AIR CUSHION SURGE TANKS

A DISSERTATION
submitted in partial fulfilment
of the requirements for the award of the Degree
of
MASTER OF ENGINEERING
in
WATER RESOURCES DEVELOPMENT

By
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C E R T I F I C A T E

Certified that the dissertation entitled 'AIR CUSHION SURGE TANKS' which is being submitted by Shri Jose C. Peter in partial fulfilment for the award of degree of Master of Engineering in 'Water Resources Development' by the University of Roorkee, is a record of the candidate's own work carried out by him under our supervision and guidance. The matter embodied in this text has not been submitted for the award of any other degree or diploma.

This is further certified that he has worked for a period exceeding nine months from October 1975 to October 1976 in connection with the preparation of this dissertation.

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C O N T E N T S

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LIST OF FIGURES

S Y M B O L S

a	=	Propagation velocity of water hammer wave	m/sec
Aa	=	Horizontal area of closed surge chamber (or Air Tank)	m^2
A_{or}	=	Cross section area of the orifice	m^2
A_o	=	Horizontal area of open surge chamber	m^2
At	=	Cross section area of tunnel	m^2
At_1, At_2	=	Cross section area of pipe (or tunnel) upstream and downstream of air cushion surge chamber	m^2
A_{Th}	=	Thoma area of surge chamber	m^2
A^*_a	=	Critical (Horizontal) area of enclosed surge chamber	m^2
A^*_o	=	Critical (Horizontal) area of open surge chamber	m^2
b, c	=	Coefficients of quadratic wave equation	
C_1, C_2	=	Wave Celerity in line upstream and downstream of air chamber	m/sec.
C_d	=	Coefficient of discharge	
D	=	Diameter of tunnel	m
E	=	Dimensionless factor depending on marginal turbine efficiency $\left(1 + \frac{u_o}{\eta_o} \cdot \frac{\Delta \eta}{\Delta u}\right)$	
f	=	Friction factor	
F	=	Dimensionless factor $\left(1 + \frac{n \cdot p_o}{\gamma \cdot l_o}\right)$	
g	=	Acceleration due to gravity	m/sec^2
h_c	=	Height of water in air chamber	m

h_f	=	Head loss due to friction	m
h_{fc}	=	Head loss in air chamber entrance	m
h_{or}	=	Head loss at orifice	m
h_r	=	Height of water in reservoir	m
h_{max}	=	Height of air chamber (hc + 1)	m
H	=	Gross head on power plant	m
H_o	=	Net head on the turbine	m
HC, HC_o, HC_1, HC_2	=	Pressure head in air chamber	m
H_{min}	=	HC_5^* - Maximum down surge adjacent to the pump	m
H_1, H_2	=	Pressure head in line	m
H_1, H_{11}, H_2, H_{22}	=	Magnitude of pressure waves	m
H_1^*, H_2^*	=	Absolute pressure head in pipe line	m
$HC^*, HC_o^*, HC_1^*, HC_2^*$	=	Absolute pressure head in air chamber	m
j	=	A fraction of time interval	m
k	=	Adiabatic constant = 1.4	
k_c	=	Air chamber entrance loss coefficient-lumped parameter solution	
K	=	$\dfrac{L_1 \cdot A_t}{g \cdot A_a \cdot HC_o^*} (v_1 - v_2)^2$	
K_c	=	Air chamber entrance loss coefficient distributed parameter solution.	
K_e	=	Entrance loss coefficient, line loss coefficient.	

K_1	=	Ratio of total power generated by the station to the grid	
K_2	=	Coefficient of head loss such that $K_2.H_1$* is the total head loss for a flow Q_1 in to the air chamber	
l, l_o	=	Length of air column in tank before load change	m
L	=	Length of tunnel	m
L_1, L_2	=	Line length upstream and downstream of air chamber	m
m	=	mass of water	
n	=	Polytropic constant	
n_o	=	normal speed of rotation of the turbine	r.p.m.
n_{max}	=	maximum speed of rotation with the transient	r.p.m.
P, P_o	=	Instantaneous pressure in air chamber	kg/m^2
P_a	=	Atmospheric pressure	kg/m^2
P_t	=	Power output of the turbine	
P_1, P_2	=	Absolute pressure in air chamber at time t_1 and t_2	kg/m^2
Q	=	Instantaneous discharge of the turbine	m^3/sec.
$Q_1, Q_2, Q_{11}, Q_{22}, Q', Q''$	=	Flow discharge in line	m^3/sec
QC_1, QC_2	=	Flow discharge in air chamber	m^3/sec
Q_{or}	=	Discharge through the orifice	m^3/sec

$\frac{dQ}{dt}$	=	Rate of heat outflow from the air mass	
t_1, t_2	=	Time	Seconds
Δt	=	incremental time	seconds
T_1, T_2	=	Temperature	degrees
T_λ	=	Penstock constant = $\frac{\Sigma L_i \cdot V_i}{g \cdot H_o}$	
v	=	Instantaneous velocity of water in tunnel	m/sec.
v_1, v_2	=	Velocity of water before and after load change	m/sec.
v'	=	Velocity of water downstream of surge chamber	m/sec.
V	=	Instantaneous volume of air cushion	m^3
ΔV	=	Incremental air volume change	m^3
V_1, V_2	=	Air volume in chamber at time t_1 and t_2	m^3
V_{max}	=	Total volume of air mass	m^3
y	=	Maximum rise of water in air tank	m
Z	=	Water level in surge chamber taken positive downwards from the water level in the Intake basin.	m
Z^*	=	Surge height corresponding to change in discharge neglecting friction and orifice losses and is given by = $v\sqrt{\frac{L}{g} \cdot \frac{At}{A_o}}$	m
α	=	Slope angle of headrace tunnel	Radians
β	=	Coefficient of friction = $\frac{h_{fo}}{v^2}$	Sec^2/m
γ	=	Specific weight of water	kg/m^3
η	=	Efficiency of the turbine	

$\tan\theta = \frac{\Delta \eta}{\Delta P_t}$ = Negative tangent of efficiency power curve.

ϕ = Transient irregular operation of the turbine

$$= \frac{n_{max} - n_o}{n_o}$$

ρ^* = Pipe line charactoristic $\frac{a \cdot V_o}{2\, g.\, H^*_1}$

σ^* = Air chamber charactoristic

$$= \frac{2g.\, V_o.\, H^*_1}{A.L.\ \ V_o^2}$$

<u>NOTE</u> : Subscript 'o' indicates steady state condition, if not, specified.

Subscript 'n' indicates the number of time intervals under study.

S U M M A R Y

Air Cushion Surge Tank is a closed surge tank which works on the principle of compressibility of gases. When air volume increases in the chamber, the pressure decreases by a certain extent and vice versa, following some definite rules.

The importance of surge tanks and the methods for eliminating them from high head plants are briefly discussed. The role which the air cushion surge tanks are likely to play in the coming decades is explained in short.

The conventional type surge tanks viz., simple, restricted orifice, differential and special types have also been discussed briefly.

Air cushion surge tank, its principle, advantages and disadvantages etc., are discussed. A historical review of this development has also been given.

A transient flow analysis with distributed parameters and lumping the same has shown that the magnitude of the short term surges does not depend on the initial volume of air in the chamber and that it is only a function of the resistance to flow at the entrance to the chamber. Hence the size of the entrance orifice of the chamber may decide the attenuating characteristics of the air chamber. Though it has not been possible to find the exact air behaviour in the chamber, it seems that the air chamber behaviour is

polytropic and the value of the exponent 'n' lies between those of adiabatic and isothermal.

The hydraulic design of the Air Cushion Surge Chamber, assumptions, limitations, etc. have been discussed and the critical area of the Air Cushion Surge Tank found out.

The prototype behaviour of the Air Cushion Surge Tank provided at Driva Power Plant in Norway is highlighted. The Salient features of the project, the reasons for providing the air cushion surge tank, the behaviour of the air chamber, practical problems encountered etc. are discussed in brief.

An economic evaluation of a scheme with Air Cushion Surge Chamber is attempted. Relevant data has been taken from the recently commissioned Idukki Hydro-electric project of Kerala State. The Water Conductor system was remodelled to suit the air cushion surge Chamber. The analysis has indicated that there will be much economy if this novel idea of air cushion surge chamber is adopted.

Concluding remarks and the suggestions for future research work form the last two chapters of the dissertation.

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CHAPTER-I

INTRODUCTION

A surge tank acts as a reservoir releasing energy for meeting the immediate demand of turbines at sudden gate opening, and for transforming the kinetic energy to potential at closure, and thereby reduces the amplitude of the pressure waves. The waves are partially reflected by the Surge Tank, which therefore protects the headrace tunnel effectively.

The great number of scientific papers dealing with surge problems in recent years are an indication of the growing importance of the surge tanks in modern hydro electric projects. Conventional type surge tanks are most popular now-a-days. But sometimes such a development becomes too costly and creates technical problems. Hence in a few cases the surge tank has been eliminated from the system by increasing the flywheel moment, the response time and the constant inertia of the pipe line and by introducing minor power restrictions.

Slowing the closure of guide vanes, however, does not always completely solve the problem since, in the case of very long penstocks, the unit may reach full run-away speed after shedding the load. A substantial increase in the regulation time of hydraulic units connected to the power system is acceptable in principle, but requires proper justification in each case in order to provide the necessary run up to power and dynamic stability of the sets in the system. In addition the operating conditions of the set should be assessed each time when it is disconnected from the net work, keeping in mind the fact that such disconnection will always entail its wake, a big spurt into speed of rotation.

Idle discharge had been previously used to reduce the water hammer effects. In practice, this solution is some-times unreliable due to non-opening of the pressure relief valves etc. and result in water losses. It is a fact that the idle discharge valve does not open with load build up in the unit and consequently there is some times a risk of relatively big magnitude of water hammer pressures. In such valves, water tightness is also a real headache.

Provision of Air Cushion Surge Tanks for hydro electric plants is a novel idea, which requires more attention on the part of water resources engineers. This technique is still in its infancy. Driva Power Plant in Norway is the only Power Station functioning with an Air Cushion Surge Chamber at present.

Air Cushion Surge Tank has got a bright future and may soon become very popular for underground developments, because of its inherent advantage of economy as well as quicker damping effect. Air Cushion Surge Tanks may be suitable for high head power projects like Punnar Hydro-Electric Scheme in Kerala State and Dibi-Bokhri Nakthan tunnel under Parbati Hydro-Electric Project (Stage-II) in Himachal Pradesh, where underground power stations are proposed. It is highly desirable that a few hydro electric schemes in the country are provided with Air Cushion Surge Tanks. Then only the complexities involved in various aspects of this new technological development will be clear to Engineers, which will lead to a new era in the history of Hydro-Power Development.

CHAPTER-II

CONVENTIONAL TYPE SURGE TANKS

2.1. DEFINITION

A Surge Tank or a surge chamber is an artificial reservoir introduced along the pressure conduit system at a suitable point upstream and/or downstream of a hydro power station fed by a long pressure conduit. The oscillations of the water levels in the surge chamber due to change in loads are damped by the frictional resistance in the conventional type Surge Chamber.

2.2. NECESSITY

In Hydro-Electric installations, where the water is brought to the machine by long pressure conduits, considerable inertia effects arise from the large mass of water in motion. The mass is of such magnitude that considerable force is necessary to accelerate or retard it. When the flow in a pipe line changes abruptly by operation of downstream control device, the dynamic energy of the water is converted into elastic energy and a series of positive and negative short period pressure waves travels back and forth in the pipe until they are damped out. This is known as water hammer, and may induce considerable stresses in the conduits. The propagation of the waves into a headrace tunnel, particularly sensitive to this type of waves may cause serious problems. Water hammer effects will result even from partial load changes, if the resulting turbine guide vane movements are rapid, which is essential to check undesirable speed rise.

The pressure rise resulting from water hammer on sudden closure of turbine guide vanes can be limited by the use of relief valves or similar devices, but they cannot help in accelerating the water column on increase of load i.e. on opening of the guide vanes. Therefore, in a long pressure conduits system, it frequently becomes necessary to introduce a surge chamber at a suitable point.

2.3. FUNCTIONS

The function of a surge tank is two fold. Firstly, the Pressure Conduit connecting the turbines to the reservoir is expediently interrupted by the tank to intercept the pressure waves due to water hammer at the free water surface thereby exempting the pressure tunnel from excessive pressures. Secondly, the surge chamber serves as a storage tank in the case of load rejection and a source of water supply in case of load demand. With a reduction or rejection of load the Surge Chamber acts as a relief valve in which the main conduit flow is partly or wholly diverted. The water level in the Surge Chamber, therefore, rises until it exceeds the level in the main reservoir, thus retarding the main conduit flow and absorbing the surplus kinetic energy. In case of starting up or on increasing load, the chamber acts as a reservoir to provide sufficient water to enable the turbines to pick up their new load safely and quickly, and keep them running at the increased load until the water level in the surge tank has fallen below the original level thereby creating sufficient head to accelerate the flow of water in the conduit until it is sufficient to meet the new demand.

2.4. DESIGN CONSIDERATIONS

2.4.1. Surge Tank Area

To ensure the hydraulic stability of the surge tank, its area should be governed by Thoma criterion. According to Thoma, the minimum area of the surge tank is given by the equation.

$$A_{Th} = \frac{L.\ At}{\beta \cdot v^2 H_o} \cdot \frac{v^2}{2g} \quad \ldots \quad (2.1)$$

Where

A_{Th} = Thoma area of Surge tank in m^2

H_o = Net head on the turbine in metres.

L = Length of the headrace tunnel in metres.

v = velocity of flow in the head race tunnel in metres per second.

β = Friction coefficient such that $h_f = \beta\ v^2$

In Equation-2.1 minimum value of β should be used.

If the power station is always to operate in a grid, the stability effect of the grid may be taken into account and the area of surge tank in that case may be worked out as follows :

$$As = A_{Th}\ (\ 1 - (3/2)\ (1 - K_1)) \quad \ldots \quad (2.2)$$

Where

A_s = Area of surge tank

K_1 = ratio of total power generated by the station to that of grid.

As the area of surge chamber given by Eq.2.1 and 2.2 is the theoretical minimum, it is usual to adopt a certain factor of safety. The usually recommended factors of safety are 2.0 for simple surge tank and 1.5 for restricted orifice and differential surge tanks.

2.4.2 Computation of Surge heights;

The surge tank should be designed to accommodate the maximum and minimum water levels anticipated under worst conditions. Normally the worst conditions to be considered for maximum upsurge are the following :-

(a) Simple load change : Full load rejection at maximum reservoir level, assuming minimum friction in the pressure tunnel (Example 100% to 0%)

(b) Combined load change : Specified maximum load acceptance followed by full load rejection at the instant of maximum positive velocity (flow from the reservoir towards surge tank) in the headrace tunnel at highest reservoir level. This condition should be tested both at minimum as well as maximum friction in the conduit (Example 50%-100%-0%, if 50% load acceptance is permitted).

For getting the minimum downsurge level the following worst conditions should be considered:

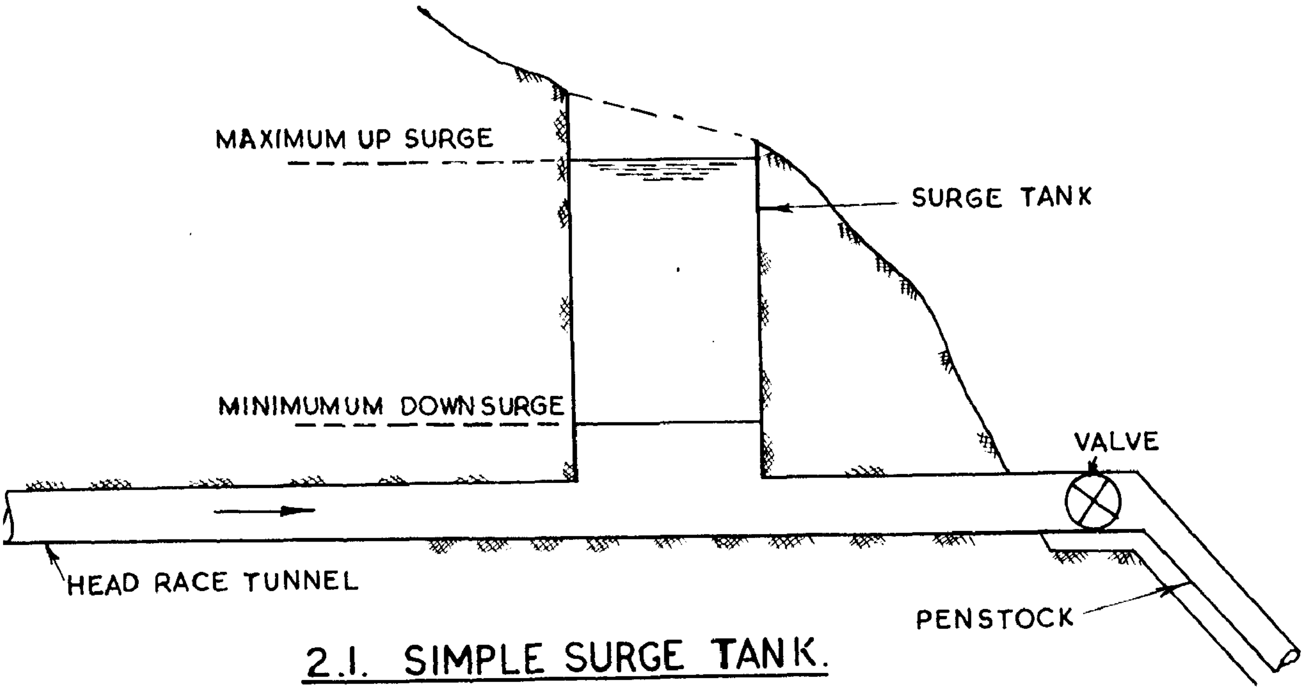
(a) Simple load change : Specified maximum load acceptance at load or speed-no-load condition at the minimum reservoir level assuming maximum friction in the conduit (Example: 0%-50% or 50%-100%, if specified maximum load acceptance is 50%).

(b) Combined load change : Full load rejection at minimum reservoir level followed by specified maximum load acceptance at the instance of maximum negative velocity in headrace tunnel (flow from surge tank towards the reservoir). This should be tested both at minimum and maximum friction in the conduit (Example 100%-0%-50%).

2.5. <u>TYPES OF SURGE TANKS</u>

2.5.1. Simple Surge Tank :

The most simple type of Surge Chamber is a plain cylindrical shaft or tank (Figure 2.1). It is usually connected to the pressure conduit by a short connecting conduit or port, the area of which is equal to or greater than that of the pressure conduit. The diameter of the shaft is governed primarily by the necessity of making the area sufficient to ensure stability and, secondly, by the necessity of keeping the surge within reasonable limits of amplitude. It will be found in general that stability will determine the diameter for low heads with short conduits, while limitation of surge amplitudes will govern those with high heads and long conduits.



