TRANSIENT SIMULATION OF A HYDROPOWER PLANT

A DISSERTATION

Submitted in partial fulfillment of the requirements for the award of the degree
of
MASTER OF TECHNOLOGY
in
ALTERNATE HYDRO ENERGY SYSTEMS

By

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CANDIDATE'S DECLARATION

I hereby certify that the work which is presented in this dissertation entitled, **"TRANSIENT SIMULATION OF A HYDROPOWER PLANT"**, in partial fulfillment of the requirement for the award of the degree of **Master of Technology** in **"Alternate Hydro Energy Systems"**, submitted in **Alternate Hydro Energy Center, Indian Institute of Technology, Roorkee** is an authentic record of my own work carried out during the period from July, 2006 to June, 2007 under the supervisions of **Prof. J. D. Sharma**, Professor, **Department of Electrical Engineering** and **Shri. M.K. Singhal** , Sr. Scientist Officer, **Alternate Hydro Energy Center, Indian Institute of Technology, Roorkee.**

I have not submitted the matter embodied in the Seminar for award of any other degree.

Date: 29th June- 2007

Place: Roorkee

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ABSTRACT

The operation of a Hydropower plant subject to several transient phenomenon due to load rejection, group start-up and shut-down, modification of operating point, earth fault, out of phase synchronization during start-up, emergency stop and so on. In order to ensure the safety of the power plant and to optimize operation parameters, a simulation model of the power plant is required to investigate normal and all worst cases. The simulation of the dynamic behavior is usually performed separately for the hydraulic and electric part of the power plant allowing determining the set of parameters related to the security of each part.

In this work, transient analysis of hydraulic systems like, penstock, valves and Pelton turbine, of hydropower plant (HPP) has been done by using Method Of Characteristics (MOC).The programming for transient analysis in penstock, valve, and turbine during load rejection has been done in C++. The graph for discharge from reservoir ,valve, pressure head at the valve and turbine speed with respect to time during load rejection have been obtained and conclusion have been drawn for better control and stability of hydraulic system in HPP.

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NOMENCLATURE

H	:	Total head or energy grade [m]
Q	:	Discharge [m3/s]
x	:	Distance along the conduit [m]
t	:	Time [s]
g	:	Gravitational constant [m/s2]
A	:	Cross sectional area of the conduit [m2]
D	:	Diameter of the conduit [m]
f	:	Darcy-Weisbach friction factor
a	:	Celerity of a compression wave traveling through the penstock [m/s]]
n	:	Number of reaches for the pipe.
α	:	Tube inclination angle [Rd]
η_o	:	Overall efficiency
N	:	Speed of turbine [rpm]
C_d	:	Discharge coefficient
A_v	:	Area of the nozzle opening [m2]
H_0	:	Head upstream of the nozzle [m]
η_{gen}	:	Generator efficiency;
T_{tur}	:	Instantaneous turbine torque [n-m]
T_{gen}	:	Instantaneous generator torque[n-m]
ω	:	Rotational speed of turbine generator [Rd/s]
WR^2	:	Total moment of inertia of the turbine and generator [kg-m ^{2}]
P_{gen}	:	Generator Load [kW]
P_{tur}	:	Power developed by turbine [kW]
Ta	:	Actuator time constant
Tr	:	Dashpot time constant
σ	:	Permanent speed droop

INTRODUCTION & LITERATURE REVIEW

1.1 General

The transients simulation is one of the important tasks for the designing a hydropower plant. The operation of a hydroelectric power plant subject to several transient phenomenon due to group start-up and shut-down, modification of operating point, earth fault, out of phase synchronization during start-up, emergency stop and so on.. In order to ensure the safety of the power plant and to optimize operation parameters, a simulation model of the power plant is required to investigate normal and all the worst cases. The simulation of the dynamic behavior is usually performed separately for the hydraulic and electric part of the power plants allowing to determine the set of parameters related to the safety of each part.

Afterward control command parameters have to be calculated considering the operation stability. However, it requires a full model of the power plant taking into account the hydraulic, electric, mechanical and control device components.
Fluid distribution systems and hydropower plants can be severely damaged by water hammer. Water hammer is the forceful slam, bang, or shudder that occurs in pipes when a sudden change in fluid velocity creates a significant change in fluid pressure. Water hammer can destroy turbo machines and cause pipes and penstocks to rupture. Water hammer can be avoided by designing and operating these systems such that unfavorable changes in water velocity are minimized.

The digital simulation for the transient process is one of the important tasks for the designing of the power plants. In order to design the water inflow system, and select the proper measures to limit the speed rise and water pressure surge cause by water hammer, transient process simulation is a necessary step. The traditional methods for the transient process calculation are diagrammatic and approximate analytical methods that can produce the maximum values of turbine speed and water pressure rise. The modern method to solve the transient process is Lagrangian approach and Impedance method. The stability of the hydraulic system plays an important rule in the stability of Speed regulation systems for the hydroelectric generating units and even effects the stability of the whole power system. Therefore not only these two maximum values but the whole transient process must be understood clearly.

Digital simulation is a very effective way to study the hydraulic transient process and the coupled transient process of hydraulic, mechanic and electric systems.

1.2 Classification of Hydraulic Transients

The transients may be classified into three categories depending upon the conduit in which the transient conditions are occurring.

1. Transients in closed conduits
2. Transients in open channels
3. Combined free-surface-pressurized transient flow.

The analysis of transients in closed conduits may be further subdivided into two types: distributed systems and lumped systems. In the former case, the fluid is considered compressible, and the transient phenomenon occurs in the form of traveling waves, e.g. like in water–supply pipes, power plant conduits and gas-transmission lines. In the analysis of lumped systems, any change in the flow conditions is assumed to take place instantaneously throughout the fluid, i.e. the fluid is considered as a solid body. Like in case of slow oscillation of water level in a surge tank following a load change on the turbine. Mathematically, the transients in the distributed systems are represented by partial differential equations, whereas the transients in the lumped systems are described by ordinary differential equations. If $L\omega / a$ is much less than 1, then the system may be analyzed as a distributed system.

Transients in open channels may be divided into two types depending upon the rate at which they occur: gradually varied flow, such as flood waves in rivers and rapidly varied flow, such as surges in power canals.

1.3 Methods for Transient Analysis

We are using following methods for transient analysis in hydropower plant.

1. Lagrangian approach
2. Eulerian approach
3. Method of characteristics
4. Method of impedance

The characteristics method is the most suitable to solve transient problems because solution is to be obtained through transformations of the momentum and continuity partial differential equations in four total differential equations, and these equations are related through finite differences. In spite of this method to be most widely used, it is difficult to employ.

1.3.1 Lagrangian Approach.

The Lagrangian approach solves the transient flow equations in an event-oriented system-simulation environment. In this environment, the pressure wave propagation process driven by the distribution system activities. The wave characteristic method (WCM) is an example of such an approach (Wood et al, 2005; conditions and the pipe-resistance term, it is not possible to obtain a direct solution. When pipe junctions, pumps, surge tanks, air vessels, and other components that routinely need to be considered are included, the basic equations are further complicated and it is necessary to utilize numerical techniques. Accurate transient analysis of large pipe networks requires computationally efficient and accurate solution techniques. Both Eulerian and Lagrangian solution schemes are commonly used to approximate the solution of the governing equations. Eulerian methods update the hydraulic state of the system in fixed grid points as time is advanced in uniform increments. Lagrangian methods update the hydraulic state of the system at fixed or variable time intervals at times when a change actually occurs. Each approach assumes that a steady-state hydraulic equilibrium solution is available that gives initial flow and pressure distributions throughout the system.

1.3.2 Eulerian Approach.

Eulerian methods consist of the explicit method of characteristics (MOC), explicit and implicit finite difference techniques, and finite element Boulos et al, 2004) and was first described in the literature as the wave plan method (Wood et al, 1966). The method tracks the movement and transformation of pressure waves as they propagate throughout the system and computes new conditions either at fixed time intervals or at times when a change actually occurs (variable time intervals). The effect of line friction on a pressure wave is accounted for by modifying the pressure wave using a nonlinear characteristic relationship describing the corresponding pressure head change as a function of the line's flow rate. Although it is true that some approximation errors will be introduced using this approach, these errors can be minimized using a distributed-friction profile (piecewise linearized scheme) . However, this approach normally requires orders of magnitude fewer pressure and flow calculations, which allows very large systems to be solved in an expeditious manner, and has the additional advantage of using a simple physical model as the basis for its development. Because the WCM is continuous in both time and space, the method is also less sensitive to the structure of the network and to the length of the simulation process, resulting in improved computational efficiency. This technique produces solutions for a simple pipe system that are virtually identical to those obtained from exact solutions.

1.3.3 Method Of Characteristics

In the strategy used by the MOC, the governing partial differential equations are converted to ordinary differential equations and then to a different form for solution by a numerical method. The equations express the head and flow for small time steps (Δt) at numerous locations along the pipe sections. Calculations during the transient analysis must begin with a known initial steady state and boundary conditions. In other words, head and flow at time $t = 0$ will be known along with head and/or flows at the boundaries at all times. To handle the wave characteristics of the transient flow, head and flow values at time $t + \Delta t$ at interior locations are calculated making use of known values of head and flow at the previous time step at adjacent locations using the ordinary differential equations expressed in different form.

1.3.4 Method of impedance

The method of impedance is based on the theory of electrical networks and uses the friction term in its non-linear development for the mean velocity value and linearized expression of the friction for the oscillatory component .This method has the advantage of being easily used in the analysis of pipe systems. A short computational time is required to obtain the solution. The model is obtained through the analogy between fundamental equation and discrete circuit.

1.4 LITERATURE REVIEW

Souza O.H. Jr., N. Barbieri and A.H.M. Santos et al [4] analyzed the discrete hydraul systems with emphasis on analog mathematical model .The results were obtained by using nonlinear analog-digital simulation method.

Nielsen T.K. and F.O. Rasmusen et al [5] developed the simulation models of hydraul turbine regulating system in a MATLAB/Simulink-based software environment. These nonline characteristics of hydraulic turbine and the no-elastic water hammer effect of pressure wat supply conduit were considered in the modeling. Some simulation tests performed, such step speed disturbance test on no-load operation and step load disturbance test on isolat operation, which will cause small hydraulic transients in turbine hydraulic system.

Vournas C.D. and G. Papaionnou et al [6], carried out modeling and the stability analys of a hydroelectric power plant with two surge tanks placed one at upstream and other downstream of the turbine. The interaction of the two surge tanks is investigated and t system is simulated for the case of sudden opening or closure of the turbine gates. T stability of the frequency control loop is assessed by using small signal analysis based linearization. The above procedures applied to a real situation and proved to be extreme useful for preliminary design purposes.

Schmitt C.,G. Pluvinage et al [11] established a numerical model in order to simulate the propagation of pressure waves in water networks. The present model formulation is based on a system of partial hyperbolic differential equations. This system has been solved via the characteristics method. The current model provides the necessary data and the necessary damping of water hammer waves, taking into account the structure of the pipe network and the pressure loss. The numerical algorithm estimates the maximum pressure values resulting from the water hammer when closing valves in the network and consequently, the maximum stresses in the pipes have been calculated. In the case of simultaneous closing of several valves, the over pressure can exceed the admissible pressure. In this case, the severity of a defect such as a corrosion crater (pit) has been estimated by computing a safety factor for the stress distribution at the defect tip. This allows the applied notch stress intensity factor to be obtained. To investigate the defect geometry effects, semi-spherical and semi elliptical defects are deemed to exist in up to one-half of the thickness of the pipe wall. The outcomes have been introduced into the structural integrity assessment procedure (SINTAP) failure diagram assessment (FAD) in order to obtain the safety factor value. Conventionally, it is considered that a failure hazard exists if this safety factor is less than two.

Tesnjak , Tomosa T, Kuzle et al[3] presented a basic non-linear mathematical model i investigation of normal and transient operations in high pressure hydroelectric power plan On the basis of this model an original digital simulating model has been developed by usi tioftware-hardware XANALOG system and the MATRIX software package because th aresuitable as a training simulator for the operators in hydroelectric power plants as well as real operating conditions. *L.* Wozniak, F. Collier et al[1991, 14] this paper discusses a digi simulation and stability study of the Bradley Lake hydroelectric project. The basic syste consists of two 45-MW, multiple-jet impulse turbines supplied by a three mile conduit. digital governor is utilized. Once produced, the simulation is used to investigate the stabil of the system. The system was found to be locally stable, but unstable to large perturbatio Due to economic constraints in plant design, the response of the system to load acceptance poor.

