PUMPS YIELDING POWER

H. RAMOS(1) and A. BORGA(2)

(1)Assistant Professor, Civil Engineering Department of IST, Technical University of Lisbon, Portugal
(2)Invited Assistant, Civil Engineering of IST, Technical University of Lisbon, Portugal

ABSTRACT: Although pumps are widely used in industrial processes, the application in water supply systems for energy production can be feasible, depending, fundamentally among other factors, upon the efficiency and the control system of the dynamic behavior, in particular, during the groups' shut-off. This solution can offer equipment cost savings and seems to be adequate to be installed in a run-of-the-river schemes of small or micro hydro developments.

The hydrotransients induced by these machines are strongly associated to the pump type, normally defined through its specific speed.

INTRODUCTION

In water supply systems, such as drinking or sewage systems, with excessive head in some zones, a considerable head could be profit, in order to produce energy. The use of pumps as turbines could be an alternative solution to Cross-flow turbines or others types, which presents reasonable efficiencies with relatively low cost investment. This solution requires a detailed study about its behavior during steady and transient state conditions. Since the pumping head and the discharge depend upon the pump speed that is, normally, controlled by the grid frequency, in case of linkage to a large grid, or through the power demand, in case of an isolated station, several instabilities can occur in the system. When a pump is installed instead of a turbine and, in particular, for low specific speed, the runner' overspeed can provoke significant hydrotransient through a sudden flow reduction [1 and 7]. Therefore, a severe safety problem, along the conveyance system, can be taken into account, whenever a full-load rejection occurs. This phenomenon can condition the pipeline design, as well as the system stability.

Based on Suter parameters the analysis of both steady and transient regimes will take place, in order to point out the behavior similitude between turbines and pumps, whenever pumps operate in turbine zone. This study intends to be a pragmatic tool for better understanding about the use of pumps to yield power, in what concerns the operation under runaway conditions that will influence the dynamic behavior of the system, and which will be the reasonable efficiency for steady-state regime.

PUMP OPERATING CONDITIONS

During pump operation the discharge, Q, is a function of the rotating speed, N, and the pumping head, H, whereas the alteration of the transient speed will depend upon the torque of the motor T [1 and 3]. When the pump operates in turbine zone the motor will operate as a generator. The curves obtained through the relationships between these four variable parameters are identified by pump characteristic curves, where the rated condition ($_R$) refers at the point of the best efficiency. By using rated values as non-dimensional reference, it will remain variables defined as the following:

$$q = \frac{Q}{Q_R} \quad h = \frac{H}{H_R} \quad n = \frac{N}{N_R} \quad b = \frac{T_G}{T_{G_R}} \tag{1}$$

The signs of q and n define four quadrants where the signs of h and b define different pump operating zones (see Figure 1) [3].

During normal pump operation q, h, n and b are all positive (Figure 1). For normal turbine operation the rotating speed (n) and discharge (q) are negatives and the head (h) and torque (b) are positives. When b is equal to zero the machine reaches the runaway and dissipation condition.

Prototype operating conditions are obtained from manufacturer data by using homologous relationships. In these relationships θ is defined as the angle of operating zone that can vary between 0º and 360º, for all range application (Figure 1) through the following equation:

$$\frac{n}{q} = \tan\theta \qquad (2)$$

The machine characteristic curves are defined by WH and WT the head and torque Suter parameters, respectively, for different operating zones (depending on θ value) which can be represented in Figure 2 through the following equations:

$$WH = h/(n^2+q^2) \qquad (3)$$

and

$$WT = b/(n^2+q^2) \qquad (4)$$

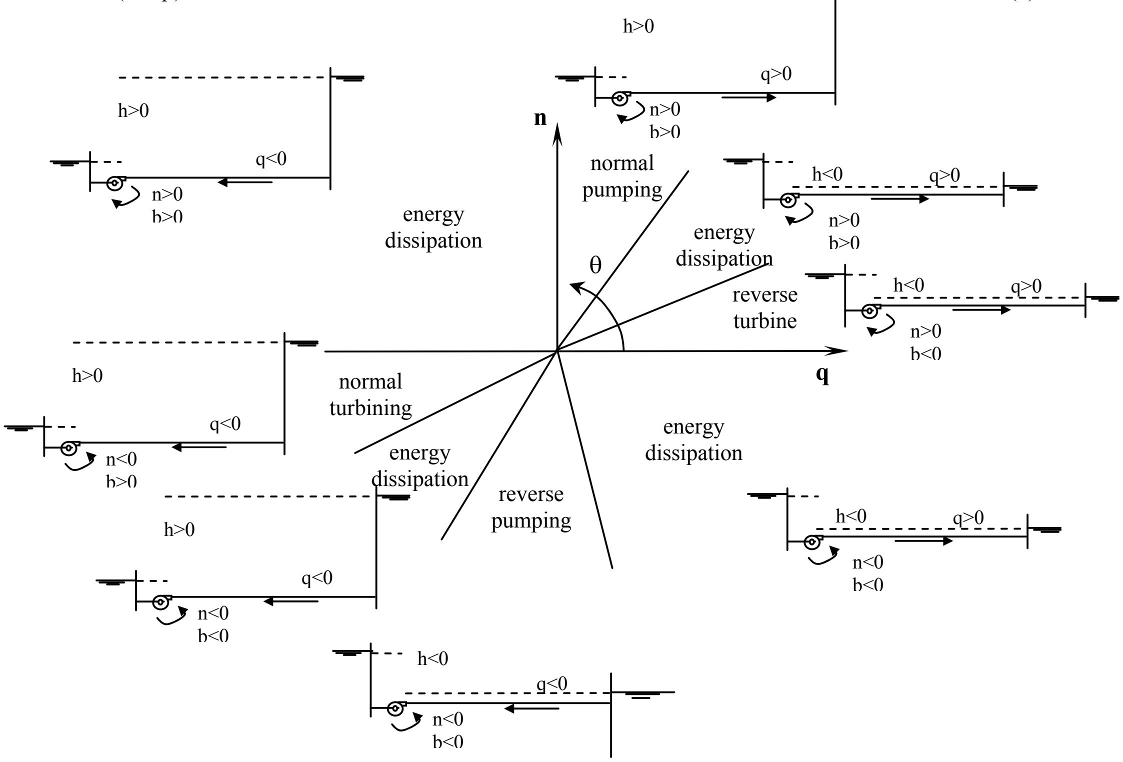

Figure 1. Operation zones for a pump with the identification of the variability of the typical characteristic parameters

Although, pump characteristic curves, in normal pumping zone are available for particular pumps, for turbines is rather scarce or there is a very incomplete data information [3 and 9].

General speaking, the specific speed is one of the main parameter of turbo-machines that characterizes the type of the runner (i.e. radial, mixed or axial flow), as well as the blade shape, the spiral casing and other design features [7]. The specific speed (N_s) can be defined by different ways. Herein N_s is obtained in m and kW by the following equation:

$$N_s = N_R \frac{\sqrt{P_R}}{H_R^{1.25}} \qquad (5)$$

in which N_R = rated wheel speed (r.p.m.), H_R = rated head (m) and P_R = rated power (kW), being $_R$ the pump design point or the best efficiency condition.

