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PIP

The "Efficiency Transient Control" Concept

for

Optimal Load Control in Kaplan Turbines

Hydroelectric Installations[1]

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Abstract

In the case of Kaplan turbines the consideration of the efficiency parameter is critical from the control strategy synthesis viewpoint. As it is not essential in hydro system analysis, explicit efficiency parameter relationships have not been deployed to date for linearized Kaplan mathematical models used in control design.

The 'efficiency transient control' concept is developed herein. Basically, this approach purports to optimize transient behavior by controlling so as to decrease efficiency during the early transient portion of load rejection and likewise maximizing efficiency for the load acceptance case. For each perturbation this method leads to minimal flow change and, therefore, minimal over- or under-pressures. As applied to a dominant gate with blade follow-up control scheme, the 'efficiency transient control' concept prescribes the appropriate delay to be employed for optimum performance within the constraint of a follow-up device. Where the early transient deviates from best efficiency operation, a return to steady-state or "cam" operation occurs as the follow-up.

The standard equations describing Kaplan turbine operation in linearized form by partial derivatives have been revised to include the efficiency parameter explicitly and now present a comprehensive and intelligible approach to Kaplan turbine control synthesis.

The effect of the 'efficiency transient control' upon system stability is analyzed and recommendations are made for desirable deployment. Application to nonlinear system operation is discussed.

The 'efficiency transient control' is compared with present day control schemes to assess their effectiveness. It is determined that a delay imposed upon the blades for the load-on case is suboptimal performancewise. The one-to-one gate-to-blade control is optimal in that case.

Introduction

With growing national concern for energy conservation and the advent and increased utility of low head installations, the Kaplan turbine with its versatility as a reasonably constant efficiency generator is becoming increasingly important in the U.S.A. as well as throughout the world. This turbine further boasts ease of adaptation to changes in reservoir level. Many studies have been carried out to date and much effort has been expended in the areas of modeling, simulation, analysis, and control synthesis for optimization of performance of hydroelectric system components and composites. The work has been directed to the general control problem as well as specifically to the Kaplan installation. Adequate representation of turbine characteristics has been a topic of continuing interest (1)[2], (2), (3) and recognized as a critically important aspect in system modeling. The notable works of Ransford, Arnaud, and Rottner (4), (5) correlate Kaplan turbine efficiency with optimal frequency regulation taking account of the grid inherent stability of the interconnected network and assuming fixed blade setting. Eilken and Lein (6) consider efficiency in their analysis of the conventionally controlled Kaplan turbine system. Borel (7), (8) analyzes the speed control process in Kaplan turbines. The theoretical efforts have been complemented by field observations of transient behavior for dynamic response and stability analysis (9). The work presented here purports to synthesize the optimal Kaplan turbine control program by consideration of the efficiency characteristic and within the constraint of a gate-dominant with blade follow-up servo system.

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[2] Underlined numbers in parentheses designate References at the end of the paper.

Analysis[3]

1. The Kaplan Turbine Control

Kaplan turbine characteristics relate the functional dependence among speed, torque, flow, pressure, efficiency, or perhaps power, and the controls. Fig. 1 exhibits a typical set giving turbine output power as a function of gate opening and blade position at constant speed and pressure. The line for steady-state operation has been superimposed on this plot and operation on the line represents the maximum efficiency condition. This figure is particularly useful for depicting the gate-dominant with blade follow-up type of control. The transfer function for this often used control mode is given in block diagram form in Fig. 2. The cam for scheduling steady-state operation is included, and the system neglects nonlinearities caused by valves and servos. The various system components are as follows:

1. Proportional plus Integral Speed Controller - relating nondimensionalized gate servo position (z) to nondimensionalized speed deviation (n),

   $z = (a + b/D)\ n$, with 'a' and 'b' as the proportional and integral governor gain settings respectively. (1

2. Blade Follow-up Cam - relating the steady-state position of blades (β) to gate position for best efficiency,

   $\beta_D = C\ (z)$, with 'C' the cam function and β and z nondimensionalized. (2

3. Follow-up Delay - permitting a dominant gate control during transient followed by a slow shift to the steady-state schedule,

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[3]See List of Symbols at end of paper.

$\beta = \left[\alpha/(\alpha + D)\right] \beta_D$, with the delay $1/\alpha$ (3

between actual and desired blade position, β_D.

4. Kaplan Turbine - relating flow (q) and torque (m) to speed (n) and pressure (h) and the controls z and β,

$m = f_1\ (n, h, z, \beta)$ and (4

$q = f_2\ (n, h, z, \beta)$ with f_1 and f_2 (5

to be found on a linearized basis later in the analysis.

The control strategy is depicted in Fig. 3 which is an enlargement of Fig. 1. On this figure path 1) represents the case with a long delay $1/\alpha$. Here a change in load demand, an increase from operation at level 'A' to level 'B', is satisfied mainly by gate action during the early transient. Then, in the follow-up period, as the blades' contribution to output is realized, the gate retreats to the new steady-state position from an originally large overshoot.

Path 3) represents quick blade follow-up with the delay parameter $1/\alpha$ compatible with gates' operational speeds. In this case the operating path is prescribed by the cam and steady-state line throughout the transient rather than just at the end points as was true with path 1). Intermediate values of $1/\alpha$ will result in paths of type 2). In the following paragraphs the linearized turbine equations corresponding to Eqs. 4) and 5) are presented explicitly and are employed for the proposition of a rational procedure for optimal selection of the follow-up delay factor $1/\alpha$.