Nicolet Christophe, Prof. François Avellan et al[15] developed an analytical approa allowing modeling transient phenomena in pipes, valves, surge tanks and Francis turbir based on impedance method. These models are implemented in a software called "SIMSEN which simulates the behavior of complex applications in the domain of adjustable spe drives and electrical power networks. This program is based on a modular structure, whi enables the numerical simulation of transient modes of systems exhibiting arbitra topologies. The numerical simulation for transient phenomena in hydropower plants w "SIMSEN" has the benefit of an algorithm that generates and solves an integrated set differential equations. This algorithm solves simultaneously the electrical, hydraulic a control equations ensuring a proper interaction between the three parts of the system. T case of a Francis turbine power plant is studied. The model of the turbine is based measured steady state characteristics. The simulation of the dynamic behavior of the pow plant under load variation was investigated.

MODELING OF HYDROPOWER PLANT

Modeling and simulation of a hydro plant (Fig. 2.1) is a valuable tool for planning operations and judging the value of physical improvement by selecting proper system parameters. This study helps in verifying safety conditions, in selecting the best alternatives in the early phase of design and to determine the requirement of special protection devices. It also helps in finding parameters of control equipments like water level regulator, governor etc. and in determining the dynamic forces acting on the system which must be considered in structural analysis of the penstock and their support. Here we have present mathematical modeling of various elements of Hydro power plant like, Reservoir, Penstock, Valve and Pelton Turbine.

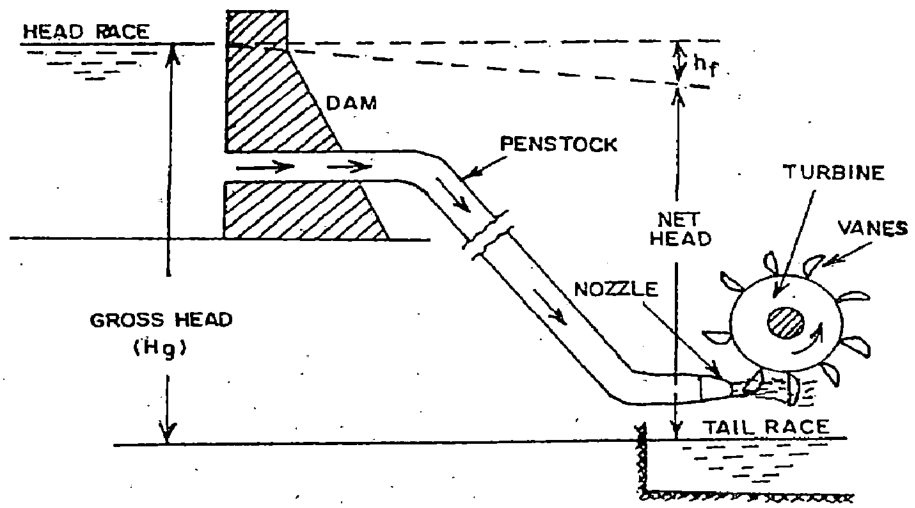

Fig.2.1: Lay-out of a Hydropower plant with Pelton turbine[10]

2.1 MODELING EQUATION FOR RESERVOIR

In modeling of reservoir, we have taken a constant head reservoir at up-stream and consequently neglect entrance losses as well as the velocity head.So we have equation[1] 2.1.

$$H_p = H_{res} \tag{2.1}$$

Fig.2.2 represents a reservoir of constant head (H_{res})

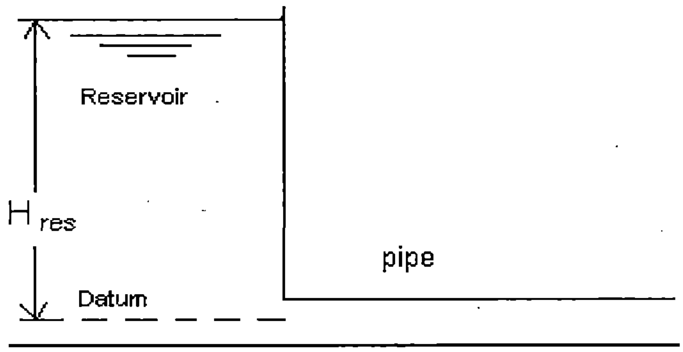

Fig. 2.2: Constant head Reservoir

2.2 MODELING EQUATIONS FOR PENSTOCK

Unsteady flow through closed conduits is described by the momentum and continuity equations.

The following assumptions are made in the derivation of the equations:

1. Flow in the conduit is one-dimensional and the velocity distribution is uniform over the cross-section of the conduit.
2. The conduit wall and the fluid are linearly elastic.
3. Formulas for computing the steady-state friction losses in conduits are valid during transient state.

The conservation of mass and momentum equations applied to the cross-flow section of fluid in a penstock can be written as in equation [1] 2.2 and 2.3.

Momentum: $$\frac{1}{gA}\frac{\partial Q}{\partial t}+\frac{\partial H}{\partial x}+\frac{fQ|Q|}{D2gA^2}=0 \qquad (2.2)$$

Continuity: $$\frac{\partial H}{\partial t}+\frac{c^2}{gA}\frac{\partial Q}{\partial x}=0 \qquad (2.3)$$

Fig. 2.3 shows the penstock of a hydropower plant.

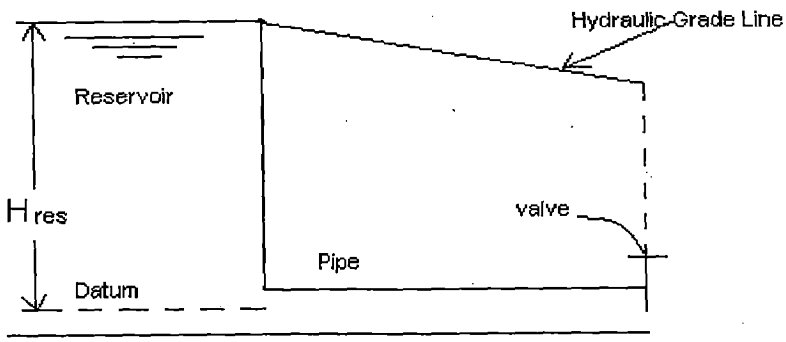

Fig.2.3: Model of penstock

2.3 MODELING EQUATIONS FOR TURBINE

PELTON TURBINE

The Pelton turbine is a tangential flow impulse turbine. The water strikes the bucket along the tangent of the runner. The energy available at the inlet of the turbine is only kinetic energy. The pressure at the inlet and outlet of the turbine is atmospheric. This turbine is used for high heads and low discharge.

From, fig.2.4. shows a Pelton turbine. The water from the reservoir flows through the penstocks at the outlet of which a nozzle is fitted.

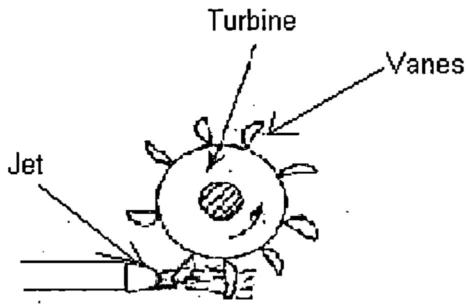

Fig. 2.4 : Pelton turbine

The nozzle increases the kinetic energy of the water flowing through the penstock. At the outlet of the nozzle, the water comes out in the form of a jet and strikes the buckets of the runner.

The main parts of Pelton turbine are:

1. Nozzle and flow regulating arrangement(spear)
2. Runner and buckets.
3. Casing.
4. Breaking jet.

2.4 CHARACTERISTIC CURVES OF PELTON TURBINE

Characteristic curves of a Pelton turbine are the curves, with the help of which the exact behaviour and performance of it under different working conditions, can be known. These curves are plotted from the results of the tests performed on the turbine under different working conditions.

The important parameters which are varied during a test on a turbine are:

1 Speed (N) 2. Head (H)

3. Discharge (Q)

4. Power (P)

5. Overall efficiency (η_o)

6. Gate opening

2.4.1 Main Characteristic Curves

Main characteristic curves(Fig 2.5 a,b,c) are obtained by maintaining a constant head and a constant gate opening (G.O.) on the turbine. The speed of the turbine is varied by changing load on the turbine. For each value of the speed, the corresponding values of the power (P) and discharge (Q) are obtained. Then the overall efficiency (η_o) for each value of the speed is calculated.

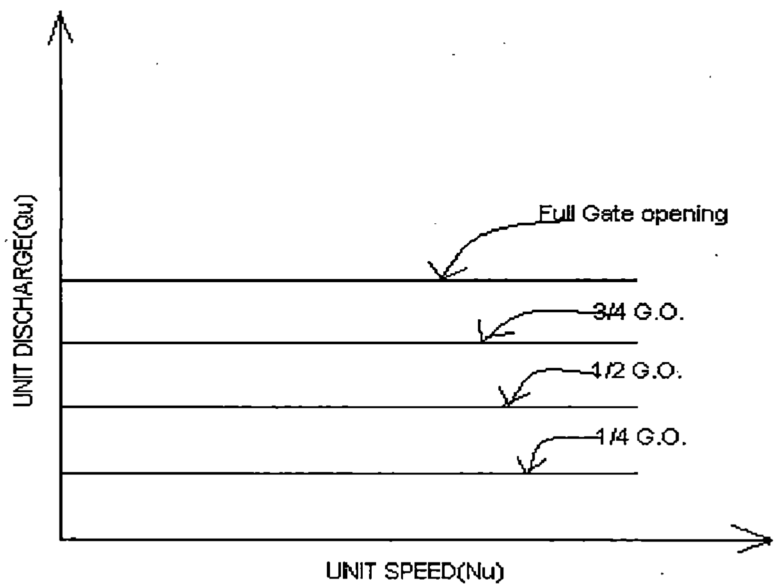

Fig. 2.5(a)

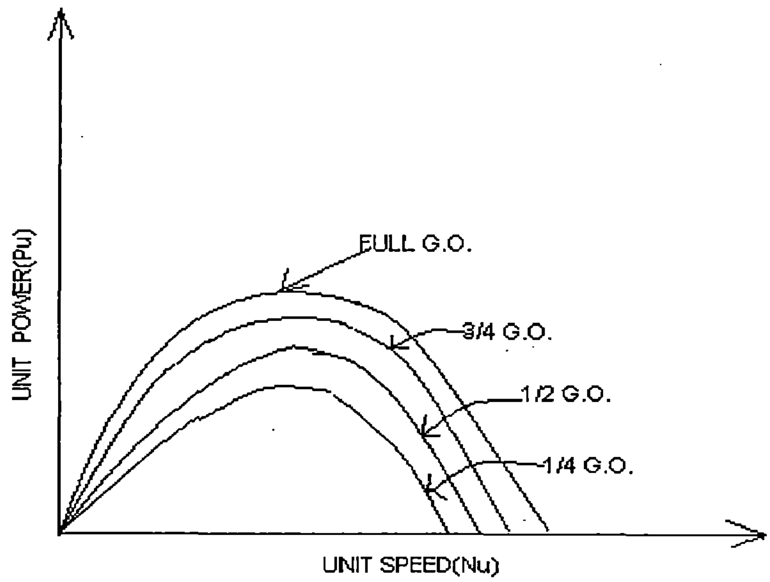

Fig. 2.5(b)

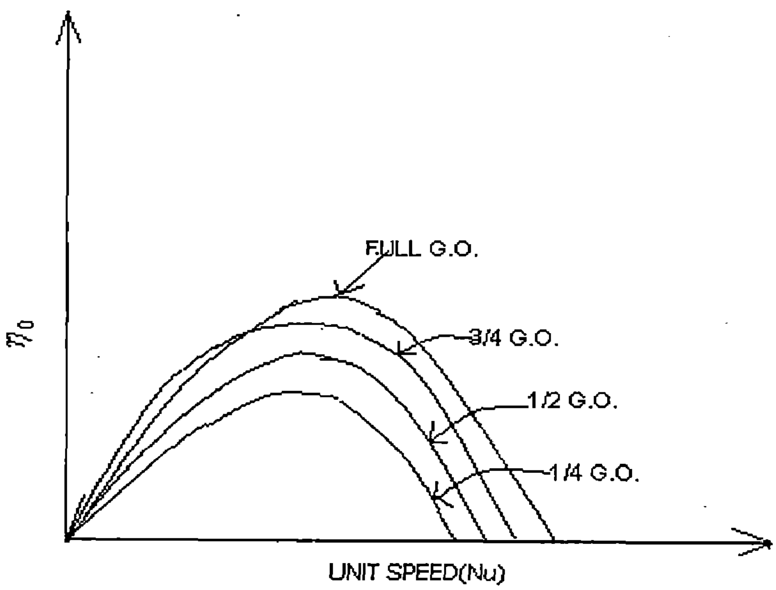

Fig. 2.5(c)

Fig.2.5: Main characteristics curves of Pelton turbine

2.4.2 Muschel Curves or Iso-Efficiency Curves

These curves(Fig 2.6) are obtained from the speed vs. efficiency and speed vs. discharge curves for different gate openings. For a given efficiency from the N_u vs. η_o curves, there are two speeds. From the N_u vs. Q_u curves, corresponding to two values of speeds there are two values of discharge. Hence for a given efficiency there are two values of discharge for a particular opening. So for a given efficiency there are two values of speeds and two values of discharge for a given opening. If the efficiency is maximum there is only one value. These two values of speed and two values of discharge corresponding to a particular gate opening are plotted. The procedure is repeated for different gate openings and the curves Q vs. N are plotted. The points having the same efficiencies are joined. The curves having same efficiency are called iso-efficiency curves. These curves are helpful for determining the zone of constant efficiency and for predicating the performance of the turbine at various efficiencies.