2.1. SIMPLE SURGE TANK.

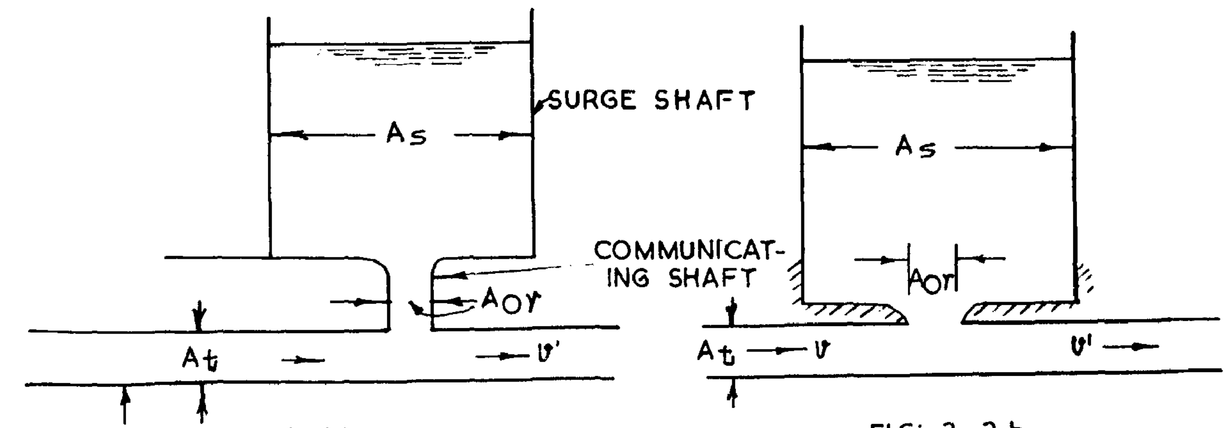


FIG: 2.2a

FIG: 2.2b

FIG. 2.2. RESTRICTED ORIFICE SURGE TANKS.

The action of the simple surge chamber is sluggish as compared to other types. The simple tank requires greater volume and is more expensive. But it is ideal, as far as ease of governing is concerned in that the head changes are so gradual that even a very slow acting governor has no difficulty in following the changes of pressure.

2.5.2. Restricted Orifice Surge Tank

The main feature of the restricted orifice or throttled surge tank is the provision of a restricted orifice installed in between the conduit and the tank. The object of the orifice is to create an appreciable friction loss when the water is flowing to or from the tank. Schematic sketches of such types of surge chambers are shown in Fig.2.2.

When load is rejected by the turbine, the surplus water passes through the restricted orifice and immediately a retarding head equal to the loss due to the restricted orifice is built up in the conduit. Under conditions of load acceptance by the turbine, the orifice tends to develop an accelerating head in the conduit more quickly than it would be developed in a simple surge chamber.

As the magnitudes of surges mainly depend upon the resistance offered by the orifice, it is essential to decide the size and shape of the orifice.

The discharge through an orifice is given by the formula

$$Q_{or} = Cd. \ A_{or} \sqrt{2g \ h_{or}} \qquad ....(2.3)$$

or

$$h_{or} = \frac{Q^2}{C_d^2 \cdot A_{or}^2 \cdot 2g} \qquad ....(2.4)$$

Where

A_{or} = Area of the orifice

C_d = Coefficient of discharge, usually varies between 0.6 and 0.9

h_{or} = head loss through the orifice

Q_{or} = discharge through the orifice

There are two criteria for fixing the area of the orifice. The first one is that the area of the orifice should be such as to satisfy the condition given by Calame and Gaden[37].

$$\frac{Z^*}{\sqrt{2}} + \frac{1}{4} h_f \leq h_{or} \leq \frac{Z^*}{\sqrt{2}} + \frac{3}{8} h_f \quad ....(2.5)$$

where

Z^* = Surge height corresponding to change in discharge neglecting friction and orifice losses and is given by

$$= v_i \sqrt{\frac{L}{g} \cdot \frac{A_t}{A_s}} \qquad ....(2.6)$$

Where

A_p = Horizontal area of open surge chamber.

A_t = Area of Cross section of the headrace tunnel

g = acceleration due to gravity.

h_f = Head loss due to friction

v = velocity of flow

The second criteria for the orifice area is that it should be kept such that the pressure on the tunnel due to water hammer caused by total load rejection is approximately equal to the pressure due to maximum rise of water level in the surge tank at the time of worst upsurge.

Regarding the shape of the orifice, any shape can be adopted, but a circular shape is preferable. If the gates are to be provided in the surge tank, the gate slots usually function as orifices and the additional area, if any, can be provided in the form of a circular hole at the top of the tunnel in the riser. Suitable stream linings should be provided at the top and bottom of orifices. Model tests are desirable to determine the required stream linings and the coefficient of discharge for entry and exit.

The more quickly the accelerating and retarding heads are applied, the more effective will be the surge chamber in the adjustment of the conduit discharge, hence less water will have to be stored in or delivered from the tank, and the tank may be smaller. The rapid creation of accelerating and decelerating heads by the restricted orifice surge chamber develops sudden fluctuations of head on the turbine, and thus complicates problems in connection with the governor mechanism. Speed control is also accomplished through the inertia of the rotating parts of the turbine and the generator. When the governor sensitivity requires the addition of this inertia to the machines, the cost of such addition may preclude the use of a restricted orifice surge chamber. On account of sudden pressure changes in the restricted orifice surge chambers, this can not be adopted for many installations where close governing is required.

2.5.3. Differential Surge Tanks

Differential surge tank is a throttled surge tank to which is added a riser pipe (Fig.2.3). The riser is usually central, but may be arranged on one side of the throttled shaft. The latter arrangement was adopted at Innertkirchen (Fig.2.4) where the construction of the inclined pressure shaft made a separation more economical. For the central riser arrangement, the riser is connected to the outer chamber by ports at its base. On change of load the water level rises or falls very rapidly in the riser, thus producing rapid deceleration or acceleration of the conduit flow, while the water level in the outer chamber moves more slowly and thus lags behind that in the riser. Though rapid in its action, the differential chamber gives reasonably low pressure rises and surges of limited amplitude.

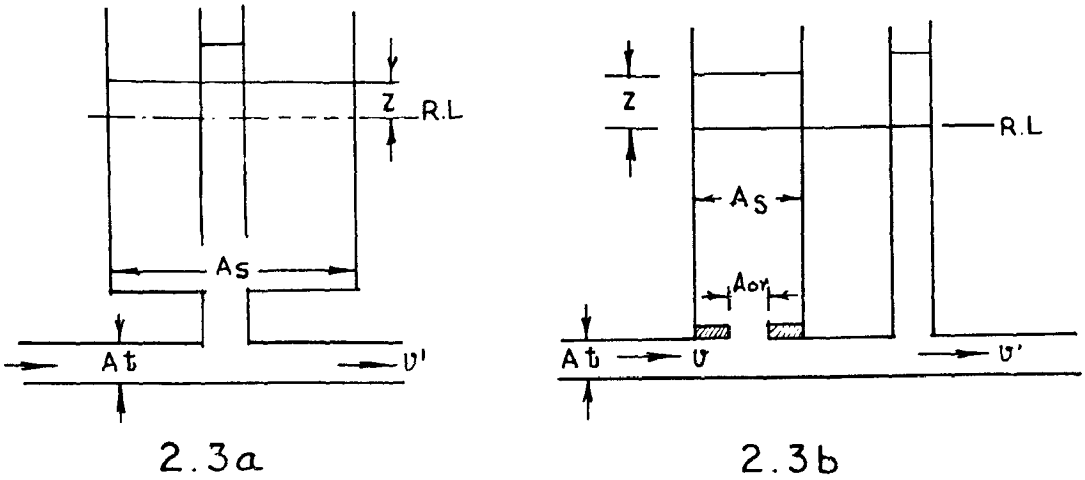


FIG. 2.3 DIFFERENTIAL SURGE TANKS.

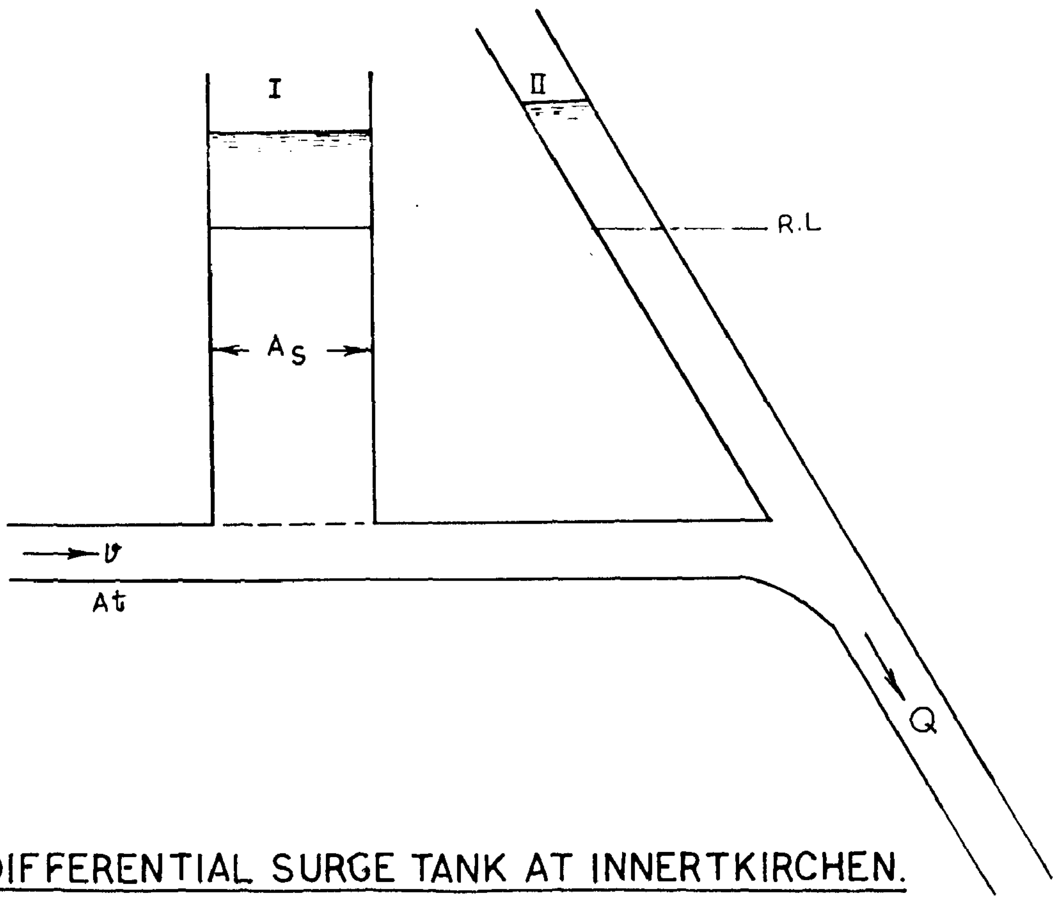


FIG. 2.4 DIFFERENTIAL SURGE TANK AT INNERTKIRCHEN.

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The magnitude of surges in the differential surge chamber will depend on the design of riser and orifice. The area of the riser should not be less than 3/4th of the area of the pressure conduit to avoid rapid changes in the water level and facilitate proper governing of machines. The height of the riser should be so chosen that the maximum level of the water spilling over the riser is less than or equal to the maximum surge level in the main shaft.

The orifice is primarily designed to supply water in case of specified load demand. In case of rejection, the area of the orifice may be kept as small as possible because the upsurge is restricted by the riser. The area of the orifice can be determined by the Eq.2.3.

The action of the differential surge chamber is similar to that of the restricted orifice Surge Chamber except that the initial pressure change and head on the turbine, instead of occurring instantly as in the case of restricted orifice surge chamber, or very slowly as in the case of simple chamber, occur quickly enough for efficient functioning of the chamber and are still spread over a period long enough to enable the governors to adjust the turbine gates to compensate for the change in head.

The salient features of differential surge chambers thus are :

a) Separation of the water supply or water storage function from the conduit acceleration or deceleration function, resulting in more rapid and efficient hydraulic action and reflecting sizeable economy in chamber diameter and capital investment.

b) Throttling action offered by the port arrangement giving the differential chamber pronounced ability to limit and suppress the surges due to synchronous load pulsations.

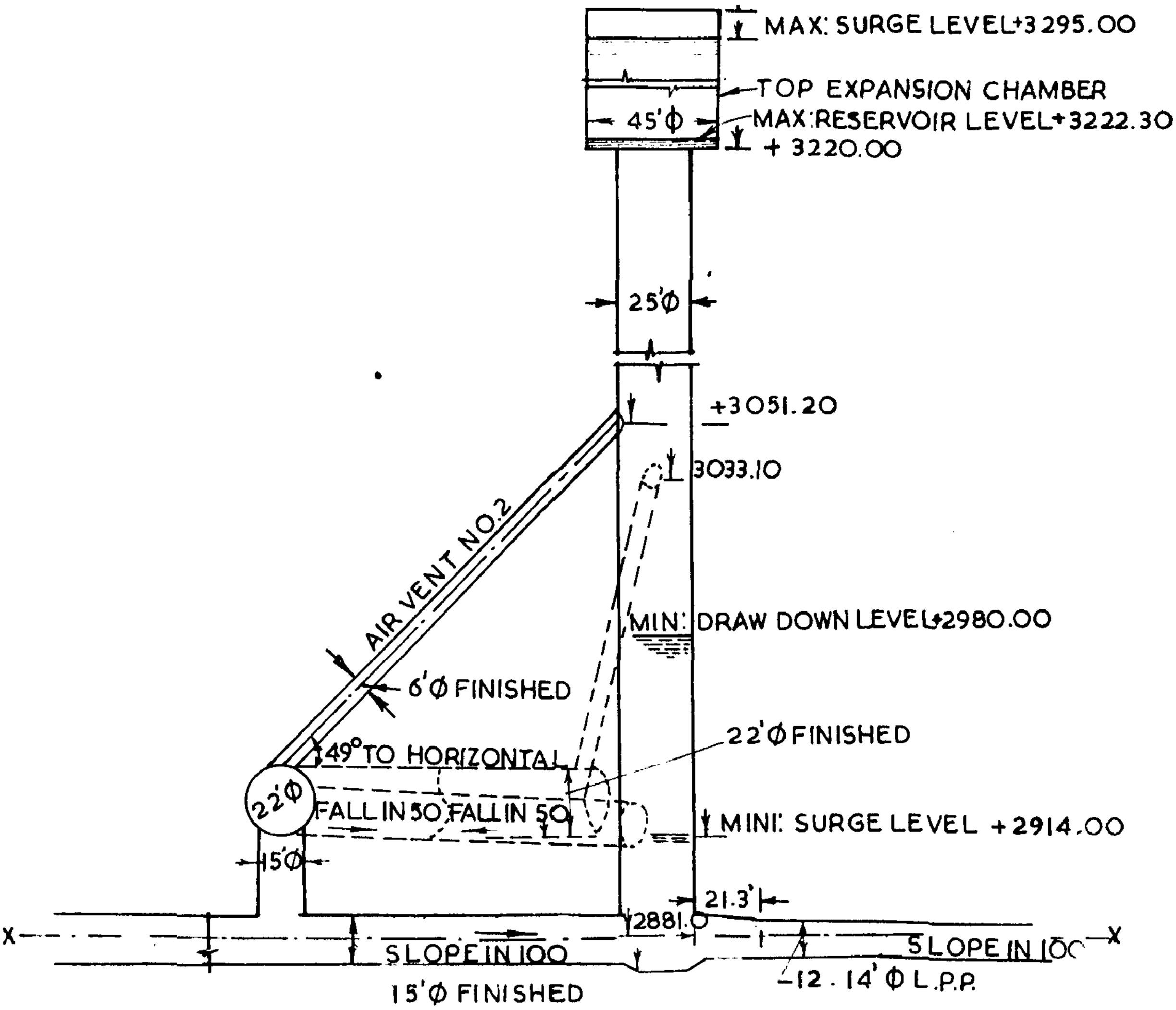
2.5.4. Special type surge tanks

2.5.4.1. Surge Chambers with expansion galleries/chambers

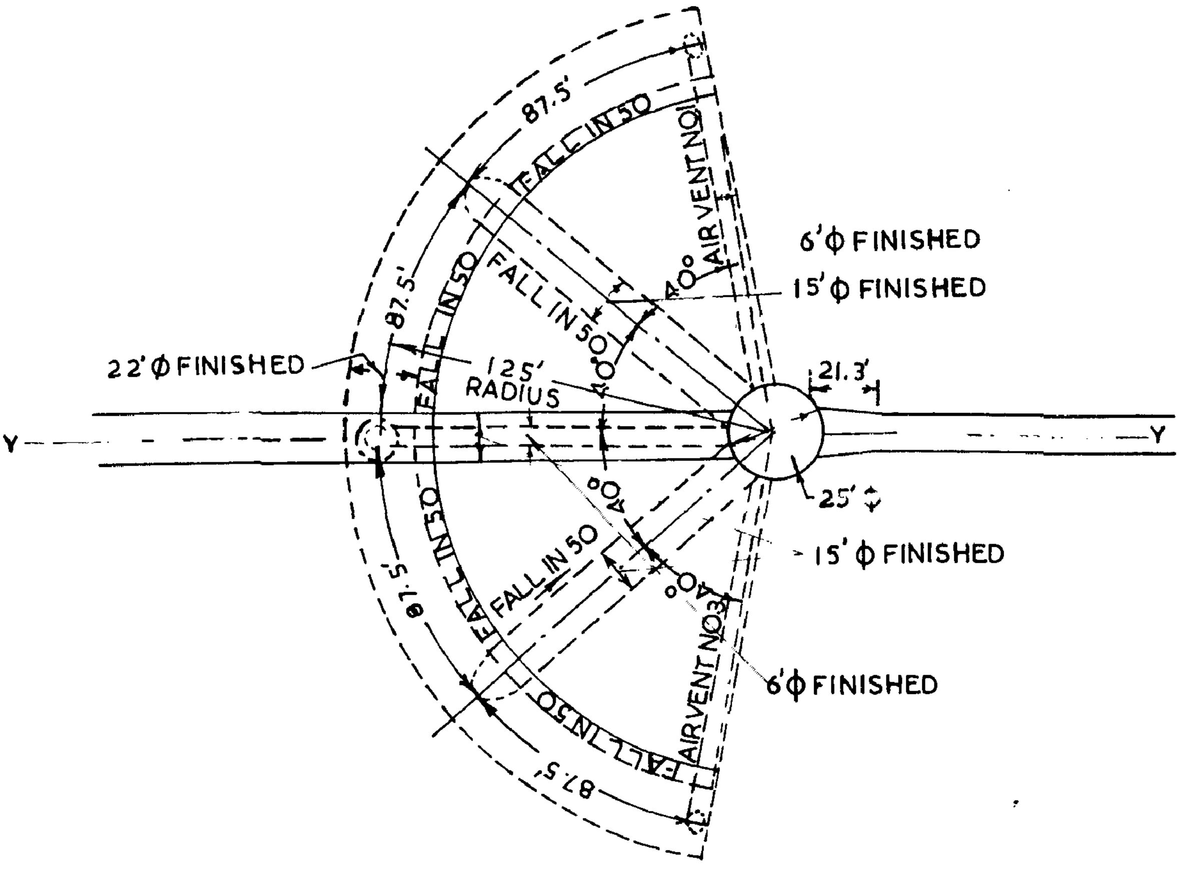
A large cylindrical (simple) surge tank will always provide an effective reduction of water hammer, thereby effective protection of the headrace tunnel and good stability by limiting the water level fluctuations, but can be expensive and of slow damping rate. Economy can be achieved by concentrating large volumes at proper elevations by means of expansion chambers or galleries (Fig.2.5) Fast damping can be ensured by increasing the losses by means of a restricted orifice (Fig.2.6) or by delaying the release of part of the accummulated energy(differential chamber). Lower and Upper expansion galleries can be provided along with any type of surge chambers. The expansion galleries (or chambers) can be of any shape, and can be provided in any direction, according to topographical and geological conditions.