2. Kaplan Turbine Characteristics and Mathematical Modeling

Though the manner in which Kaplan characteristics are presented in Fig. 1 is convenient for control mode analysis and appreciation, not sufficient information is present for complete determination of the turbine operation states.

For example, flow information is not available there. A more appropriate layout of characteristics is given in Fig. 4. Here the speed function (N) and the moment function (M) of flow (Q) are given for variable gate. The plot is for constant blade which signifies that a set of these plots, each for a different but constant blade setting, is required. The unity head condition is typical for model characteristics and is accommodated through the similitude relations. (See Appendix A.) The choice of independent variables is inconsequential from the viewpoint of definition of the characteristics' "solid" (consider several blade plots). Hence moment (M) and flow (Q) will be made to depend upon speed (N), gate position (Z) and blade position (B). Having chosen a reference point of operation, the appropriate partial derivatives are computed directly from Fig. 4 and Eqs. 4) and 5) may be written explicitly for the unity head condition,

$$m = (\partial m/\partial n)\ n + (\partial m/\partial z)\ z + (\partial m/\partial \beta)\ \beta \qquad (6$$

$$q = (\partial q/\partial n)\ n + (\partial q/\partial z)\ z + (\partial q/\partial \beta)\ \beta \text{ , where lower} \qquad (7$$

case letters denote the nondimensionalized deviation from reference of the associated quantities. By taking pressure into account through the similitude relations for the stepup from model to prototype, the turbine description becomes, (See Appendix A.)

$$m = (\partial m/\partial n)\ n + (\partial m/\partial z)\ z + (\partial m/\partial \beta)\ \beta + \left[M'/M_R - (1/2)\ (\partial m/\partial n)\right] h \qquad (8$$

$$q = (\partial q/\partial n)\ n + (\partial q/\partial z)\ z + (\partial q/\partial \beta)\ \beta + 1/2 \left[Q'/Q_R - (\partial q/\partial n)\right] h \qquad (9$$

where the subscript (R) denotes the point of reference while the prime(') denotes the point of operation about which linearization is made. For coincidence of operation with reference and assuming standard partials,

$$\partial m/\partial n = -1 \qquad (10$$

$$\partial q/\partial n = 0 \qquad (11$$

the Eqs. 8) and 9) reduce to

$$m = (\partial m/\partial z)\ z + (\partial m/\partial \beta)\ \beta - n + (3/2)\ h \qquad (12$$

$$q = (\partial q/\partial z)\ z + (\partial q/\partial \beta)\ \beta + (1/2)\ h \qquad (13$$

The simplification is justified inasmuch as the analysis concentrates on the effect of the controls z and β. For the sake of completeness the conduit equation is included as

$-h = T_w Dq$ where the nondimensionalized flow (14
derivative (Dq) is related to pressure (h) through the water time constant T_w. And finally,

$m - \ell = T_m Dn$ relates system acceleration (Dn) to the (15
net torque, developed (m) minus load (ℓ), through the rotor time constant, T_m.

3. Introduction of the Efficiency Parameter

The equations given in the preceeding paragraphs are sufficient for modeling the system for simulation and analysis. Indeed, they represent a natural approach to the control analyst. Unfortunately, an important parameter, efficiency (E) is not explicitly present. Clearly efficiency is a factor in the flow parameter, but even there it is implicit within the characteristic functions of Fig. 4. The efficiency parameter will be introduced explicitly in the linearized turbine model in the next paragraph. At this time the criterion for selection of appropriate delay $1/\alpha$ should be examined. Following load demand change there exists a torque imbalance, $m - \ell$. This fact suggests controlling Eq. 12) to null the imbalance as quickly as possible. However, too fast a control action would cause a quick change in flow through Eq. 13) and thus overpressure, Eq. 14). Can the delay parameter be used effectively for optimizing generated torque deviations subject to pressure constraints?

Consider net turbine output (P). This quantity may be evaluated in terms of speed (N) and net torque (M) or in terms of pressure (H) and flow (Q) with appropriate efficiency (E). Thus,

$$P = MN = HQE \tag{16}$$

With given assumptions (Appendix B) the above equation linearizes to,

$m = q + \eta$, relating the previously (17

defined torque (m) and flow (q) nondimensionalized deviations to that newly defined for efficiency (η). Repeating Eq. 13) and substituting Eq. 17) into Eq. 12) the resulting form is,

$$m = (\partial q/\partial z)\ z + (\partial q/\partial \beta)\ \beta + (\partial \eta/\partial z)\ z + (\partial \eta/\partial \beta)\ \beta - n + (3/2)\ h \tag{18}$$

$$q = (\partial q/\partial z)\ z + (\partial q/\partial \beta)\ \beta + (1/2)\ h \tag{19}$$

4. <u>The 'Efficiency Transient Control' Concept</u>

These last equations contain the efficiency partial derivatives explicitly thus permitting insight as to selection of the optimal control strategy with relation to choice of the follow-up delay $1/\alpha$. Indeed for the case of load rejection $1/\alpha$ should be chosen so as to minimize $\partial\eta/\partial z$ and $\partial\eta/\partial\beta$. Thus a torque decrease is achieved by a combination of a minimum flow decrease and a maximum efficiency decrease. On the contrary, load acceptance is controlled so as to maximize $\partial\eta/\partial z$ and $\partial\eta/\partial\beta$. Here a minimum flow increase is accomplished through a maximum efficiency increase. In all cases the follow-up terminates on the steady-state line thus guaranteeing an eventual return to good operating efficiency.