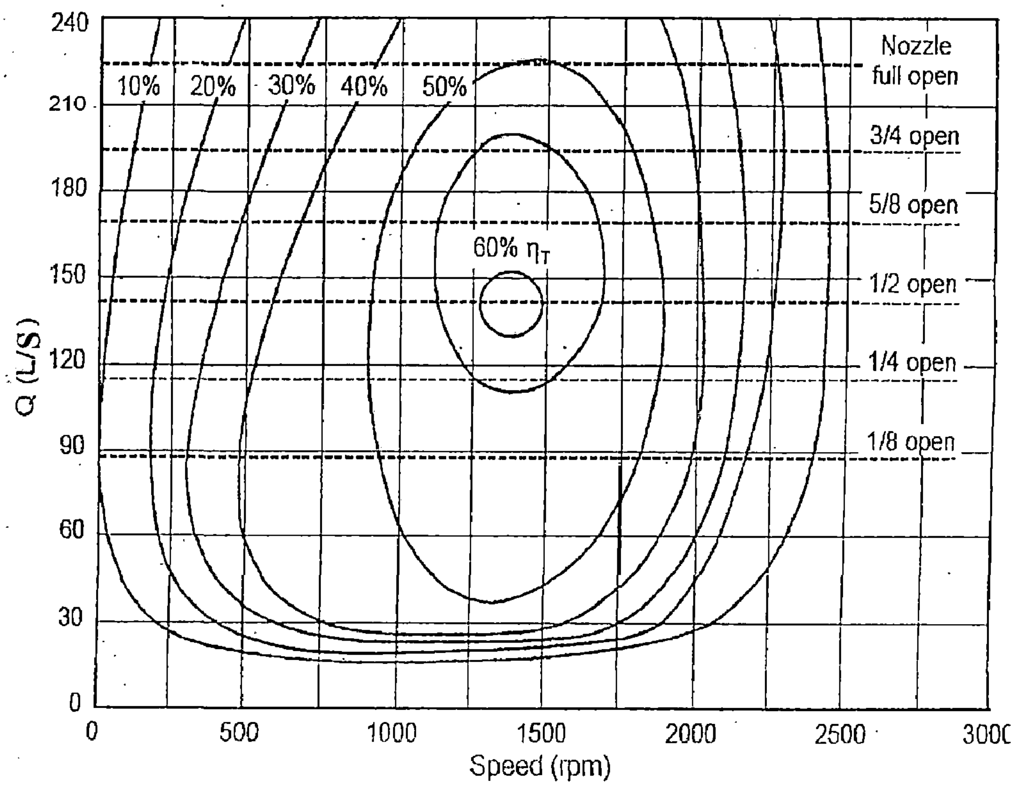

Fig. 2.6: Constant efficiency curve for Pelton turbine

2.4.3 GOVERNING EQUATION

The conservation of mass and energy equations applied to the cross-flow section of fluid in turbine can be written as in equation[1] 2.4 and 2.5.

Energy : $f(H_iQ - H_jQ) = 0$ (2.4)

Continuity: $Q_i = Q_j$ (2.5)

2.5 HYDRAULIC TURBINE GOVERNORS

Governors are provided to keep the turbo generator speed at desired speed. When the load on the generator decreases the speed of the generator increases beyond the normal speed (constant speed). The speed of the turbine also increases beyond the normal speed. If the turbine or the generator is to run at constant(normal) speed, the rate of flow of water to the turbine should be decrease it–till the speed becomes normal. So, the process by which the speed of the turbine (and hence of generator) is kept constant under varying condition of load is called governing.

The main component of the governor is a speed sensing device and a servomechanism for opening and closing the nozzle.

Three types of governors are used for the hydro turbines:

(1) Dashpot

(2) Accelerometric

(3) Proportional –integral –derivative (PID)

In dashpot governors, the corrective action of the governor is proportional to the speed deviation, n; in the accelerometric governor, it is proportional to dn/dt. In the PID, it is proportional to n. dn/dt, and time integral of n. In this work dashpot governor is used.

2.5.1 Dashpot governor for Pelton turbine

The main components of a dashpot governor are a speed-sensing device and a servomechanism for opening or closing the nozzle. Fig. 2.7 shows the position of the piston in the relay cylinder, position of control or relay valve and fly-balls of the centrifugal governor, when the turbine running at the normal speed.

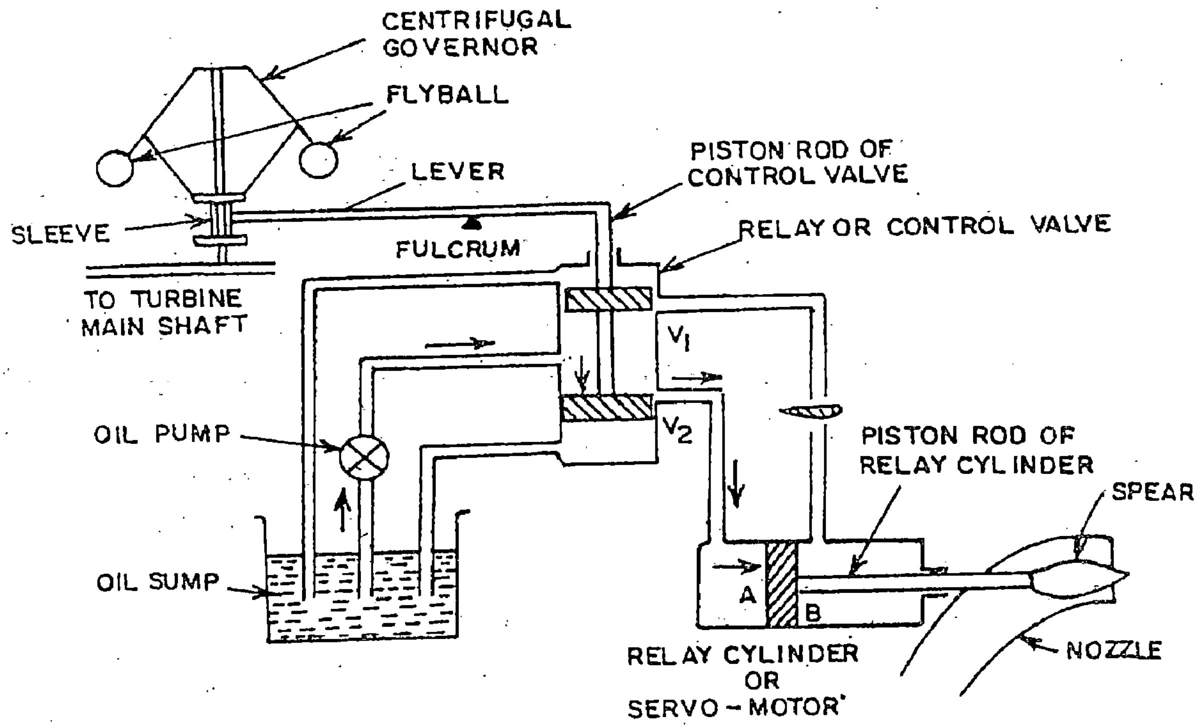

Fig. 2.7: Governing of Pelton turbine.

When the load on the generator decreases, the speed of the generator increases. This increases the speed of the turbine beyond the normal speed. The centrifugal governor, which is connected to the turbine main shaft, will be rotating at an increased speed. Due to increase in the speed of the centrifugal governor, the fly-balls move upward due to the increased centrifugal force on them. Due to the upward movement of the fly-balls, the sleeve will also move upward. A horizontal lever, supported over a fulcrum and the piston rod of the control valve move downward. This closes the valve V_1 and open the valve V_2. The oil, pumped from the oil pump to the control valve or relay valve, under pressure will flow through the valve V_2 to the servomotor (or relay cylinder) and will exert force on the face A of the piston of the relay cylinder. The piston along with piston rod and spear will move towards right. This will decrease the area of flow of water at the outlet of the nozzle. This decrease of area of flow will reduce the rate of flow of water to the turbine which consequently reduces the speed of the turbine. When the speed of the turbine becomes normal, the fly-balls, sleeve, lever and piston rod of control valve come to its normal position as shown in Fig.2.7.

When the load on the generator increases, the speed of the generator and hence of the turbine decreases. The speed of the centrifugal governor also decreases and hence the centrifugal force acting on the fly-balls also reduces. This brings the fly-ball in the downward direction. Due to this, the sleeve moves downward and the lever turns about the fulcrum, moving the piston rod of the control valve in the upward direction. This close the valve V_2 and opens the valve V_1. The oil under pressure from the control valve, will move through valve V_1 to the servomotor and will exert a force on the face B of the piston. This will move the piston along with the piston rod and spear toward left, increasing the area of flow of water at the outlet of the nozzle. This will increases the rate of flow of water to the turbine and consequently, the speed of the turbine will also increases, till the speed of the turbine becomes normal.

The speed *droop* is a governor characteristic that requires a decrease in the speed to produce an increase in the nozzle opening (Fig.2.8). For hydraulic turbines, a large value of speed droop is required for stability. However, this is not permissible from the point of view of system operation. Therefore, two types of speed droop are provided: permanent and temporary. Permanent speed droop is usually about 5 percent, is fixed during transient, and is utilized for sharing loads on parallel units. The value of temporary droop may be large. It is made temporary by means of a dashpot.

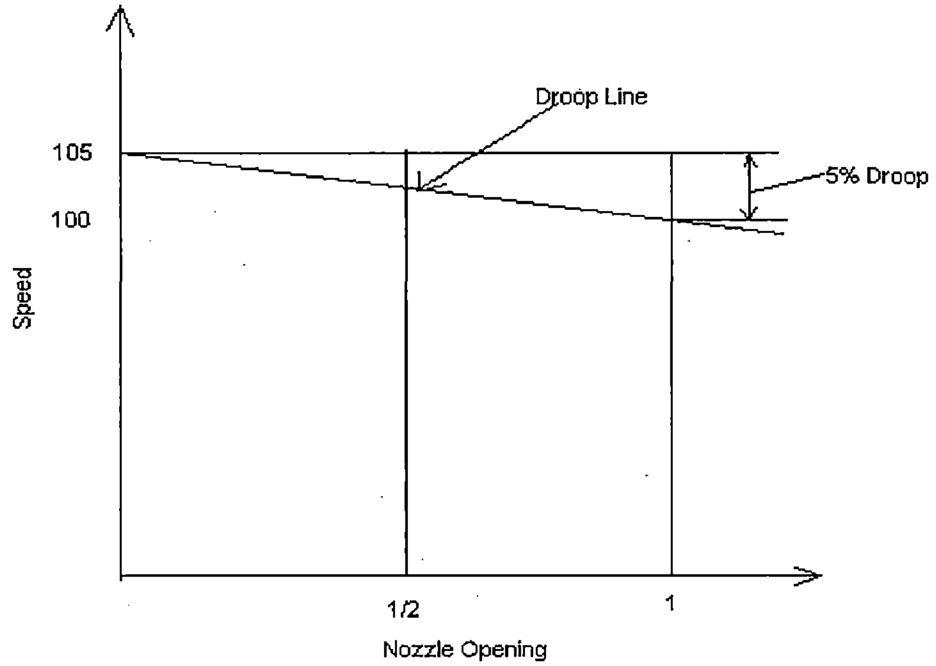

Fig. 2.8: Speed droop

The differential equations for different components of the governor is obtained using its transfer function (Fig.2.9).

