign of the pipeline layout ([1] to [4]). Operational, safety and economic factors are decisive for the optimum selection of the method for controlling transients.

Water hammer is described by a set of hyperbolic partial differential equations, the continuity equation and the equation of motion. The method of characteristics is used for solving the water-hammer equations. The hydraulic turbomachine is treated as a boundary condition within the staggered grid of the method of charactersitics. Liquid-column separation and distributed cavitation are briefly introduced. Cavitating flow occurs when the pressure in the pipe drops to the liquid vapour pressure.

The paper concludes with two case studies validating the theoretical model. The first system under consideration, the Toro II hydro-electric powerplant (Costarica), is fitted with two 34-MW Francis turbines (rated net head 376 m, discharge 10 m^3/s). Results from the sudden load rejection of the two units are presented. Water hammer is controlled by appropriate adjustment of the wicket-gate closing manoeuvre. The second case study considers a centrifugal-pump rundown in the pumping system of a mine in Velenje (Slovenia). The computational results match reasonably well with the field test results in both systems.

1 MODEL VODNEGA UDARA

1 WATER-HAMMER MODEL

Vodni udar popisuje potovanje tlačnih valov vzdolž cevovoda. Tlačni valovi so vzbujeni s spremembo pretočne hitrosti. Neustaljeni tok v zaprtih ceveh popišemo z dvema enačbama v eni razsežnosti, kontinuitetno enačbo in gibalno enačbo ([1], [3] in [5]):

Water hammer is the transmission of pressure waves along the pipeline resulting from a change in flow velocity. Unsteady flow in closed conduits is described by two one-dimensional equations: the continuity equation and the equation of motion ([1], [3] and [5]):

$$\frac{\partial H}{\partial t} + v\frac{\partial H}{\partial x} - v\sin\theta_p + \frac{a^2}{g}\frac{\partial v}{\partial x} = 0 \qquad (1)$$

in and

$$g\frac{\partial H}{\partial x} + \frac{\partial v}{\partial t} + v\frac{\partial v}{\partial x} + \frac{\lambda v|v|}{2D} = 0 \qquad (2),$$

kjer so H – piezometrična višina, t – čas, v – pretočna hitrost v cevi, x – koordinata, θ_p – strmina cevovoda, a – hitrost vala, g – zemeljski pospešek, λ – Darcy-Weisbachov koeficient trenja in D – premer cevi. V večini inženirskih uporab so člen strmine cevovoda in konvekcijski členi v enačbah (1) in (2) majhni in jih zanemarimo ([1], [3] in [5]). V obravnavanih enačbah običajno vpeljemo pretok $Q = vA$ namesto pretočne hitrosti v; A – prečni prerez. Enačbi (1) in (2) sestavljata sistem navidezlinearnih hiperboličnih parcialnih diferencialnih enačb. Splošna rešitev za obravnavane enačbe ne obstaja. Običajna rešitev

in which H – the piezometric head, t – the time, v – the pipe flow velocity, x – the distance, θ_p – the pipe slope, a – the wave speed, g – the gravitational acceleration, λ – the Darcy-Weisbach friction factor, and D – the pipe diameter. For most engineering applications, the pipe slope and convective acceleration terms in Equations (1) and (2) are small and can be neglected ([1], [3] and [5]). Usually, the discharge $Q = vA$ replaces the flow velocity v; A – pipe area. Equations (1) and (2) are a set of quasi-linear hyperbolic partial differential equations. A general solution to these equations is not available. The common method of solving equations enačb (1) in (2) je metoda karakteristik ([1] in [3]). Sprememba po metodi karakteristik da združljivostne enačbe vodnega udara, ki veljajo vzdolž karakterističnih krivulj. Numerično stabilne združljivostne enačbe vodnega udara, zapisane v obliki končnih razlik, se glasijo (majhne člene zanemarimo) ([3], [6] in [7]):

- vzdolž C^+ karakteristike ($\Delta x/\Delta t = a$):

(1) and (2) is by the method-of-characteristics transformation ([1] and [3]). The transformation by the method of characteristics gives the water-hammer compatibility equations, which are valid along the characteristic curves. The numerically stable water-hammer compatibility equations, written in a finite-difference form, are (small terms are neglected) ([3], [6] and [7]):

- along the C^+ characteristic line ($\Delta x/\Delta t = a$):

$$H_{j,t} - H_{j-1,t-\Delta t} + \frac{a}{gA}\left[(Q_u)_{j,t} - (Q_d)_{j-1,t-\Delta t}\right] + \frac{\lambda \Delta x}{2gDA^2}(Q_u)_{j,t}\left|(Q_d)_{j-1,t-\Delta t}\right| = 0 \qquad (3)$$

- vzdolž C^- karakteristike ($\Delta x/\Delta t = -a$):

- along the C^- characteristic line ($\Delta x/\Delta t = -a$):

$$H_{j,t} - H_{j+1,t-\Delta t} - \frac{a}{gA}\left[(Q_d)_{j,t} - (Q_u)_{j+1,t-\Delta t}\right] - \frac{\lambda \Delta x}{2gDA^2}(Q_d)_{j,t}\left|(Q_u)_{j+1,t-\Delta t}\right| = 0 \qquad (4),$$

kjer so j – indeks računske točke, Q_u – pretok na navzgornjem koncu računske točke, Q_d – pretok na navzdolnjem koncu računske točke, Δx – dolžina cevnega odseka in Δt – časovni korak. V enačbah (3) in (4) uporabimo nespremenljivo vrednost Darcy-Weisbachovega koeficienta trenja λ. V primeru hitrih prehodnih pojavov to postavko popravimo z vpeljavo neustaljenega člena trenja v zgornjih enačbah ([7] do [9]). V primeru vodnega udara sta pretok na navzgornjem koncu računske točke Q_u in pretok na navzdolnjem koncu računske točke Q_d enaka ($Q_u \equiv Q_d$), tlak v računski točki je večji od parnega tlaka kapljevine. Na robu enačba robnega pogoja nadomesti eno od združljivostnih enačb vodnega udara.

in which j – the computational section index, Q_u – the discharge at the upstream side of the computational section, Q_d – the discharge at the downstream side of the computational section, Δx – the reach length, and Δt – the time step. A constant value of the Darcy-Weisbach friction factor λ is used in Equations (3) and (4). This assumption may be corrected for the case of rapid transients by introducing an unsteady friction term in the above equations ([7] to [9]). Discharge at the upstream side of the computational section Q_u and the discharge at the downstream side of the section Q_d are identical for the water-hammer case ($Q_u \equiv Q_d$), i.e. the pressure at a section is greater than the liquid vapour pressure. At a boundary, the boundary equation replaces one of the water-hammer compatibility equations.

