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PIPE SYSTEMS WITH MICRO-TURBINES: WATER HAMMER CONSIDERATIONS

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ABSTRACT: Micro and small turbines, as a means of producing clean and renewable energy by transforming hydropower to electricity, can be used extensively in pipe systems. With respect to hydraulic transient modeling, governed turbines have two additional features compared to the more familiar pump boundary condition: namely, wicket gate adjustments and more complicated device characteristics. Based on head balance (or nodal flow balance) considerations, torque (or speed change) relations, and the governor equation, a numerical model of the turbine boundary condition in a pipe system is established. The combinations of the three basic equations under specific situations are then discussed. To verify in a general way this numerical model, a penstock failure at Lapino Power Plant (Poland) is simulated. The current work sets the stage for a more comprehensive analysis of turbines and related unsteady flow issues in topologically complex pipe networks.

INTRODUCTION

Although hydropower has long been an important resource, only a few recent projects in developed countries such as Canada and US have been reported. This relative neglect may partly reflect the reality that much of the available large-scale hydropower has long-since been exploited, while small-scale projects have had proportionately higher unit costs and have thus often been considered uneconomical. However, with improvements to Small Hydropower (SHP) and Micro/Mini Hydropower (MHP), involving new materials, updated devices and improved design and operation, the cost of small hydropower units has become more competitive. Particularly, when no dam has to be constructed at many hydraulic potential sites, hydropower can be considered as one of the most cost-effective and environmentally sound alternatives for power generation [1]. Oil, as an energy source, by contrast, is not only increasingly expensive and a growing source of international conflict, but its consumption leads directly to greenhouse gases and other environmental burdens.

All this has led to a more enthusiastic search for alternative energy sources, including small-scale hydropower. In fact, early in the last decade, a United Nations report pointed out that, as a clean and renewable energy, SHP/MHP ought to be developed as a priority both for its economic benefits and for its contribution to other project objectives [2]. These other objectives might include its complimentary role in irrigation, water supply, fish breeding and various ecological enhancement projects. In the light of these economic, social and environmental incentives, even some countries that stopped SHP/MHP work have again been exploring its potential. Overall, there is a renewed interest in hydropower projects, especially involving SHP/MHP and micro-turbines.

Micro-turbines can be applied not only in dedicated hydroelectric plants, but also in multiple-use systems and for power recovery purposes. Installation of micro-turbines in many water conveyance and irrigation systems is gradually becoming a popular alternative to the wasteful dissipation of energy through devices such as pressure reducing valves (PRV) and head loss chambers. The use of micro-turbines in zones with a significant topographic difference (i.e., elevation difference) not only produces renewable energy, but also can avoid the use of stronger (more expensive) downstream pipes and may even reduce downstream water leakage. Moreover, extensive (and sometimes oversized) storage tanks in urban water supply systems may have a valuable hydraulic potential first for energy storage and then for subsequent electricity production during periods of high demand, thus allowing the municipal system to function somewhat as a pumped storage scheme. In cases like these, it is sometimes desirable to use centrifugal pumps (operated in reverse) as turbines, since pumps in such contexts are often cheaper and more readily available than turbines.

An important issue in any micro-turbine installation is that of system hydraulics. In particular, water hammer in systems with turbines can cause numerous problems, especially in systems with long pressurized pipes. These concerns originate largely from the complexity of turbine operation whose performance is influenced by the operation of the governor, generator, control valve, and wicket gates, from the properties of the conveyance network, and from the properties of the power grid connected to the unit. Yet, even allowing for this complexity, there are strict requirements for accuracy in prediction and calculation.

Regardless of the specific application, three key issues arise. The first is to represent the complete characteristics of the power unit (turbine/pump), typically using Suter parameters. The characteristic curves of a turbine are different for every gate position, and shape functions are used to interpolate a turbine's characteristics between different gate openings for various combinations of the independent variables.

The second issue is to calculate the transient scenarios within the conduit system. This problem may be more difficult in some systems, owing to the possibly complex nature of the both the pipe system and the devices within it. When micro-turbines are applied in multi-purpose pipe systems, the connected pressurized pipe systems can neither be ignored nor simplified into the simple conduits often assumed in traditional treatments. Rather, the transient equations have to be carefully coupled to both the upstream and downstream conduit systems with all the devices and device-interactions they contain.

Finally, a variety of transient events must be taken into account, including load rejection and load variation (reduction or increase) in three states: (i) with normal governor operation, (ii) with governor malfunction and (iii) without a governor. It is also important to distinguish whether it is necessary to account for the effect of rotation speed change when the generating unit is connected to either a large power utility network or a small isolated network.