The synchronous speed of the turbo generator set , N_R is used to normalized the turbine speed , i.e. n= N/N_R. If τ_o be the initial steady state wicket gate opening, then $n_{ref} = 1 + \sigma\tau_0$.

opening

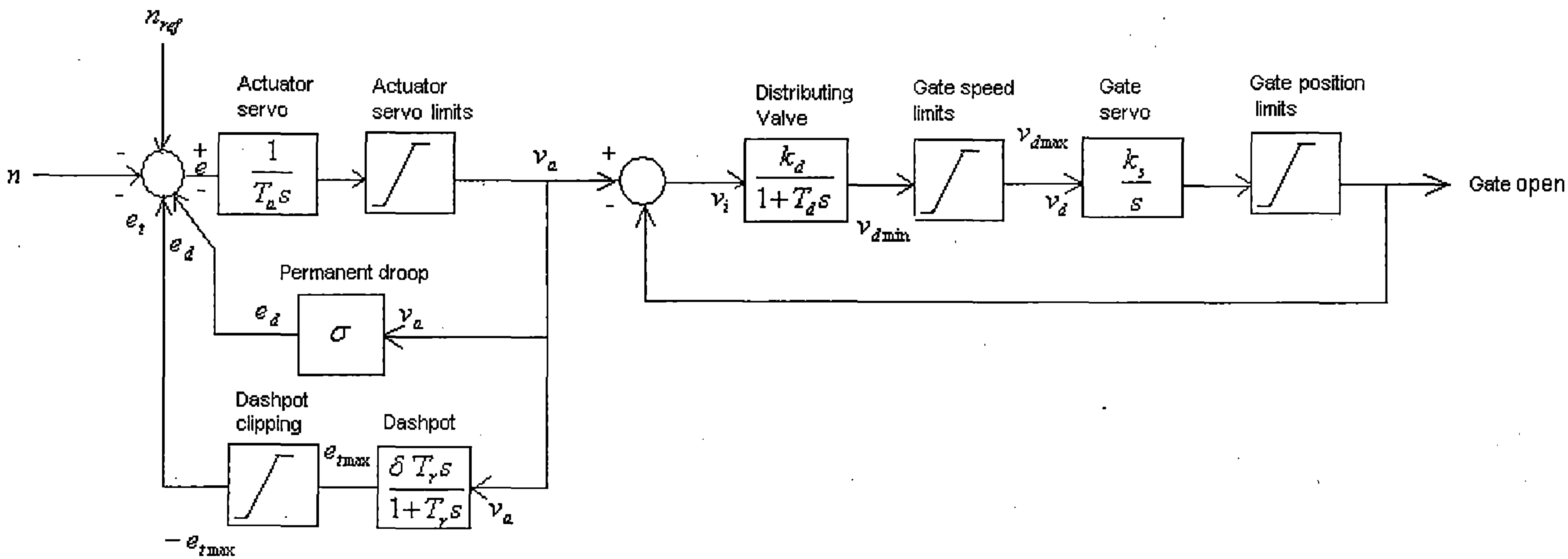

Fig 2.9: Block diagram for a Dashpot governor

Actuator

$$v_a = \frac{1}{T_a s} e$$

$$e = T_a \frac{dv_a}{dt} \qquad 0 \le v_a \le 1 \tag{2.6}$$

Dashpot

$$e_t = \frac{\delta T_r s}{1 + T_r s} v_a$$

or $$e_t + T\ \mathrm{r}\ \frac{de_t}{dt} - \delta T_r \frac{dv_a}{dt} = 0 \qquad -e_{t\max} \le e_t \le e_{t\max} \tag{2.7}$$

Permanent Drop

$$e_d - \sigma v_a = 0 \tag{2.8}$$

Distributing valve

$$v_a = \frac{k_d}{1 + T_d s} v_i$$

$$T_d \frac{dv_d}{dt} + v_d - k_d v_i = 0$$

$$T_d \frac{dv_d}{dt} + v_d - k_d v_i = 0 \qquad v_{d\min} \le v_d \le v_{d\max} \tag{2.9}$$

$$\tau = \frac{k_s}{s} v_d$$

Gate Servomotor *thus* $$0 \le \tau \le 1 \tag{2.10}$$

$$\frac{d\tau}{dt} - k_s v_d = 0$$

The following equations may be written because of two feedbacks

$e = n_{ref} - e_d - e_t - n$

and

$v_i = v_a - \tau$

Output of various components may saturate and that these saturation limits must be taken in to consideration in analysis of large load changes.

By eliminating $e.and.e_d$ v_i and rearranging above equations we get

$$\frac{dv_a}{dt} = \frac{1}{Ta}(n_{ref} - n - e_t - \sigma v_a) \tag{2.11}$$

$$\frac{de_t}{dt} = \frac{1}{T_r}(\delta T_r \frac{dv_a}{dt} - e_t) \tag{2.12}$$

$$\frac{dv_d}{dt} = \frac{1}{T_d}[k_d(v_a - \tau) - v_d] \tag{2.13}$$

$$\frac{d\tau}{dt} = k_s v_d \tag{2.14}$$

The preceding four differential equations in four variables, namely $, v_a, e_t, v_d, and..\tau,$ may be integrated by any standard numerical technique; a close form of solution is not possible because of nonlinearities introduced by the saturation of various variables.

METHOD OF CHARACTERISTICS FOR TRANSIENT FLOW ANALYSIS

In the strategy used by the MOC, the governing partial differential equations are converted to ordinary differential equations and then to a different form for solution by a numerical method. The equations express the head and flow for small time steps (Δt) at numerous locations along the pipe sections. Calculations during the transient analysis must begin with a known initial steady state and boundary conditions. In other words, head and flow at time $t = 0$ will be known along with head and/or flows at the boundaries at all times. To handle the wave characteristics of the transient flow, head and flow values at time $t + \Delta t$ at interior locations are calculated making use of known values of head and flow at the previous time step at adjacent locations using the ordinary differential equations expressed in different form.

3.1 STABILITY AND CONVERGENCE CONDITION

The finite-difference scheme termed as *convergence scheme* if the exact solution of the differential equations approaches that of the original differential equations as Δt and Δx approach zero. If the round-off error due to representation of the irrational number by a finite number of significant digits grows as the solution progresses, the scheme is called *unstable*; if the error decays, the scheme is *stable. e.*

The finite-difference scheme will be *stable* if,

$$\frac{\Delta t}{\Delta x} < \frac{1}{a} \tag{3.1}$$

This condition implies that the characteristics through point P in Fig 2.1. should not fall outside the segment AB.

For neutral scheme,

$$\frac{\Delta t}{\Delta x} = \frac{1}{a} \tag{3..2}$$

The criteria for convergence indicate that the most accurate solutions are obtained if above condition is satisfied.

$$\frac{\Delta t}{\Delta x} \leq \frac{1}{a} \tag{3.3}$$

This is called *Courant's stability* condition

3.2 CHARACTERISTIC EQUATIONS

The momentum and continuity equations for penstock are as follows,

$$L_1: \quad \frac{1}{gA}\frac{\partial Q}{\partial t} + \frac{\partial H}{\partial x} + \frac{fQ|Q|}{D2gA^2} = 0 \tag{3.4}$$

$$L_2: \quad \frac{\partial H}{\partial t} + \frac{c^2}{gA}\frac{\partial Q}{\partial x} = 0 \tag{3.5}$$

We have a linear combination of above equations 3.4. and 3.5 ,

$$L = L_1 + \lambda L_2$$

Or, $$\left(\frac{1}{gA}\frac{\partial Q}{\partial t}+\frac{\partial H}{\partial x}+\frac{fQ|Q|}{D2gA^2}\right)+\lambda\left(\frac{\partial H}{\partial t}+\frac{c^2}{gA}\frac{\partial Q}{\partial x}\right)=0 \quad (3.6)$$

$$\left(\frac{\partial Q}{\partial t}+\lambda a^2\frac{\partial Q}{\partial x}\right)+\lambda gA\left(\frac{\partial H}{\partial t}+\frac{1}{\lambda}\frac{\partial H}{\partial x}\right)+\frac{f}{2DA}Q|Q|=0 \quad (3.7)$$

If H=H(x, t) and Q=Q(x, t) are solutions of Eqs. 3.5. and 3.6 then the total derivative may be written as.

$$\frac{dQ}{dt}=\frac{\partial Q}{\partial t}+\frac{\partial Q}{\partial x}\frac{dx}{dt} \quad (3.8)$$

$$\frac{dH}{dt}=\frac{\partial H}{\partial t}+\frac{\partial H}{\partial x}\frac{dx}{dt} \quad (3.9)$$

Now, the unknown multiplier λ is as

$$\frac{1}{\lambda}=\frac{dx}{dt}=\lambda a^2 \quad (3.10)$$

Or, $$\lambda=\pm\frac{1}{a} \quad (3.11)$$

and by using Eqs. 3.8. and 3.9., Eqs. 3..7. can be written as,

$$\frac{dQ}{dt}+\frac{gA}{a}\frac{dH}{dt}+\frac{f}{2DA}Q|Q|=0 \quad (3.12)$$

If $$\frac{dx}{dt}=a \quad (3.13)$$

And

$$\frac{dQ}{dt} - \frac{gA}{a}\frac{dH}{dt} + \frac{f}{2DA}Q|Q| = 0 \tag{3.14}$$

If
$$\frac{dx}{dt} = -a \tag{3.15}$$

the partial differential equations have been converted into ordinary differential equations in the independent variable t.

In the x-t plane, Eqs.3.12. and 3.14. represent two straight lines having slopes $\pm\frac{1}{a}$. These are called *Characteristics Lines*. Mathematically, these lines divide the x-t plane into two regions, which may be dominated by two different kinds of solution. Physically, they represent the path traversed by a disturbance. For example, a disturbance at point A (Fig.3.1)at time t_0 would reach point P after time Δt.

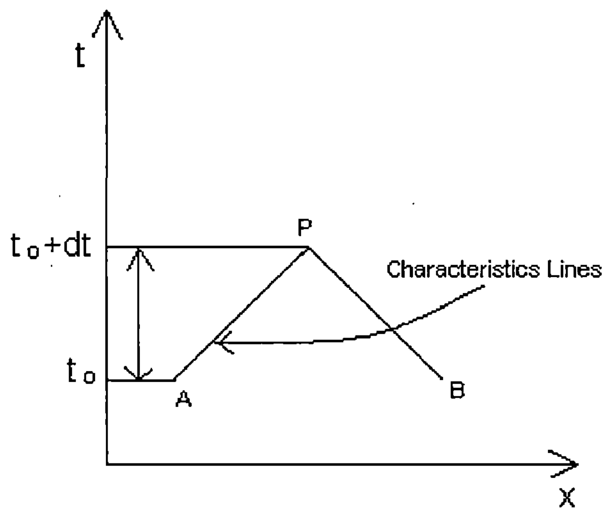

Fig. 3.1: Characteristics lines in x-t plane.

3.3 SOLUTION OF MOC EQUATIONS

To solve Eqs.3.12. and 3.14., we use a first-order finite-difference technique. Because the time interval used in solving these equations for practical problems are usually small, a first-order technique is sufficiently accurate. However, if the friction losses are large, then a first-order approximation may yield unstable results. For such cases, a second-order approximation should be used to avoid instability of the finite-difference scheme.