Prehodni kavitacijski tok

Prehodni kavitacijski tok se pojavi, ko se tlak kapljevine zniža na parni tlak kapljevine. Kavitacija se lahko pojavi v dveh oblikah ([10] do [12]). Prva oblika je krajevna diskretna kavitacija s paro z velikim kavitacijskim razmernikom (pretrganje stebra). Druga oblika kavitacije je neprekinjen kavitacijski tok pri parnem tlaku kapljevine, ki se ustvari na daljši dolžini cevovoda (majhen kavitacijski razmernik). Za prehodni kavitacijski tok običajna metoda reševanja vodnega udara ne velja. V prispevku obravnavamo diskretni parni kavitacijski model ([3] in [13]).

Diskretni kavitacijski model dovoljuje stvaritev kavitacije s paro v vseh tistih računskih točkah numerične mreže metode karakteristik, kjer se tlak zniža na parni tlak kapljevine. V cevnih odsekih med računskimi točkami postavimo kapljevinsko fazo s stalno hitrostjo širjenja udarnega vala a. Dinamiko diskretne kavitacije s paro v poljubni računski točki j vzdolž cevovoda v celoti popišemo z dvema združljivostnima enačbama vodnega udara (3) in (4), kjer višini H priredimo vrednost $z+h_v$ (z – geodetska višina, h_v – parna tlačna višina), in s kontinuitetno enačbo prostornine diskretne kavitacije s paro:

Transient Cavitating Flow

Transient cavitating flow in a pipeline system occurs when the pressure drops to the liquid vapour pressure. Two basic flow situations may occur ([10] to [12]). The first type is a localised, discrete vapour cavity with a large void fraction (column separation). The second type is a distributed, vaporous cavitation, which may extend over long sections of the pipe (small void fraction). The standard water-hammer solution is no longer valid. This paper deals with a discrete vapour-cavity model ([3] and [13]).

The discrete vapour-cavity model allows vapour cavities to form at all computing sections in the method-of-characteristics numerical model when the pressure drops to the liquid vapour pressure. A liquid phase with a constant wave speed a is assumed to occupy the full reach length between the computational sections. The behaviour of the discrete vapour cavity at an arbitrary computational section j along the pipeline is fully described by the two water-hammer compatibility Equations (3) and (4) with H set to $z+h_v$ (z – the elevation above datum, h_v – the vapour pressure head), and the continuity equation for the discrete vapour-cavity volume:

$$V_v = \int_{t_{in}}^{t} (Q - Q_u)\, dt \qquad (5),$$

kjer sta V_v – prostornina diskretne kavitacije s paro in t_{in} – čas nastanka kavitacije.

Numerična rešitev enačbe (5), zapisane v deltoidni mreži metode karakteristik, se glasi [14]:

in which V_v – discrete vapour-cavity volume, and t_{in} – the time of cavitation inception.

The numerical solution of equation (5) within the staggered grid of the method of characteristics is [14]:

$$(V_v)_{j,t} = (V_v)_{j,t-2\Delta t} + \left(\psi\left((Q_d)_{j,t} - (Q_u)_{j,t}\right) + (1-\psi)\left((Q_d)_{j,t-2\Delta t} - (Q_u)_{j,t-2\Delta t}\right)\right)2\Delta t \quad (6)$$

kjer je ψ – utežni koeficient (ψ = 0,5 do 1). Kavitacija se zruši, ko je zbirna prostornina kavitacije manjša od nič. Ponovno se vzpostavi kapljevinski tok in s tem tudi veljavnost rešitve enačb vodnega udara (3) in (4). Priporoča se, da največja prostornina diskretne kavitacije ne preseže 10 % prostornine cevnega odseka [13].

in which ψ = the weighting factor (ψ = 0.5 to 1). The cavity collapses when the cumulative cavity volume becomes less than zero. The liquid phase is re-established and the water-hammer solution using equations (3) and (4) is valid. It is recommended that the maximum size of the discrete vapour cavity at a section is less than 10 % of the reach volume [13].

2 MODELIRANJE HIDRAVLIČNEGA TURBOSTROJA

2 MODELLING A HYDRAULIC TURBOMACHINE

Hidravlični turbostroj lahko preide turbinsko, črpalno ali črpalno-turbinsko področje obratovanja. Dinamični odziv krmiljene črpalke – turbine popišemo z enačbami črpalke – turbine (energijska enačba, enačba vrtenja turboagregata), krmilnika (enačba za popis spremembe vrtilne frekvence črpalke – turbine v odvisnosti od giba krmilnega mehanizma (ov)) in cevovoda (enačbe vodnega udara in prehodnega kavitacijskega toka). Razmerje med vplivnimi veličinami turbostroja upodobimo v obliki eksperimentalno določenih karakteristik črpalke – turbine (višina, moment, vzdolžna osna sila). V literaturi so na voljo številne metode reševanja zgoraj navedenih enačb ([1], [3] in [15]). V primeru spremembe obremenitve, ko je vrtilna frekvenca črpalke – turbine krmiljena, moramo v teoretičnem modelu upoštevati enačbe črpalke – turbine, krmilnika in cevovoda. Enačbe krmilnika ne upoštevamo v primeru analize trenutne razbremenitve turbine ali izklopa črpalke, pri kateri je sprememba vrtilne frekvence agregata odvisna od čistega hidravličnega momenta turbostroja. Robna pogoja za popis trenutne razbremenitve francisove turbine in izklopa centrifugalne črpalke sta definirana, kakor sledi.