2.5.4.2. Surge Chambers in Series or multiple surge chambers:

For economic reasons, some times more than one surge chamber can be provided in the system depending upon the topography. If topography permits, spilling arrangements



SECTION Y.Y



SECTION.X.X

FIG. 2.5 SABARIGIRI SURGE SHAFT-KERALA

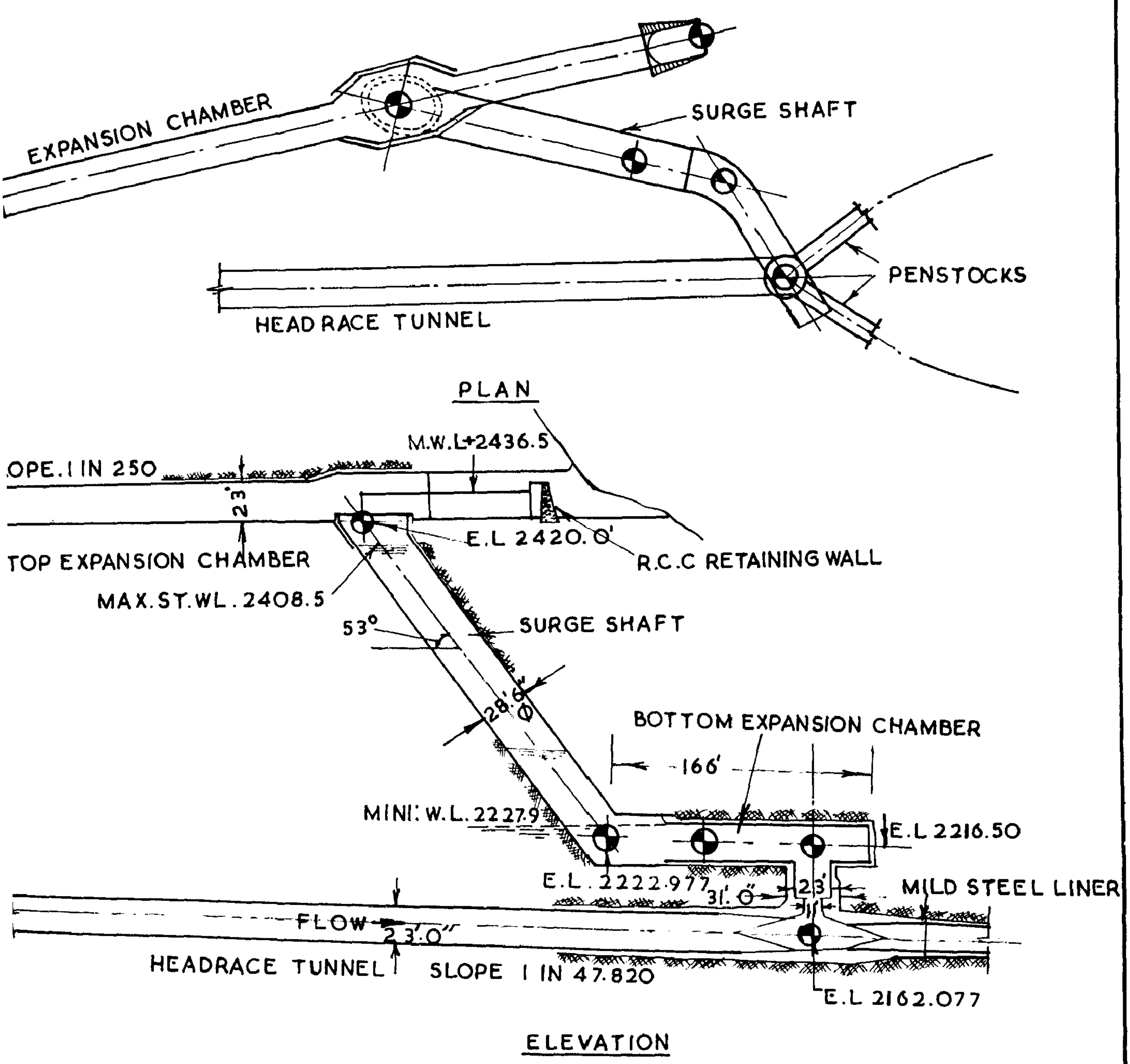


FIG. 2.6 IDUKKI SURGE SHAFT- KERALA.

can also be provided in a surge system along with any of the above developments, if one can afford occasional loss of water due to upsurge. This type of arrangement will help to reduce the volume of excavation of the surge chamber considerably. Khodri Surge Chamber under Yamuna Hydel Scheme Stage-II is a good example of twin surge chambers one of them having spilling arrangement (Fig.2.7). The first surge chamber which is connected to the pressure conduit can be of any type, but the other chambers are usually of simple type to accommodate the fast coming mass oscillations and hence the volume of water so released to avoid or reduce loss of water due to spilling over the chamber.

"Multiple Surge Tanks" or shafts is a term used to denote a system in which two or more tanks or shafts rise from one pressure tunnel (Fig.2.8). Usually this type of arrangement becomes necessary where the capacity of the Power Scheme is extended and the original surge tank would be too small for the enlarged scheme; or two shafts may be built where the diameter of the single equivalent shaft would be excessive (as occurred at Coblari in Italy), or where constructional advantages are obtained.

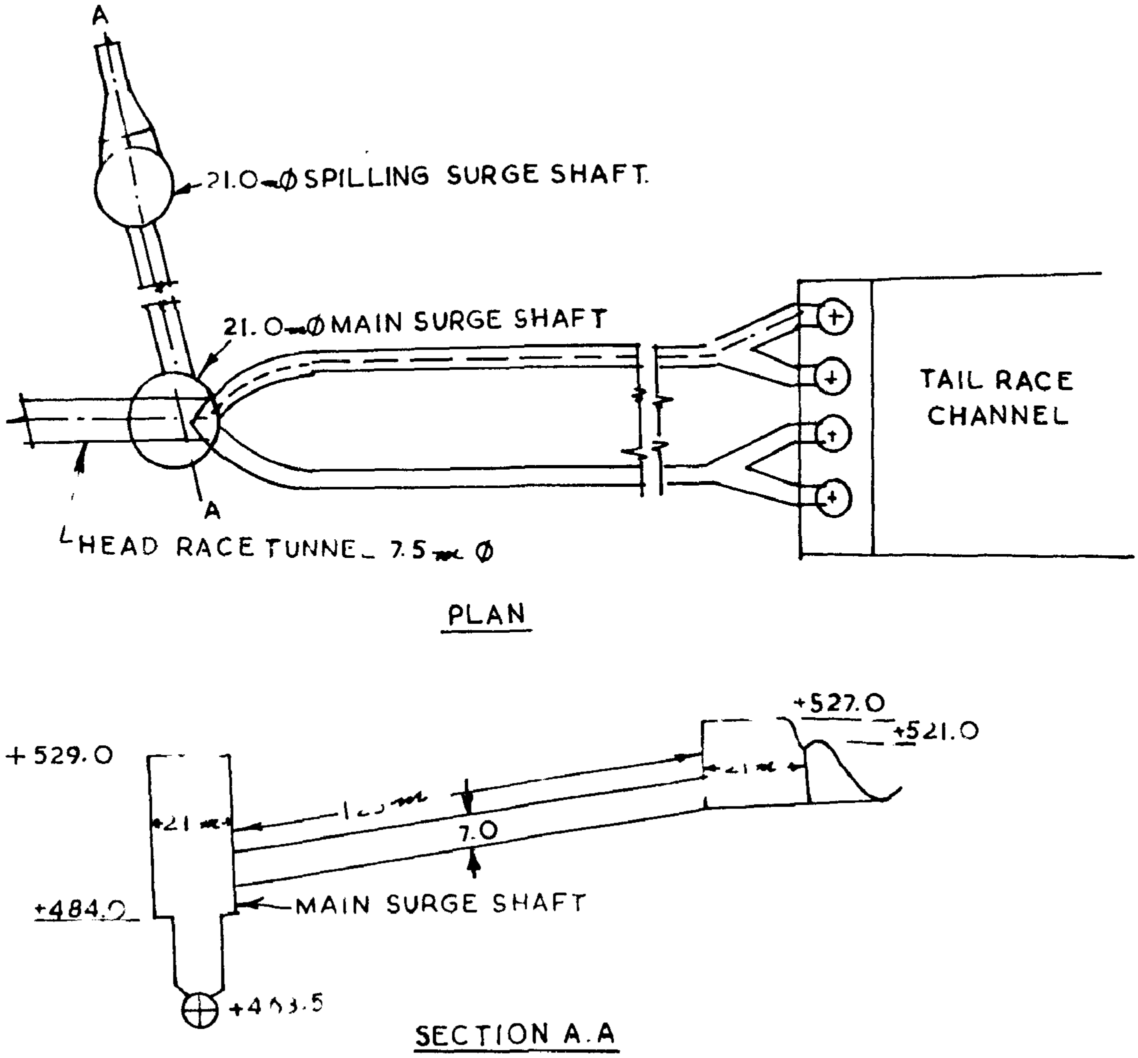


FIG. 2.7 LAY OUT OF KHODRI SURGE CHAMBERS-U.P

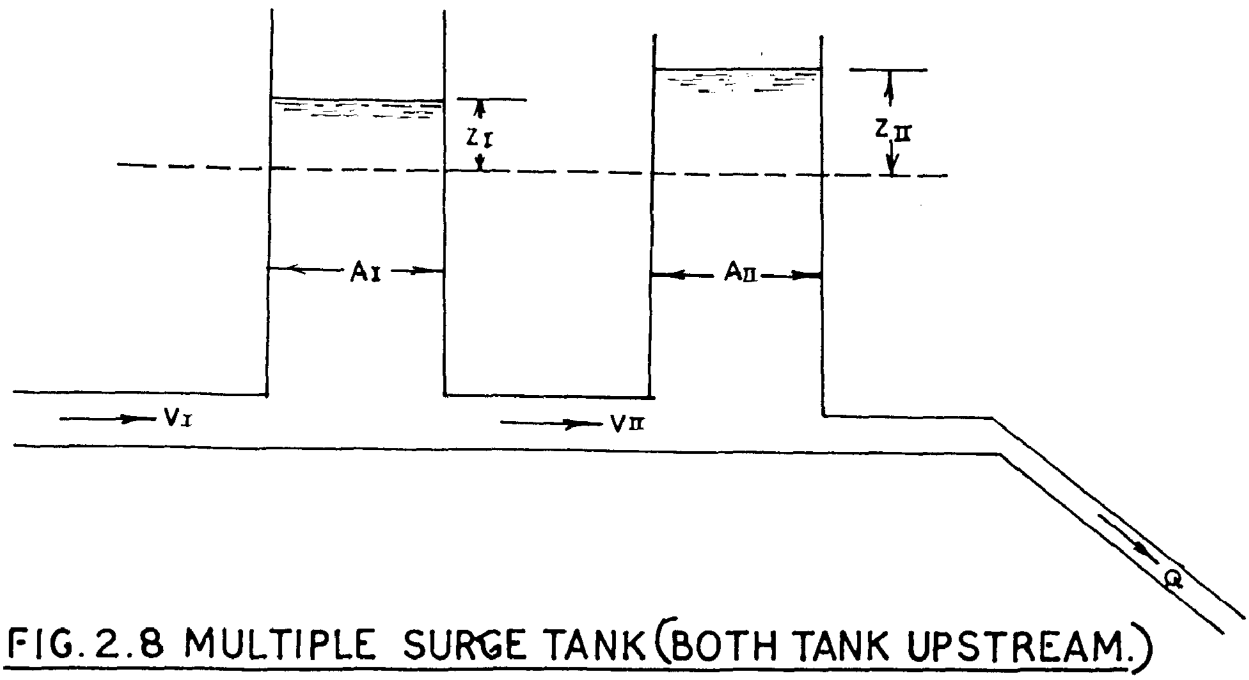


FIG. 2.8 MULTIPLE SURGE TANK (BOTH TANK UPSTREAM.)

CHAPTER - III

AIR CUSHION SURGE TANKS

3.1 DEFINITION

The conventional type surge tanks described in Chapter II are located in between the head race tunnel and the pressure shaft with its top either open to atmosphere (Figures 2.1 to 2.5) or connected to it through a suitable conduit (Figure 2.6). Such an arrangement will necessitate an adit at the top of the pressure shaft, the access to which may be difficult and costly.

Development in recent years has made it economical to excavate tunnels with slopes upto 1 in 8. Consequently, it seems attractive to replace the conventional horizontal (or fairly horizontal) head race tunnel and pressure shaft with one inclined tunnel running straight from the intake to the power station. This, however, requires a long shaft to reach the free water surface of a conventional open surge chamber.

In such cases, a surge tank with an enclosed air cushion offers an economic alternative to a conventional surge chamber. Such an alternative solution allows the distance between the turbines and the free water surface to be reduced considerably, thus reducing the problem of elastic pressure surges on the speed regulation. The schematic profile of a development with a conventional and an Air Cushion Surge Chamber is shown in Figure 3.1.

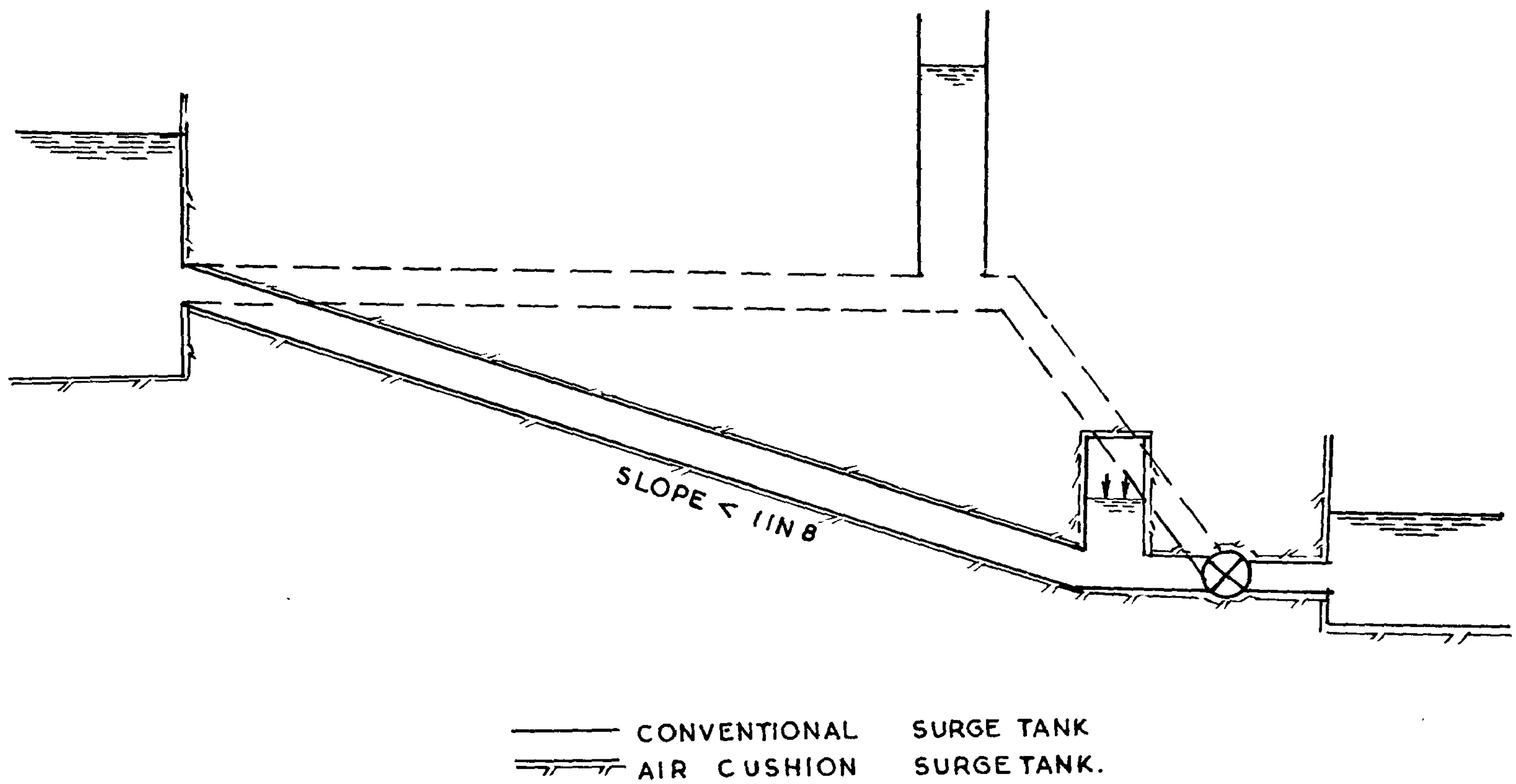


FIG.3.1. SCHEMATIC PROFILE OF CONVENTIONAL & AIR CUSHION SURGE TANKS.

An Air Cushion Surge tank can, therefore, be defined as a closed surge chamber filled partially with compressed air and located close to the Power Station in which the mass oscillations are mainly absorbed by the variation in the pressure of the compressed air.

3.2 PRINCIPLE

An air charged accummulator can be used effectively to reduce or eliminate pressure surges in flow systems. These devices can also be employed to control periodic pressure and flow fluctuations in flow systems. When the water level in the surge chamber has been disturbed by the change in load, the disturbance can be absorbed by the air chamber as the air volume can compress or expand adiabatically or isothermally. The mass oscillations in surge systems provided with air cushion surge chambers can be analysed as in the case of simple surge tanks with slight modifications.

The idea behind the air cushion surge tanks is nothing but utilisation of accummulated energy - not kinetic, but potential. For example in case of load rejection, the mass of water flowing in the pressure tunnel will start to discharge into the air chamber, raising the air pressure. The growing excess pressure, together with the friction drag along the tunnel walls, will exert a steadily growing retarding force on the flowing waters, until the water starts to flow back. The water level in the chamber will then oscillate with damped motion around a new equilibrium.