Referring to Fig.3.1, let the condition at time $t = t_0$ be known. These are either initially known (i.e. at t = 0, these are initial steady state conditions) or have been calculated for previous time step. To compute the unknown conditions at $t_0 + \Delta t$. From Fig. 3.1 , we can write along the positive characteristic line AP.

$$dQ = Q_P - Q_A \tag{3.16}$$

$$dH = H_P - H_A \tag{3.17}$$

Similarly, from Fig. 3.1, we can write along the negative characteristics line BP.

$$dQ = Q_P - Q_B \tag{3.18}$$

$$dH = H_P - H_B \tag{3.19}$$

Substituting the value of dQ and dH in Eqs. 3.12. and 3.14. computing the friction term at the points A and B, and multiplying throughout by Δt, we obtain.

$$(Q_P - Q_A) + \frac{gA}{a}(H_P - H_A) + \frac{f\Delta t}{2DA} Q_A |Q_A| = 0 \tag{3.20}$$

$$(Q_P - Q_B) + \frac{gA}{a}(H_P - H_B) + \frac{f\Delta t}{2DA} Q_B |Q_B| = 0 \tag{3.21}$$

Equation 3.19. can be written as

$$Q_P = C_P - C_a H_P \tag{3.22}$$

and Eq. 3.20. as

$$Q_P = C_n + C_a H_P \tag{3.23}$$

in which

$$C_P = Q_A + \frac{gA}{a} H_A - \frac{f\Delta t}{2DA} Q_A |Q_A| \tag{3.24}$$

$$C_n = Q_B - \frac{gA}{a} H_B - \frac{f\Delta t}{2DA} Q_B |Q_B| \tag{3.25}$$

And

$$C_a = \frac{gA}{a} \tag{3.26}$$

The Eq.3.22. is valid along the positive characteristic line AP and Eq.3. 23. along the negative characteristics line BP. The values of the constants C_P and C_n are known for each time step, and the constant C_a depends upon the conduit properties. In Eqs. 3.22. and 3.23., we have two unknowns, namely, H_P and Q_P. The values of these unknowns can be determined by simultaneously solving these equations. By adding Eqs. 3.22 and 3.23 we have,

$$Q_P = 0.5(C_P + C_n) \tag{3.27}$$

Now the value of H_p can be determined either from Eq. 3.22 or Eq. 3.23. Thus, by using Eqs. 3.22. and 3.23. , conditions at all interior points at the end of the time step can be determined. However, at the boundaries, either Eq. 3.22. or 3. 23. is available.

From, Fig.3.2. the pipeline is divided into n equal reaches(Fig.3.3.) and the steady- state conditions at the grid points at t= t_0 are first obtained. Then, to determine the conditions at t =$t_0+\Delta t$, Eqs 3.22. and 3.23 are used for the interior points, and special boundary conditions are used for end conditions.

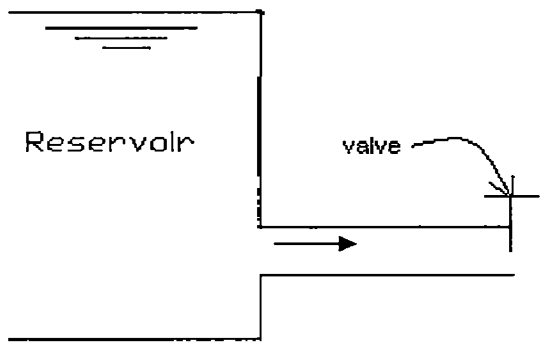

Fig. 3.2: Single Pipeline

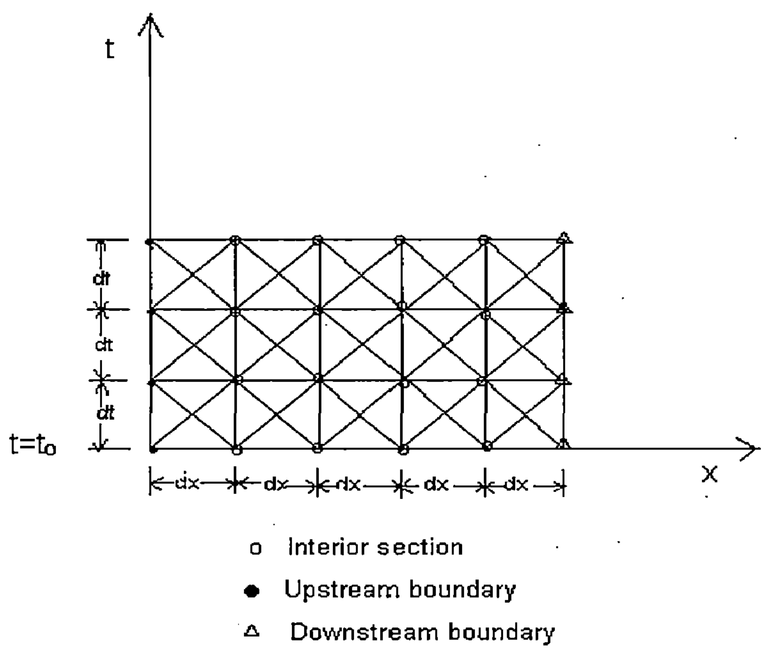

Fig. 3.3: Characteristic grid

Fig. 3.3 shows that the conditions at the boundaries at $t = t_0 + \Delta t$ must be known for calculating the conditions at $t = t_0 + 2\Delta t$ at the interior points adjacent to the boundaries. Now condition at $t = t_0 + \Delta t$ are known at all the grid points, and the conditions at $t = t_0 + 2\Delta t$ are determined by following the procedure. In this manner, transient condition are computed step-by-step for the required time.

3.4 BOUNDARY CONDITIONS

The special boundary conditions are required to determine the conditions at the boundaries. These are developed by solving Eq 3.22. and 3.23, or both, and the conditions imposed by the boundary. Equation 3.19. Is used for the downstream boundaries and Eq. 3.26 for the upstream boundaries.

3.4.1 Constant-Head Reservoir at Upstream End.

In fig 3.4 the reservoir has been shown with penstock pipe , E.G. and H.G.L. have also been shown.

If the entrance losses as well as the velocity head are negligible, then

$$H_P = H_{res} \tag{3.28}$$

In which H_{res} = height of the reservoir water surface above the datum.

$$Q_P = C_n + C_a H_{res} \tag{3.29}$$

However, if the velocity head or the entrance losses are not small, then these may be also considered in the analysis.

Let the entrance losses is

$$h_e = \frac{kQ_P^2}{2gA^2} \tag{3.30}$$

Where, k = entrance loss coefficient.

and the head losses is,

$$h_l = \frac{Q_P^2}{2gA^2} \tag{3.31}$$

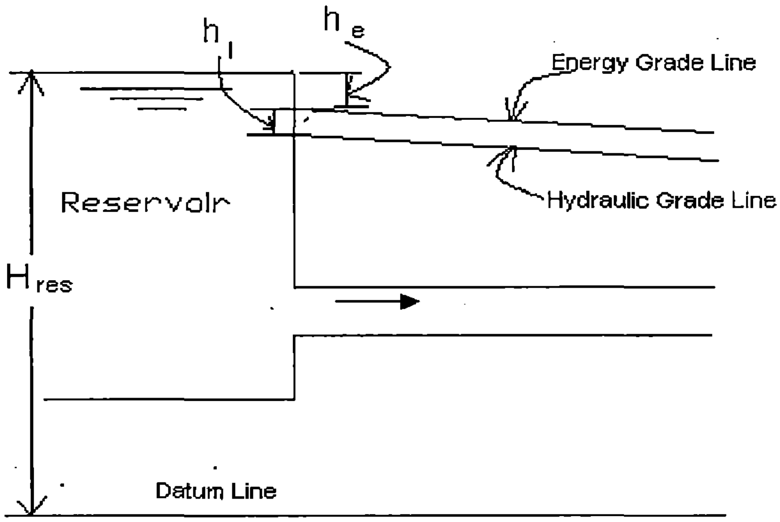

Fig. 3.4: Constant-level upstream reservoir

From Fig. 3.4, we have,

$$H_P = H_{res} - (1+k)\frac{Q_P^2}{2gA^2} \tag{3.32}$$

By, solving Eq. 3.31 and the negative characteristic equation 3.20. simultaneously, we have.

$$Q_P = \frac{-1+\sqrt{1+4k_1(C_n + C_a H_{res})}}{2k_1} \tag{3.33}$$

in which

$$k_1 = \frac{C_a(1+k)}{2gA^2} \tag{3.34}$$

Now H_P can be determined from Eq. 3.31.

3.4.2 Valve at Downstream End (Nozzle in case of Pelton turbine).

In fig 3.5, the valve has been shown at the downstream end of the penstock pipe. In fig 3.5(b) and fig.3.5(c), the curve has been plotted for position of valve w.r.t. time during opening and closing of valve respectively.

The steady-state flow through a valve can be written as,

$$Q_0 = (C_d A_v)_0 \sqrt{2gH_0} \tag{3.35}$$

in which subscript $_0$ indicate steady-state conditions, C_d= coefficient of discharge, H_0= head upstream of the valve, and A_v= area of the valve opening. For transient state condition,

$$Q_P = (C_d A_v)\sqrt{2gH_P} \tag{3.36}$$

Dividing Eq. 3.35. by Eq. 3.31., taking square of both sides and the relative valve opening $\tau = (C_d A_v)/(C_d A_v)_0$, we obtained.

$$Q_P^2 = \frac{(Q_0\tau)^2}{H_0} H_P \tag{3.37}$$

Substitution for H_P from the positive characteristics equation 3.22. into Eq 3.36. we have.

$$Q_P^2 + C_v Q_P - C_P C_v = 0 \tag{3.38}$$

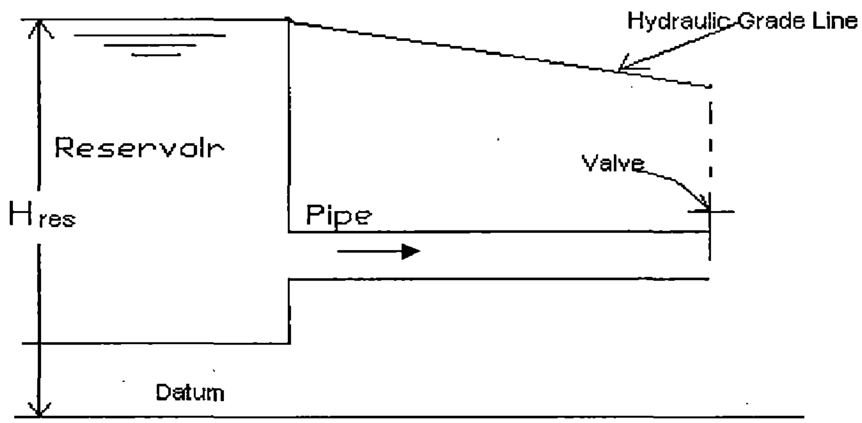

Fig. 3.5(a)

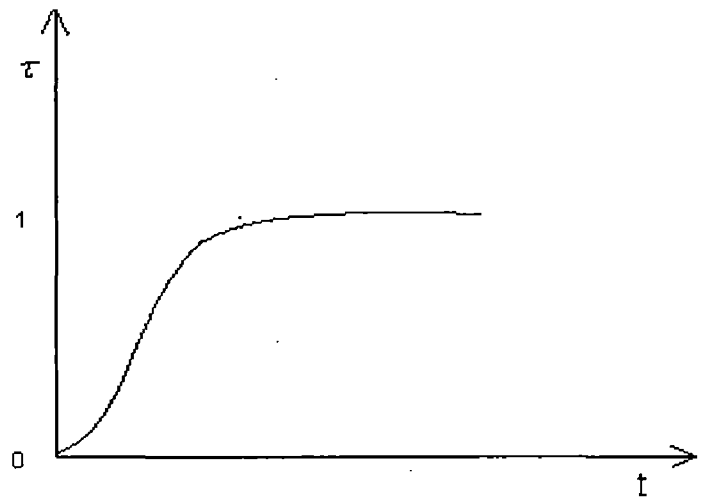

Fig. 3.5 (b): Opening

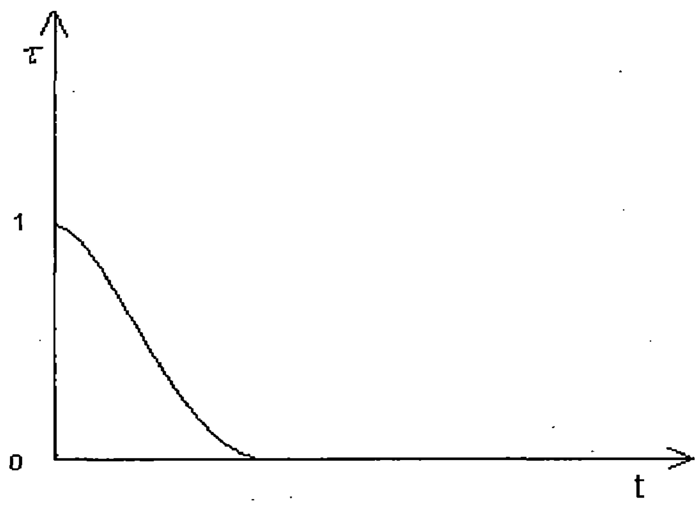

Fig. 3.5(c): Closing

Fig. 3.5: Valve at Downstream

In which $C_v = \frac{(Q_0\tau)^2}{(C_a H_0)}$. Solving for Q_P and neglecting the negative sign with the radical term, we have.

$$Q_P = 0.5(-C_v + \sqrt{C_v^2 + 4C_P C_v}) \quad (3.39)$$

Now H_P can be determine from Eq.3.22.