The hydraulic turbomachine may undergo turbine, pump or pump-turbine operating modes. The dynamic behaviour of a governed pump-turbine is described by the pump-turbine (head balance equation, dynamic equation of rotating masses), the governor (dynamic equation that relates the pump-turbine rotational speed change to the position of the regulating mechanism(s)) and the pipeline equations (water-hammer and column-separation equations). The relationship among the influential turbomachine variables is presented in the form of the experimentally predicted pump-turbine characteristics (head, torque, axial force). Different methods for handling and solving the system of dynamic equations are available in the literature ([1], [3] and [15]). The complete set of hydraulic turbomachine-governor-pipeline equations should be used for the case of load reduction in which the pump-turbine speed is controlled by the governor. The governor equations are omitted in the analysis for the case of a turbine sudden load rejection or pump rundown in which the unit-speed change depends only on the net torque. Boundary conditions defining the sudden load rejection of the Francis turbine and the rundown of the centrifugal pump are as follows.

(1) Robni pogoj za trenutno razbremenitev francisove turbine

Robni pogoj za trenutno razbremenitev francisove turbine, vgrajene v cevovodu, zapisan v deltoidni mreži metode karakteristik, definiramo z naslednjimi enačbami (prehodna kavitacija na vstopnem in izstopnem robu francisove turbine ni dovoljena):

- združljivostni enačbi vodnega udara (3) in (4)
- energijska enačba:

(1) Boundary Condition for a Sudden Load Rejection of the Francis Turbine

The Francis turbine inline boundary condition for the case of sudden load rejection, which is incorporated into the staggered grid of the method of characteristics, is described by the following equations (no column separation is allowed at the Francis turbine inlet and outlet):

- water-hammer compatibility equations (3) and (4)
- head-balance equation:

$$H_u - H_r\left(\left(\frac{n}{n_r}\right)^2 + \left(\frac{Q}{Q_r}\right)^2\right)W_H\left(y(t),x\right) - H_d = 0 \quad (7)$$

- enačba vrtenja sklopa turbine in generatorja:

- dynamic equation of the turbine-unit rotating masses:

$$\left(\left(\frac{n}{n_r}\right)^2+\left(\frac{Q}{Q_r}\right)^2\right)W_T(y(t),x)+\left(\frac{T}{T_r}\right)_{t-2\Delta t}-I\frac{\pi}{30}\frac{n_r}{T_r}\frac{1}{\Delta t}\left(\left(\frac{n}{n_r}\right)-\left(\frac{n}{n_r}\right)_{t-2\Delta t}\right)=0 \qquad (8),$$

kjer so H_u – višina na navzgornjem koncu turbine, H_r – imenski padec turbine, n – vrtilna frekvenca turbine (pozitivna v turbinski smeri vrtenja), r – imenski pogoji, $W_H(y(t),\ x)$ – brezrazsežna turbinska tlačna karakteristika, $y(t)$ – brezrazsežno odprtje vodilnika, $x = \mathrm{tg}^{-1}((Q/Q_r)/(n/n_r))$ – turbina v polarnem karakterističnem digramu, definiranem za turbinsko in disipacijsko področje (samo v enem kvadrantu), H_d – višina na navzdolnjem koncu turbine, $W_T(y(t), x)$ – brezrazsežna turbinska momentna karakteristika, T – moment in I – polarni vztrajnostni moment vrtečih se delov.

in which H_u – the head at the upstream side of the turbine, H_r – the rated turbine head, n – the turbine rotational speed (positive in turbine direction), r – the rated conditions, $W_H(y(t), x)$ – the dimensionless turbine head characteristic, $y(t)$ – the dimensionless wicket-gates position, x – $\mathrm{tg}^{-1}((Q/Q_r)/(n/n_r))$ – the angular position of the turbine characteristic curve in generating and dissipating mode (only in one quadrant), H_d – the head at the downstream side of the turbine, $W_T(y(t), x)$ – the dimensionless turbine torque characteristic, T – the torque, and I – the polar moment of inertia of rotating parts.

(2) Robni pogoj za izklop centrifugalne črpalke

Robni pogoj za izklop centrifugalne črpalke, vgrajene v cevovodu, zapisan v deltoidni mreži metode karakteristik, določimo z naslednjimi enačbami (prehodna kavitacija na vstopnem in izstopnem robu centrifugalne črpalke ni dovoljena):
- združljivostni enačbi vodnega udara (3) in (4)
- energijska enačba:

(2) Boundary Condition for a Rundown of the Centrifugal Pump

The centrifugal pump inline boundary condition for the case of rundown, which is incorporated into the staggered grid of the method of characteristics, is described by the following equations (no column separation is allowed at the centrifugal pump inlet and outlet):
- water-hammer compatibility equations (3) and (4)
- head-balance equation:

$$H_u+H_r\left(\left(\frac{n}{n_r}\right)^2+\left(\frac{Q}{Q_r}\right)^2\right)W_H(x)-H_d=0 \qquad (9)$$

- enačba vrtenja sklopa črpalke in elektromotorja:

- dynamic equation of the pump-unit rotating masses:

$$\left(\left(\frac{n}{n_r}\right)^2+\left(\frac{Q}{Q_r}\right)^2\right)W_T(x)+\left(\frac{T}{T_r}\right)_{t-2\Delta t}+I\frac{\pi}{30}\frac{n_r}{T_r}\frac{1}{\Delta t}\left(\left(\frac{n}{n_r}\right)-\left(\frac{n}{n_r}\right)_{t-2\Delta t}\right)=0 \qquad (10),$$

kjer so H_u – višina na navzgornjem koncu črpalke, H_r – imenska črpalna višina, n = vrtilna frekvenca črpalke (pozitivna v črpalni smeri vrtenja), $W_H(x)$ – brezrazsežna črpalna tlačna karakteristika, $x = \pi + \mathrm{tg}^{-1}((Q/Q_r)/(n/n_r))$ – črpalka v polarnem karakterističnem diagramu, določenem za vse štiri kvadrante, H_d – višina na navzdolnjem koncu črpalke in $W_T(x)$ – brezrazsežna črpalna momentna karakteristika.