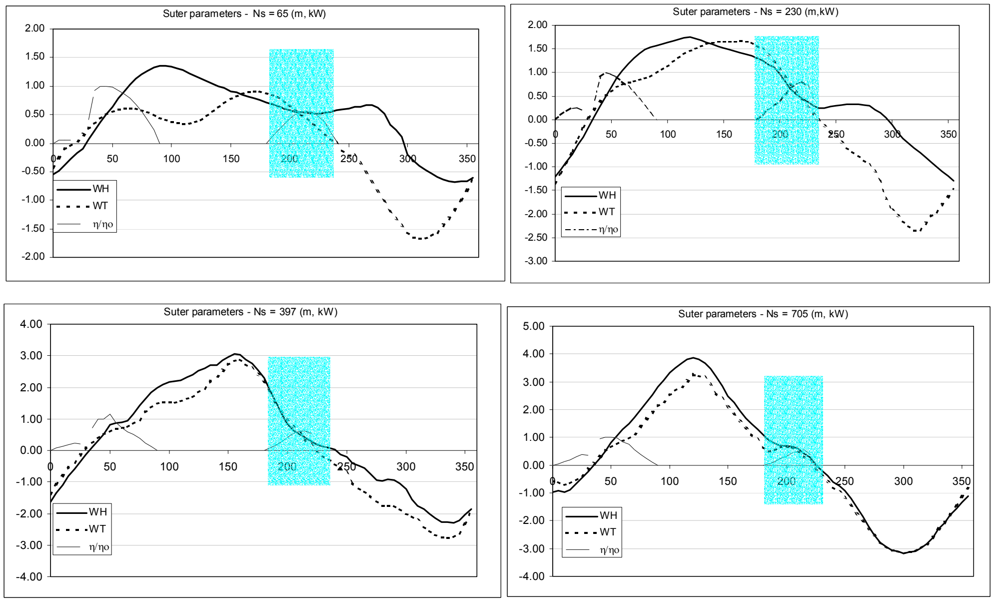

Figure 2. Suter parameters for pumps, for different specific speeds

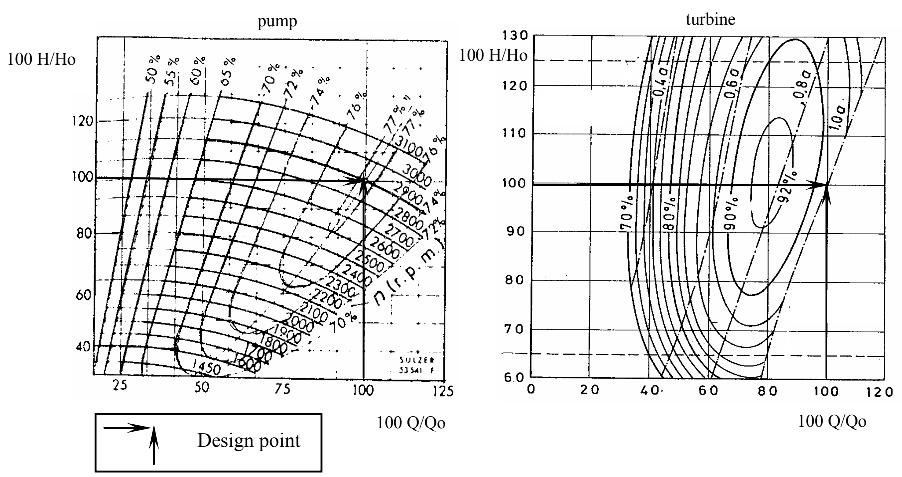

Figure 3. Design point in a pump and in a turbine hill charts

(adapted from [8])

In order to compare pump and turbine behaviors will be used available information data (Suter parameters [3]) for a radial-pump flow (N_s = 65 r.p.m.(m,kW)), a mixed-pump (N_s = 230 r.p.m.(m,kW)), and two axial-pumps (N_s= 397 and 705 r.p.m.(m,kW)), which N_s, for turbine must be corrected to the normal turbine operating point (e.g. given through $N_{sT} = \dfrac{N_s}{\sqrt{Q/Q_R}}$, for Q = Qmax and not for the maximum efficiency). It corresponds to verify that the maximum efficiency, for reaction turbines, generally, is not for the total opening position of the guide vane (Figure 3 and 4). In this way, the turbine zone will be thorough analyzed and relevant conclusions will be presented whenever a system claims to yield power.

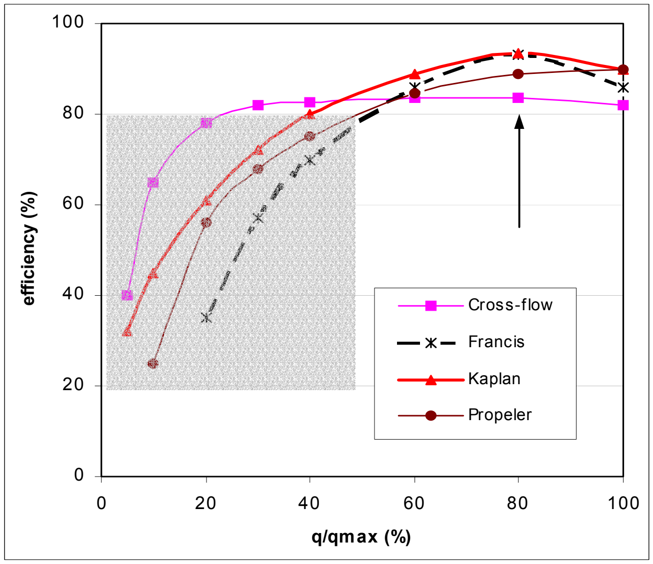

Figure 4. Turbines efficiency for discharge variation

EVALUATION OF STEADY – STATE REGIME

In a pump, when operating as a turbine, could be differentiated two types of power [3]:

- **Flow power** (P_{flow}) - h and q represent the transmitted power from the flow to the pump;
- **Generator power** ($P_{generator}$) - n and b define the transmitted power by the pump to the generator.