3.3 ADVANTAGES

(a) An air cushion surge tank eliminates the necessity of an opening at the top of surge tank or a suitable connecting conduit. Hence costly approach roads to that point can be eliminated.

(b) Where the rock surface is below the hydraulic gradient line, a conventional type of surge tank cannot be provided economically. In such cases, air cushion surge chamber provides the best alternative solution.

(c) By providing an air cushion surge chamber, the head race tunnel can be excavated at a steeper slope, say upto 1 in 8 with modern tunnelling techniques thereby reducing quantity of excavation considerably.

(d) As the Surge Chamber can be provided nearer to the power station, length of costly pressure shafts can be reduced considerably.

(e) For developing countries like India, there will be considerable savings in foreign exchange, by way of reducing or even eliminating the import of penstock steel.

3.4 DISADVANTAGES

(a) Necessary air compressors with sufficient number of stand-by units with remote control facilities have to be installed near the surge chamber to replenish possible loss of air due to leakage. Separate high pressure air receivers may also be necessary.

(b) There will be recurring expenditure (though it is negligible when compared to the savings in capital investment) for the maintenance and upkeep of compressors.

(c) If air chambers were provided without proper evaluation of the accummulator, they may yield totally unacceptable results, which may prove dangerous to the machines.

(d) The larger the air volume, the more efficient will it be for regulating purposes. But care should be taken always, not to fill the tank with air to such an extent that when a sudden load is thrown on, all the water in the chamber may be exhausted and the turbines may suck air. Therefore automatic safety arrangements need to be provided.

(d) The air content of an air chamber has to be maintained within acceptable results, by water level actuated switches to control the compressor plant, plus vent valves to release any excess air.

3.5 HISTORICAL REVIEW

Though the concept of air tanks on pipe lines for water wheel regulation was known to Engineers even seven to eight decades back, the literature on this subject is very limited even now. This is perhaps due to two reasons, firstly because of general belief that the problem is very complex and needs complicated mathematical solution, and secondly because of a popular idea that in order to be effective, the air tank must be so large as to be prohibitive in cost. However, as will be evident from the following discussion, neither of the two contentions is true.

WARREN[76] was the first to show authentically that if properly built, air tanks could be successful for regulation purposes on pipe lines, and had got a great practical value in improving regulation and preventing water hammer. He developed equations to find out the maximum/minimum water level and pressure rise in air tank, with the help of Newton's second law of motion and the physical laws governing the expansion and compression of air, and found that they were sufficiently accurate for most practical purposes. He also showed that the true values of 'y' and HC_1^* (Figure 3.2) would lie between the values obtained for isothermal and for adiabatic compression (or expansion) but would be nearer to the latter values.

He obtained the following equations for water level oscillations in the air charged surge chamber.

(1) Assuming isothermal compression or expansion

$$y = \sqrt{K.\ l_o + \frac{K^2}{4}} \mp \frac{K}{2} \qquad \ldots\ldots (3.1)$$

Where $K = \dfrac{L \cdot A_t}{g \cdot A_a \cdot HC_o^*} (v_1 - v_2)^2$

$$HC_1^* = \frac{HC_o^* \cdot l_o}{l_o \mp y} \qquad \ldots\ldots (3.2)$$

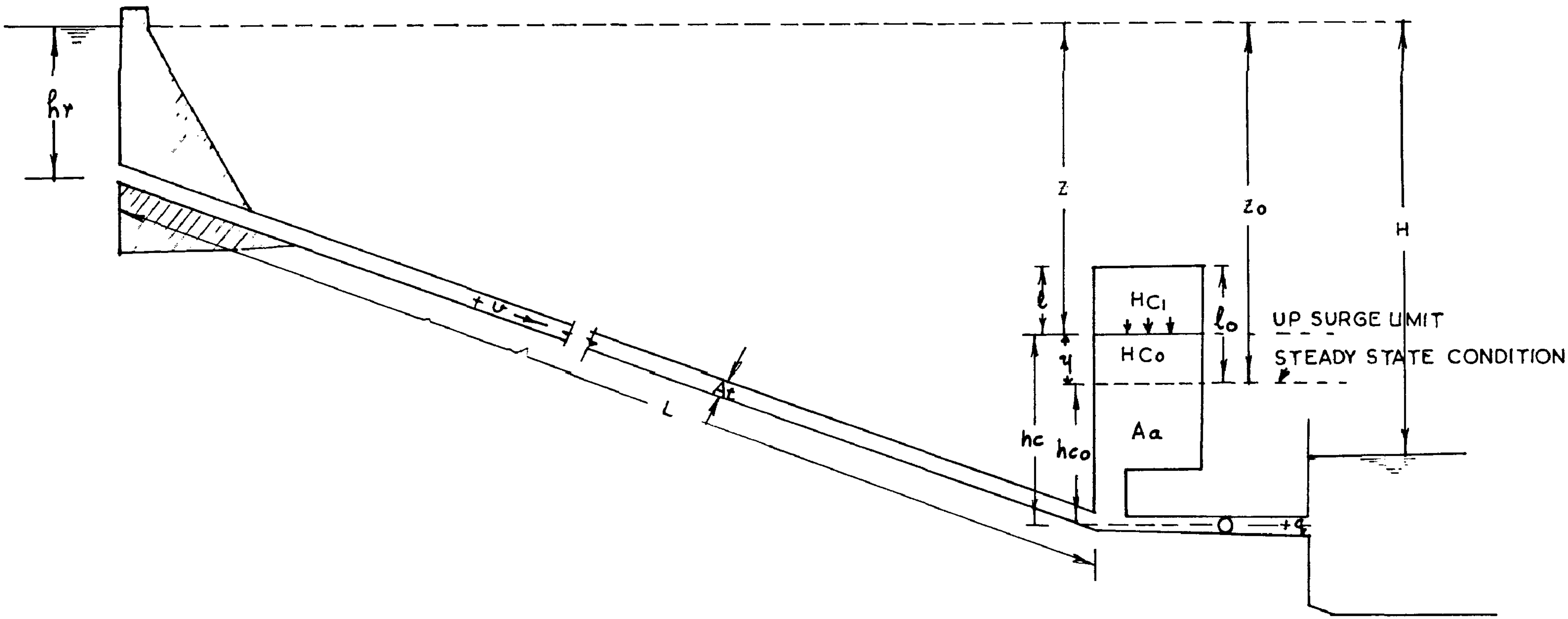


FIG. 3·2 LONGITUDINAL SECTION OF A DEVELOPEMENT WITH AN AIR TANK.

(ii) Assuming adiabatic compression or expansion

(1) For loads thrown off :

$$(l_o - y)^{1.4} = \frac{y \cdot l_o^{1.4}}{K + y} \qquad \text{..... (3.3)}$$

(iii) For loads thrown on,

$$(l_o + y)^{1.4} = \frac{y \cdot l_o^{1.4}}{K - y} \qquad \text{..... (3.4)}$$

The pressure rise can be expressed as

$$HC_1^* = HC_o^* \left(\frac{l_o}{l_o \mp y} \right)^{1.4} \qquad \text{..... (3.5)}$$

Where

A_a = Cross Sectional area of the air chamber in m^2

A_t = Cross sectional area of pipe (tunnel) in m^2

HC_o^* = Air pressure in air tank before load change including atmospheric pressure in m.

HC_1^* = Maximum or minimum air pressure in air tank including atmospheric pressure in m.

L = Length of pipe (or tunnel) between open reservoir and air chamber in m.

l_o = Length of air column in tank before load change in m.

v_1, v_2 = Velocity of water in pipe before and after load change in m/sec.

y = Maximum rise or fall of water in air tank measured from water level before load change in m.

(Where plus and minus sign appears, the plus sign is for load thrown on and the minus sign applies to load thrown off condition).

For deriving the above mentioned equations, the following assumptions were made :

(1) Pressure in tank rises or falls at a constant rate

(2) Water level in tank rises or falls at a constant rate.

(3) Water pressure due to change of water level in tank is neglected.

(4) Loss of head between tank and pipe line is negligible.

(5) Time necessary to open or close wheel gates is neglected

(6) Friction in pipe line is neglected.

(7) Time necessary for a pressure wave to travel the length of pipe is neglected.

These assumptions may result in cummulative error, which may defeat the purpose totally. Therefore, he revised his analysis after deleting assumptions 1 to 3 and obtained the following expressions :-

(i) Isothermal Compression or Expansion

(a) For loads thrown off:

$$y^2 - 2\,HC_o^* \left\{ l_o \log_e\left(1-\frac{y}{l_o}\right) + y \right\} = HC_o^* . K \qquad \ldots\ldots(3.6)$$

(b) For loads thrown on :

$$y^2 - 2\,HC_o^* \left\{ l_o \log_e(1+y/l_o) - y \right\} = HC_o^* . K \qquad \ldots\ldots(3.7)$$

(ii) Adiabatic Compression or Expansion

(a) For loads thrown off:

$$y^2 - 2\,HC_o^* \left\{ \frac{l_o}{0.4} - \frac{l_o^{1.4}}{0.4(l_o-y)^{0.4}} + y \right\} = HC_o^* . K \qquad \ldots\ldots(3.8)$$

(b) For loads thrown on :

$$y^2 - 2\,HC_o^* \left\{ \frac{l_o}{0.4} - \frac{l_o^{1.4}}{0.4(l_o+y)^{0.4}} - y \right\} = HC_o^* . K \qquad \ldots\ldots(3.9)$$

$$t = \frac{\pi}{2}\sqrt{\frac{L.\ A_a\ .\ y}{A_t . g . (HC_1^* - HC_0^*)}} \qquad \ldots\ldots (3.10)$$

Where t is the time required for y to reach a maximum CHURCH[18] modified the Eq.3.10 as below :

$$t = \sqrt{3}\sqrt{\frac{L.\ A_a\ .\ y}{A_t .\ g\ (HC_1^* - HC_0^*)}} \qquad \ldots\ldots (3.11)$$

In order to bring out the comparative variations between different equations, a specific case from an actual plant was analysed and the results were noteworthy. The difference in the results of the equations for adiabatic and isothermal compression was 14%, whereas the difference between the results obtained by the more approximate formulae (Equations 3.1 and 3.2) and the results obtained from Eq. 3.6 and 3.8 was from 3 to 6% only.

For loads thrown on, the use of Equations 3.7 and 3.9 resulted in a variation of 6% for the two relations, namely isothermal and adiabatic, and the more approximate Eq.3.1 and 3.4 gave results differing from Eq.3.7 and 3.9 by 3 to 4%.

Warren also pointed out that if a restricted orifice were inserted between the pipe line and the air tank, the air chamber will prove to be more effective.

JOHNSON[18] developed an equation for finding the critical size of a simple air tank by way of supplement to Warren's equations. His equation reads :

$$\frac{A_o .\ l_o}{HC_o^* + l_o} = \frac{A_t\ .\ L}{2g . \beta\ H_o} \qquad \ldots\ldots (3.12)$$

Where H_o = Net head on turbine corresponding to HC_o in metres.
β = Friction coefficient = h_f/v^2 in sec^2 per m.

This equation may yield absurd results, if A_a and l_o are comparatively small and (or) HC_o^* is large. Then the design becomes theoretically an impossible one.

If the negative tangent of the efficiency Power curve (i.e. $\frac{\Delta\eta}{P_t}$) where η is the efficiency of the turbine and P_t is the Power output, be called $\tan\theta$, then we get an expression

$$\frac{A_a \cdot l_o}{HC_o^* + l_o} = \frac{A_t \cdot L(\eta + 3/2\, P_t \tan\theta)}{2g \cdot \beta\, H_o \cdot \eta} \qquad \ldots\ldots(3.13)$$

$$\therefore \quad A_a^* = (HC_o^* + l_o)\,\frac{A_t \cdot L(\eta + 3/2\, P_t \tan\theta)}{2g \cdot \beta \cdot H_o\, \eta \cdot l_o} \qquad \ldots\ldots(3.14)$$

Where A_a^* is the critical (horizontal) area of an enclosed air cushion simple surge tank.

An everlasting wave is a constant invitation of trouble, due to partial synchronous conditions between the period of load changes and that of the pressure wave, and ordinarily a commercially practicable simple air tank would need to be much larger than indicated by Eq.3.14.

Incidentally, when l_o becomes infinite, the Eq.3.12 is seen to agree with the Thoma formula for the simple surge tank.

Studies conducted on the air tanks during the period 1920 to 1950 were mainly limited to the rising pipe lines and pumping plants. However, these studies were of great help in giving a clearer picture of the air tank behaviour, because the principle behind the air tanks fitted on Pumping mains and Hydro Plants is one and the same.

FOTCH[38] derived a formula for equating the change in velocity preceding and succeeding the air tank, with rate of change of height of water column in air tank.

ENGER[22] proved that the air chambers are very effective in preventing water hammer, when they are large enough, properly located, and kept full of air. He derived an equation to calculate the energy stored in an air chamber when the pressure has reached its maximum value and expressed it in terms of (i) the kinetic energy of the water in the pipe line before the valve was closed (ii) plus the work done by the flow head after the valve was closed (iii) minus the energy required to compress the water and to stretch the walls of the pipe; (iv) minus the energy lost in pipe friction and in the passages to the air chamber after the valve was closed. (v) minus the work done in lifting the water into the air chamber.

The pressure volume relationship for both adiabatic and isothermal compression, in case of an air chamber has been worked out by him and plotted in Fig.3.3. The shaded areas represent the energy stored when the pressure increases from P_1 to P_2. It can be seen that the areas under curve II (Isothermal) is greater than the area under curve-I(Adiabatic)

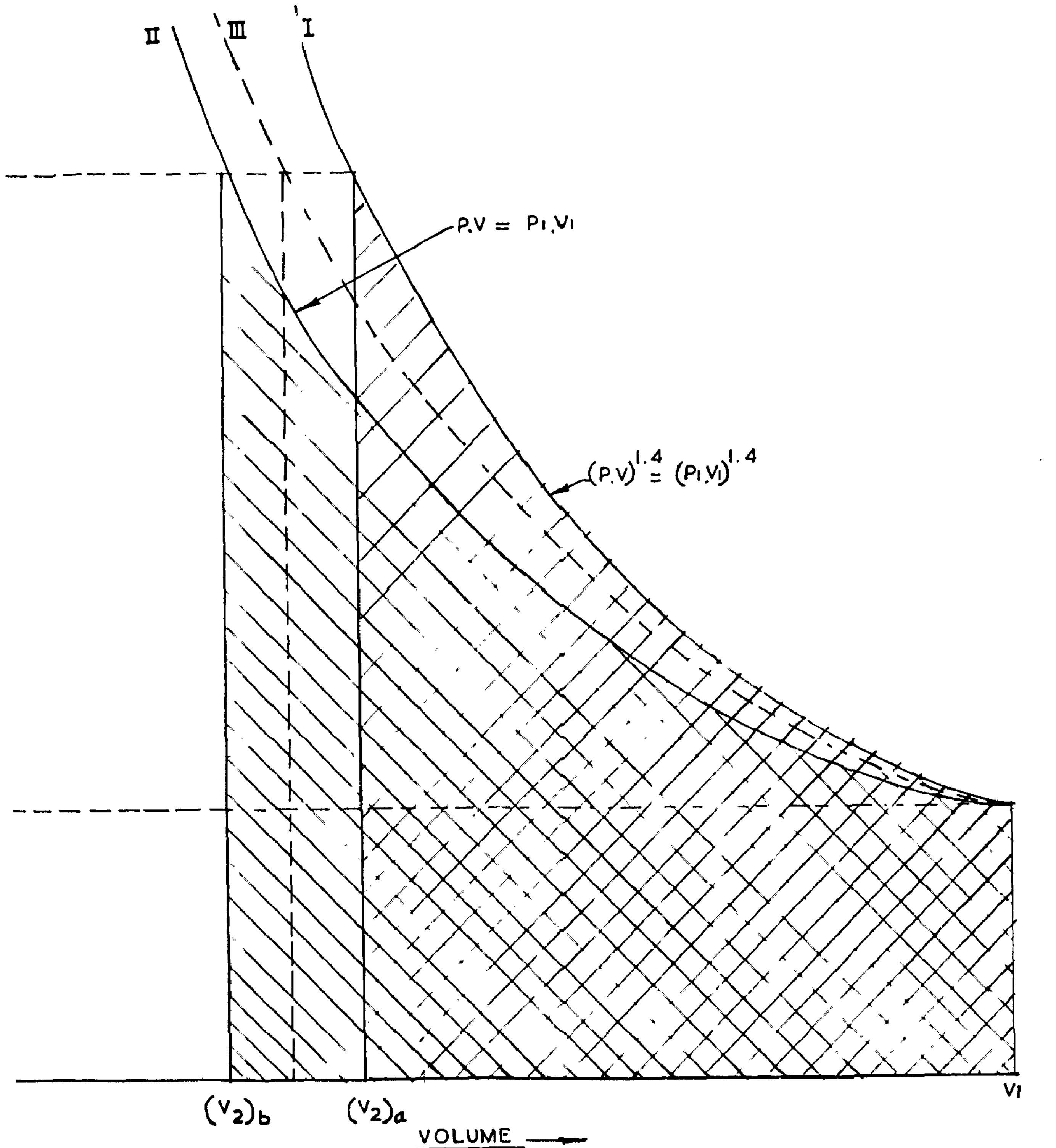


FIG. 3.3 COMPARISON OF ADIABATIC & ISOTHERMAL COMPRESSION

The difference being equal to the energy dissipated in the form of heat in the isothermal compression. The law of compression in an air chamber cannot be represented by either curve I or II, but is probably by some intermediate curve-III. When the time of compression is short, the compression will be nearly adiabatic and the curve-III will lie near curve-I. When the Compression requires a long time, a large amount of heat would be dissipated, and the compression would be more close to isothermal, hence curve-III would lie nearer curve-II. The actual law of compression in an air chamber is further complicated by rapid changes of vapour content at the various pressures and temperatures. It seems safe to think that adiabatic and isothermal compressions represent the limiting conditions between which the actual compression will fall. This aspect will be discussed again in subsequent Chapters.