5. <u>Application of the Efficiency Transient Control to Small Load Perturbations</u>

In the remaining discussion the 'efficiency transient control' concept presented above will be applied to a typical set of Kaplan characteristics and discussed for various operative modes. As efficiency is now of prime interest for transient control the form in which turbine characteristics are presented should include that parameter. Fig. 5 is an especially useful display. Here the model efficiency is plotted as a function of power output with gate and blade settings as variable parameters. The line joining the peaks of the various constant blade curves represents the follow-up cam schedule or steady-state operation. To facilitate control synthesis an enlargement of Fig. 5 near reference is shown in Fig. 6. The load rejection case will be considered first and then the load acceptance.

Consider steady-state operation at level A followed by a step load-off demand of magnitude A - B. Paths 1), 2), and 3) on Fig. 6 represent possible control modes dependent upon the follow-up delay $1/\alpha$. Path 1) represents a large delay with operation following a constant blade path during the early portion of the transient. Eventually, blade action will shift the operation to the new power level on the steady-state curve. On the other hand, if blading control is sufficiently responsive, the transient occurs with controls following the steady-state line. This is path 3) and an intermediate value of delay is represented by path 2). And these paths are identical to the three paths of Fig. 3. However, on the latter plot the appropriate control strategy is evident in view of prior analysis. Path 1) presents the optimal choice in that early transient efficiency decreases most. Thus flow change will be minimal and likewise overpressure. For the case of load acceptance, operation at level A followed by a demand for a C - A increase in power, the 'efficiency transient control' concept selects path 5) in Fig. 6 as optimal over path 4) and all other intermediate paths. Thus for load-on cases a one-to-one blade-to-gate control along the best efficiency line assures minimum flow increase and least under-pressure. It should be noted that this analysis is not valid while operating within the "maximum output" region of Fig. 5 which is beyond the best efficiency line.

6. A Note on System Stability

The effect on system stability resulting from deployment of the 'efficiency transient control' is examined here. Pertinent equations are Eqs. 1), 2), 3), 12), and 13). For small perturbations Eq. 2) may be linearized to,

$$\beta_D = C\,z, \qquad \text{which results in} \qquad (20$$

$$\beta = \left[\alpha/\left(\alpha + D\right)\right] C\,z \qquad \text{when substituted} \qquad (21$$

into Eq. 3). Equation 21) is now substituted into Eqs. 12) and 13),

$$m = (\partial m/\partial z)z + C\,(\partial m/\partial\beta)\left[\alpha/(\alpha + D)\right]z - n + (3/2)\,h \qquad (22$$

$$q = (\partial q/\partial z)\,z + C\,(\partial q/\partial\beta)\left[\alpha/(\alpha + D)\right]z + (1/2)\,h \qquad (23$$

It is concluded that stability will be affected to the extent of the delay operator. For the load rejection case, the long follow-up delay, ($\alpha \to 0$), nulls the blade contribution to turbine gains, rendering effective gains $\partial m/\partial z$ and $\partial q/\partial z$ respectively. For the load-on case, the quick follow-up ($\alpha \to \infty$) results in total gains ($\partial m/\partial z + C\,\partial m/\partial\beta$) and ($\partial q/\partial z + C\,\partial q/\partial\beta$) respectively. It is clear that this gain differential directly affects system stability. The proper design procedure will then be to adjust the governor constants 'a' and 'b' to differing values depending upon the direction of load change, with the load-off settings being the same as those for steady-state operation.

7. Application of the Efficiency Transient Control to Large Load Perturbations

Where changes in load demand exceed some predetermined value (approximately 10%), governor Eq. 1) is no longer valid and operation shifts to a non-linear mode. Here gate control speed is held at some feasible value. The phenomenon is one of gate speed saturation. With a sufficiently large transient the blade velocity will likewise saturate. For this type of operation the follow-up delay $1/\alpha$ is not an effective parameter because of the relatively long transient time. The more reasonable parameter to assess is that of blade timing. Blade timing refers to the time required for a full blade excursion and is the inverse of relative blade travel rate during saturation. The control problem here is to determine the appropriate blade timing given a specified gate timing, or determination of a timing ratio. The answer is readily available in consideration of the 'efficiency transient control' concept. For the case of load rejection a quick efficiency drop is desired. Fig. 5 indicates that a slow blade timing or high ratio will take operation along a constant blade curve to achieve optimal results within the constraint of the follow-up device. Clearly, the load-on case demands responsive blade action with a one-to-one blade-to-gate schedule for optimal performance.

8. Evaluation of Present Day Kaplan Control Deployment

In concluding it is of interest to evaluate present day control practice in view of the 'efficiency transient control' method. The accepted standard controller for Kaplan turbines today is the gate direct control governor with a slower rate blade follow-up. This approach agrees with the efficiency control prediction for the case of load rejection. It is clear that any operation of control other than along the cam schedule will decrease efficiency and better response. For the load-on case delayed blading operation is suboptimal. Some attempts have been made at speeding up blade action for load acceptance. Any delay whatsoever will detract from system performance, and the operation should be carried out along the best efficiency line within the limitation of the servo installation. The standard control at low power output levels is a gate control at locked blade. Interestingly enough this type of operation is quasi-optimal for both load-on and load-off cases. Constant blade setting curves exhibit high slope on the efficiency versus output power plane (reference Fig. 5). This fact results in a power demand change being accommodated to a great extent through change in efficiency with a minimum flow change and an accompanying minimum pressure peak. Finally, for operation within the maximum output region of Fig. 5, blocked blade operation will result in poor performance. If it is essential that control be exercised within the region, and often it is not, a better approach can be achieved by employing an additional cam specifically designed for the operation.