Throughout the current work, the implications of the characteristics for the various types of turbines must be kept in mind.

TRANSIENTS CAUSED BY TURBINES

Compared with pumps, models of a governed turbine must contend with two additional complications [3]: (a) The governor will produce a change in wicket-gate position or blade angle in response to a speed change and an equation is thus needed to describe the dynamic response of the governor; (b) The turbine characteristic curves are different for every gate position and therefore additional characteristic curve data must be available for the numerical solution. For a pump, only one curve is required for both the head H and torque T characteristics, since pumps having variable characteristics are still rare. For a turbine, by contrast, two families of curves represent H and T characteristics with different gate positions.

When we calculate the water-hammer caused by load-rejection or load acceptance, the variation of turbine speed N must be taken into account under the following two cases:

(1) The turbine-generator is initially operating under load in an isolated network, whether with a governor or not, whether considering load-rejection or load-acceptance.

(2) The unit is on-line (in a large electrical network) initially, and when the total load is rejected, the unit will be disconnected from the power network.

The maximum allowable speed rise/decrease in the turbine is typically 45-65 percent. When the unit rejects the load, reducing the duration of regulation (i.e., valve/wicket gates closure time) can decrease the speed rise but will bring a bigger water hammer (pressure rise), so the reasonable closure time should be chosen to consider simultaneously both pressure and speed rise.

However, if a unit is on line and its power is a small portion of a power network, the variation of speed is small and can be neglected. In this case, the speed here must correspond to the nearest synchronous speed N_E whenever its load increases or decreases:

$$N_E=60\,f\,/\,p$$

in which f is the frequency of the AC supply system (i.e., usually 50 to 60 Hz) and p is number of the pairs of generator poles. The permitted variation of frequency is typically ±0.2 Hz for large systems, and ±0.5 Hz for small systems.

NUMERICAL MODELING

Head Balance Equation and Nodal Continuity Equation

Referring to Fig.1, and neglecting the changes in kinetic energy between the upstream node **A** and downstream node **B**, the energy equation (head balance equation) for this configuration is

$$F_H = H_A - H - H_B = 0 \qquad (1)$$

in which H is the net head difference across the turbine, and H_A, H_B are the instantaneous piezometric head (HGL) at nodes **A** and **B** respectively. The head at any node i can be calculated from the method of characteristics (MOC) equation [4] and the nodal continuity equation:

$$H_i = C' - B'(Q + Q_V) \qquad (2)$$

in which

$$B' = \left(\sum_{i \in N_1} \frac{1}{B_{P_i}} + \sum_{i \in N_2} \frac{1}{B_{M_i}} \right)^{-1}; \quad C' = B' \left(\sum_{i \in N_1} \frac{C_{P_i}}{B_{P_i}} + \sum_{i \in N_2} \frac{C_{P_i}}{B_{M_i}} \right) \qquad (3)$$

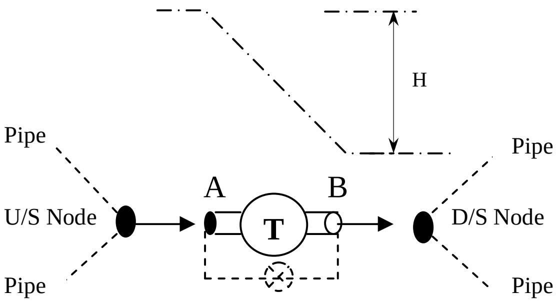

Fig. 1 Schematic of Turbine System with/without PRV

The summed terms account for any pipes connected at the turbine. Specifically, at node *i*, the set N_1 represents the pipes whose assumed direction is toward the node and N_2 the set of pipes whose assumed direction is away from the node; B_P, B_M, C_P, C_M are characteristic (pipe) constants; Q is the discharge through the turbine; and Q_V is the discharge through the bypass PRV (Pressure Regulation Valve), if present. When there is a PRV installed and activated, $Q_V = \tau\, Es\, H^{0.5}$ where τ is the known relative opening of the PRV at every time step; if no PRV is present or if it is inactive, then $Q_V = 0$.

For a typical turbine system in a conventional hydropower plant, the pressurized downstream water passage is usually short and its transient effect can be largely neglected. That is, H_B is known from the water level of tailrace; and in a network context, H_B can itself be determined as part of a complex system.