ALLIEVI[1] investigated the case of a Pump situated at the foot of a rising pipe with an air vessel (Figure 3.4). If the area of cross section at the neck of the air tank is the same as that of incoming line, then the volume of water in the air tank at any time interval t_2 is given by

$$V_2 = V_1 + A_t \int_{t_1}^{t_2} v \, dt \qquad \ldots\ldots(3.15)$$

Where v is the incoming velocity and

V_1, V_2 = air volume in chamber at time t_1 and t_2

The isothermal law was used as auxilliary equation

$$V_1 \cdot HC_0^{*} = V_2 \cdot HC_2^{*} \qquad \ldots\ldots(3.16)$$

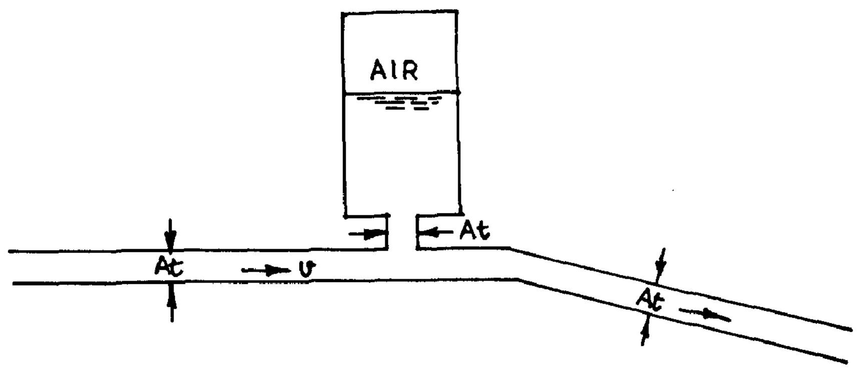
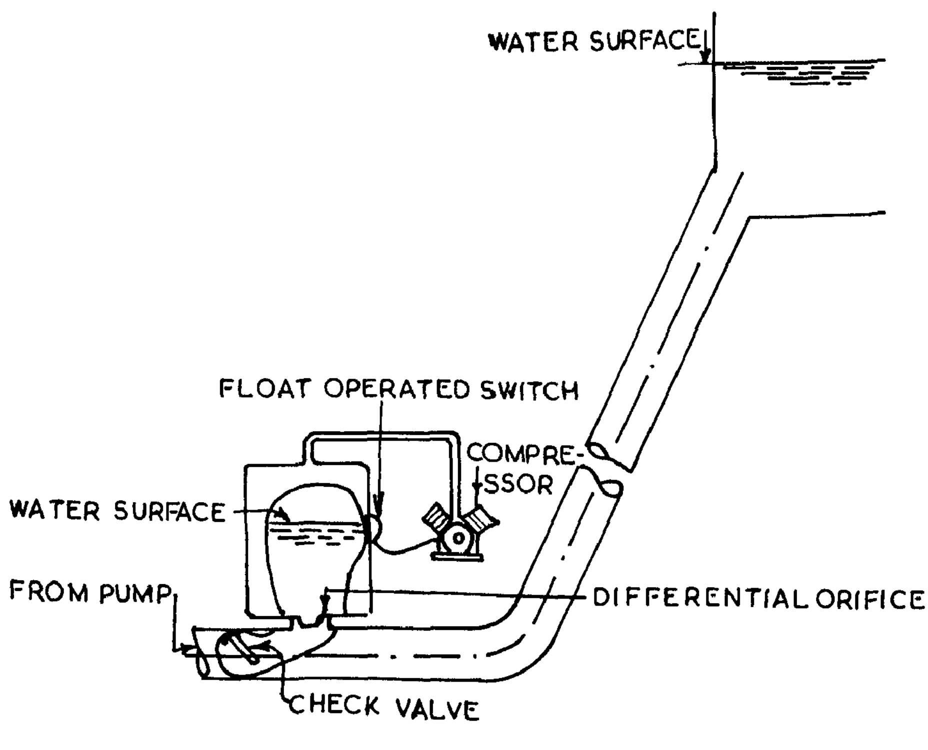


FIG. 3.4 ALLIEVI'S AIR VESSEL.



G.3.5. AIR CHAMBER AND CHECK VALVE INSTALLATION ON PUMP LINE

Allievi concluded that large vessels had to be analysed by means of step by step method, but necessary additional term for the slower surge in the air vessel has to be included. Till the studies conducted by Allievi, only a vague and inadequate solution was found to the problem, because the earlier investigators neglected the consideration of the potential energy of the pipe line as compared to the analogous energy of the chamber. In fact the ratio of these two quantities of energy constitutes the parameter of the laws of pressure variation, the crux of the problem. For arriving at such a solution, he first tried the problem disregarding liquid friction and established a system of pendular relations of finite differences which rigorously define the laws of pressure at intervals of the phase. He then introduced in such a system, terms representing the resistance of liquid friction, and found by a method of corrections, the new values of pressure dependant on such resistance. He showed that the pressure surges in a pipe line equipped with an air chamber depended on two parameters ρ^* and σ^*, neglecting friction. He, however, also showed that without frictional effects, chambers of normal size were ineffectual in controlling upsurges.

$$\rho^* = \frac{a \cdot V_0}{2g \cdot H_1} \qquad \text{.....(3.17)}$$

$$\sigma^* = \frac{2g \cdot V_0 \cdot H_1^*}{a \cdot L \cdot V_0^2} \qquad \text{.....(3.18)}$$

Where a = Propagation velocity of water hammer wave in m/sec.

H_1^o = Normal absolute pressure head in the pipe line in metres at the entrance of the chamber.

ρ^o is dimensionless and is a function of the ratio of the steady state kinetic energy to the total potential energy in a unit length of conduit where as Eq.3.18 expresses the ratio of steady state potential energy of the air in the air chamber to the steady state kinetic energy of the water in the discharge line.

Finally with the help of approximate energy balances, he proposed rational rules for the selection of values to be assigned in individual cases, to the volume of air chamber and to the necessary resistance of liquid friction for discharging flow.

BERGERON(9) in a discussion of ALLIEVI's paper, described a simple, effective differential orifice for use in conjunction with the air chamber.

During the same period ANGUS(2) came forward with equations which are necessary for the complete analysis of surge conditions in pump discharge lines, when data for the discharge line conditions are presented graphically. He selected several types of discharge line and presented graphs which indicate water hammer conditions therein.

BINNIE(11) tried to work out a quick procedure for calculating the maximum pressure and expansion of air resulting from a sudden shutdown of a plant, because, according to him, the procedure available for predicting the oscillations in a closed Surge tank were very lengthy. He took the friction into account both in the pipe and also in any arrangement placed between the pipe and the tank to damp the oscillations. The compression (or expansion) was assumed as isothermal. Small scale experiments confirmed the expectation that the observed maximum pressures would be greater than the theoretical results obtained from his formulae.

The complete shut down (or load acceptance) of a big Power Plant is not instantaneous. He claimed that his theories could be extended to yield results sufficiently accurate for all practical purposes.

His investigations were on simple surge tanks with no differential action. The results of his analysis were identical with those obtained by him for air tank fitted to a rising main(12).

In 1945 BLAIR(13) showed that it was possible to establish a relationship between the volume of air in the air vessel and the frequency and amplitude of the surge. It was found that the surge could be considerably diminished and more rapidly damped out by what is believed to be a novel method namely the introduction of damping which is effected by constructing the opening between the air vessel and the main. Though he dealt with pipe lines and pumps, it can be safely applied to Hydro-Power Plants.

EVANS AND CRAWFORD[23] developed design charts for air chambers on Pump lines in the year 1954. When an air chamber and check valve are introduced in the line near the pump, power failure causes the head developed by the pump to drop rapidly, and a head difference is thus created across the air chamber outlet (Fig.3.5). The air chamber begins to discharge into the pipe line to maintain the head and the flow, Soon, the head produced by the pump is less than that maintained by the air chamber, and the check valve closes; the pump comes to a stop. Water will continue to be discharged from the pipe line at a diminishing rate, the air chamber supplying both the water and the energy. The water in the discharge line will reverse its direction and flow into the air chamber. While the water is flowing into the air chamber, the pressure in the discharge line will increase to exceed normal operating head and will produce the maximum head for the transient. Resurges in the pipe line will ensure with diminishing intensity.

A frictional resistance is essential to the effective use of an air chamber in a pump discharge line, another variable K_2 is introduced so that $K_2.H_1^*$ is the total head loss for a flow of Q, down the pipe line and into the air chamber were Q_1 is the initial rate of flow in the pipe line. A differential orifice will be used to create the head loss. Because of the differential orifice design, the head loss for flow from the air chamber will be less than that for flow in the chamber.

In order to get a graphical solution, they modified equations (3.17) and (3.18) and developed another equation

$$V_1 = \sigma^* \cdot \rho^* \cdot \frac{Q_1 \cdot L}{a} \qquad \text{.....(3.19)}$$

If the subscript n indicate the number of time interval under study one may write,

$$V'_n = \frac{V_n}{V_1} \qquad \text{.....(3.20)}$$

To fit the graphical solution, the interval of time used will be a fraction of the travel time of wave $j. \frac{L}{a}$ in which j may take values 2, 1, 1/2, 1/4 as the problem requires.

Similarly if Q'_n is defined as Q_n/Q_1 then

$$V'_2 = V'_1 + \frac{(Q'_2 + Q'_1)}{2\sigma^* \cdot \rho^*} \qquad \text{.....(3.21)}$$

and for time interval n,

$$V'_n = V'_{n-1} \quad \frac{(Q'_n + Q'_{n-1})}{2\sigma^* \cdot \rho^*} \qquad \text{.....(3.22)}$$

which is the desired relationship.

For this study, the following assumptions were made

(1) The air chamber is located near the pump.

(2) The check valve at the pump closes immediately upon power failure.

(3) The pressure volume relationship for the compressed air in the air chamber is $HC^*.V^{1.2}$ = a constant.

(4) The ratio of the total head loss for the same flow into and out of the air chamber is 2.5:1; $K_2 \cdot H^*_1$ is the sum of the hydraulic losses in the discharge line and the throttling losses at the differential orifice when a reverse flow equal to Q_1 is passing into the air chamber.

(5) The head loss varies with the square of the velocity.

(6) Water column remains intact throughout the length of the line.

To ensure that air will not enter the discharge line when the maximum die downsurge is attained, the total volume of the air chamber must be greater than

$$V_{max} > \frac{V_o \cdot HC^*_o}{H_{min}} \qquad \text{.....(3.23)}$$

Where H_{min} = HC^*_o minus maximum downsurge adjacent to the pump.

K_2 = Coefficient of head loss such that $K_2 . H^*_1$ is the total head loss for a flow of Q_1 into the air chamber.

With the help of the charts prepared by them the preliminary design of an air chamber on pump line can be easily worked out fairly accurately.

A gas filled accummulator has more recently been advocated and utilised for controlling pressure and flow transients in rocket propellent feed system and protecting flow systems by DORSCH[21] and others, against the destructive effects of large magnitude pressure surges which occur in nuclear reactors. Devices or techniques which serve the same purpose as a gas accummulator such as relief valves[52], compensating belbws[51] and gas injection[78] are closely related methods of surge control.

In the year 1970, WOOD[79] conducted a systematic and detailed analysis of an air cushion surge tank, which will be discussed in the next Chapter.

The design of air cushion surge chamber with respect to the mass oscillations and stability considerations has been studied at length at the River and Harbour Laboratory at the Technical University of Norway., TRONDHEIM[68]. This led to the provision of an air cushion surge chamber on the Driva Power Station (Fig.3.6) which is the first and the only Hydro power Plant equipped with air cushion Surge Chamber. The Power Plant has since come in operation and has been functioning satisfactorily.

From the differential equations describing the hydraulic transient response of the conduit and air chamber, GARDNER and GUTIER[25] derived the stability limits in 1973 and presented in a form from which direct estimates can be made of the air chamber and orifice dimensions, required to limit the amplitude of resonant pressure oscillations to any speed. For that they have considered the value of n, the polytropic constant of expansion as 1.2.

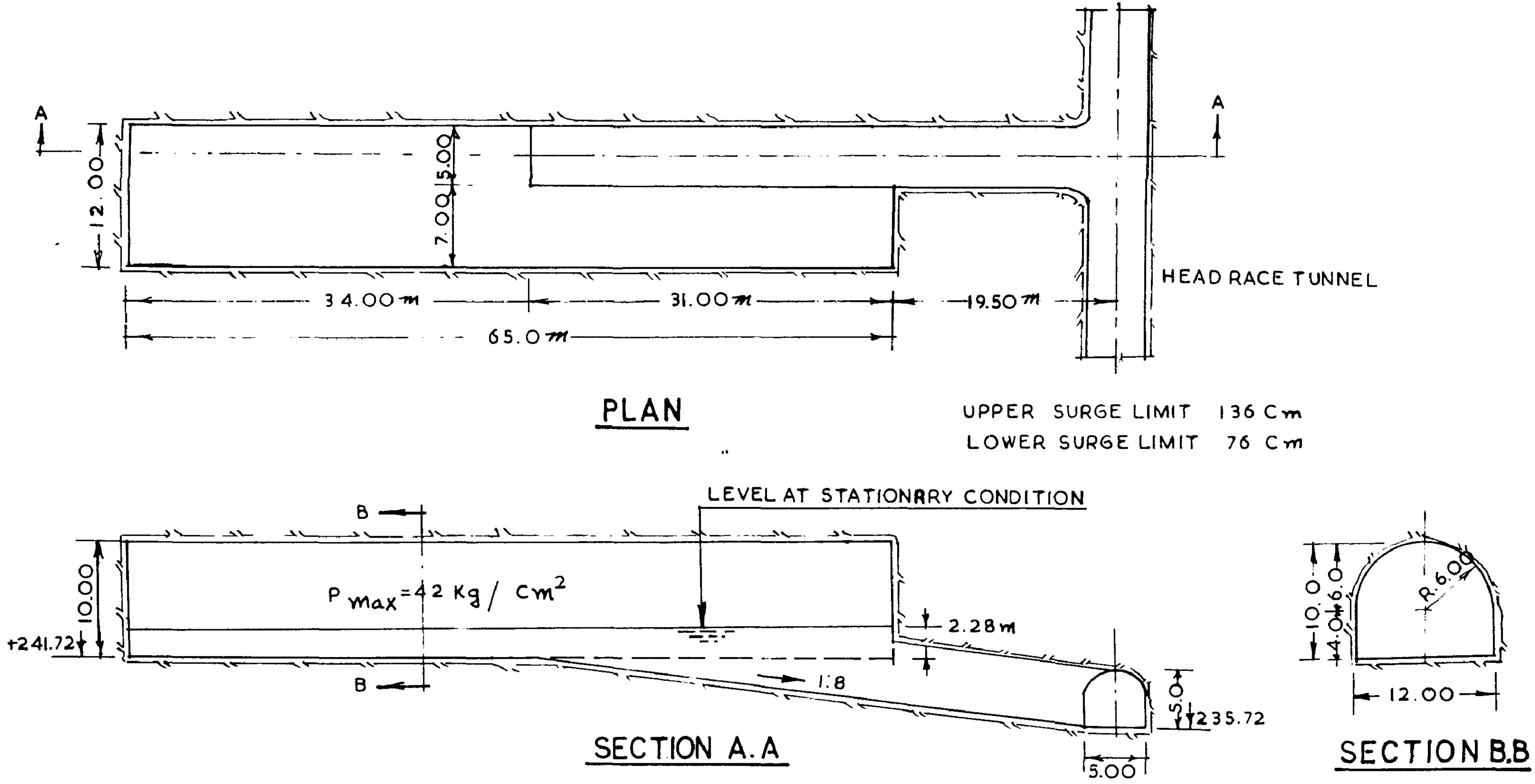


FIG. 3.6 AIR CUSHION SURGE CHAMBER AT DRIVA - NORWAY

CHAPTER-IV

TRANSIENT FLOW ANALYSIS OF AN AIR CUSHION SURGE CHAMBER

4.1. EFFECT OF AIR CHAMBER ON TRANSIENT FLOW

For the proper evaluation of the effect of an air chamber on transient flow in a system, it is essential to have a transient flow analysis of the flow system including the air chamber taking into consideration, the types of disturbances expected. An accurate analysis of this may be difficult and time consuming. Therefore, several simplifications are often employed. These include lumping parameters and linearizing various non linear effects. These simplifications may obscure short term effects or instabilities which occur in the flow system. These short term effects can be predicted by a more comprehensive analysis. It is also not possible to completely describe the performance of an air chamber or any surge control device without closely relating to the configuration of the entire flow system and the types of transients to which it will be subjected.

4.2. STUDY OF A SINGLE AIR CHAMBER

Let Fig.4.1 represent a schematic profile of the flow system to be analysed. The flow is from the reservoir to the exit valve. The exit valve is closed rapidly thus generating a pressure surge. The surge propagates towards

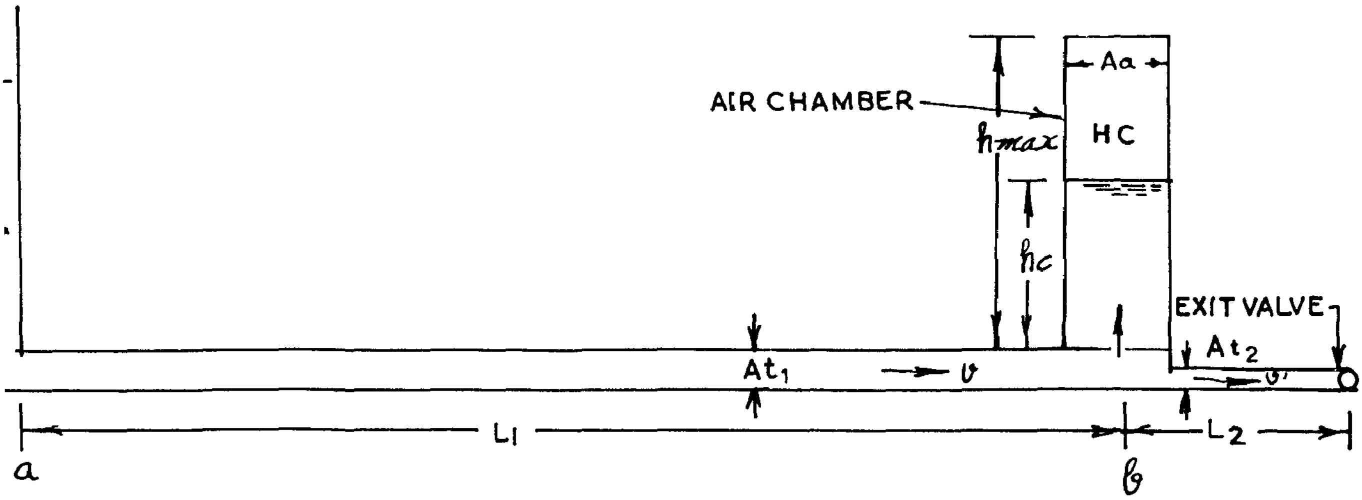


FIG. 4.1 SCHEMATIC PROFILE OF A FLOW SYSTEM

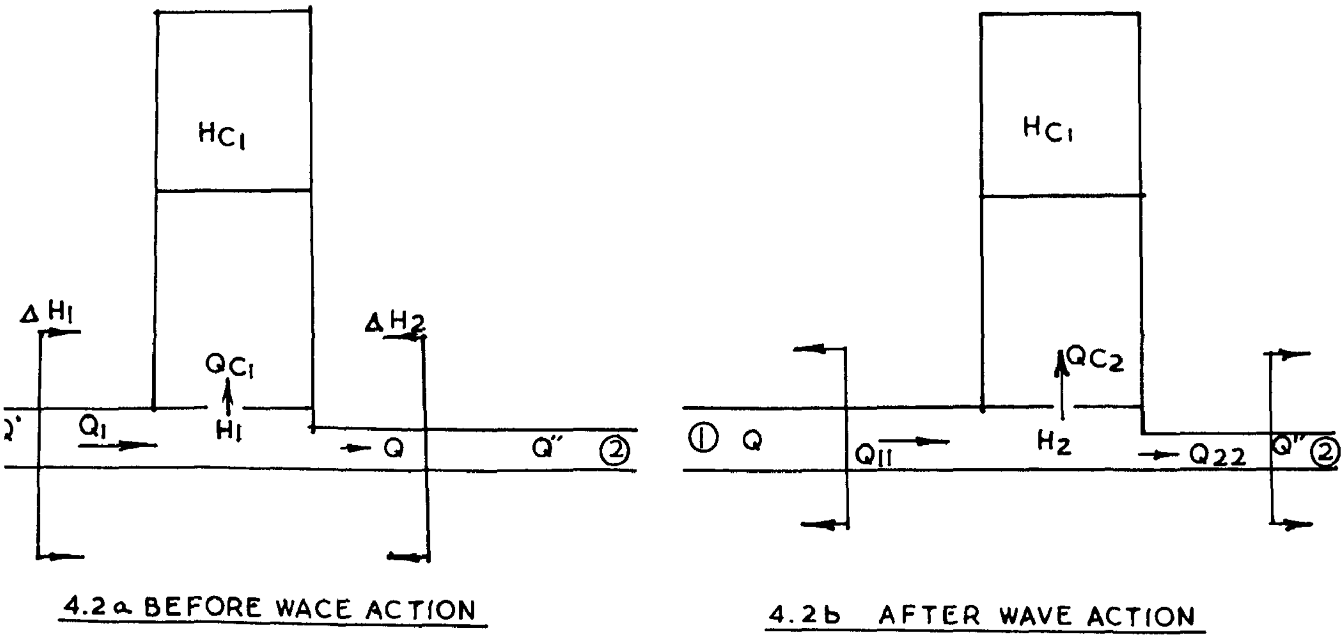


FIG. 4.2. CONDITIONS AT AIR CHAMBER

the reservoir and is influenced by the air chamber positioned at a distance L_2 from the valve. The performance of the air chamber is measured by the ability of the chamber to reduce the magnitude of the pressure surge which propogates upstream past the chamber. This is evaluated as a function of the initial volume of air in the chamber and the entrance resistance to flow into the chamber which is dependent on the diameter of the chamber entrance orifice.