Conclusion

Currently used methods for Kaplan turbine hydro system analysis are adequate for system modeling but should be improved for utilization in control synthesis. The turbine efficiency parameter is often buried within flow information and not expressed explicitly. Efficiency can be made to appear explicitly by modeling the turbine linearly in terms of the nondimensionalized deviations of its parameters as given in Eqs. 18) and 19). Examination of these equations indicates that it may be feasible to 'efficiency transient control' for best system performance. On a gate primary control with blade follow-up, the 'efficiency transient control' concept can be used to select follow-up time delay $1/\alpha$ for optimal response. This concept proposes that operation be momentarily shifted to lower efficiency during rejection and be held at maximum efficiency during load acceptance. In either case, flow change and thus pressure peaks will be minimal. For the rejection case, return to peak efficiency is effected as the follow-up. For the load-on case the controls remain on the steady-state line throughout the transients. It is noted that basically the same control approach is predicted for optimal performance for nonlinear system operation following large power demand perturbations. Here, the relative blade timing is of major significance as opposed to the follow-up delay for linear operation.

Analysis of system stability shows a shift in turbine gain depending upon the follow-up delay value. This shift can be completely compensated for by appropriate choice of governor gains.

As for present day practice, the gate-dominant with slower follow-up blade timing controller is optimal for the load rejection operation and for operation at low power levels for both load-on and load-off. At intermediate-to-high power outputs a control schedule that permits dephasing of gate-to-blade for load acceptance transients is performancewise suboptimal. The rational control strategy will permit follow-up delays for load rejection while implementing a quick and responsive blade action for load acceptance. This control scheme will be optimum within the constraints of the blade follow-up approach and of a linearized analysis. The 'efficiency transient control'

LIST OF SYMBOLS

N	Turbine speed of rotation
P	Net power output
M	Net torque output
Q	Flow
H	Pressure
Z	Gates position
B	Blades position
E	Turbine efficiecny
Δ M Δ Q etc.	Change in torque, flow, etc.
ℓ	Nondimensionalized load demand
m	Nondimensionalized torque output
n	Nondimensionalized speed deviation
q	Nondimensionalized flow deviation
h	Nondimensionalized pressure deviation
z	Nondimensionalized gates deviation
β	Nondimensionalized blades deviation
η	Nondimensionalized efficiency deviation
D	Derivative operator
∂	Partial deviative operator
a	Proportional gain of governor
b	Integral gain of governor
C	Blades follow-up cam function
$1/\alpha$	Blades position servo delay
T_w	Conduit total water time constant
T_m	Rotor total time constant

concept is applicable for the evaluation of control in the case of both linear and nonlinear operation.

APPENDIX A

The constant head, static turbine characteristics are described linearly as follows:

$$q_m = (\partial q/\partial n)\ n_m + (\partial q/\partial z)\ z + (\partial q/\partial \beta)\ \beta \tag{24}$$

$$m_m = (\partial m/\partial n)\ n_m + (\partial m/\partial z)\ z + (\partial m/\partial \beta)\ \beta, \text{ with the m subscript} \tag{25}$$

identifying model quantities. Corresponding prototype relations exhibiting head dependence are established thru consideration of the similitude relations with the subscripted (R) denoting reference conditions and the prime (') denoting the point of operation:

A. Moment (valid for model at unity head)

$$M \sim M_m H \tag{26}$$

$$M = kM_m H \tag{27}$$

$$M/M_R = (kM_{mR}H_R/M_R)\ (M_m/M_{mR})\ (H/H_R) \tag{28}$$

$$(M_R' + \Delta M)/M_R = \left[(M_{mR}' + \Delta M_m)/M_{mR}\right]\left[(H_R' + \Delta H)/H_R\right] \tag{29}$$

$$\text{since, } kM_{mR}H_R/M_R \equiv 1 \tag{30}$$

then,

$$m = (M_{mR}'/M_{mR})\ h + m_m, \tag{31}$$

$$\text{considering } H_R' = H_R. \tag{32}$$

Likewise for speed and flow,

B. Speed

$$n_m = n - h/2 \tag{33}$$

C. Flow

$$q_m = q - 1/2\ (Q_R'/Q_R)\ h \tag{34}$$

Elimination of q_m, m_m and n_m from Eqs. 24) and 25) thru Eqs. 31), 33), and 34) establishes Eqs. 8) and 9).

APPENDIX B

Equation 16) relates net power output (P) to torque (M), speed (N), pressure (H), flow (Q), and efficiency (E). Repeated for convenience, it is:

$$P = MN = HQE \qquad (35$$

This expression in linearized in terms of the nondimensionalized deviations as follows:

$$(M_R + \Delta M)\ (N_R + \Delta N) = (H_R + \Delta H)\ (Q_R + \Delta Q)\ (E_R + \Delta E), \qquad (36$$

and,

$$M_R N_R\ (\Delta M/M_R + \Delta N/N_R) = H_R Q_R E_R\ (\Delta H/H_R + \Delta Q/Q_R + \Delta E/E_R), \qquad (37$$

or,

$$m = h + q + \eta - n \qquad (38$$

Eq. 38) is used to substitute into the $\partial m/\partial z$ term of Eq. 12). Hence, the h and n terms of Eq. 38) need not be considered, and Eq. 17) results.