These powers will influence the turbine efficiency during normal operation. Whereas in a pump the efficiency is defined as

$$\eta_P = \frac{P_{flow}}{P_{motor}} \tag{6}$$

since the pump can operate as a turbine, the turbine efficiency (η_T) will be defined, in an equivalent way, through the following equation:

$$\eta_T = \frac{P_{generator}}{P_{flow}} = \frac{T_G N}{\gamma Q H} \tag{7}$$

yielding the following dimensionless equation

$$e = \frac{\eta_T}{\eta_{TR}} = \frac{T_G/T_{GR} \quad N/N_R}{Q/Q_R \quad H/H_R} = \frac{b\,n}{q\,h} = \frac{p}{qh} \tag{8}$$

or in an equivalent way

$$e = \frac{\eta_T}{\eta_{TR}} = \tan\theta \quad \frac{WT}{WH} = \tan\theta \frac{b}{h} \tag{9}$$

being e a percentage of the maximum efficiency.

Figure 5 shows the variability of relative efficiency for different types of pumps and Table 1 presents the conditions were occur the maximum values. The curve which presents the best efficiency conditions, corresponds to N_s = 230 r.p.m. (m, kW).

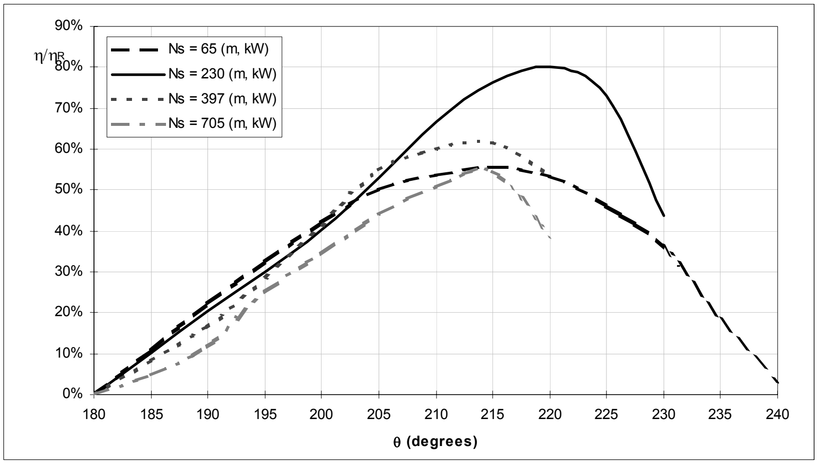

Figure 5. Efficiency variation for different type of pumps for normal turbining

Table 1. Maximum efficiency for normal turbining under rated conditions

N_s (m,kW)	p_{flow} = -q h	$p_{generator}$ = -b n	θ	e = η/η_R
65	1.04	0.57	215	0.55
230	0.73	0.58	220	0.80
397	0.37	0.23	214	0.62
705	0.06	0.03	214	0.54

Solving numerically and in simultaneously equations (3), (4) and (8) for a pump with N_s = 230 (m, kW), as a function of q, for different values of p and e, h values are obtained (Figure 6-A). Directly from equation (3) Figure 6-B shows curves of equal relative rotating speed.

The main feature of the analysis of Figure 6 enables concluding, that independently of head value and save for scale effects, it is possible to obtain a maximum relative efficiency (η/η_R) up to 80%, when the pump operates with a discharge close to condition of $q=q_{max}$.

In a real turbine, with a guide vane installed, normally, the rotating speed is fixed to the nominal one (n=1 or $N=N_R$) and the discharge is regulated by the guide vane opening, in order to obtain the desired output power. In this way, the turbine characteristic curve is defined for constant N, but variable guide vane aperture. In a pump there is not this device (guide vane), therefore is equivalent to consider the total opening of the guide vane and the rotating speed varying according to the adjustable output power. For isolated machines, the electric load will fix the output power and nothing will fix the speed, as in same way with turbines without guide vane. Nevertheless, in a real pump operation and in machine connected to a large grid, this problem is not applied due to the grid frequency will control the rotating speed [6 and 9].

The operating point is defined by the interception between the characteristic curve of the machine and the characteristic curve of the hydraulic system (e.g. depending on the way of machine control).