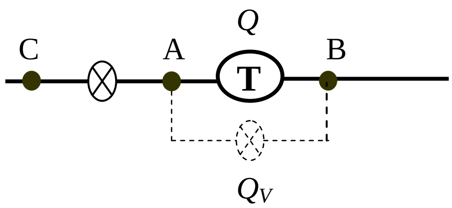

Fig. 2 Schematic of the turbine system with valves

As sketched in Fig.2, if there is a Shut-off Valve or Inlet Valve, it is usually installed in front of the turbine unit (about a 2-3 m distance). To avoid a short pipe in the system, the shut-off valve or inlet valve, with the connector between the valve and turbine, can be treated as one boundary condition; thus, node **A** connects two different devices directly, i.e., the Valve-in-Line (**CA** section) and Turbine Unit (**AB** section). Numerically, the discharge at node **A** (i.e., Q_{ext}) can be computed within the **CA** section and then used in the solution of turbine unit. In this case, the continuity equation at node **A** will replace the head balance equation. For convenience, the first control equation can be generalized as:

$$F_1(\alpha, v, y) = 0 \tag{4}$$

Here, y is the relative wicket gate opening; α is the relative rotational speed (i.e. $\alpha = N/N_R$); the parameter x is defined as $x = \tan^{-1}(\alpha/v)$; and $v = Q/Q_R$ is the relative flow rate.

As has become standard practice in derivations, all of the control equations in this paper are expressed in terms of relative parameters α, v, relative head $h = H/H_R$ and relative torque $\beta = T/T_R$. The subscript "R" represents the rated parameters. Then two dependent variables of x, that is Suter parameters W_H and W_B, are defined as:

$$W_H(x) = \frac{h}{\alpha^2 + v^2} \; ; \quad W_B(x) = \frac{\beta}{\alpha^2 + v^2}$$

By virtue of the shape function, the expressions of $H = H_R (\alpha^2 + v^2)(a_0 + a_1 x + a_2 y + a_3 x\, y)$ and $\beta = (\alpha^2 + v^2)(b_0 + b_1 x + b_2 y + b_3 x\, y)$ are used to represent the head and torque characteristic relationships of turbine, respectively. The parameters a_0, a_1, a_2, a_3 and b_0, b_1, b_2, b_3 are interpolation coefficients of the shape functions. For different types of turbines, different characteristic relationships are applied into calculations.

Speed Change Equation (Torque Balance Equation)

The speed change equation can be expressed as [3]:

$$T_m \frac{d\alpha}{dt} = \beta - \frac{P_g}{P_R \alpha} \tag{5}$$

where P_R is the rated power; P_g is the power absorbed by the generator; $T_m = I\,\omega_R^2 / T_R \omega_R$ is the mechanical starting time; and $\omega_R = 2\pi N$ is the angular speed of the turbine. Implicit integration of (5) over the time step Δt, results in

$$F_2(\alpha, v, y) = \beta - \frac{1}{P_R}\frac{P_g}{\alpha} - \frac{T_m}{\Delta t}(\alpha - \alpha_0) = 0 \tag{6}$$

The subscript "0" denotes variables evaluated at the previous time step.

Governor Equation

An equation of a Proportional-Integral-Derivative speed-control governor can defined as [3]:

$$T_d T_\alpha \frac{d^2 y}{dt^2} + T_\alpha' \frac{dy}{dt} + \sigma(y-1) + (\alpha - 1) + T_d \frac{d\alpha}{dt} = 0 \tag{7}$$

In which T_d = dashpot time constant; T_α = promptitude time constant without temporary speed droop; $T_\alpha' = T_\alpha + \delta T_d$ is the promptitude time constant with temporary speed droop; σ = permanent speed droop and δ = temporary speed droop.

Assuming Z is the rate of turbine opening, i.e. Z= *dy/dt*, then using difference approximation and integrating the governor equation (7) over Δt, we have the following form of governor equation:

$$\left(\frac{y-y_0}{\Delta t}-Z_0\right)+\frac{T_\alpha^{'}}{T_dT_\alpha}(y-y_0)+\frac{\sigma\Delta t}{T_dT_\alpha}(y-1)+\frac{\Delta t}{T_dT_\alpha}(\alpha-1)+\frac{1}{T_\alpha}(\alpha-\alpha_0)=0$$

$$i.e. \qquad F_3(\alpha, y)=0 \tag{8}$$

In this governor equation, the subscript "0" again represents the values at previous time step. As long as the coefficients in the equation are properly identified, this equation effectively describes mechanical, electric or digital systems. If the turbine speed is effectively known, one can solve this equation and thus obtain the dimensionless opening y. The term y represents the position of main servomotor and may itself control a needle valve, wicket gates, runner blades or a combination of these or other elements; for a Kaplan turbine, both the wicket gates opening and the runner blade angle have to be continuously updated.