Acknowledgements

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LIST OF FIGURES

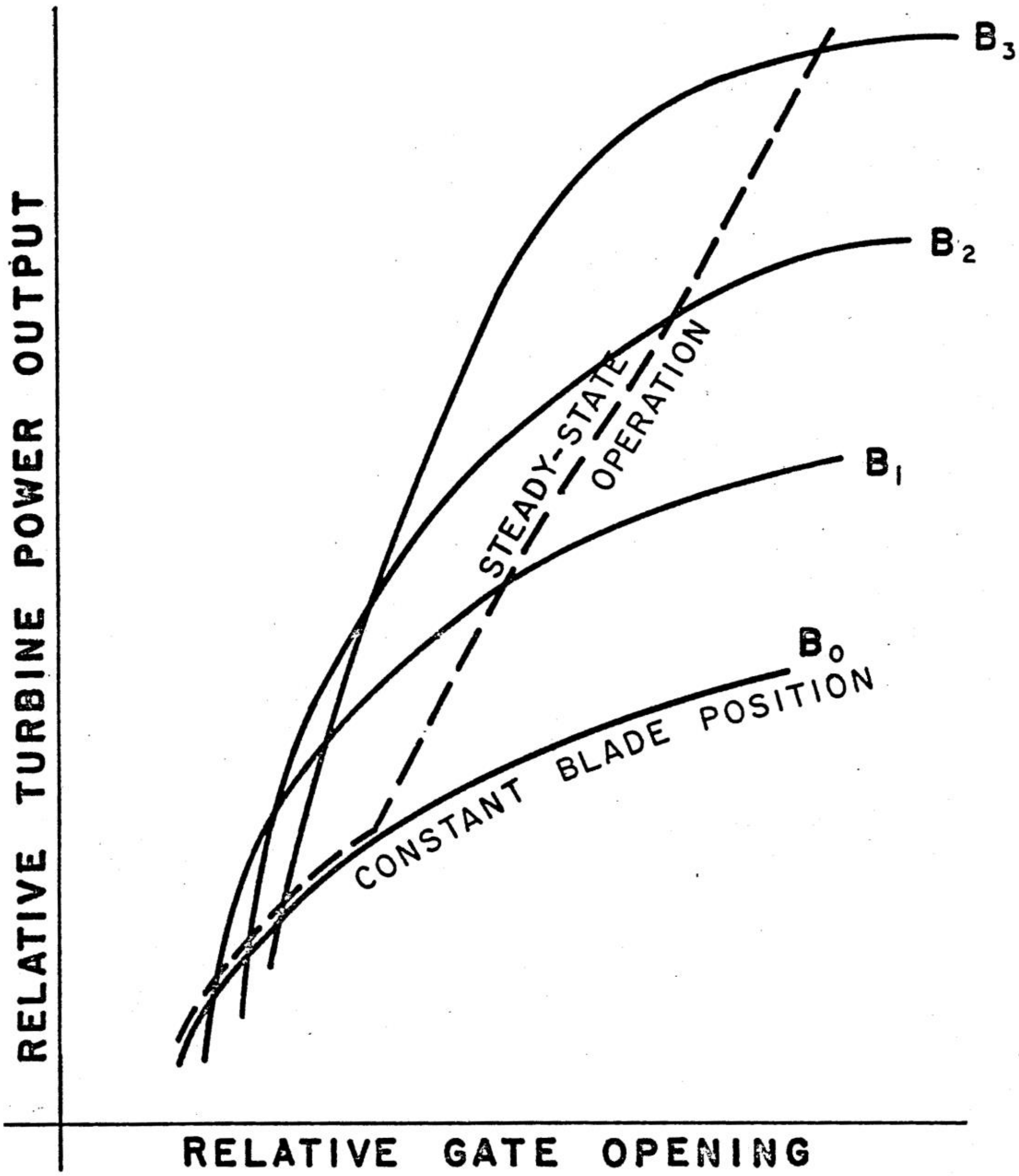

FIG.1 MODEL KAPLAN TURBINE POWER CHARACTERISTIC AT CONSTANT SPEED AND PRESSURE