4.3. DISTRIBUTED PARAMETER ANALYSIS

As with all distributed parameter solutions which consider arbitrary inputs and non-linear boundary conditions, the practical application of this technique to all but very simple flow systems requires the use of a digital computer. The technique consists basically of approximating system disturbances by a series of incremental step changes, each of which produces a small pressure wave. The pressure waves are propogated through out the flow system at sonic velocity and are transmitted and reflected at each system discontinuity. A solution for transient flow is obtained following each of the waves generated at the point of the disturbance and computing the effect of system discontinuities on the propagation of the waves. Finally the effect of these waves at any point in the flow system may be summed up in time. In order to extend this computational method to a flow system with an air chamber equations describing the response of an air chamber to pressure waves must be developed.

Fig.4.2 shows the condition of Air Chamber, before and after impingement of pressure waves. The following assumptions are made in this analysis.

i) The pressure waves produce step changes in the line pressure and flow and these conditions remain unchanged until the next pressure wave impinges.

ii) The step change in flow into the chamber can be taken as constant over the short time period between the impingement of waves.

iii) The flow into the chamber over the time period creates a slight change in the air volume which will determine the pressure in the air chamber at the time the next set of waves impinges.

At time t_1 an instant before the incoming pressure waves ($\triangle H_1$ and $\triangle H_2$) reach the air chamber, let

H_1 = Head in the line at the chamber in metres.

HC_1 = Pressure head in the air chamber in metres.

Q_1 = Flow in the line towards the chamber in m^3/sec.

Q_2 = Flow in the line past the chamber in m^3/sec

QC_1 = Flow in to the air chamber in m^3/sec.

V_1 = Volume of air in the chamber in m^3

At time t_2, an instant after the incoming pressure pulses have reached the air chamber and the pressure pulses have been transmitted past and reflected from the air chamber, let

H_2 = Head in the line at the chamber in metres.

Q_{11} = Flow in the line towards the chamber in m^3/second.

Q_{22} = Flow in the line past the chamber in m^3/second.

QC_2 = Flow entering the air chamber in m^3/second.

The head in the chamber HC_1 is assumed to remain constant during the instant that the pressure pulses strike the chamber and new pulses are emitted. However, the head in the chamber does take a new value before the next set of pressure pulses reach the chamber due to the flow into the chamber over a brief period of time.

The equations for the pressure pulses coming into ($\triangle H_1$) and leaving ($\triangle H_{11}$) the chamber on the left side pipe at upstream of surge chamber can be written as,

$$\triangle H_1 = \frac{C_1}{g \cdot \Delta t_1} (Q' - Q_1) \qquad \ldots(4.1)$$

$$\triangle H_{11} = \frac{C_1}{g \cdot \Delta t_1} (Q' - Q_{11}) \qquad \ldots(4.2)$$

Combining Eq.4.1 and 4.2 one gets,

$$\triangle H_{11} = \triangle H_1 + \frac{C_1}{g \cdot \Delta t_1} (Q_1 - Q_{11}) \qquad \ldots(4.3)$$

Similarly, equation for the pressure pulses for the right of the air chamber can be written as :

$$\triangle H_{22} = \triangle H_2 + \frac{C_2}{g \cdot \Delta t_2} (Q_{22} \cdot Q_2) \qquad \ldots(4.4)$$

Where

At_1 = Area of the pipe 1 in m^2

At_2 = Area of the pipe 2 in m^2

C_1 = Wave velocity in pipe 1 in metres/second.

C_2 = Wave velocity in pipe 2 in metres/second.

Q' = Flow discharge in pipe 1 in m^3/second.

The relationship for orifices obtained from steady state energy considerations gives :

$$QC_2 = K_c (H_2 - HC_1)^{\frac{1}{2}} \quad \text{....(4.5)}$$

where K_c = chamber entrance loss coefficient.

The incremental pressure changes can be expressed as

$$H_2 = H_1 + \triangle H_1 + \triangle H_{11} = H_1 + \triangle H_2 + \triangle H_{22} \quad \text{...(4.6)}$$

Continuity equation gives

$$Q_{11} = Q_{22} + QC_2 \quad \text{....(4.7)}$$

Solving the Eq.4.3 to 4.7 one gets

$$(QC_2)^2 + b\, QC_2 + c = 0 \quad \text{...(4.8)}$$

$$\text{in which } b = \frac{C_1}{g.\ At_2} \left\{ \frac{K_c^2 .\ C_2 .\ At_1}{At_1 . C_2 + At_2 . C_1} \right\} \quad \text{...(4.9)}$$

$$\text{and } c = - K_c^2 \left\{ H_1 + 2 \triangle H_1 + \left(\frac{C_2 . At_1}{At_1 . C_2 + At_2 C_1} \right) \left(2 \triangle H_1 - 2 \triangle H_2 + \frac{C_2 . Q_2}{g.\ At_2} + \frac{C_1 . Q_1}{g.\ At_1} \right) - \frac{C_2 .\ Q_2}{g.\ At_2} - HC_1 \right\} \quad \text{...(4.10)}$$

The positive root of the Eq.4.8 gives the correct solution for QC_2, if the flow is into the chamber as was assumed. However, when the head inside the chamber, HC_1, becomes greater than the head in the line at the chamber, H_2, the flow through the chamber orifice will reverse and equation 4.8 will change into

$$(QC_2)^2 - b \quad QC_2 - c = 0 \qquad \ldots\ldots (4.11)$$

After calculating QC_2, Eqs. 4.3, 4.4, 4.6 and 4.7 can be solved to give values for all other pressure and flow conditions.

The change in volume of liquid in the air chamber over a time period Δt, is due to the inflow into the chamber and is given by

$$\Delta V = QC_2 . \Delta t \qquad \ldots\ldots (4.12)$$

Hence the volume of air in the chamber V_2 at the end of the time period, Δt, in terms of the volume of air at the beginning of the time period is

$$V_2 = V_1 - \Delta V \qquad \ldots\ldots (4.13)$$

Now the pressure in the chamber has to be related to volume of air by using the ideal gas law

$$\frac{H_1{}^{*} \cdot V_1}{T_1} = \frac{H_2{}^{*} \cdot V_2}{T_2} \qquad \ldots\ldots (4.14)$$

in which T_1 and T_2 are absolute temperatures in Rankine.

Eq.4.14 can be simplified if the system acts adiabatically or isothermally. If the system is found to act isothermally Eq.4.14 gives

$$H^*_1 . V^*_1 = H_2^* . V_2^*$$

and if it is assumed to act adiabatically, then

$$H^*_1 . (V_1)^k = H^*_2 \ (V_2)^k \qquad ..... (4.16)$$

where k is adiabatic constant for the chamber gas.

Utilizing the equations developed for wave action at an air chamber, a distributed parameter transient analysis of the flow system can be performed. Equations are written for a downstream valve which is closed in a prescribed manner over a designed period of time. Line viscous effects may be accounted for through the use of friction orifices distributed along the line.

WOOD et al[80] developed equations needed for the downstream orifice and the friction orifice. A digital programme was developed combining the equations for the downstream valve, friction orifices and the air chamber formulating an analytical model for the flow system.

4.4 LUMPED PARAMETER ANALYSIS

The system shown in Fig.4.1 can be analysed assuming that the fluid column in the pipe moves as a slug. For this situation, the momentum equation for the liquid column in the pipe between points a and b can be written as

$$A_t . \gamma \left\{ h_r - h_f - (HC - h_c - h_{fc}) \right\} = \frac{A_t . \gamma . L_1}{g} . \frac{dv}{dt} \qquad ..... (4.17)$$

Where

h_f = head loss in the line and can be assumed as

$$h_f = \left(\frac{f \cdot L_1}{D} + Ke \right) \frac{v^2}{2g} = \beta . v^2 \qquad .....(4.18)$$

D = line diameter in metres

f = friction factor

h_c = height of liquid column in the air chamber in metres.

h_{fc} = energy loss at the chamber entrance

$$= k_c \left(\frac{dh_c}{dt} \right)^2 \qquad .....(4.19)$$

h_r = height of liquid in the constant head reservoir in meters.

HC = Air chamber pressure in metres.

k_c = Loss coefficient for the air chamber entrance orifice.

Ke = entrance loss coefficient

v = liquid velocity in the line upstream of air tank in m/sec.

γ = specific weight of water in kg/m^3

The continuity equation for the pipe junction with the air chamber is given by

$$v . At_1 = v' . At_2 = \frac{dh_c}{dt} . A_a \qquad .....(4.20)$$

Where v' = velocity of water downstream of surge chamber in metres/second.

If isothermal compression is assumed for the air chamber, then the equation of state gives

$$V. HC = V_o . HC_o \qquad \text{.....(4.21)}$$

Where

HC_o = initial air chamber pressure in metres

V_o = initial air volume in m^3

This can be expressed in terms of height of air chamber and height of water in it.

$$HC = HC_o \quad \frac{h_{max} - h_{co}}{h_{max} - h_c} \qquad \text{.....(4.22)}$$

where

h_{co} = the initial height of liquid in the air chamber in metres

h_{max} = height of air chamber in metres = (h_c+1)

For the situation under consideration, when $t = 0$, $v = v(0)$ and $h_c = h_{co}$ and when $t > 0$, $v' = 0$. The last condition states that at time $t = 0$, the downstream valve is closed and the flow downstream from the valve is following the closure. Hence the continuity equation can be written as :

$$v = \frac{A_a}{At_1} \left(\frac{dh_c}{dt} \right) \qquad \text{.....(4.23)}$$

differentiating one gets,

$$\frac{dv}{dt} = \frac{A_a}{At_1} \cdot \frac{d^2 h_c}{dt^2} \qquad \text{.....(4.24)}$$

Combining equations 4.16 to 4.24 yields a nonlinear equation for h_c

$$\frac{g \cdot A_a}{L_1 \cdot At_1}\left\{h_r - \left(\frac{d\,h_c}{dt}\right)^2\left(\frac{\beta \cdot Aa^2}{At_1^2} + k_c\right) - HC_a\left(\frac{h_{max} - h_{ca}}{h_{max} - h_c}\right) - h_c\right\} = \frac{d^2 h_c}{dt^2} \qquad (4.25)$$

The solution of this equation with the prescribed initial condition gives the lumped parameter response of the system. It is not possible to solve the Eq.4.25 in a closed form. Therefore, the solution has to be carried out by utilizing an analog computer.

4.5. VERIFICATION OF THE FORMULAE

4.5.1. Properties of flow system

The above formulae were verified by WOOD(79) analytically as well as experimentally in the laboratory for a set up shown in Fig.4.2. The physical properties of the pipe line and air chamber are given below :

L_1 = 10.92 8 mtr. C_1 = 1219.188 m/sec.

L_2 = 1.2192 mtr. C_2 = 1219.188 m/sec.

D_1 = 2.6035 cm. h_{max} = 32.4119 cms.

D_2 = 2.6035 cm. h_r = 15.8496 mtr.

Dia. of inlet orifice = 1.27cm. v = 1.5768 m/sec.

Dia. of downstream orifice=0.9525 cm.

With the flow system operating under steady state conditions, the downstream orifice was closed in approximately 2/1000 sec. The pressure generated by this action in the line upstream from the air chamber (point b of Fig.4.1) was theoretically computed and obtained experimentally.

4.5.2. Analytical results

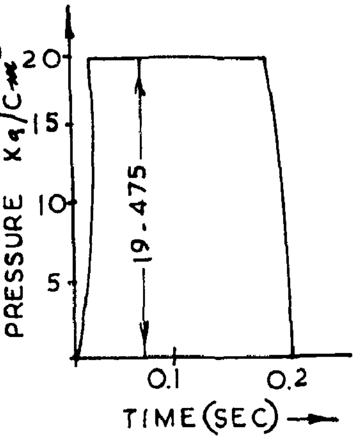
4.5.2.1. Response with no air chamber

With no air chamber in the system or the chamber initially closed, the usual water hammer response was obtained. The maximum pressure rise was 19.476 kg/cm^2. The pressure was maintained until negative reflections from the reservoir returned to the valve ($t = \frac{2 L_1}{c}$) = 0.010 sec.). The pressure diagram so obtained has been plotted in Fig.4.3.

4.5.2.2. Air Chamber behaviour

The pressure response for sudden closure of the valve was computed for different initial air volumes for chambers having entrance orifice diameters 6.35 mm, 9.525 mm and 12.7 mm and in the case of no orifice at the chamber entrance.

A few typical digital computer plots, which are the results of distributed parameter solution when initial air volume is kept constant as 33.1 cm^3 and assuming isothermal expansion for different orifice diameters are shown in Fig.4.4.



.4.3. PRESSURE RESPONSE
WITH NO AIR CHAMBER.

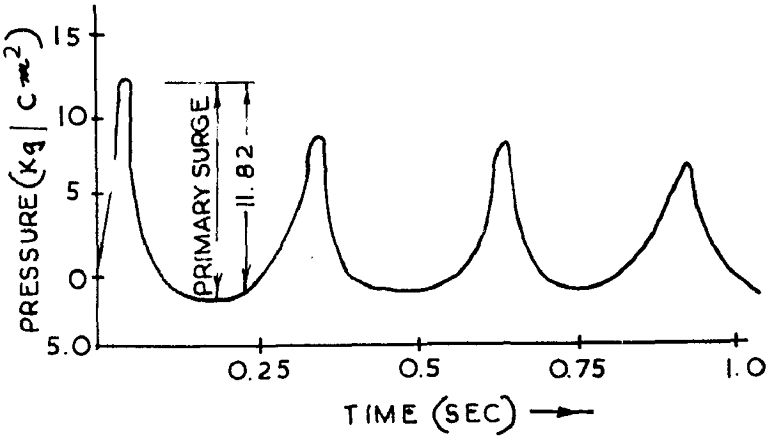


FIG. 4.4a ORIFICE DIAMETER. 26.035 mm

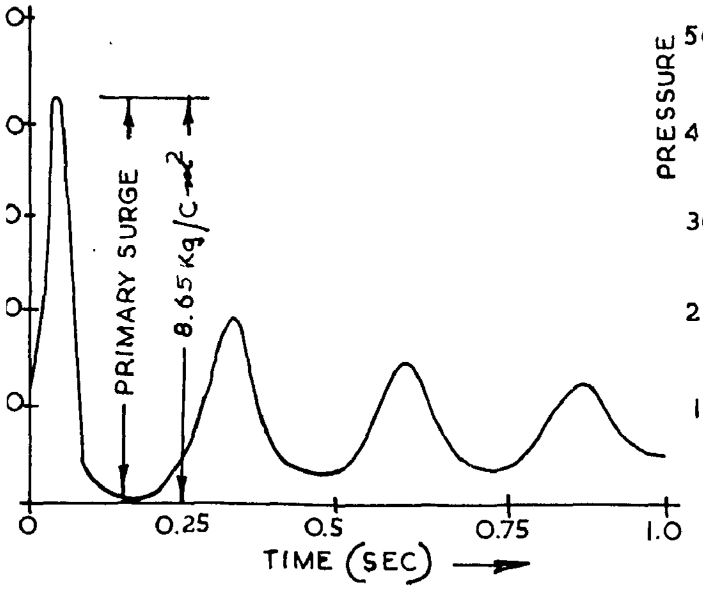


FIG. 4.4b ORIFICE DIA. 9.525 mm

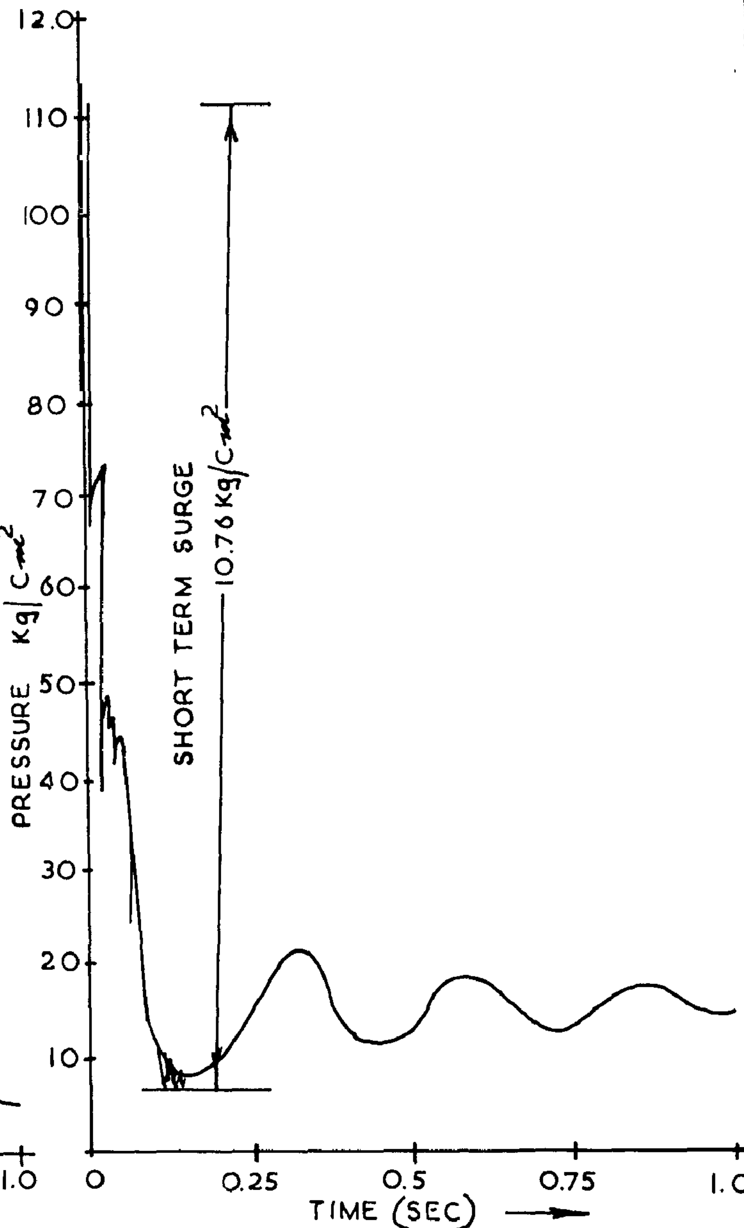


FIG. 4.4c ORRIFICE DIA. 6.35 mm

FIG. 4.4 PRESSURE RESPONSE WITH AIR CHAMBER.