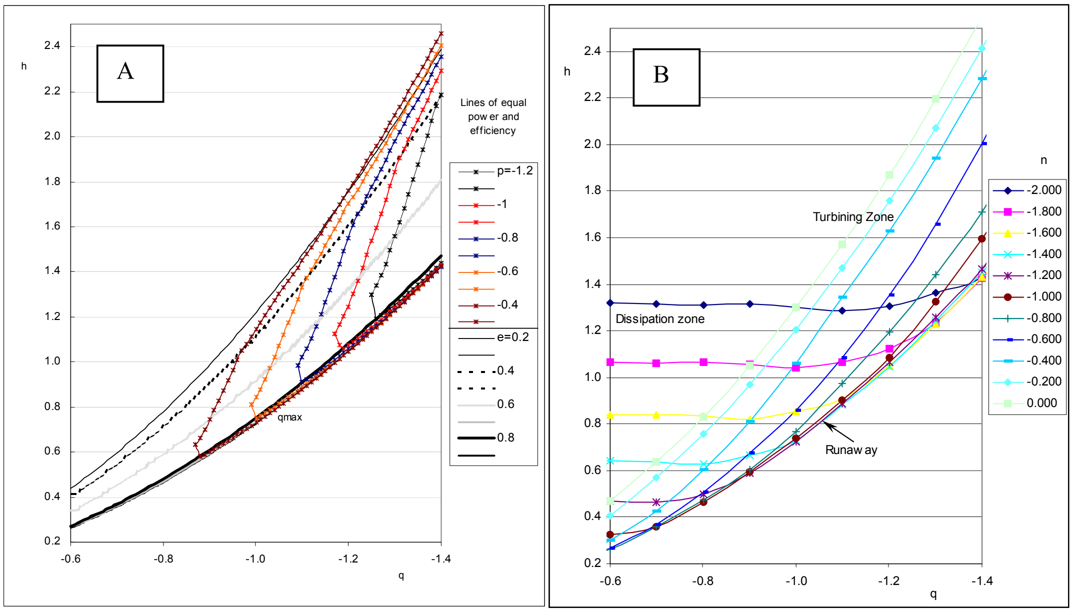

Figure 6. Characteristic curves of a pump operating in turbine zone. A) curves of equal relative power and efficiency for different values of relative head and discharge; B) curves of equal relative rotating speed

In order to avoid instability situations from turbine zone operating, the pump might operate in the point of the characteristic curve correspondent to the maximum power, normally close to the maximum efficiency. The reason of instabilities occurrence can be explained through the interception between the characteristic curve of the hydraulic system and the line of equal power, which means, for each power value, that would have two possible operating points.

Schematically, in Figure 7 is represented the operating point, supposing the output power control, characterized through the dash line, correspondent to p_{max}, which has only one solution (in the peak of relative power curves (p)).

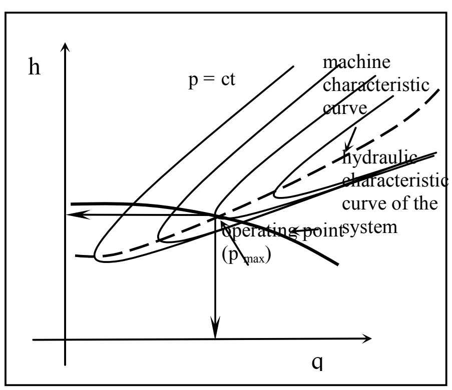

Figure 7. A scheme to explaine the operating point of a pump in turbine zone (interception between characteristic curves of the hydraulic system and the machine)

Anyway to explain the operating point for these hydropower schemes, a mathematical formulation is required through the definition of the hydraulic characteristic curve:

- <u>Hydraulic characteristic curve</u>

$$H = Z - KQ^2 \tag{10}$$

being Z the difference topographic water levels.

Transforming this equation dividing it by the pump rated head $H_{R,}$ which is generally defined through the following relationship:

$$H_R = Z + KQ_R^2 \tag{11}$$

yields

$$\frac{H}{H_R} = \frac{Z}{H_R} - \frac{KQ^2}{H_R} \tag{12}$$

which is equivalent to

$$h = z - kq^2 \tag{13}$$

being $z = \frac{Z}{H_R}$, $k = \frac{KQ_R^2}{H_R}$, h and q are defined in equation (1).