Numerical Solution for Typical Applications

At each time step, the value of dimensionless opening y is first estimated by integrating (8) explicitly; then the operating point from the turbine characteristics is located by y and the values of α and v at the previous iteration step. Next, an implicit method (Newton-Raphson) is used to solve for α and v based on equations (4) and (6). Finally, the opening y is corrected by the new condition obtained. These steps are repeated until the solution has converged. This overall computational framework is sketched in the Appendix, and discussed in more detail next.

TURBINE OPERATION

Isolated Unit or Isolated Unit With Governor

The term "isolated unit" refers to a unit operating in an isolated (or small) electric network. In this case, speed-control governing is essential. However what is not so clear is the criteria for determining whether a system is large or small. The general rule of thumb is that any unit on a system supplying 40% or more of the system load [5] should be designed as an isolated unit, but the relationship of the system capacity to the load change should also be compared. Any load change that is 10% or more of the system capacity should be analyzed to determine its effect on frequency [5].

In (6), the variation of unit load P_g with time is known (or assumed), including load-rejection or load-decrease, load-acceptance or load increase. Thus, either the head balance equation or upstream node continuity equation (4), speed change equation (6), and governor equation (8) constitute the basic governing equations for the isolated turbine unit, so the three unknowns α, v and y can be simultaneously determined.

Isolated Units without Governor or Governor Malfunction

Without the governor, the opening movement $y=y(t)$ or the variation of relative flow rate $v=v(t)$ is set in advance, typically based on external circumstances, when the load of unit increases or decreases.

If the governor is out of operation, the wicket gates cannot be closed, so $y=y_0$ and thus the governor relation (8) is removed from the equation solution set. Under this situation the emergency valve (i.e., shut-off valve) may or may not be actuated. Thus one has to choose head balance equation, or node continuity equation, as the first governing equation and combine it with the speed change equation (6) to solve for the two unknowns (α, v) at each time step.

Connected Units with Governor

Having the unit connected to the network, the unit will keep a constant synchronous speed, at least if the permanent speed droop σ and dynamic process of adjustment can be neglected. With constant speed ($N = N_R$, i.e., $\alpha=1$) the relative speed is clearly known, there are only two unknowns v and y, and two governing equations are required. Specifically, the first governing equation reduces to $F_1(v, y) = 0$ and the governor equation (8) reduces to $F_3(y)=0$; the speed change equation is not used; rather the governor adjusts the opening y to keep the dynamical power produced by the turbine equal to the applied load: i.e., $T\omega = \eta\gamma QH = P_g$.

Connected Unit under Total Load-rejection

When the generator is disconnected from the network, the power, $\boldsymbol{P_g}$, absorbed by the generator is zero and the speed change equation (6) reduces to:

$$F_2 = \beta - \frac{T_m}{\Delta t}(\alpha - \alpha_0) \tag{9}$$

Typically, the governor is still controlling the unit, so the same process as for "isolated unit with governor" can be used.

Connected Units without Governor

With constant speed ($N=N_R$), α is known (α=1); in addition, the turbine opening or the turbine discharge is set in advance, i.e., $y=y(t)$ or $v=v(t)$ is also known. At each time step, the known value is substituted into the first governing equation $F_1(v, y)=0$, thus allowing the other to be obtained.

For different types of turbines, and even for the same type of turbine with different specific speeds N_s, different characteristic data exist for both the head/energy equation and torque/speed-change equation. Similarly, for different types of governor, different governor equations should be employed. For impulse turbines having jet deflectors, the controlling and working principle of the deflector needs to be mathematically expressed. Particularly for transients controlled by valve/gates operations, the governor equation (the relationship of speed change and opening) will be replaced by known opening or discharge $y=y(t)$ or $v=v(t)$; however, the operating process of control valves/gates (e.g., wicket gates, the pressure relief valve, shut-off valve) must be clear. In summary, although the basic equations are somewhat variable depending on the details of the specific event, but no significant differences exist in the solution approach among the various equipment or devices.