It can be inferred that the short term effects which occur directly after valve closure are especially noticeable when the size of the entrance orifice is reduced. This is due to the water hammer surge downstream from the chamber, it dies out very rapidly. This phenominon can be predicted by using a distributed parameter analysis.

Short term surges can be clearly distinguished from primary surges. The primary surge is the difference between the maximum and minimum surge pressures, neglecting short term effects. The short term pressure surge is defined as the difference between the initial surge peak and the subsequent minimum. The short term pressure surge is not always well defined and is of little significance, if it is smaller than the primary pressure surge.

The analytical results are summarised in Fig.4.5, which shows both the magnitude of the primary pressure surge and the short term surge for the range of entrance conditions and initial air volume studied.

4.5.2.3. Inferences

The main inferences from these studies are listed below :-

1) The magnitude of the short term surge does not depend on the initial volume of air in the air chamber and is only a function of the resistance to flow at the entrance to the chamber. This is already expected since the pressure in the chamber can not be significantly changed over the short

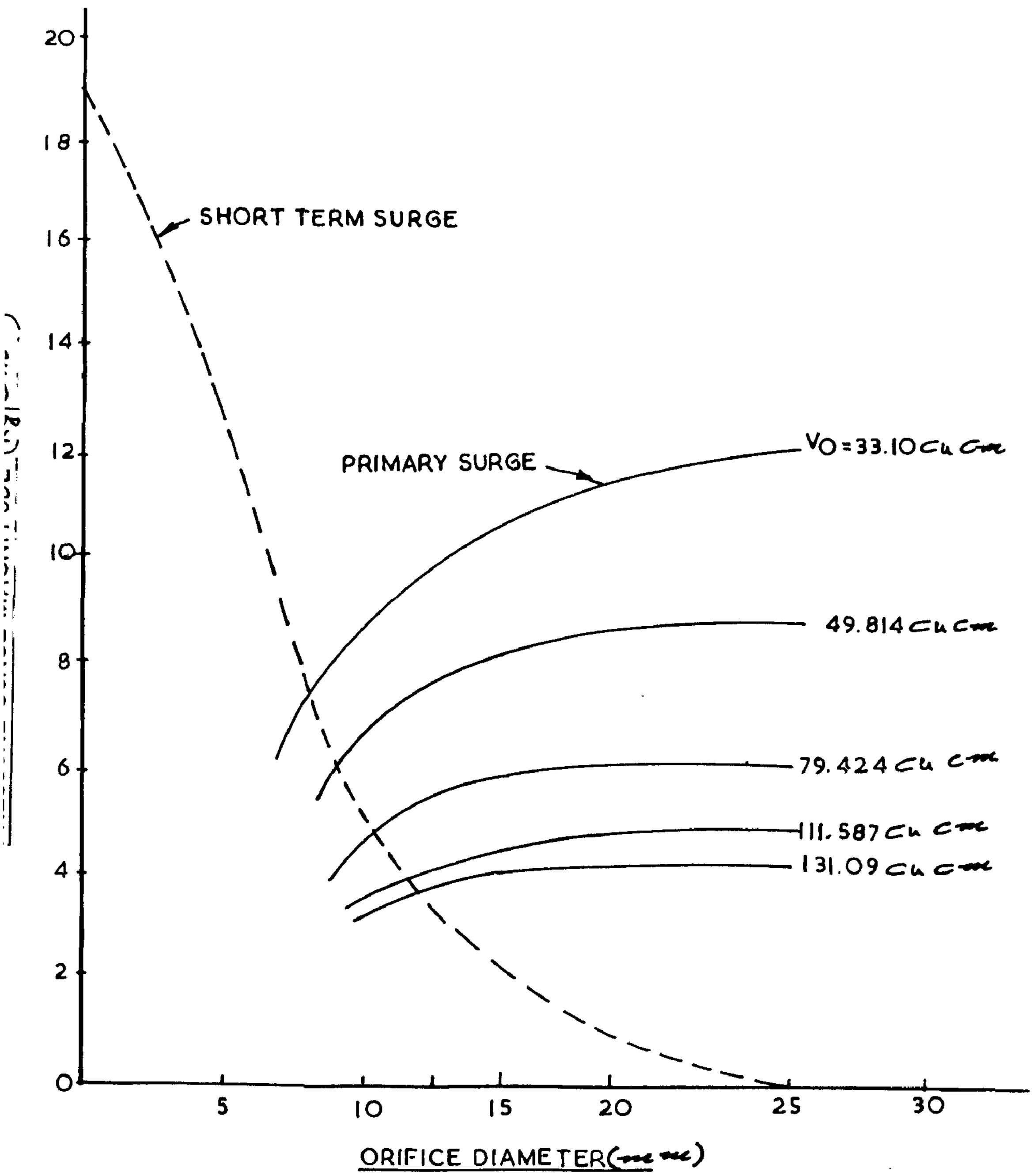


FIG. 4.5. SHORT TERM AND PRIMARY PRESSURE SURGE MAGNITUDES

time involved and the flow into the chamber is controlled by the size of the orifice only.

ii) Reducing the size of the chamber entrance orifice will improve the attenuating characteristics of the air chamber.

However, the second inferance will be true only if the size of the orifice is not reduced below the point where the short term surge exceeds the primary surge. Below this point, the attenuating characteristics of the air chamber will rapidly deteriorate. The optimum size occurs at the transition point which gives a minimum surge in the line.

A comparative study with the equations for adiabatic compression were also studied and the results were found slightly on the higher side, say about 10% with a slightly reduced period. This shows that the type of compression assumed can have a significant effect on the primary surge. However, it could not be expected to significantly affect the short term surge.

Also when analysed with the lumped parameter solution no short term surges are noticed. Regarding primary pressure surges, both the lumped parameter results as well as a distributed parameter results are in nearly perfect agreement.

4.5.3. Experimental Analysis

The same system used for the theoretical analysis was constructed in the laboratory also. The pressure response in the line at the chamber was obtained experimentally for the same range of initial air volume and entrance conditions. The oscilloscope traces obtained using piezo electric quartz pressure transducers and rapidly closing valve were found to be in good agreement with the analytical results. The shape of the traces and the frequency of the surges also had resemblance.

From the above discussions it can be concluded that an optimum size exists for the chamber entrance orifice, and the performance of the air chamber is quite sensitive to this parameter.

In the foregoing paragraphs, the possible indiscrete use of air chamber and the consequent need of system design for finding the transient flow conditions have been discussed. It has not been possible to find the exact air chamber behaviour because the experimental values tally neither with isothermal nor with adiabatic law.

GREEZE(31) is of the opinion that in such set ups in the laboratory, the casing of the air chamber has to be regarded as a heat sink and that the consequent adoption of the polytropic equation HC . V^n = constant will be valid where n is the polytropic constant. But to describe the air behaviour is totally true in principle.

Further considerations reveal that a polytropic equation of the stated form cannot adequately describe the air behaviour since the initial change in the air is of an adiabatic nature while the final equilibrium position exhibits an isothermal situation when referred to the initial air mass conditions. The expected air behaviour is schematically shown in Fig.4.6 and if the time history of the air is to be defined by an equation similar to the polytropic one, then n must be a variable with time.

$$\text{i.e. } HC^{*}.\ V^{n(t)} = \text{constant} \qquad (4.26)$$

which is a function, which cannot be predetermined.

4.5.4. Fundamental Analysis

Fundamental analysis has, however, shown[30] that for a mass of air acting as a perfect gas, the following general equation is valid

$$\frac{dHC^*}{dt} = -k\frac{HC^*}{V}\frac{dv}{dt} - \frac{(k-1)}{V}\frac{du}{dt} \qquad (4.27)$$

in which k is the ratio of specific heat = 1.4 and

$\frac{du}{dt}$ = rate of heat outflow from the air mass.

Further analysis indicated that the rate of heat outflow, $\frac{du}{dt}$, for the set up under consideration can satisfactorily be represented by convective heat transfer coefficients so that

$$\frac{dQ}{dt} \text{ / Unit area of surface} = 0.19/T - T_{ex}/0.333(T.T_{ex}) \qquad (4.28)$$

Where T = Temperature of the air mass.

T_{ex} = Temperature of the heat sink.

This is the Rational Heat Transfer process. The R.H.T. process avoids the guess work for the value of 'n'.

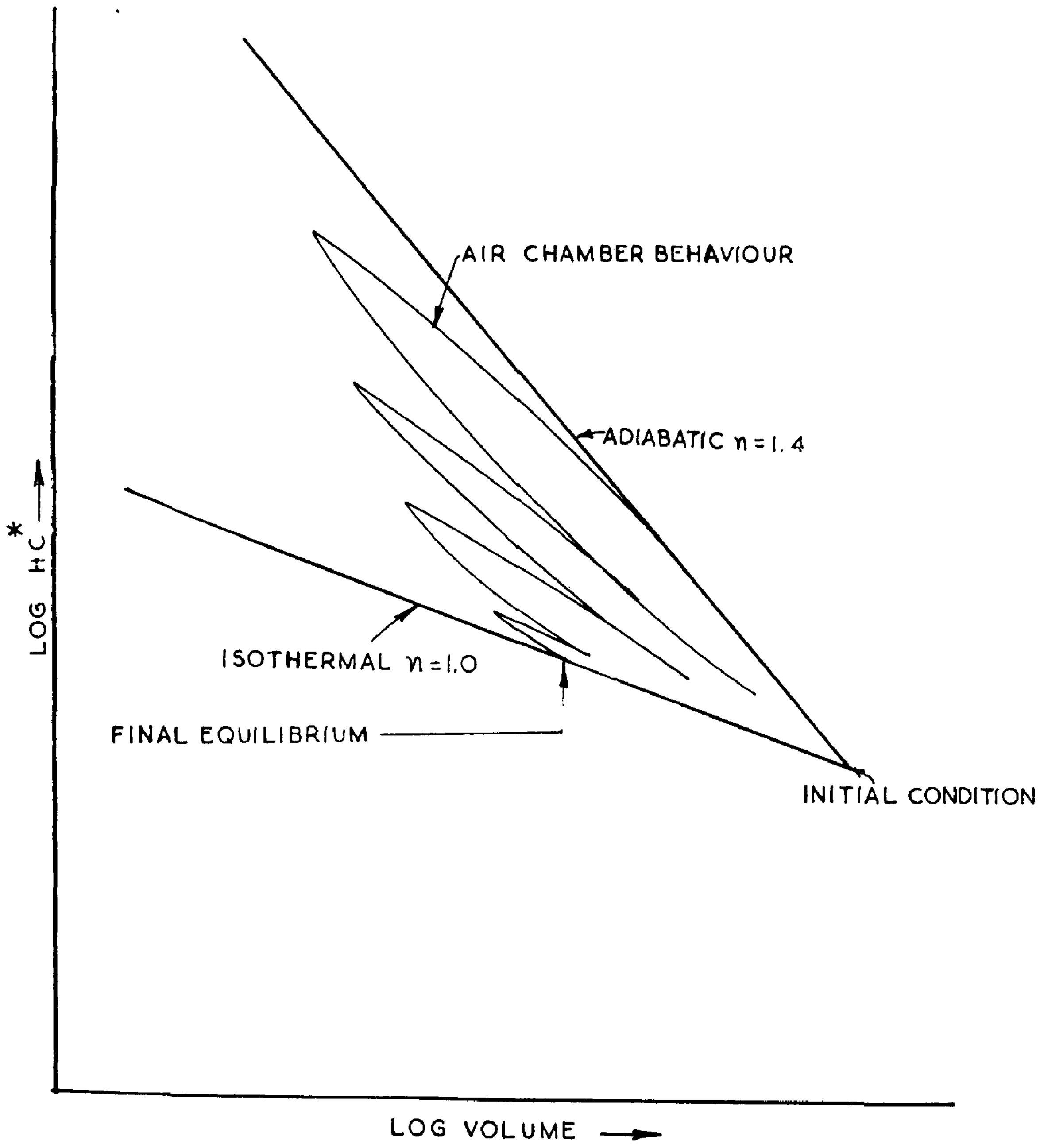


FIG. 4.6 EXPECTED AIR BEHAVIOUR.

CHAPTER - V

DESIGN OF AIR CUSHION SURGE CHAMBER

5.1 GENERAL

Resonance is a potential source of danger to any closed conduit filled with liquid, and will develop whenever boundary conditions exist which cause a net inflow of energy to some part of the conduit. Surge tanks and air chambers are the cheap methods of pressure control devices that will ensure that the amplitudes developed under resonant conditions do not exceed acceptable limits.

Even though the provision of an Air Cushion Surge Chamber practically solves the problem of pressure surge stability and also seems rather attractive from an economical point of view, the major question arises as to how such an air cushion will influence the mass oscillation stability.

An air chamber is primarily an energy dissipating device, in which the air provides an elastic boundary which deforms sufficiently with pressure changes for the flow through the orifice to dissipate a substantial amount of energy.

In high head Power Plants, a surge chamber with an enclosed air cushion offers an excellent alternative to a surge tank extending to the surface. This alternative allows the distance between the turbines and the free water surface to be reduced considerably, thus reducing the problem of pressure surge stability.

SVEE[74] established stability criteria of the mass oscillations for an air cushion surge chamber by applying the theory of small oscillations. The critical surge chamber area is shown to be equal to the required area of a conventional surge chamber multiplied by a factor considerably larger than unity. The multiplying factor depends on the initial pressure in the air cushion (for stationary conditions) as well as the initially enclosed air volume.

5.2 ASSUMPTIONS

The basic assumptions made for a conventional type surge chamber also hold good for this case, since the problem has to be treated as the stability of mass surge oscillations caused by any gate movement with increasing or decreasing net head across the turbine. They are reproduced again :

(i) An ideal regulation is assumed i.e. $\eta . Q . H_o$ = constant in all phases of the oscillations, in order to maintain a constant net work output.

(ii) The inertia of the water in the Surge Chamber is neglected in the direction normal to the turbine axis.

(iii) The water mass in both the shaft and the chamber is neglected.

(iv) Pressure differences are transmitted with infinite velocity i.e. no time lag exists between a water wheel alternation in the Surge Chamber and its effect on the oscillation of the tunnel watermass.

(v) Velocity is constant across the tunnel area.

5.3 BASIC EQUATIONS

The basic equations of this surge system (Fig.3.2) are given below:-

(1) Dynamic Equation $P.dt = d(m.v.)$ .....(5.1)

(2) Continuity Equation ~~P.dt =~~

$$Q = A_a \frac{dZ}{dt} + A_t.v \qquad .....(5.2)$$

(3) Power equation $\eta.Q.H_o$ = Constant .....(5.3)

in which

A_a = Horizontal area of enclosed air chamber in m^2

H_o = Net head of the Power Plant (Steady state condition) in m.

m = mass of water

P = Instantaneous pressure in air chamber in kg/m^2

Q = Instantaneous discharge of the turbine in m^3/sec.

v = Instantaneous velocity of water in tunnel in m/sec.

Z = Water level in surge chamber taken positive downwards from the water level in the intake basin in m.

Eliminating Q and v from the equations, one gets a non linear differential equation of second order in Z, which cannot be solved analytically.

To examine the stability, a small equilibrium disturbance is imposed on the system stationary conditions. For that it is imagined that a layer of thickness dZ is placed on the water surface in the Surge Chamber at stationary conditions. The effect of such an equilibrium disturbance is then examined.

In these calculations the Δ values are considered to be very small, but finite deviations from the respective stationary values. Now by neglecting small terms of second or high order, the three basic equations lead to a linear homogeneous differential equation with constant coefficients.

5.3.1. <u>Dynamic Equation</u>

$$(P_a \cdot A_t + h_r \cdot \gamma \cdot A_t + \gamma \cdot A_t \cdot L \text{ Sin } \alpha - \beta \cdot v^2 \cdot \gamma A_t - P A_t - h_c \gamma A_t)dt$$
$$= (m + dm)(v + dv) - mv \qquad \ldots\ldots(5.4)$$

Where h_c = height of water in air chamber in metres

h_r = height of water in the reservoir in metres.

P_a = Atmosphere pressure in kg/m^2

α = Slope angle of the Head race tunnel with Horizontal in radians.

The water mass dm is that entering the tunnel from the shaft during oscillation. It is here assumed that this portion has no velocity component in the direction of the tunnel before entering the tunnel.