Neznanke v zgornjem sistemu enačb za francisovo turbino oziroma za centrifugalno črpalko so: višini H_u in H_d, pretok Q ($Q_d \equiv Q_u$) in vrtilna frekvenca turbostroja n. Sistem nelinearnih enačb (3), (4), ter (7) in (8) za turbino oziroma (9) in (10) za črpalko rešimo s Newton-Raphsonovo metodo [16]. V primeru prehodnega kavitacijskega toka dodamo zgornjemu sistemu enačb spremenjeno enačbo (6), zapisano za navzgornji ali navzdolnji rob turbostroja.

in which H_u – the head at the upstream side of the pump, H_r – the rated pump head, n – the pump rotational speed (positive in pump direction), $W_H(x)$ – the dimensionless pump head characteristic, $x = \pi + \mathrm{tg}^{-1}((Q/Q_r)/(n/n_r))$ – the angular position of the pump four-quadrant characteristic curve, H_d – the head at the downstream side of the pump, $W_T(x)$ – the dimensionless pump torque characteristic.

The unknowns in the above system of equations for the Francis turbine and the centrifugal pump respectively, are the heads H_u and H_d, discharge Q ($Q_d \equiv Q_u$) and turbomachine rotational speed n. The system of non-linear equations (3), (4), and (7) and (8) for the turbine, and (9) and (10) for the pump, respectively, is solved by the Newton-Raphson method [16]. The modified equation (6) is added to the above system of equations for the column-separation case, either at the upstream or the downstream side of the turbomachine.

3 PREVERITEV TEORETIČNEGA MODELA

Preveritev teoretičnega modela je prikazana za dva primera iz industrije. Prvi primer obravnava hidroelektrarno Toro II (Kostarika), ki ima vgrajeni dve 34 MW francisovi turbini [17]. Podajamo

3 VALIDATION OF THE THEORETICAL MODEL

Two case studies validating the theoretical model are presented. The first system under consideration, the Toro II hydro-electric powerplant (Costarica), is fitted with two 34-MW Francis turbines rezultate za trenutno razbremenitev obeh turbin. Drugi primer obravnava izklop centrifugalne črpalke v drenažnem črpalnem sistemu v rudniku Velenje [2]. Izračuni so bili izdelani z uporabo računalniških programov za analizo prehodnih pojavov v Litostroju ([18] do [20]). V teh programih so zajeti elementi pretočnega sistema hidroelektrarn in črpalnih postaj (hidravlični turbostroj, ventil, izravnalnik, tlačni kotel itn.).

[17]. Results from the sudden load rejection of the two units are presented. The second case study considers a centrifugal-pump rundown in Velenje (Slovenia) in the pumping system of a mine [2]. Calculations were performed with the aid of computer programs for hydraulic transient analysis in Litostroj ([18] to [20]). The hydropower plant and the pumping-station elements are included in these programs (hydraulic turbomachine, valve, surge tank, air receiver, etc.).

Primer 1 – HE Toro II

Hidroelektrarno Toro II (Kostarika) sestavljajo navzgornji zbiralnik, cevovod z ustreznim premerom D = 2,23 m in skupno dolžino L = 1577,3 m (sl. 1), dve 34 MW francisovi turbini z navpično gredjo (imenski padec H_r = 376 m, pretok Q_r = 10 m^3/s) priključeni na odvodni predor s premerom D = 2,5 m in dolžino L = 22,8 m ter navzdolnji zbiralnik. Gladina vode v navzgornjem zbiralniku z_u se giblje od 1069,5 m do 1075,0 m, gladina vode v navzdolnjem zbiralniku z_d pa od 689,7 m do 690,5 m. Imenska vrtilna frekvenca turbine je n_r = 720,0 min^{-1}, polarni vztrajnostni moment vrtečih se delov stroja pa I = 47,2×10^3 kgm^2.

Prevzemne meritve na terenu so zajemale naslednje preizkuse: start, obremenjevanje, zmanjšanje obremenitve ter trentno razbremenitev ene ali dveh turbin. Nastali vodni udar blažimo z ustrezno nastavitvijo časov zapiranja in odpiranja vodilnika.

Case Study 1 – Toro II HPP

The Toro II hydro-electric powerplant (Costarica) is comprised of an upstream reservoir; a penstock of equivalent diameter D = 2.23 m and total length L = 1577.3 m (see Fig. 1); two vertical-shaft 34-MW Francis turbines of rated head H_r = 376 m and discharge Q_r = 10 m^3/s, which are connected to the outlet tunnel of diameter D = 2.5 m and length L = 22.8 m; and a downstream reservoir. The water level in the upstream reservoir z_u is in the range from 1069.5 m to 1075.0 m; the level in the downstream reservoir z_d is in the range from 689.7 m to 690.5 m. The rated speed of the turbine is n_r = 720.0 min^{-1} and the polar moment of inertia of the unit rotating parts I = 47.2×10^3 kgm^2.