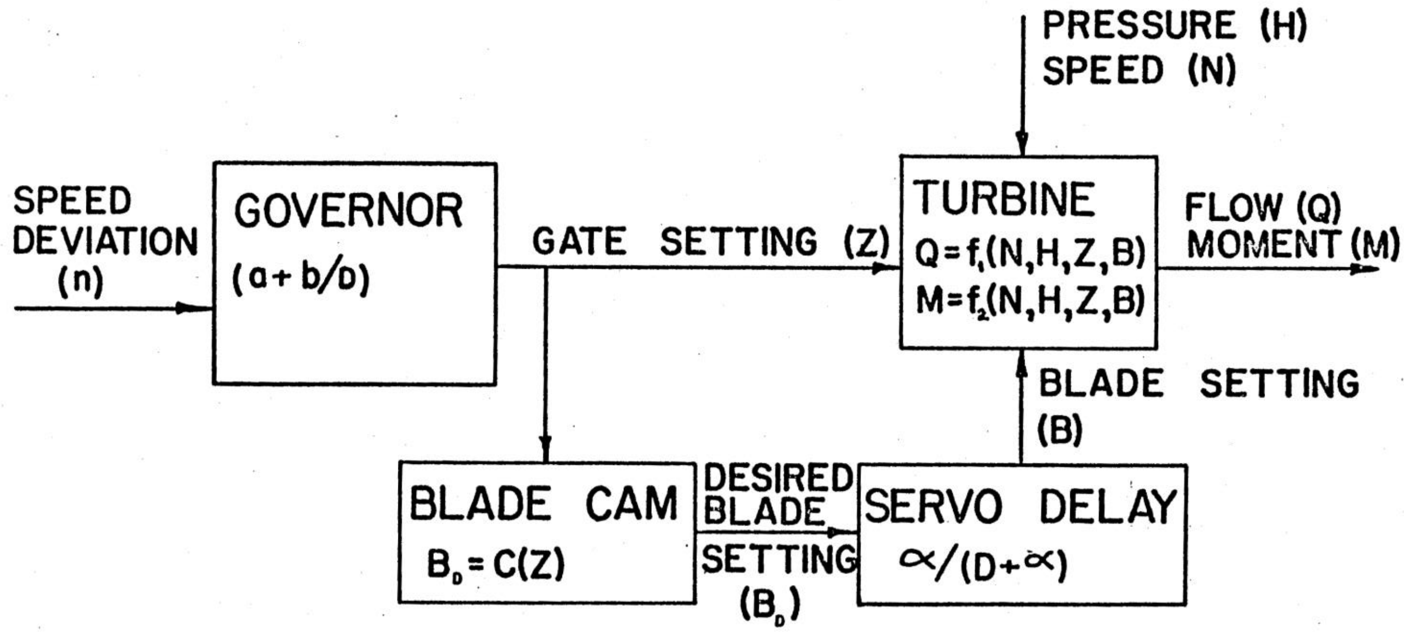

FIG. 2 TYPICAL FOLLOW-UP BLADE SETTING CONTROL FOR KAPLAN TURBINES.

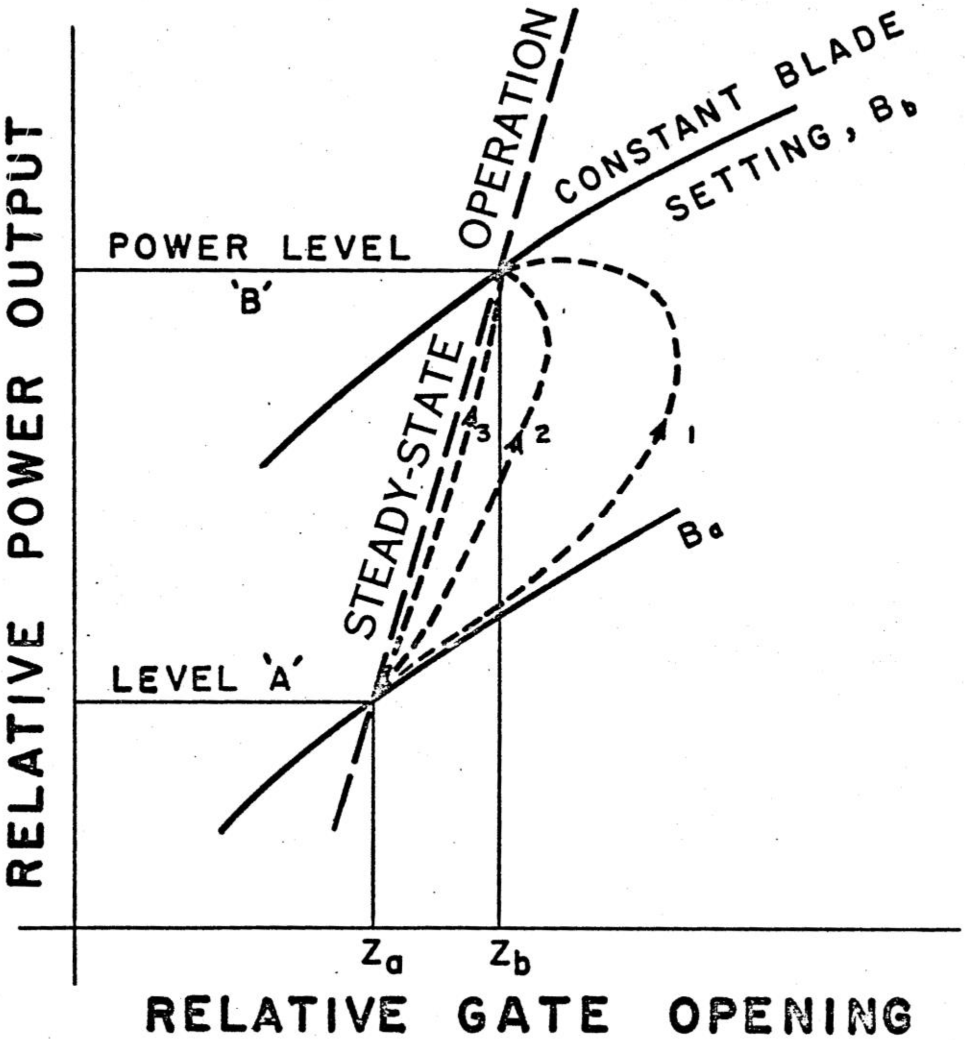

FIG. 3 KAPLAN TURBINE CONTROL STRATEGIES.