To compute the transient-state conditions for an opening or a closing valve, τ versus t curve (Fig.3.5(b) or (c)) may be specified either in a tabular form or by an algebraic expression. $\tau = 1$ corresponding to a valve opening at which the flow through the valve is Q_0 under a head of H_0.

In case of pelton turbine, the discharge at Δt will be same as derived for the nozzle in Eqs. 3.22 and to compute the transient-state conditions for load rejection nozzle opening and closing i.e. value of τ is control by the turbine governor.

3.5 TRANSIENT SIMULATION FOR TURBINE

Because of instantaneous unbalanced torque, $T_u = T_{tur} - T_{Gen}$,the speed of turbine and generator will in accordance with the unbalance torque as detailed below.

$$T_u = WR^2 \frac{d\omega}{dt}$$

or (3.40)

$$T_{tur} - T_{gen} = WR^2 \frac{2\pi}{60}\frac{dN}{dt}$$

$$P_{tur} - \frac{P_{gen}}{\eta_{gen}} = WR^2 (\frac{2\pi}{60})^2 N \frac{dN}{dt} \quad (3.41)$$

Integrating both side we have,

$$\int_{t1}^{tp}(P_{tur} - \frac{P_{gen}}{\eta_{gen}})dt = 1.097\times10^{-2} WR^2 \int_{N1}^{Np} NdN$$

simplifying (3.42)

$$\left(\frac{P_{tur1} + P_{turP}}{2} - \frac{P_{gen1} + P_{genP}}{2\eta_{gen}}\right)\Delta t = 0.548\times10^{-12} WR^2 (N_P^2 - N_1^2)$$

In which subscript 1 and P indicate the values of the variables at the beginning and at the end of time step. Solving for N_p,

$$N_P = \{N_1^2 + 182.38\frac{\Delta t}{WR^2}\left[0.5(P_{tur1} + P_{turP}) - \frac{0.5}{\eta_{gen}}(P_{gen1} + P_{genP})\right]\}^{0.5} \quad (3.43)$$

Which is the required governing equation.

TRANSIENT ANALYSIS IN CLOSED CONDUIT

In analysis of transients in closed conduits, we considered the fluid is compressible, and the transient phenomenon occurs in the form of traveling waves, e.g. like in water–supply pipes, power plant conduits. Mathematically, the transients in the distributed systems are represented by partial differential equations.

4.1 GOVERNING EQUATIONS FOR HORIZONTAL PIPE

Rapidly varying pressure and flow conditions in pipe networks are characterized by variations that are dependent on both position, *x*, and time, *t*. These conditions are described by the continuity equation (3.5)

$$\frac{\partial H}{\partial t}+\frac{a^2}{gA}\frac{\partial Q}{\partial x}=0$$

and the momentum (Newton's second law) equation (3.4)

$$\frac{\partial H}{\partial x}+\frac{1}{gA}\frac{\partial Q}{\partial t}+f(Q)=0$$

By solving the above two equations by using MOC , we have two solutions, eqs. 3.22 and 3.23.

$$Q_P = C_P - C_a H_P$$

$$Q_P = C_n + C_a H_P$$

Where

$$C_P = Q_A + \frac{gA}{a} H_A - \frac{f\Delta t}{2DA} Q_A |Q_A|$$

$$C_n = Q_B - \frac{gA}{a} H_B - \frac{f\Delta t}{2DA} Q_B |Q_B|$$

And

$$C_a = \frac{gA}{a}$$

Where as , from eq. 3.27

$$Q_P = 0.5(C_P + C_n)$$

Putting the value of Q_P in 3.22 or 3.23, we can determine H_p.

We follow the following procedure to determine the pressure head and discharge at the valve. From, Fig.3.2. the pipeline is divided into n equal reaches(Fig.3.3.) and the steady- state conditions at the grid points at t= t_0 are first obtained. Then, to determine the conditions at t =$t_0 + \Delta t$, Eqs 3.22. and 3.23 are used for the interior points, and special boundary conditions are used for end conditions. Fig. 3.3 shows that the conditions at the boundaries at t =$t_0 + \Delta t$ must be known for calculating the conditions at $t = t_0 + 2\Delta t$ at the interior points adjacent to the boundaries. Now condition at t =$t_0 + \Delta t$ are known at all the grid points, and the conditions at $t = t_0 + 2\Delta t$ are determined by following the procedure. In this manner, we compute transient condition step-by-step for the required time.

The first investigated case is taken from . M.H. Chaudhry [1]. The effect of the closure of a valve on a plant (figure 4.1) constituted by a reservoir, a pipe and the valve is studied.

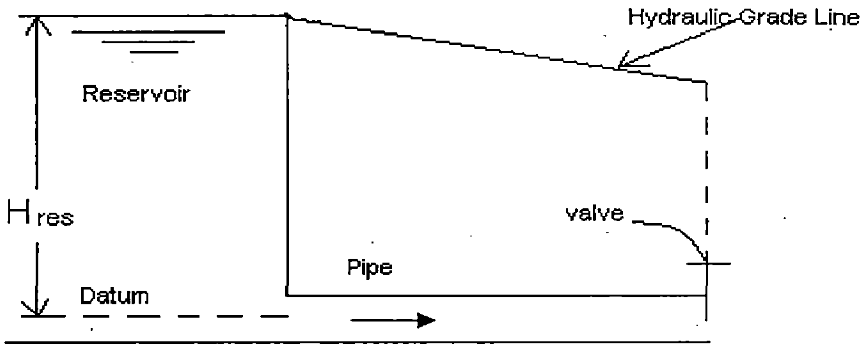

Fig. 4.1: Shows HPP[1]

$$H_{res} = 67.7m$$

$$Q_0 = 1m^3/s \quad , \quad \tau = 1 - (t/2.1)^{0.75}$$

$$L_1 = 550m \qquad D_1 = 0.75m$$

$$a_1 = 1100m/s \qquad f_1 = 0.010$$

As the pipe system becomes more complex, the number of required calculations increases, therefore, a computer program of Appendix A has been done for solving the model equations in C++ language. For the simulation, the pipe is decomposed in 2 elements and the time step of 0.25 sec. have been taken.

4.1.1 FLOWCHART FOR PIPING SYSTEM.

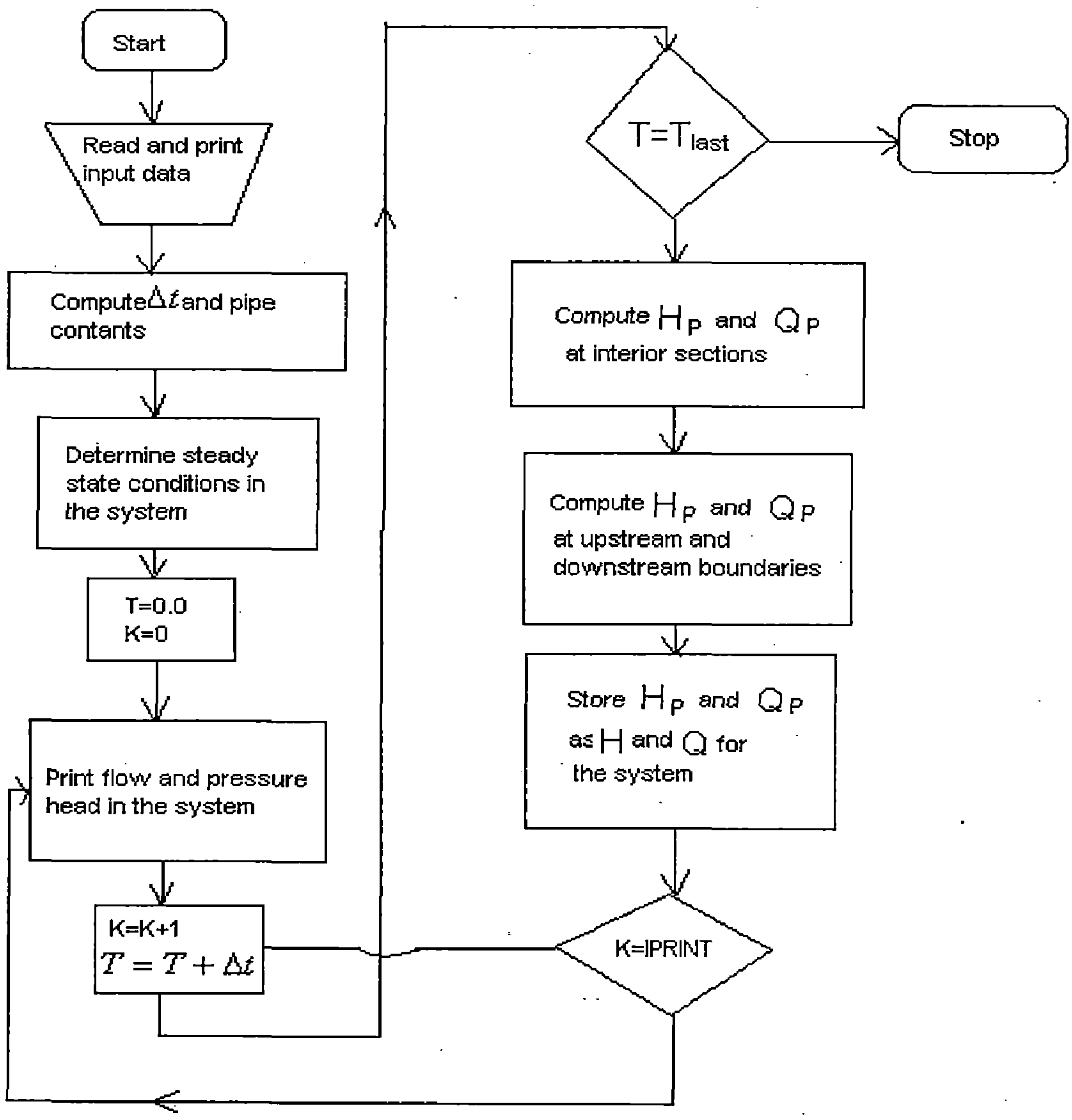

Fig. 4.2: Flowchart for a Piping system

4.1.2 ALGORITHM

1. Read and input data.
2. Calculate various Pipe constant and Δt.

$$\Delta t = \frac{L}{an} \tag{4.1}$$

3. Calculate Steady state value i.e. Head (H_0) and Discharge (Q_0) at t = 0.
4. Calculate the value of τ for time step of Δt, for the given time t.
5. Calculate Head (H_P) and Discharge (Q_P) at t = Δt, 2Δt, 3Δt,............, nΔt. at reservoir boundary condition and store H_P and Q_P as H and Q for the system.
6. Calculate Head (H_P) and Discharge (Q_P) at t = Δt, 2Δt, 3Δt,............, nΔt .at interior points of the pipe and store H_P and Q_P as H and Q for the system
7. Calculate Head (H_P) and Discharge (Q_P) at t = Δt, 2Δt, 3Δt,............, nΔt. at the valve and store H_P and Q_P as H and Q for the system
8. Repeat the steps 3., 4.,5.,6.,7. until the given condition in for loop is satisfied.
9. Print the output data.

4.2 GOVERNING EQUATIONS FOR INCLINED PIPE

Rapidly varying pressure and flow conditions in pipe networks are characterized by variations that are dependent on both position, x, and time, t. These conditions are described by the continuity equation [4]

$$\frac{\partial H}{\partial t}+\frac{c^2}{gA}\frac{\partial Q}{\partial x}-\frac{Q}{A}\sin\alpha=0 \tag{4.2}$$

and the momentum (Newton's second law) equation[4]

$$\frac{\partial H}{\partial x}+\frac{1}{gA}\frac{\partial Q}{\partial t}+f(Q)=0 \tag{4.3}$$

In which $f(Q)$ represents a pipe-resistance term that is a nonlinear function of flow rate.

By solving above two equations using MOC , we have two solutions eqs. 3.22 and 3.23.

$$Q_P=C_P-C_aH_P$$

$$Q_P=C_n+C_aH_P$$

Where

$$C_P=Q_A+\frac{gA}{a}H_A-\frac{f\Delta t}{2DA}Q_A|Q_A|+\frac{\Delta tgQ_A}{a}\sin\alpha \tag{4.4}$$

$$C_n=Q_B-\frac{gA}{a}H_B-\frac{f\Delta t}{2DA}Q_B|Q_B|-\frac{\Delta tgQ_B}{a}\sin\alpha \tag{4.5}$$

And

$$C_a=\frac{gA}{a}$$

Where as , from eq. 3.27

$$Q_P = 0.5(C_P + C_n)$$

Putting the value of Q_P in 3.22 or 3.23, we can determine H_p.