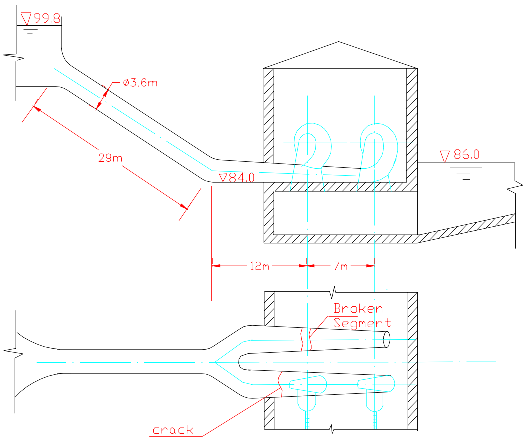

Fig. 3 Layout of Flow System at Lapino Power Plant

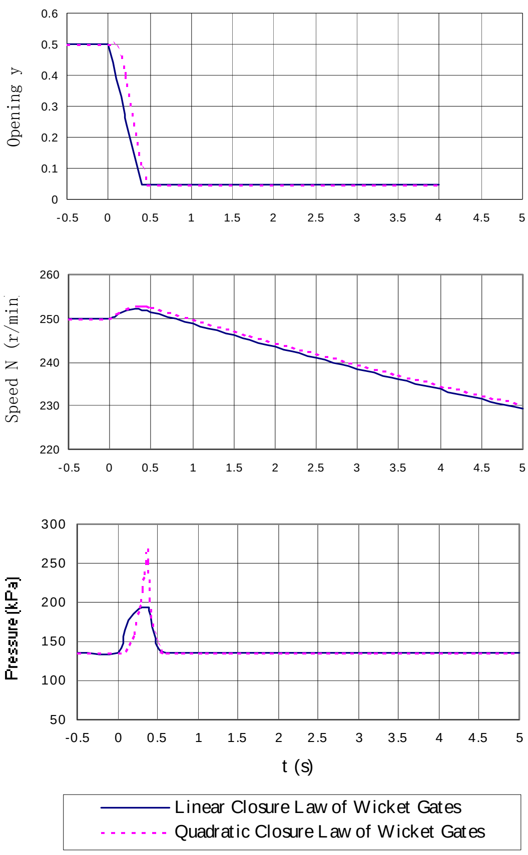

Fig.4 Transients Calculated for Failure Conditions

APPLICATION OF THE NUMERICAL MODEL

Adamkowski [6] investigated the event of a penstock rupture at the Lapino hydropower plant, Poland, in December 1997. An analysis of associated transients events under the condition of failure was carried as well.

The power plant with total capacity of 2 MW is equipped with two identical generator sets, each including two horizontal Francis turbines and a generator. The rated head and discharge of the turbine are 13.8 m and 22 m^3/s, respectively. The flow system is sketched in Fig. 3. The main rupture of the penstock shell took place in the left branch (looking downstream) that supplied water to generator set No.1. A load rejection test of the genenetor set No. 2 (from a load of about 50% of the rated power output) had been carried out immidiately before the pipe failure. Generator set No.1 was disconnected from the network during this test yet, significantly, it was not hydraulically disconnected form the system since there was no isolation valve.

To simulate load rejection, the rotational speed change of the turbine has to be taken into account. Generator set No.2 is equipped with a new electro-hydraulic governor. Because of a lack of appropriate data, and also for comparing with the simulation situation performed by Adamkowski, the variation of wicket gates was assumed to follow either a linear or a quadratic law. The parameters and data concerning turbines were formulated based on rated power, rated flow, rated speed (250 r/min) and constant head. First, the specific speed can be estimated (*Ns*=300 r/min), and then the corresponding characteristics of Francis turbine could be applied.

The results obtained from the numerical modeling in this paper, which is shown in Fig.4, fit well with the simulation in [6]. The maximum water hammer at the instant of penstock rupture is 194.76 kPa (210.8 kPa in [6]) using linear closure law and increases to 267.95 kPa (260.0 kPa in [6]) when using quadratic closure law. These results imply that the estimated values of critical breach pressure (250-300 kPa) can be exceeded if the closure time of the wicket gates from 50% opening is less than the values used in simulation. The closure law and closure time period of wicket gates are very sensitive to the simulated water hammer results, though they do not significantly influence the speed change. The difference in the maximum values of water hammer and the wave attenuation processes between this paper and Adamkowski's might arise from small differences in wicket gates closure, turbine parameters or pipe parameters used during the simulation.

CONCLUSIONS

1) As a cost-effective and environmentally sound alternative for power generation, a significant amount of micro hydropower potential remains untapped in North America and elsewhere and could be realized and developed [1]. In this quest, the importance of critical hydraulic and transient issues must be appreciated.

2) Transient hydraulic modeling of turbines is an important aspect in the design of pipe systems. Even for systems with short penstocks, water hammer events associated with caused fast shut downs and load rejections can lead to serious consequence; the closure of the wicket gates is the key factor in this determination, and should thus be controlled and simulated carefully.

3) The numerical model of a turbine system presented here, and illustrated for a simple pipe system, is general enough to be used in complex pipe systems for many turbine applications and situations. The issues of turbines and transients in more topologically complex systems require further exploration.

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APPENDIX – Flow Chart of Turbine Modeling