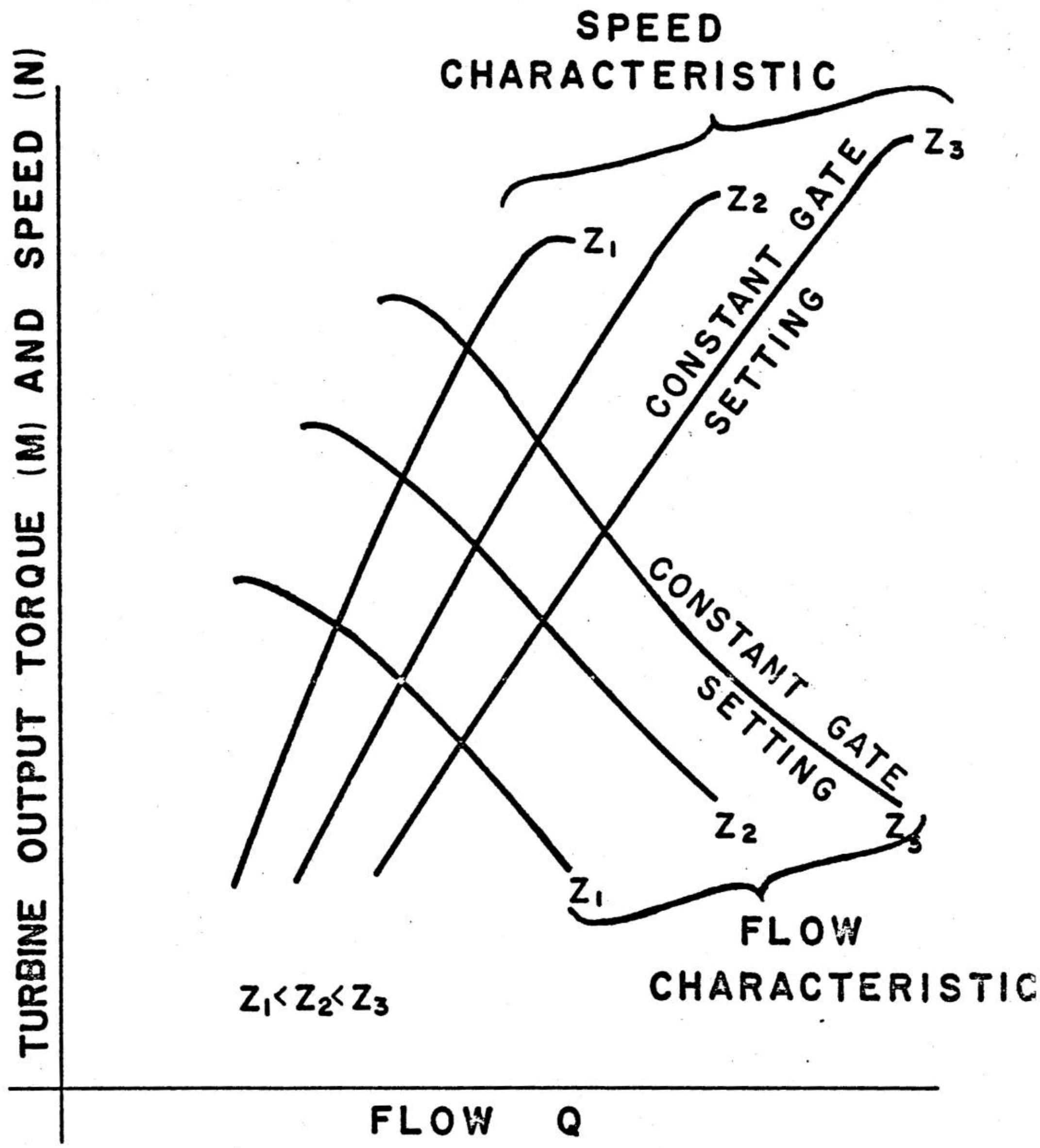

FIG. 4 SPEED AND MOMENT REPRESENTATION OF MODEL KAPLAN TURBINE DATA AT CONSTANT BLADE SETTING AND UNITY HEAD.