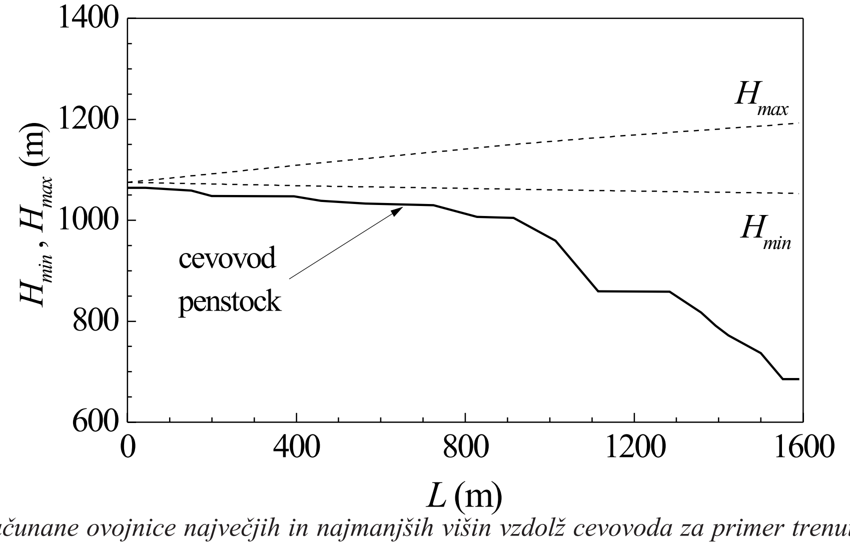
Various operating regimes were performed during the commissioning tests, including turbine start-up, load acceptance, load reduction and sudden load rejection of one or two turbines. The resulting water hammer was controlled by appropriate adjustment of wicket-gates closing and opening manoeuvres.

Trenutna razbremenitev dveh turbin

Trenutna razbremenitev dveh turbin je najbolj nevaren prehodni režim med normalnimi obratovalnimi razmerami [1]. Turbinska agregata izklopimo iz električnega omrežja, temu sledi hkratno polno zaprtje vodilnikov obeh turbin.

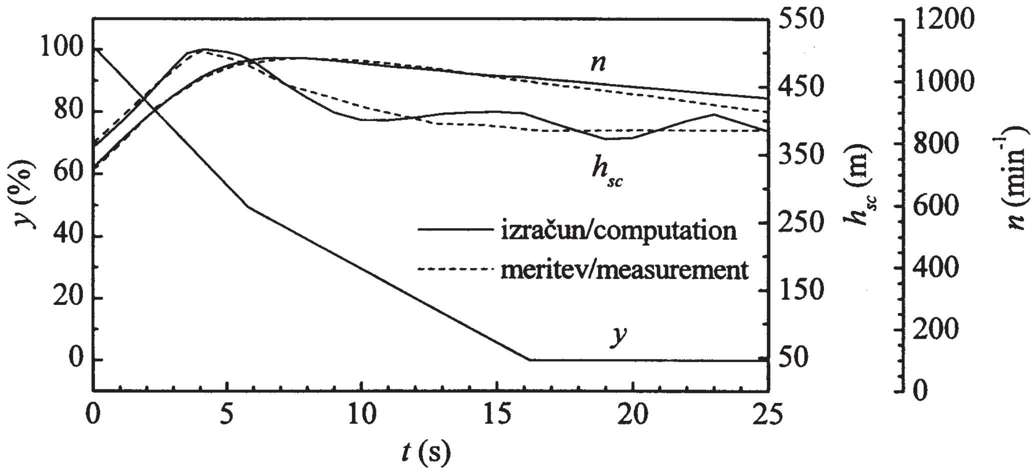
Sudden Load Rejection of Two Turbines

The sudden load rejection of two turbines is the most severe transient regime that occurs during normal operating conditions [1]. The turbines are simultaneously disconnected from the electrical grid followed by the simultaneous full closure of the wicket gates.



Sl. 1. *Izračunane ovojnice največjih in najmanjših višin vzdolž cevovoda za primer trenutne razbremenitve dveh turbin (H_{max} – največja višina, H_{min} – najmanjša višina, L – dolžina cevovoda)*

Fig. 1. *Computational envelopes of maximum and minimum heads along the penstock after the sudden load rejection of two turbines (H_{max} = maximum head, H_{min} = minimum head, L = penstock length)*



Sl. 2. *Primerjava izračunanih in izmerjenih rezultatov za primer trenutne razbremenitve obeh turbin (y - brezrazsežno odprtje vodilnika, h_{sc} - tlačna višina v spirali, n - vrtilna frekvenca turbine, t - čas)*

Fig. 2. *Comparison of computational and experimental results after the sudden load rejection of two turbines (y - dimensionless wicket gates position, h_{sc} - scroll-case pressure head, n - turbine rotational speed, t - time)*

Slika 1 prikazuje izračunane ovojnice največjih in najmanjših piezometričnih višin vzdolž profila cevovoda. Ta diagram omogoča inženirju načrtovanje varnega in gospodarnega cevnega sistema. Iz ovojnice najmanjše višne (H_{min}) izluščimo nevarnost pretrganja kapljevinskega stebra, ko se tlak zniža pod koto cevovoda. V našem primeru je izračunana višina zadosti nad vzdolžnim profilom cevovoda.

Največji tlak v spirali in največja vrtilna frekvenca turbine pomembno vplivata na izmere elementov turbine. Slika 2 prikazuje tlačno višino v spirali na vstopu v turbino h_{sc} (geodetska višina z = 685,0 m) in vrtilno frekvenco turbine n. Rezultati izračuna in meritev se dobro ujemajo. Izračunana največja višina $h_{sc\ max,c}$ = 504,2 m je rahlo večja od izmerjene višine $h_{sc\ max,m}$ = 501,0 m. Izračunana največja vrtilna frekvenca turbine $n_{max,c}$ = 1075 min^{-1} je rahlo manjša od izmerjene hitrosti $n_{max,m}$ = 1082 min^{-1}. Odstopanja med rezultati se povečajo pri majhnih odprtjih.