We follow the same procedure as in case of horizontal penstock.

The effect of the closure of a valve on a plant (figure 4.3) constituted by a reservoir, a penstock and the valve is studied

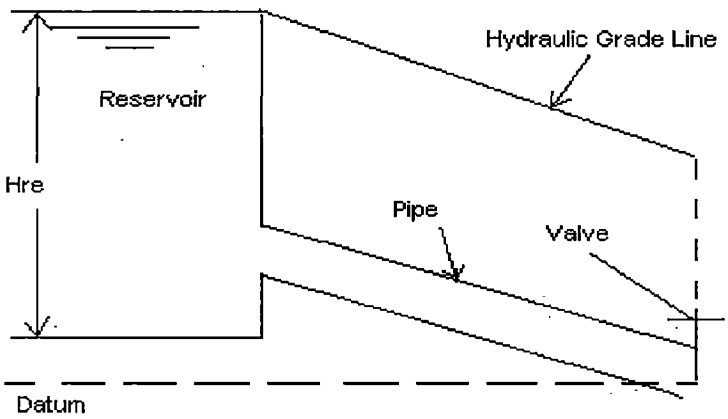

Fig. 4.3: The HPP with inclined penstock

$$H_{re} = 150m$$

$$Q_0 = 1m^3/s \quad , \quad \tau = 1 - (t/2.1)^{0.75}$$

$$L_1 = 45m \quad , \quad D_1 = 0.5m$$

$$a_1 = 900m/s \quad , \quad f_1 = 0.02$$

$$\alpha = 1.046667 \text{ (Radian)}$$

As the pipe system becomes more complex, the number of required calculations increases, therefore, a computer program of Appendix B has been done for solving the model equations in C++. For the simulation, the pipe is decomposed in 4 elements and the time step of 0.0125 sec. have been taken.

4.3 TRANSIENT ANALYSIS OF PELTON TURBINE

The operation of a hydropower plant subject to several transient phenomenon due to load rejection, group start-up and shut-down, modification of operating point, earth fault, out of phase synchronization during start-up, emergency stop and so on. In load rejection, when the load on the generator decreases, the speed of the generator increases. This increases the speed of the turbine beyond the normal speed. Governors are provided to keep the turbine speed at desired speed.

4.3.1 GOVERNING EQUATION FOR TURBINE

Rapidly varying turbine speed during load rejection is characterized by variations that are dependent on both, turbine output (P_{tur}) and generator load (P_{gen}) at time, t_0 and $t_0 + \Delta t$. These conditions are described by the equation (3.43).

$$N_P = \left\{N_1^2 + 182.38 \frac{\Delta t}{WR^2}\left[0.5(P_{tur1} + P_{turP}) - \frac{0.5}{\eta_{gen}}(P_{gen1} + P_{genP})\right]\right\}^{0.5}$$

The effect of the load rejection on a plant (figure 4.4) constituted by a reservoir, a pipe , a nozzle, a Pelton turbine and a Generator is studied. We have taken input data in tabular form, from the hill chart shown in Fig.2.6

Discharge(m3/s)	Speed(rpm)	Efficiency(pu)
0.223	1500	0.5
0.223	1600	0.49
0.223	1700	0.48
0.223	1800	0.45
0.223	1900	0.4
0.223	2000	0.35
0.195	1500	0.6
0.195	1600	0.57
0.195	1700	0.53
0.195	1800	0.48
0.195	1900	0.43
0.195	2000	0.39
0.17	1500	0.6

0.17	1600	0.6
0.17	1700	0.58
0.17	1800	0.49
0.17	1900	0.47
0.17	2000	0.4
0.14	1500	0.6
0.14	1600	0.6
0.14	1700	0.54
0.14	1800	0.51
0.14	1900	0.45
0.14	2000	0.41
0.115	1500	0.6
0.115	1600	0.58
0.115	1700	0.53
0.115	1800	0.5
0.115	1900	0.45
0.115	2000	0.401
0.087	1500	0.54
0.087	1600	0.52
0.087	1700	0.51
0.087	1800	0.48
0.087	1900	0.44
0.087	2000	0.39
0.06	1500	0.51
0.06	1600	0.501
0.06	1700	0.496
0.06	1800	0.44
0.06	1900	0.36
0.06	2000	0.3
0.03	1500	0.4001
0.03	1600	0.4
0.03	1700	0.31
0.03	1800	0.29
0.03	1900	0.2
0.03	2000	0.101
0.02	1500	0.1
0.02	1600	0.099
0.02	1700	0.095
0.02	1800	0.0854
0.02	1900	0.0765
0.02	2000	0.0675
0.01	1500	0.0556
0.01	1600	0.054
0.01	1700	0.0453
0.01	1800	0.0234
0.01	1900	0.0221

Table 1: Input data for turbine analysis.

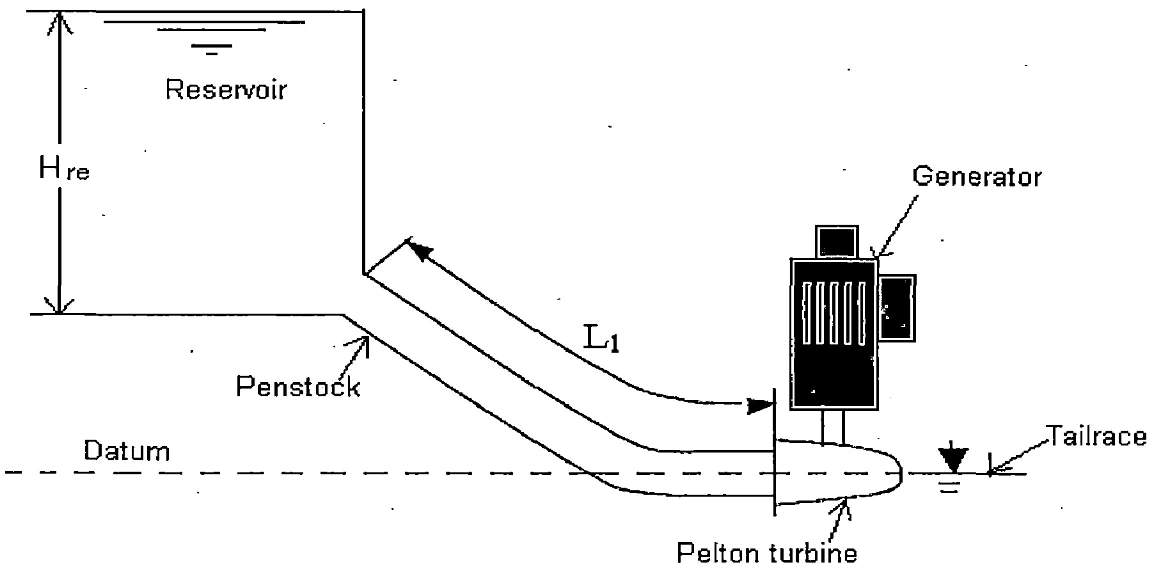

Fig. 4.4: Hydropower plant with Pelton turbine.

$$H_{re} = 150m, \; Q_0 = 0.223m^3/s$$

$$L_1 = 45m \;, \quad D_1 = 0.5m$$

$$a_1 = 900m/s \;, \quad f_1 = 0.02$$

$$\alpha = 1.046667 \text{(Radian)}$$

As the load is decomposed in small time steps, so the number of required calculations increases, therefore, a computer program of Appendix C has been done for solving the model equations in C++ language. For the simulation, load is reduces from 372 KW to 170 KW in 3 sec. The load is decomposed in the time step of 0.0125 sec.

4.3.2 ALGORITHM

1 Read input data. Taking $\tau = 1.0$.

2. Calculate various Pipe constant and Δt from the equation (3.1).

$$\Delta t = \frac{L}{an}$$

3. Reading Discharge, Speed and efficiency values from the input file "lal.txt".
4. Division of load in time step of Δt for the given time t.
5. Calculating Steady state value i.e. Head (H_0) and Discharge (Q_0) at t = 0.
6. Calculating turbine power at t = t_0 sec.
7. Calculate Head (H_P) and Discharge (Q_P) at t $=\Delta t$, $2\Delta t$, $3\Delta t$,............., nΔt. at reservoir boundary condition and store H_P and Q_P as H and Q for the system.
8. Calculate Head (H_P) and Discharge (Q_P) at t $=\Delta t$, $2\Delta t$, $3\Delta t$,............., nΔt.at interior points of the pipe and store H_P and Q_P as H and Q for the system
9. Calculate Head (H_P) and Discharge (Q_P) at t $=\Delta t$, $2\Delta t$, $3\Delta t$,............., nΔt. at the valve

   and store H_P and Q_P as H and Q for the system.
10. Searching for efficiency, for calculated Q_P and N_1 from the input file "lal.txt" and using Newton forward interpolation method for interpolating.
11. Calculating turbine power at t $= t_0 + \Delta t$.
12. Calculating turbine speed at t $= t_0 + \Delta t$ from the equation (4.9).
13. calculating the value of n. Where, n=(np-n1)/nrat. If n<0.002 , calculating new value of τ by solving governor equation through fourth order *Runge-kutta* method.
14. Store np and pp as n1 and p1 for the system.
15. Repeat the steps 7, 8, 9, 10, 11, 12, 13 and 14. until the given condition in for loop is satisfied.
16. Print the output data.

Chapter-5

RESULTS AND DISCUSSION

The results have been obtained for horizontal penstock pipe, inclined penstock pipe and turbine during load rejection.

5.1 HORIZONTAL PENSTOCK PIPE

The graph for discharge from reservoir ,valve and pressure head at the valve with respect to time have been obtained and significant results have been pointed out so as to, better control and stability of hydraulic system in hydropower plant

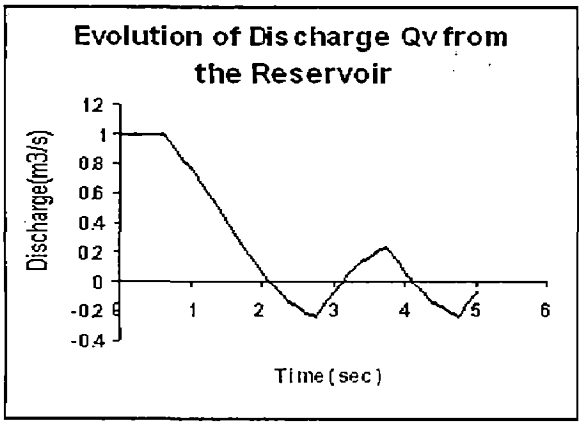

Fig. 5.1(a)

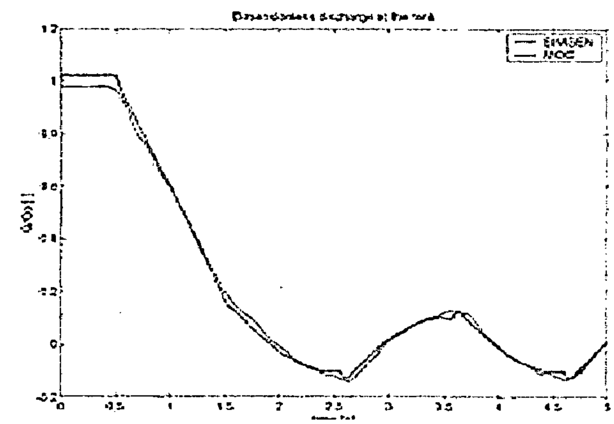

Fig.5.1(a1)[15]

In Fig. 5.1(a)., we observe that as the valve being closed, discharge through the reservoir decreases from 1m3/s to 0m3/s during 0 to 2.1 sec. After that discharge is –ve i.e., the direction of flow is reverse. Because the pressure head at the valve is greater than the reservoir head. After 2 sec. onward an oscillation takes place. By comparing with Fig. 5.1(a1), we observe that both curve are similar in nature.