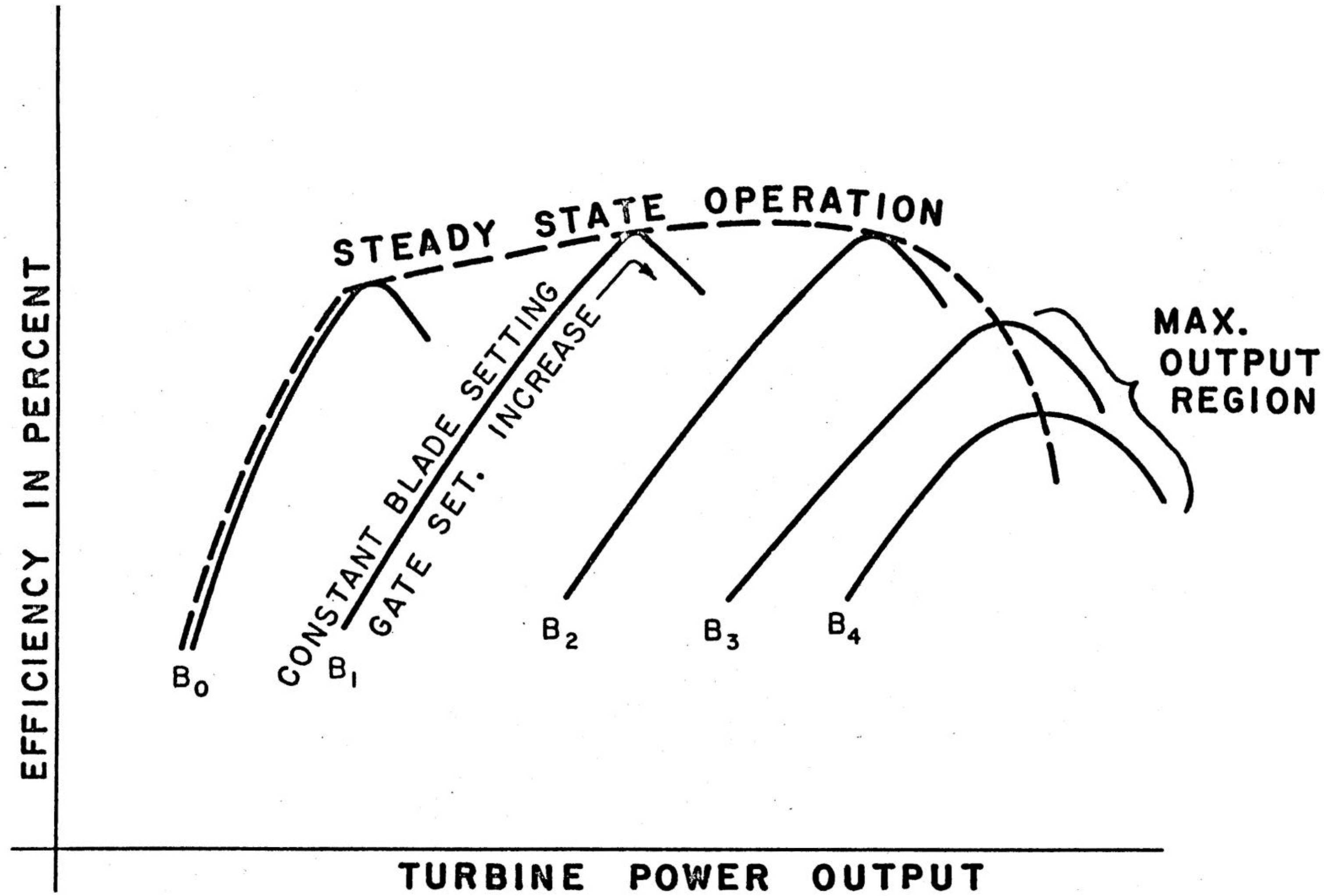

FIG. 5 MODEL KAPLAN TURBINE EFFICIENCY VERSUS POWER OUTPUT AT UNITY PRESSURE.

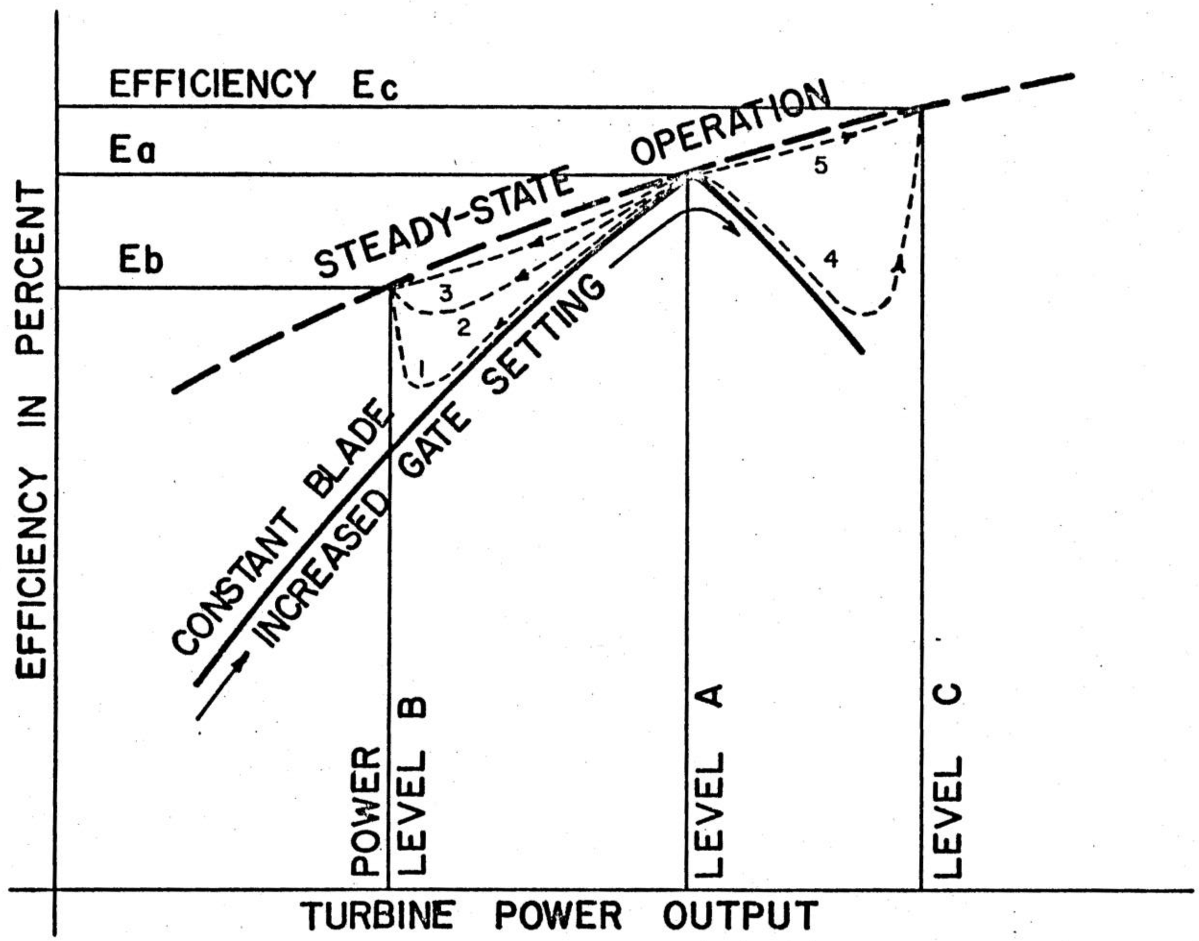

FIG. 6 KAPLAN TURBINE CONTROL MODE AS FUNCTION OF BLADE FOLLOW-UP DELAY.