The dimensionless of equation (12) based on the pump rated head H_R allows obtaining a relationship between parameters k and z:

$$1 = \frac{Z}{H_R} + \frac{KQ_R^2}{H_R} \tag{14}$$

or equivalent equations

$$1 = z + k \qquad \text{or} \qquad k = 1 - z \tag{15}$$

Thus, the equation (13) can be replaced by the following one:

$$h = z - (1 - z)q^2 \tag{16}$$

- <u>Turbine characteristic curves</u> (based on Suter parameters)

$$WH = h/(n^2+q^2) \tag{3}$$
$$WT = b/(n^2+q^2) \tag{4}$$

That can be transformed according to h and p parameters in the following way:

$$\begin{aligned} h &= WH\left(n^2 + q^2\right) \\ p &= WT\left(n^2 + q^2\right)n \end{aligned} \tag{17}$$

since,

$$p = b \, . \, n \tag{18}$$

Equalizing h of equations (16) and (17) and using the p relationship from the same last equation (17) yields the following functions:

$$\begin{aligned} &WH\left(n^2 + q^2\right) - z + (1 - z)q^2 = 0 \\ &p - WT\left(n^2 + q^2\right)n = 0 \end{aligned} \tag{19}$$

Through the analysis of functions presented through equations (19), and for different values of q and n, lines of equal power (p) and relative difference topographic water levels (z) were performed in Figure 8.

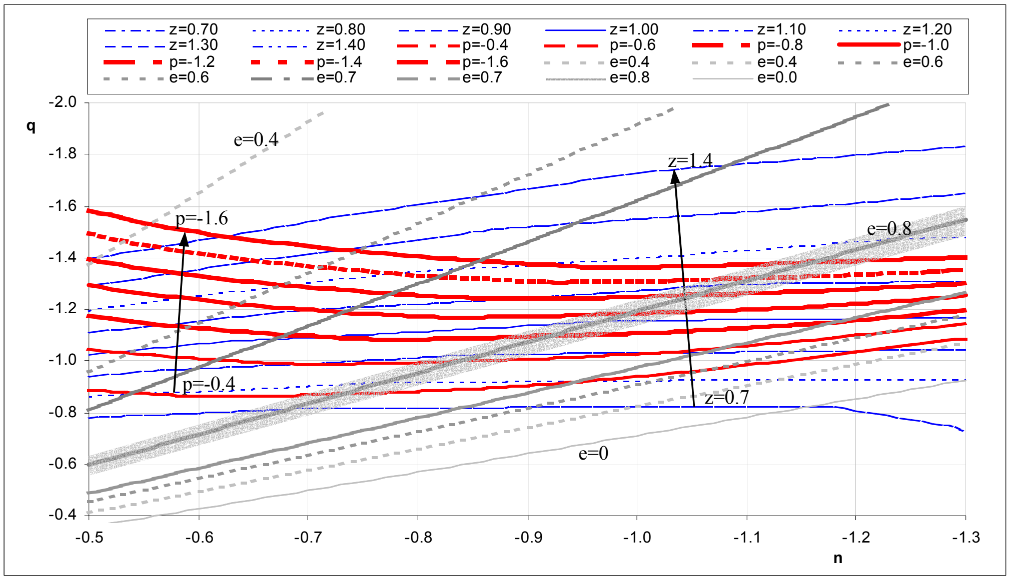

Figure 8. Characteristic curves of equal relative difference topographic water level (z), power (p), and efficiency (e), for different values of relative discharge, q, and rotating speed, n.

In this way, the machine will operate over the hydraulic system characteristic curve (z), in the point of interception with output demand curve (p). The developed analysis allows concluding the following important remarks:

- For a given z a small output power variation will provoke large variations in the rotating speed value. Thus, the machine shall have a frequency transformer to avoid producing a fix frequency power.
- In order to guarantee the system stability, the machine shall work with the maximum power that must be totally used or dissipated.
- A pump, with Ns = 230 (m, kW) when operating as turbine in maximum output power condition, presents a relative efficiency e ≈ 80% for any z value.