Computed envelopes of the maximum and minimum piezometric heads along the penstock profile are shown in Fig. 1. This diagram is important for design engineers to help them design a safe and economic pipeline system. The envelope of the minimum head (H_{min}) indicates the danger of liquid column separation when the pressure drops below the penstock profile. The computed minimum head is well above the penstock profile.

The maximum pressure in the scroll case and the maximum turbine rotational speed are two important parameters in turbine design. The pressure head in the scroll-case at the turbine inlet h_{sc} (datum level z = 685.0 m) and the turbine rotational speed n are depicted in Fig. 2. There is a reasonable agreement between the results of the computation and the measurement. The computed maximum head $h_{sc\ max,c}$ = 504.2 m is slightly higher than the measured one $h_{sc\ max,m}$ = 501.0 m. The computed maximum turbine rotational speed $n_{max,c}$ = 1075 min^{-1} is slightly lower than the measured speed $n_{max,m}$ = 1082 min^{-1}. The discrepancies between the results increase at small wicket-gates openings.

Primer 2 – Rudnik Velenje

Črpalni sistem v rudniku Velenje je visokotlačni sistem, kjer vodoravna večstopenjska centrifugalna črpalka potiska vodo v skoraj vertikalni cevovod s premerom D = 0,205 m in skupno dolžino L = 441,5 m (glej sl. 3). Voda prosto izteka v okolico. Koncentracijski razmernik trdnin v vodi je zanemarljiv. Na navzdolnjem robu črpalke je vgrajena nedušena povratna loputa, ki prepreči nasprotno vrtenje črpalke. Črpalka obratuje na imenski višini H_r = 382 m, pretoku Q_r = 0,05 m^3/s in vrtilni frekvenci črpalke n_r = 720 min^{-1}.

Na terenu smo preizkusili zagon in izklop črpalke. Vodni udar v cevovodu je v veliki meri vplivan z dinamičnim odzivom povratne lopute [21].

Case Study 2 –Velenje Mine

The Velenje mine pumping system (Slovenia) is a high head-system with a horizontal multistage centrifugal pump forcing water into a nearly vertical pipeline of diameter D = 0.205 m and total length L = 441.5 m (see Fig. 3). The water discharges freely into the atmosphere. The concentration ratio of solids in the water is negligible. An undamped swing-type check valve is installed at the downstream side of the pump to prevent pump-flow reversal. The pump operates at rated head H_r = 382 m, discharge Q_r = 0.05 m^3/s and pump rotational speed n_r = 720 min^{-1}.

Pump start-up and rundown tests were performed in-situ. The water hammer in the pipeline was controlled by the dynamic action of the check valve [21].

Izklop centrifugalne črpalke

Izklop centrifugalne črpalke je najbolj nevaren prehodni režim v obravnavanem črpalnem sistemu. Izklopimo elektromotor črpalke, povratna loputa se zapre v času t_{cv} = 1,1 s, črpalka pa se zaustavi v času t_{ps} = 40 s po izklopu.

Izračunane ovojnice največjih in najmanjših piezometričnih višin vzdolž profila cevovoda so prikazane na sliki 3. Iz slike 3 razberemo stvaritev področja neprekinjenega kavitacijskega toka na navzdolnjem koncu cevovoda, ki pa ima zanemarljiv vpliv na obliko obeh ovojnic. Obravnavani cevovod je projektiran tudi za podtlak.

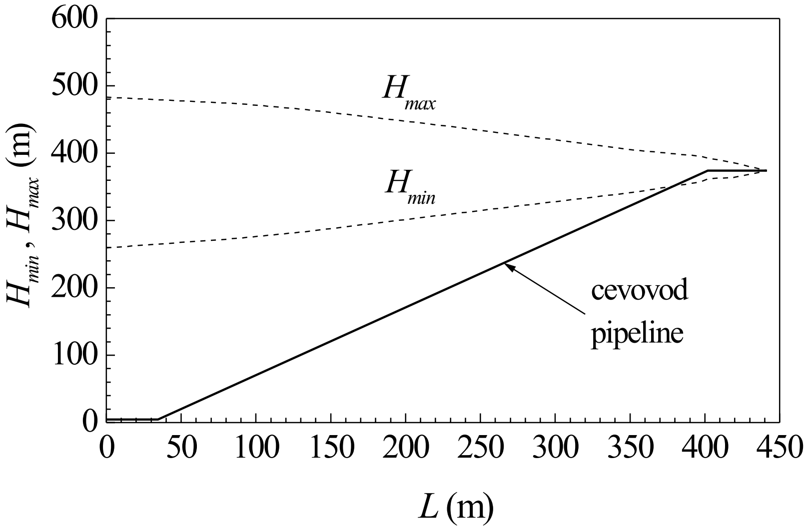
Na sliki 4 primerjamo izračunane in izmerjene tlačne višine $h_{cv,d}$ na navzdolnjem koncu povratne lopute, ki je priključena k črpalki. Izračunana največja višina $h_{cv,d\,max,c}$ = 481,4 m se dobro ujema z izmerjeno

Centrifugal Pump Rundown

Pump rundown is the most severe transient regime in the considered pumping system. The pump-electromotor was switched off, the check valve shut in time t_{cv} = 1.1 s and the pump stoppage time after power loss was t_{ps} = 40 s.