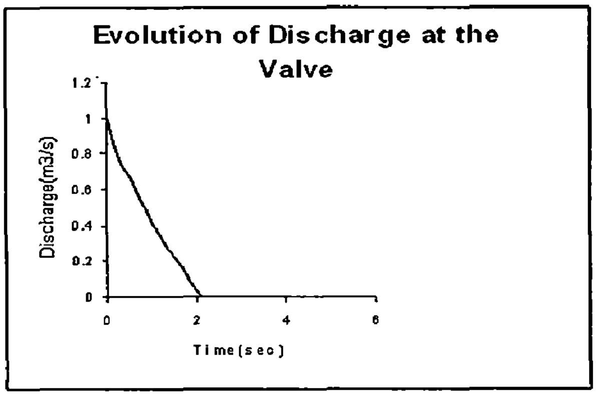

Fig5.1(b)

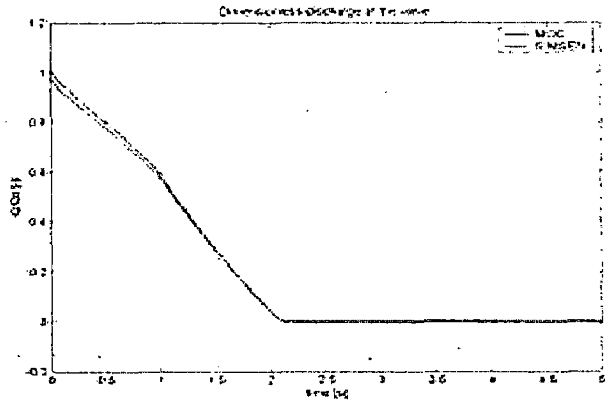

Fig. 5.1(b1)[15]

Fig. 5.1(b), shows the discharge through the valve. As the valve being closed discharge through it decreases from 1m3/s to 0m3/s during 0 to 2.1 sec and after 2.1 sec. discharge is zero. By comparing with Fig.5.1(b1), we observe that both curve are similar in nature.

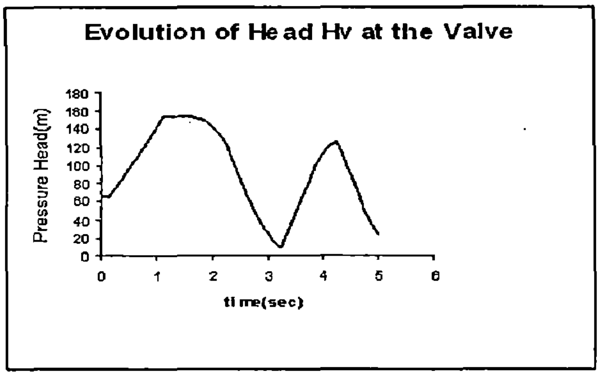

Fig.5.1(c)

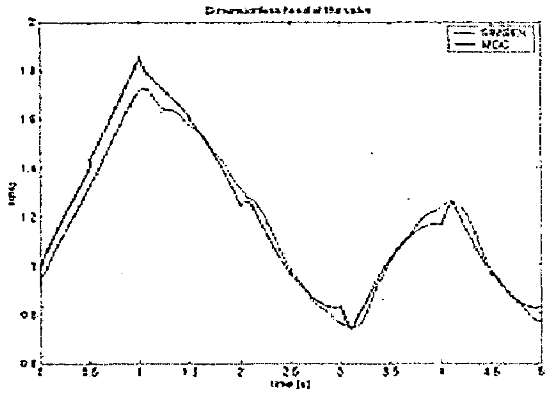

Fig. 5.1(c1)[15]

In Fig. 5.1(c), we observe that the pressure at the valve varied as the valve being closed. First it increases from 67.7m to 159.78m in 1 sec. linearly then its value approximately remains constant at 159.78m between 1 to 1.75 sec. After 1.75 sec it decreases from 159.78m to 14.45m by following a curve path till 3.25 sec. Again it increases linearly from 14.45m to 139.67m till 4.35 sec. and decreases to 21.34m in 5 sec. Due to the friction and head losses at each point in the penstock peak of the pressure head decreases as time as time increases. By comparing with Fig.5.1(c1), we observe that both curve are similar in nature.

5.2 INCLINED PENSTOCK PIPE

For the simulation purpose, we have taken a prototype model in Fig. 4.4 and Fig. 5.2(a), (b), (c). shows the results.

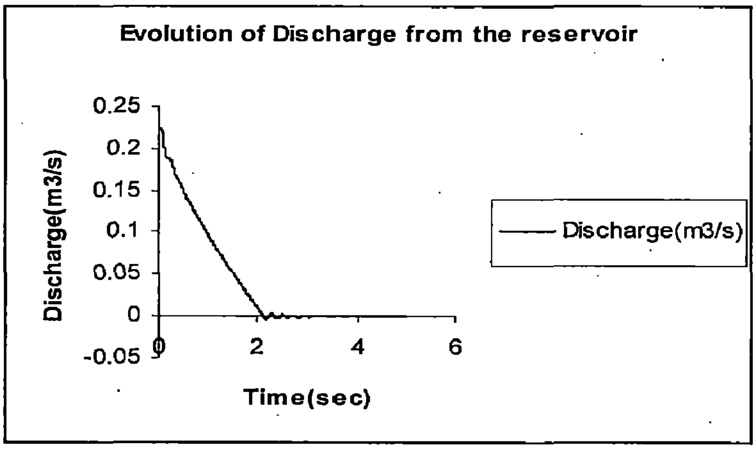

Fig. 5.2(a).

In Fig. 5.2(a).we observe that as the valve being closed, discharge through the reservoir decreases from 0.223m3/s to 0 m3/s during 0 to 2.1 sec. After that discharge is –ve i.e., the direction of flow is reverse because the pressure head at the valve is greater than the reservoir head. After 2.1 sec. an damped oscillation takes place till 3.25 sec due to the variation in head. After 3.25 sec discharge is zero because pressure at the valve and at the reservoir is approximately same.

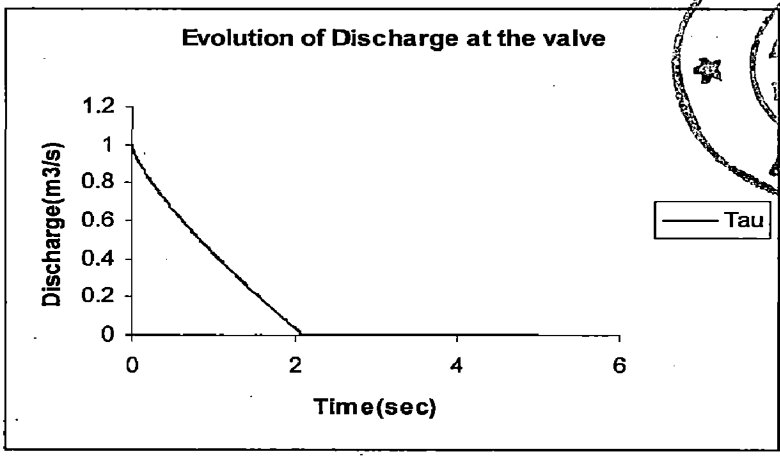

Fig.5.2(b).

Fig. 5.2(b), shows the discharge through the valve. As the valve being closed discharge through it decreases from 0.223m3/s to 0 m3/s during 0 to 2.1 sec and after 2.1 sec. discharge is zero.

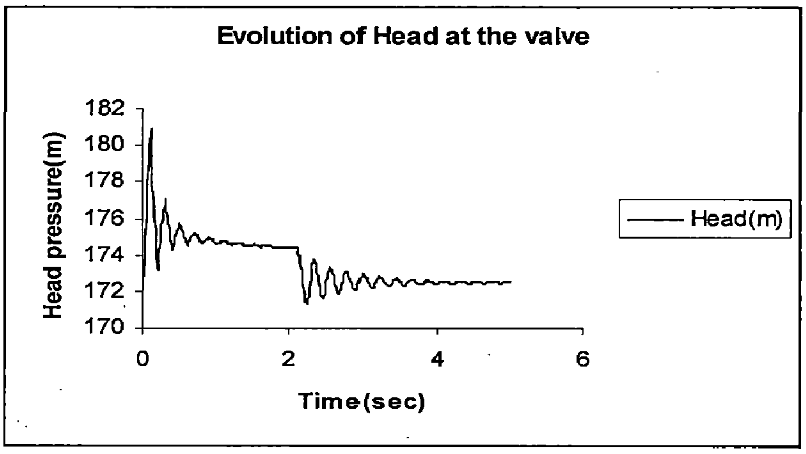

Fig. 5.2(c)

In Fig. 5.2(c), we observe that the pressure head at the valve varied as the valve being closed. First there is step rise in from 172.1342m to 180.8451m in 0 to 0.1125sec then its value decreases sharply to 174.0998m at 0.2125sec. Again it raises sharply 177.0567 in

0.3125 sec. and an damped oscillation takes place till 1.125 sec. Its value decreases linearly till 2.1 sec.Again an damped oscillation takes place till 5 sec with final value 172.5042m. We observe that peak of the pressure head decreases as time increases. It is only due to the friction and head losses at each points. In this simulation an error of 0.215% occurred.

The above simulation results gives information about water hammer effect in the in the system which helps in designing penstock thickness, diameter and other parameter . So that it can withstand the sudden rise in pressure without being damage.

5.3 TURBINE DURING LOAD REJECTION

The graph for turbine speed with respect to time during load rejection have been obtained and significant results have been pointed out so as to, better control and stability of hydraulic system in hydropower plant.

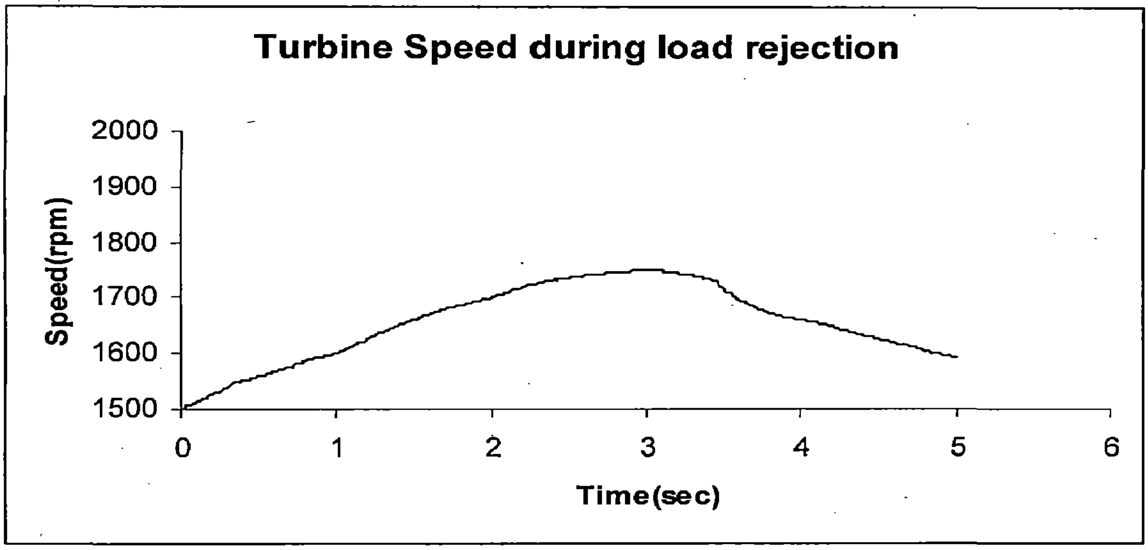

Fig.5.3: Turbine speed during load rejection.

From Fig. 5.3, we observe that during load rejection the turbine speed first increases after attaining a certain value, it start decreasing. During the period of 0 to 3 sec. speed increases because the mechanical power output of turbine is greater than the electrical power output of generator. After that speed decreases because the mechanical power output of turbine is lesser than the electrical power output of generator.

CONCLUSION

This study give information about water hammer effect in the in the system which helps designing the various elements of the hydropower plant. A hydraulic model of penstock h been used and the analysis of this model permits the determination of flow, pressure head a pressure oscillations in some point of the system.. A hydraulic turbine model has been us not only as a characteristic valve, but also as a turbine whose characteristics curve a coefficients were extracted from the test model. The results obtained through simulati when confronted with results obtained from examples literature, prove the accuracy of t models, in which the main merit is the accurate and fast response of simulation.

SCOPE FOR FUTURE WORK

Other hydraulic structures (such as Flume, Tunnel and Surge tank etc.) also can be added l appropriate changes in Programming. Options for governors can be increased by addi digital governor. Here we have considered only load rejection but other transient conditio like 3-∅ symmetrical fault (with and without impedance), other faults (i.e. L-N, L-L-N, L- faults), out of phase synchronization during start-up and emergency stop, can be simulat easily.

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