DYNAMIC EFFECTS

During operational turbine conditions, transients caused by a suddenly full-load rejection (i.e. due to a sudden power failure) can be very severs, and the pipeline should be designed to withstand positive and negative pressures. Following a power failure, the runner speed increases, since the pump/turbine inertia is small [5 and 7]. Depending on the runner type (e.g. characterized by low, medium or high specific speed), when the speed increases the flow changes, producing pressure variations along the hydraulic grade line. For low specific speed, when the runner speed increases until reaching the runaway speed, the flow through the machine will reduce rapidly, producing a choking effect, with successive positive and negative pressure waves propagating in the discharge line to upstream and downstream of the flow direction, respectively. Special concerns must be given to the pipeline profile, such that the transient hydraulic grade line never falls below the pipeline, in order to avoid vacuum pressure and water column separation. During design stages, this transient phenomenon should be analyzed, such as mitigation measures should be taken, considering specific controlling devices [7].

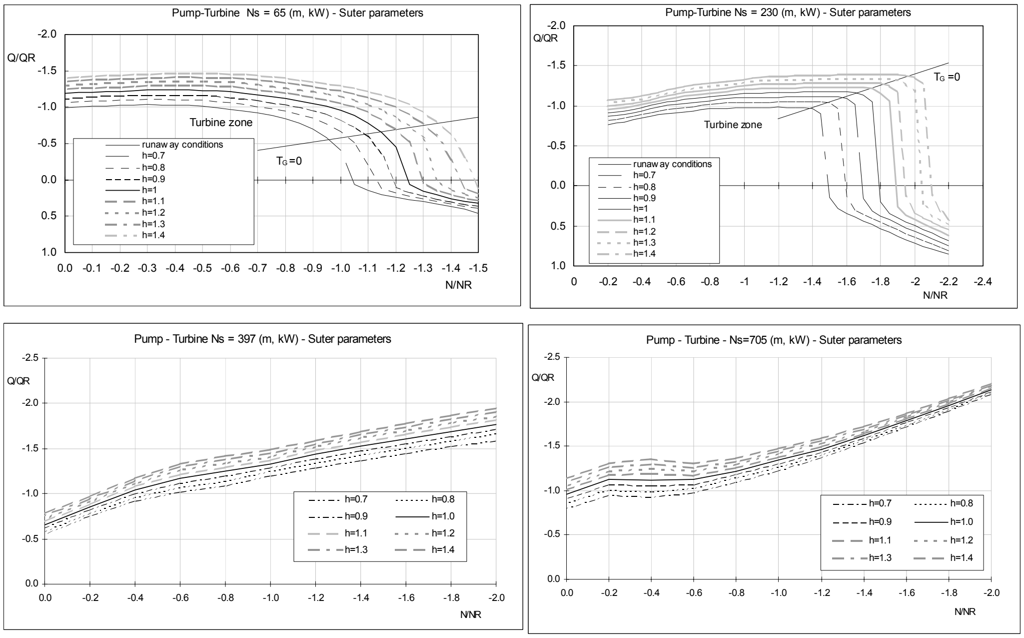

Figure 9. Analysis of discharge variation with overspeed of a pump in turbine zone

Both pumps and turbines dynamic behavior information are restricted available. Therefore, based on data obtained during steady-state manufacturer tests, important information can be extrapolated and utilized for some specific transient state conditions. This analysis requires a mathematical transformation of available data of pumps (based on Suter parameters – Figure 2) into characteristic curves of discharge variation, Q/Q_R, with the rotating speed, N/N_R (Figure 9). This procedure makes easy understanding the dynamic pump/turbine behavior on waterhammer effects.

Figure 9 shows the influence of the specific runner pump speed on the discharge variation when the pump operates in turbine zone, whenever the runner is submitted to speed variations at a constant head (being h defined according to equation (1)).

A good estimation for discharge variation with rotating runner speed was obtained through Euler application to a control volume, defined between inlet and outlet of a turbine runner, that was validated based on reaction turbines lab tests, according to the following equation:

$$\frac{Q}{Q_R} = \frac{A}{N/N_R} + B\,.\,N/N_R \qquad (20)$$

in which A and B are parameters that depend on the type of the runner (N_s) (e.g. A is always positive, and B<0 for low specific speed and B>0 for high specific speed – Table 2).

The application of equation (21) allows concluding that for reaction turbines with specific runner speeds between 80 and 350 (m, kW) B is negative and conversely for specific speed uper than 350 (m, kW) B is positive. Parameter B is the main factor responsible for discharge reduction or increasing and for all runner types A is always positive (being A>1 for low specific speed and A<1 for high specific speed). There is a notorious tendency to diminish A value, while specific speed increases, excepting for the pump with N_s = 230 (m, kW), which has characteristic curves with a peculiar shape, which can not be extrapolated for turbines (Figure 9) when comparing with N_s = 65 and 397 (m, kW).