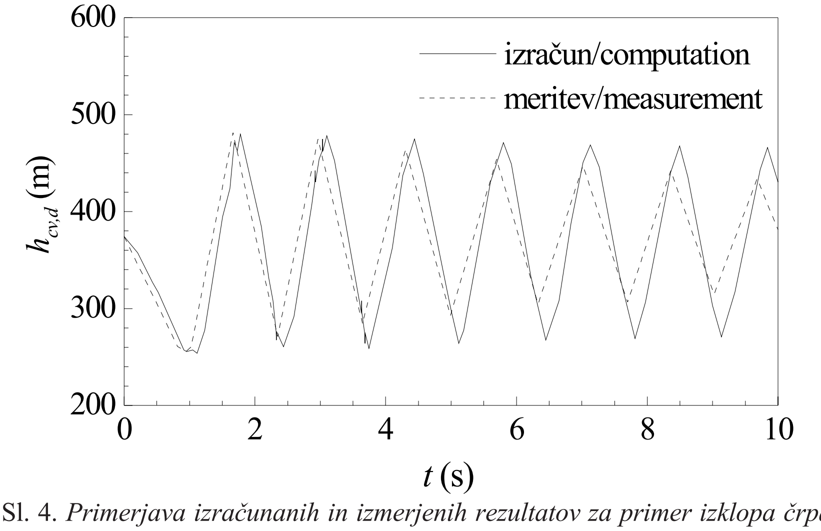
Computed envelopes of the maximum and minimum piezometric heads along the pipeline profile are shown in Fig. 3. As can be seen from Fig. 3, there is a distributed vaporous cavitation at the upper part of the pipeline, which does not significantly affect the shape of both envelopes. The pipeline is designed to withstand the underpressure.

Fig. 4 shows the comparison between computed and measured pressure heads $h_{cv,d}$ at the downstream end of the check valve connected to the pump. The computed maximum head $h_{cv,d\,max,c}$ = 481.4 m reasonably



Sl. 3. *Izračunane ovojnice največjih in najmanjših višin vzdolž cevovoda za primer izklopa črpalke (H_{max} = največja višina, H_{min} = najmanjša višina, L = dolžina cevovoda)*

Fig. 3. *Computational envelopes of maximum and minimum heads along the pipeline after pump rundown (H_{max} = maximum head, H_{min} = minimum head, L = pipeline length)*



Sl. 4. *Primerjava izračunanih in izmerjenih rezultatov za primer izklopa črpalke ($h_{cv,d}$ – tlačna višina na dolvodnem koncu povratne lopute, t – čas)*

Fig.4. *Comparison of computational and experimental results after pump rundown ($h_{cv,d}$ – pressure head at the downstream end of the check valve, t – time)*

višino $h_{cv,d\ max,m}$ = 483,3 m. Časovni odmik med izračunanim in izmerjenim potekom tlačne višine izvira iz zapletenega modeliranja dinamičnega odziva povratne lopute. Na prvem in drugem računskem tlačnem nihanju razberemo šibke utripe. Ti utripi so inducirani z zrušitvijo šibkih diskretnih kavitacij na navzdolnjem delu cevovoda. Iz izmerjene tlačne višine ne razberemo obravnavanih kavitacijskih učinkov. Kondenzacija izračunanih diskretnih kavitacij generira večje tlake kakor pa kondenzacija dejanskega področja neprekinjenega kavitacijskega toka vzdolž cevovoda [12].

matches the measured one $h_{cv,d\ max,m}$ = 483.3 m. The time shift between the calculated and measured head trace is mainly due to difficulties in the modelling of the dynamic behaviour of the check valve. The computed head exhibits low-amplitude pressure spikes superimposed on the first and the second pressure pulse. These spikes are due to discrete multi-cavity collapse at the upper part of the pipeline. The measured head does not exhibit such cavitating effects. Condensation of the computed discrete vapour cavities produces larger pressures than the condensation of the actual distributed vaporous cavitation zone along the pipeline [12].

4 SKLEP

Analiza prehodnih pojavov v pretočnih sistemih hidroelektrarn in črpalnih sistemih mora zajeti kritične obratovalne dogodke, da obremenitve, vzbujene z vodnim udarom, ne presežejo dopustnih vrednosti. Trenutna razbremenitev dveh francisovih turbin je najbolj nevaren prehodni režim v visokotlačni hidroelektrarni Toro II (Kostarika). Podobno vzbudi največje obremenitve izklop centrifugalne črpalke v visokotlačnem cevnem sistemu v rudniku Velenje. Rezultati izračuna, dobljeni z metodo karakteristik, se dobro ujemajo z rezultati meritev na terenu. Metodo karakteristik priporočamo za analizo prehodnih pojavov v hidravličnih sistemih, v katerih je dolžina cevovoda mnogo večja od premera cevi.

4 CONCLUSION

Transient analysis in hydro-electric power plants and pumping systems should include critical operating conditions such that the loads induced by water hammer are kept within the prescribed limits. Sudden load rejection of two Francis turbines is the most severe transient regime in the Toro II (Costarica) high-head hydro-electric powerplant. Similarly, centrifugal pump rundown is the most severe transient regime in the Velenje (Slovenia) high-head mine pumping system. The method-of-characteristics computational results agree reasonably with the results of the measurements for both cases. The method is recommended for the transient analysis in hydraulic systems where the pipeline length is much larger than the pipe diameter.

5 OZNAČBE
5 SYMBOLS

prečni prerez	A	pipe area
hitrost vala	a	wave speed
premer cevi	D	pipe diameter
zemeljski pospešek	g	gravitational acceleration
piezometrična višina	H	piezometric head
tlačna višina na navzdolnjem koncu povratne lopute	h_{cv}	pressure head at the downstream end of the check valve
tlačna višina v spirali	h_{sc}	scroll-case pressure head
parna tlačna višina	h_v	vapour pressure head
polarni vztrajnostni moment vrtilnih delov	I	polar moment of inertia of rotating parts
dolžina cevi	L	pipe length
vrtilna frekvenca turbostroja	n	turbomachine rotational speed
pretok	Q	discharge
moment	T	torque
čas	t	time
čas zapiranja povratne lopute	t_{cv}	check-valve closure time
čas nastanka kavitacije	t_{in}	time of cavitation inception
čas zaustavitve črpalke	t_{ps}	pump stoppage time
prostornina diskretne kavitacije pare	V_v	discrete vapour-cavity volume
pretočna hitrost	v	pipe flow velocity
brezrazsežna tlačna karakteristika turbostroja	W_H	dimensionless turbomachine head characteristic
brezrazsežna momentna karakteristika turbostroja	W_T	dimensionless turbomachine torque characteristic