Table 2. Estimation of parameters A and B, from pump curves when operating in turbine zone and under runaway conditions

N_s (m, kW) pump	N_{sT} (m, kW) turbine	A	B
65	73	4.52	-2.90
230	257	18.80	-5.80
397	444	0.66	0.71
705	788	0.19	1.08

The unbalanced torque between turbine (H) and generator (G) changes according to the general angular momentum equation for rotating masses, through the following equation:

$$T_H - T_G = I\frac{d\omega}{dt} = I\frac{2\pi}{60}\frac{dN}{dt} \qquad (21)$$

in which T_H is the net hydraulic turbine torque, T_G is the electromagnetic resistant torque, I is the total polar moment of rotational mass inertia ($I = WD^2 / 4g$), ω is the angular velocity of rotating masses and N is the rotating speed of the runner.

After a full load rejection the electromagnetic resistant torque, T_G (or b of equation (1)), can be set equal to zero. According to equation (21), the polar moment has a significant influence on the speed variation of rotating masses. For low inertia units (e.g. associated to small or micro hydro turbines) the runner speed increases rapidly till attaining runaway conditions. Analyzing Suter curves presented in Figure 2, the runaway conditions correspond to WT = 0. According to the low specific pump speed N_s = 65 r.p.m. (m, kW) (or 24 r.p.m.(m, m^3/s)) θ will be 240º and WH = 0.57. For these conditions, the relationship between the discharge and the rotating speed, for any head value, can be obtained by:

$$\tan\theta = \tan 240^{\circ} = 1.73 = \frac{n}{q} \quad \Rightarrow \quad n = 1.73\,q \qquad (22)$$

based on WH relationship presented on equation (3):

$$(1.73\,q)^2 + q^2 = \frac{h}{WH} = \frac{h}{0.57} \qquad (23)$$

that yields

$$\begin{aligned} q &= 0.66\sqrt{h} \\ n &= 1.14\sqrt{h} \end{aligned} \qquad (24)$$

In a similar way, and now, for N_s = 230 r.p.m. (m, kW) (or 85 r.p.m. (m, m3/s)) it is obtained the following relationships:

$$\begin{aligned} q &= 1.17\sqrt{h} \\ n &= 1.67\sqrt{h} \end{aligned} \qquad (25)$$

Equations (24) and (25) are for two available data of low specific pump speeds, in which is evident the discharge reduction whenever the rotating speed increases (for runaway and dissipation conditions). This phenomenon shall be considered in all hydropower analysis, being the main factor of design purposes due to small inertia and runner shape of these machines that is a particular characteristic of low specific speeds.

CONCLUSIONS

The feasibility of pumps operating as turbines seems to be proved, should some control stability systems were performed. Through developed analysis it is possible to conclude that the use of pumps as turbines allows obtaining a maximum relative efficiency up to 80%, depending on the type of the runner. The dynamic behavior of those machines are quite similar to the reaction turbines, in what concerns the flow variation due to runner type, generally characterized by its specific speed (N_{sT}).

These type of solutions can be adopted at upstream of energy dissipation devices in long conveyance gravity systems with excess available energy at some profile sections. The main idea subjacent of this analysis consists in obtaining a more economic solution to recover part of dissipated energy, in point of view of reduced dimensions imposed for turbo-machine equipment. On the other hand, the use of reverse pumps in drinking water or sewage systems, in some cases, can be an advantageous solution, because can take the advantage of the available head, whenever the flow reverses. Nevertheless, this can be an alternative in systems whenever the circulation due to treatment purposes or environmental constrains are inevitable. In what concerns the isolated operation, the use of synchronous motors can avoid problems of instabilities and/or the necessity of regulation, in order to produce a fixed electric frequency. Nevertheless, there are other economic alternatives to synchronous motors, such as alternators or direct current motors.

These schemes are particularly adequate to produce power from excess energy conditions, or in isolated rural communities, whenever are available natural or artificial heads, which turbine discharge are too small to make economically the installation of real turbines. Imposing the maximum output power condition (no fix the rotating speed) it leads to obtain a solution with relative efficiency up to 80% independently of the hydraulic system (difference topographic water levels) under certain valid limits.

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