koordinata, turbostroj v polarni karakteristiki	x	distance, angular position in turbomachine characteristic curve
brezrazsežno odprtje vodilnika	y	dimensionless wicket-gates position
geodetska višina	z	elevation above datum
časovni korak	Δt	time step
dolžina cevnega odseka	Δx	reach length
Darcy-Weisbachov koeficient trenja	λ	Darcy-Weisbach friction factor
strmina cevovoda	θ_p	pipe slope
utežni koeficient	ψ	weighting factor
Indeksi	*Subscripts*	
izračun	c	computation
navzdolnje	d	downstream
indeks računske točke	j	computational section index
meritev	m	measurement
največje	max	maximum
najmanjše	min	minimum
imensko	r	rated
čas	t	time
navzgornje	u	upstream

6 LITERATURA
6 REFERENCES

[1] Chaudhry, M.H. (1987) Applied hydraulic transients. *Van Nostrand Reinhold Company,* New York, USA.

[2] Bergant, A., A.R. Simpson, E. Sijamhodžić (1991) Water hammer analysis of pumping systems for control of water in underground mines. *Proceedings of the 4th International Mine Congress,* Ljubljana, Slovenia & Pörtschach, Austria, Vol. 2, 9 - 20.

[3] Wylie, E.B., V.L. Streeter (1993) Fluid transients in systems. *Prentice Hall,* Englewood Cliffs, USA.

[4] Bergant, A., E. Sijamhodžić (1997) Water hammer problems related to refurbishment and upgrading of hydraulic machinery. *Hydropower into the Next Century,* Portorož, Slovenia, 611 - 622.

[5] Bergant, A., A.R. Simpson (1997) Development of a generalised set of pipeline water hammer and column separation equations. *Report n. R149*, Dept. of Civil and Envir. Engrg., University of Adelaide, Adelaide, Australia.

[6] Wylie, E.B. (1983) Advances in the use of MOC in unsteady pipeline flow. *BHRA Conference on Pressure Surges,* Bath, England, 349 - 356.

[7] Anderson, A., M. Arfaie, R. Sandoval-Pena, K. Suwan (1991) Pipe-friction in water hammer. *XXIV IAHR Congress,* Madrid, Spain, D23 - D30.

[8] Bergant, A., A.R. Simpson (1994) Estimating unsteady friction in transient cavitating pipe flow. *BHR Group Conference on Water Pipeline Systems,* Edinburgh, Scotland, 3 - 16.

[9] Bergant, A., A.R. Simpson, V. Vítkovský (2001) Developments in unsteady pipe flow friction modeling. *IAHR Journal of Hydraulic Research,* 39(3), 249 - 257.

[10] Simpson, A.R., E.B. Wylie (1991) Large water hammer pressures for column separation in pipelines. *ASCE Journal of Hydraulic Engineering,* 117(10), 1310 - 1316.

[11] Bergant, A., A.R. Simpson (1992) Interface model for transient cavitating flow in pipelines. *Conference on Unsteady Flow and Fluid Transients,* Durham, England, 333 - 342.

[12] Bergant, A., A.R. Simpson (1999) Pipeline column separation flow regimes. *ASCE Journal of Hydraulic Engineering*, 125(8), 835 - 848.

[13] Simpson, A.R., A. Bergant (1994) Numerical comparison of pipe-column-separation models. *ASCE Journal of Hydraulic Engineering,* 120(3), 361 - 377.

[14] Wylie, E.B. (1984). Simulation of vaporous and gaseous cavitation. *ASME Journal of Fluids Engineering,* 106(3), 307 - 311.

[15] Krivčenko, G.I., N.N. Aršenevskij, E.V. Kvjatovskaja, V.M. Klabukov (1975) Hydromechanical transient processes in hydro power plants (in Russian). *Energija,* Moscow, Russia.

[16] Carnahan, B., H.A. Luther, J.O. Wilks (1969). Applied numerical methods. *John Wiley & Sons,* New York, USA.

[17] Bergant, A., E. Sijamhodžić (1998) Water hammer flow regimes in a high-head Francis turbine hydro powerplant. *Hydroturbo'98,* Loučná nad Desnou, Czech Republic, 237 – 245.

[18] Fašalek, J. (1985) Unsteady phenomena at complete rundown of the pump turbine with particular reference to the pumping mode of operation (in Slovene). *Master of Science Thesis*, Dept. of Mech. Engrg., University of Ljubljana, Ljubljana, Slovenia.
[19] Bergant, A. (1986) Review of modern theoretical methods for hydraulic transient analysis applied in Litostroj (in Serbian). *Transient Processes in Hydrotechnical Systems*, Belgrade, Serbia and Montenegro, Vol. 2, 8 – 20.
[20] Bergant, A. (1992) Transient cavitating flow in piping systems (in Slovene). *Dissertation*, Dept. of Mech. Engrg., University of Ljubljana, Ljubljana, Slovenia.
[21] Kruisbrink, A.C.H. (1988) Check valve closure behaviour. Experimental investigation and simulation in water hammer computer programs. *BHR Group Conference on Developments in Valves and Actuators,* Manchester, England, 21 pp.

Avtorjev naslov: doc.dr. Anton Bergant
Litostroj E.I. d.o.o.
Litostrojska 40
1000 Ljubljana
anton.bergant@litostroj-ei.si

Author's Address: Doc.Dr. Anton Bergant
Litostroj E.I. Ltd.
Litostrojska 40
1000 Ljubljana, Slovenia
anton.bergant@litostroj-ei.si

Prejeto:
Received: 13.12.2002

Sprejeto:
Accepted: 29.5.2003

Odprt za diskusijo: 1 leto
Open for discussion: 1 year