During the time dt, the water mass entering the tunnel is

$$dm = -\rho \cdot A_c \cdot \frac{dZ}{dt} \qquad \ldots\ldots(5.5)$$

Whereas the watermass in the tunnel is

$$m = \rho \cdot L \cdot A_t \qquad \ldots\ldots(5.6)$$

As seen from Fig.5.1

$$h_r - h_c = Z - L \text{ Sin } \alpha \qquad \ldots\ldots(5.7)$$

The expansion and contraction of the enclosed air is governed by the relation

$$P . V^n = P_o . V_o^n \qquad .....(5.8)$$

Neglecting the small terms dm, dv in Eq.5.4 letting $\rho .g. = \gamma$ and eliminating P, h_c, h_r and α from Eq.5.4 to 5.8 one gets

$$\frac{L}{g} . \frac{dv}{dt} = Z - \beta v^2 + \frac{v . A_a}{g . A_t} . \frac{dZ}{dt} + \frac{1}{\gamma}(P_a - \frac{V_o^n . P_o}{V^n}) \qquad ..(5.9$$

5.3.2. Continuity Equation

With the direction defined in Fig.5.1 the continuity equation becomes :

$$Q = A_a . \frac{dZ}{dt} + A_t . v \qquad .....(5.10)$$

5.3.3. Power Equation

For ideal regulation, Eq.5.3 gives,

$$\eta . Q(H - Z + \frac{P}{\gamma} - \frac{P_a}{\gamma} + \frac{v^2}{2g}) = \eta_o . Q_o \; (H - Z_o + \frac{P_o}{\gamma} - \frac{Pa_o}{\gamma} + \frac{v_o^2}{2g}) \qquad .....(5.11)$$

Where $$\frac{P_o}{\gamma} = Z_o + \frac{P_a}{\gamma} - h_{fo} \qquad .....(5.12)$$

H = Gross head in Power Plant.

Combining Eq.5.11 and 5.12 and substituting for P from Eq.5.8, yields

$$\eta . Q . (H - Z + \frac{V_o^n}{V^n} . \frac{P_o}{\gamma} - \frac{P_a}{\gamma} + \frac{v^2}{2g}) = \eta_o Q_o (H - h_{fo} + \frac{v_o^2}{2g}) \qquad .....(5.13)$$

5.4 STABILITY ANALYSIS

As previously stated, $\triangle$ values are very small, but finite deviations from stationary equilibrium.

Let $v = v_o + \Delta v$ ..... (5.14)

$Z = Z_o + \Delta Z$ ..... (5.15)

$Q = Q_o + \Delta Q$ ..... (5.16)

$\eta = \eta_o + \Delta \eta$ ..... (5.17)

$P = P_o + \Delta P$ ..... (5.18)

$V = V_o + \Delta V = V_o + A_a \cdot \Delta Z$ ..... (5.19)

Eq.5.8 gives, for small deviations from the stationary values,

$$\Delta (P.V)^n = \frac{\delta\, P.V^n}{\delta P} \Delta P + \frac{\delta\, P.\ V^n}{\delta V} \cdot \Delta V = 0 \qquad ..... (5.20)$$

or

$$V^n . \Delta P + P.\ V^{n-1} \cdot \Delta V = 0 \qquad ..... (5.21)$$

Combining Eq.5.18, 5.19 and 5.21 and neglecting small terms of higher orders yields

$$\Delta P = -n \cdot \frac{P_o}{V_o} \cdot A_a \cdot \Delta Z \qquad ..... (5.22)$$

Let further

$$l_o = \frac{V_o}{A_a} \qquad ..... (5.23)$$

$$F = 1 + \frac{n.\ P_o}{l_o\, \gamma} \qquad ..... (5.24)$$

$$E = 1 + \frac{Q_o}{\eta_o} \cdot \frac{\Delta \eta}{\Delta Q} \qquad ..... (5.25)$$

$$H_o = H - h_{fo} + \frac{v_o^2}{2g} \qquad ..... (5.26)$$

Combining Eq.5.14 to 5.18 and Eq. 5.22 with the basic equations i.e. equation 5.9, 5.10 and 5.13 and neglecting terms of second or higher order, one gets,

$$\frac{L}{g} \cdot d\, \frac{(\Delta v)}{dt} = F . \Delta Z - 2 . \beta \cdot v_o . \Delta v + \frac{v_o . A_a}{g . A_t}\, d\, \frac{(\Delta Z)}{dt} \qquad ...(5.27)$$

$$\Delta Q = A_a \cdot d \frac{(d\ Z)}{dt} \cdot A_t \cdot \Delta v \qquad \text{.....(5.28)}$$

$$E. \quad Q - F\frac{Q_o}{H_o} \cdot \Delta Z + \frac{Q_o \cdot v_o}{H_o \cdot g} \cdot \Delta v = 0 \qquad \text{.....(5.29)}$$

Eliminating v and Q in the above three equations, gives the following second order differential equation in Z with constant coefficients :

$$\frac{L.\ A_a}{g} d^2 \frac{(\Delta Z)}{dt^2} + 2\beta . v_o . A_a - \frac{LF . Q_o}{g . E H_o} + \frac{v_o\ A_a}{g . A_t} (A_t + \frac{Q_o . v_o}{E\ H_o g} \quad d \frac{(\Delta Z)}{dt}$$

$$+ F \left\{ A_t + \frac{Q_o . v_o}{E . H_o g} - 2\beta . \frac{v_o\ Q_o}{E\ H_o} \right\} \Delta Z = 0 \qquad \text{.....(5.30)}$$

If all the coefficients in the characteristic equation, Eq.5.30 are greater or equals to zero, oscillations cannot grow. The decisive criteria is given by the coefficient in front of the term $d \frac{(\Delta Z)}{dt}$, then

$$\left\{ 2\beta . v_o . A_a - \frac{L.F.Q_o}{g.E\ H_o} + \frac{v_o . A_a}{g} + \frac{v_o^2 . A_a\ Q_o}{g^2 . A_t\ E . H_o} \right\} \geq 0 \qquad \text{...(5.31)}$$

or rearranging

$$A_a \geq \frac{L.\ A_t \quad F}{2g(\beta + \frac{1}{2g})E.H_o + 2\frac{v_o^2}{2g}} \qquad \text{..... (5.32)}$$

$$\therefore \text{ The critical area } A_a^* = \frac{L.\ A_t \quad F}{2g(\beta + \frac{1}{2g})\ E.H_o + 2v_o^2/2g} \qquad \text{....(5.33)}$$

In the case of an open surge chamber, the factor l_o = infinity, and F becomes unity. Thus Eq.5.33 gives the following critical area A_s^* for the open surge tank.

$$A_s^* = \frac{L.\ A_t}{2g\ (\beta + \frac{1}{2g})\ E.H_o + \frac{2\ v_o^2}{2g}} \qquad .....(5.34)$$

or when substituting for E and H_o from Eqs.5.25 and 5.26 one gets;

$$A^*_s = \frac{L.\ A_t}{2g(\beta + \frac{1}{2g})(H_o - h_{fo} + \frac{v_o^2}{2g})\ (1 + \frac{Q_o}{\eta_o}.\frac{\Delta\eta}{\Delta Q}) + \frac{2v_o^2}{2g}}$$

$$.....(5.35)$$

The critical area A_s for the enclosed surge chamber may in accordance with Eqs.5.33, 5.34 and 5.24 be written as

$$A^*_s = A^*_s\ (\ 1 + \frac{n.P_o}{\gamma.l_o}\ ) \qquad .....(5.36)$$

The latter equation may also be obtained by applying the condition of equal pressure deviation from the stationary condition in an open and an enclosed chamber due to a discharge variation from Q_o to $(Q_o + \Delta Q)$ for ideal regulation.

The pressure increase due to a small rise Δs in an open chamber is $\gamma . \Delta s$, whereas the corresponding pressure rise in an enclosed chamber is completed by a water rise by Δh_c and by an air pressure rise ΔP. Thus the condition of equal pressure rise in the two case leads

$$\Delta P + \gamma . \Delta h_c = \gamma . \Delta s \qquad .....(5.37)$$

Where $\Delta h_c = - \Delta Z$ and ΔP is given by Eq.5.22.

For identical regulation in the two cases, the same amount of water enters the two surge chambers for a given load variation. Thus

$$\Delta h_c \cdot A_a = \Delta s \cdot A_s \qquad \text{.....(5.38)}$$

By combining Eqs. 5.37, 5.22 and 5.38, one will get

$$A_a = A_s \left(1 + \frac{n \, p_o}{l_o} \right) \qquad \text{..... (5.39)}$$

which for the critical area is the same equation as Eq. 5.36.

5.5 LIMITATIONS

(1) The calculation of critical area A^*_a for an enclosed compressed air cushion surge chamber can according to Eq. 5.39 first be carried out for an open surge chamber giving A^*_s for the system in question. The critical area of an enclosed surge chamber is then given by Eq. 5.36. It should be emphasized that such a procedure is only possible when determining critical areas i.e. for small oscillations.

(2) For calculations of large oscillations, such as determination of the upper and lower mass oscillation limits by sudden closure or start of the Power Plant, a numerical integration of the basic equations is required. This is due to the non-linear working diagram for the air cushion. It should be mentioned, however, that both the upper and the lower oscillation limits turn out to be relatively moderate for the air cushion design.

(3) The amount of air leakage represents an uncertain factor as far as the practical aspect of the air cushion solution is concerned.

(4) As stated earlier, the action of air is neither isothermal nor adiabatic. The expansion (or compression) is of polytropic nature. But the value of the polytropic constant has not been worked out correctly so far. However, a value of 1.2 for the constant appears to be reasonable.

(5) GREEZE[33] proposed a rational approach (RHT) governing the behaviour of the air in the chamber. He found that the temperature parameter, while being completely absent in the polytropic equation approach, but obviously present in actual installations, plays a vital part in the rational approach. The R.H.T. equation may be considered as a replacement of the polytropic equation.

(6) The drop in temperature below freezing point tends to release latent heat from the water vapour present in the air. This additional form of energy which may be regarded as booster energy, can make the behaviour of the air chamber some what different than the theoretical conclusions.

CHAPTER- VI

PROTOTYPE PERFORMANCE OF AIR CUSHION SURGE CHAMBER

6.1. INTRODUCTION

At this stage it will be interesting to know the prototype behaviour of an Air Cushion Surge Chamber. Though a number of pumping mains in the different parts of the world, were fitted with Air Chamber, the only Power Plant now functioning with an Air Cushion Surge Chamber is Driva Power Plant in Norway. Several high head plants are presently being designed and constructed with Air chamber in Norway. The largest one so far, is the 1200 M.W. Kvilldal Power Station which comprises a 120,000 m^3 air chamber. In the following paragraphs the prototype behaviour of Driva Power Plant will be briefly discussed.

6.2. SALIENT FEATURES

The Driva Power Plant is situated in Trondelag in the North West part of Southern Norway, about 150 kilometres south of Trondheim. The salient features of the project are shown in Fig.6.1.

The difference between H.F.L. and L.W.L. on lake Gjevilvatn is 15 m. thus providing storage of 280 hm^3. The total catchment area is 411 km^2 and the Annual Average runoff 441 hm^3.

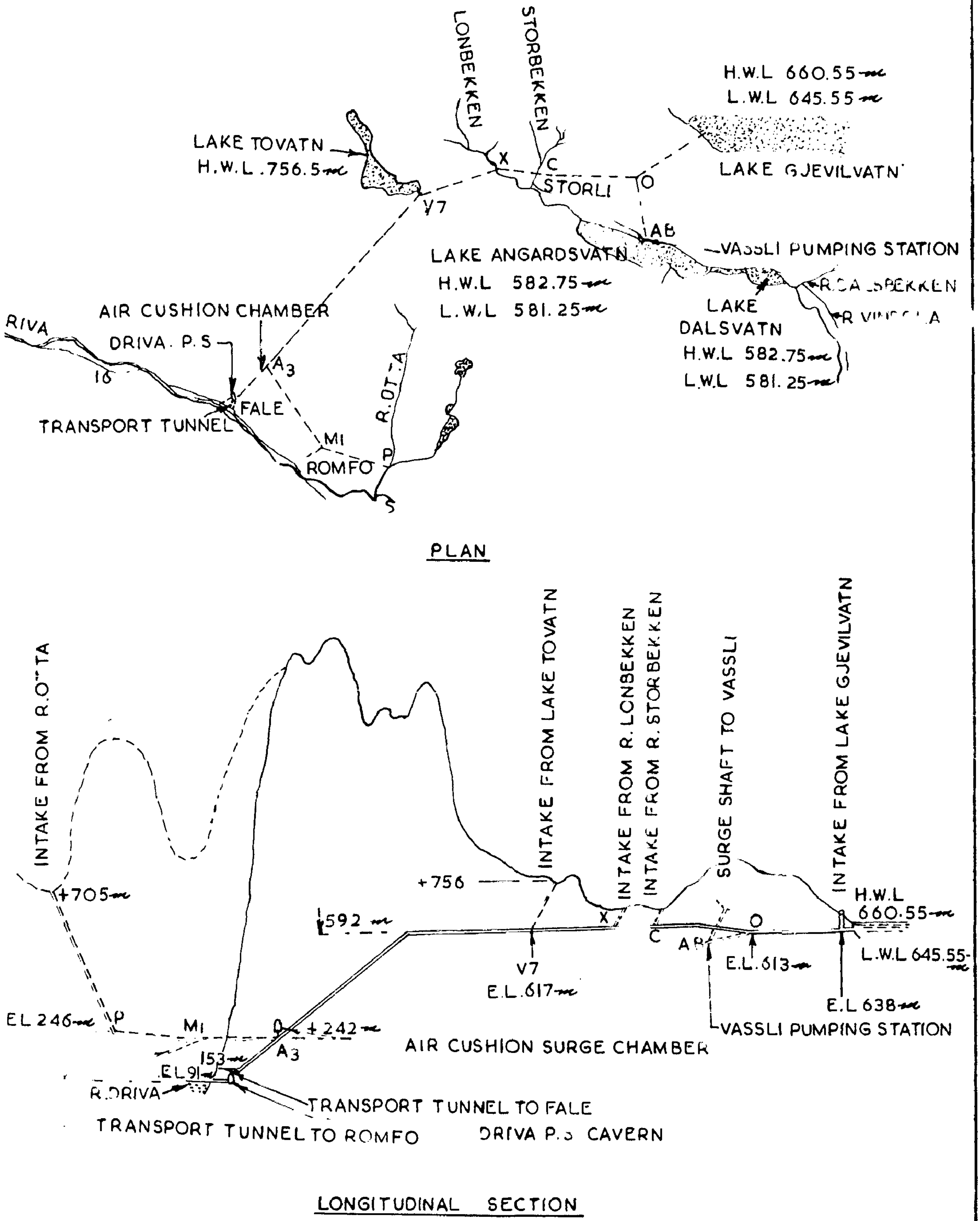


FIG. 6.1 DRIVA POWER SYSTEM - NORWAY.

6.3. WATER CONDUCTOR SYSTEM

As usual several alternatives were studied and a conventional type of surge chamber has been outlined at first for the head race tunnel which is shown in Fig.6.2(a).

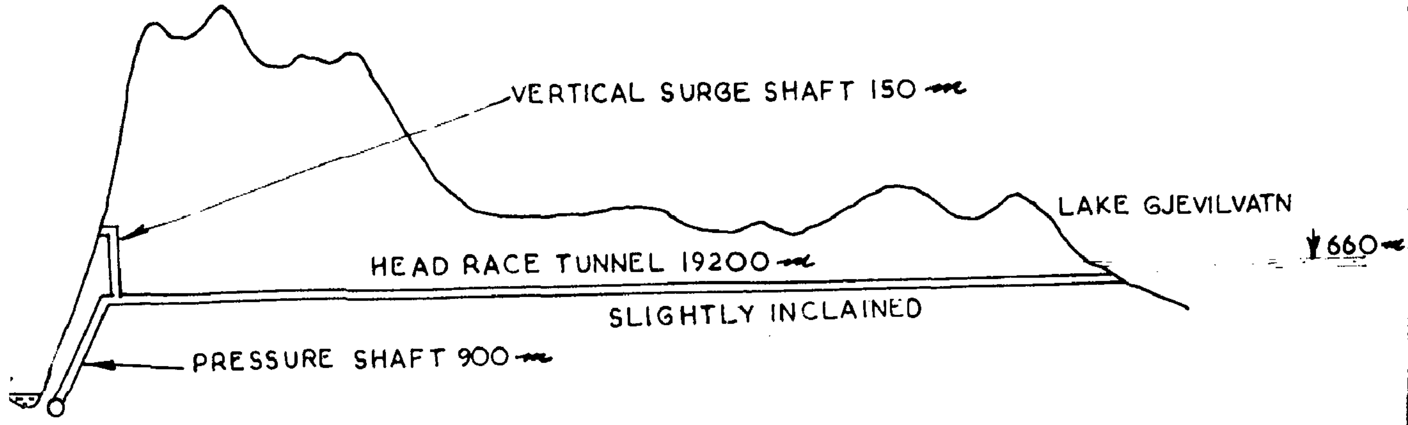
This solution was not acceptable as the head race tunnel had to be excavated from both sides simultaneously. This necessitated construction and maintenance of an access road to El.600, about 500 m above the Power House elevation.

As the sides of the valley are precipitous with frequent rock slides and avalanches, this proposal was very difficult to tackle. Hence the plans had to be changed.

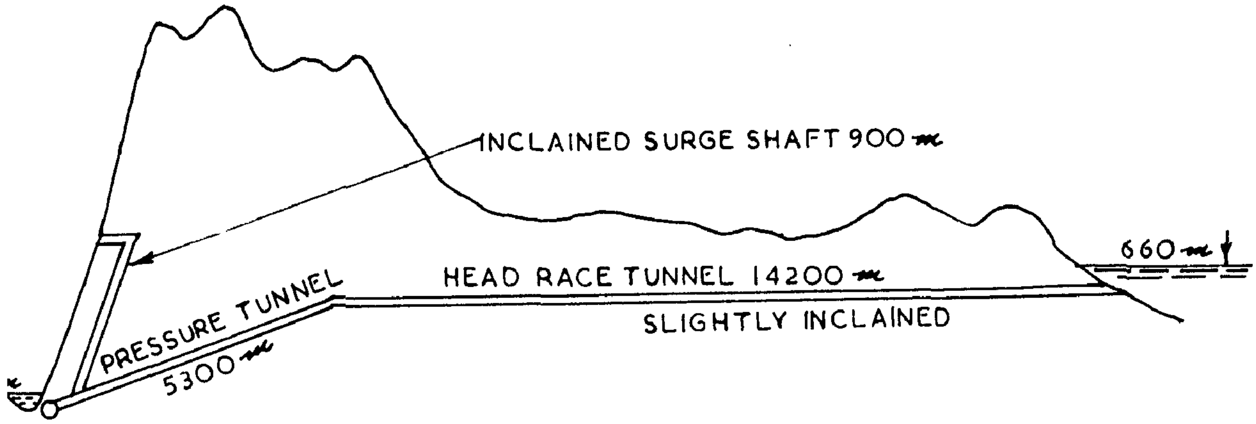
The second proposal was then worked out as shown in Fig.6.2(b). But this proposal had some economical drawbacks. The long surge shaft, which for stability reasons required a minimum cross sectional area of 20 m^2 proved to be very costly.

Then the Chief Engineer of the planning team introduced the idea of replacing the long surge shaft by a short closed air chamber (as shown in Fig.6.2(c) partly filled with compressed air. The compressed air would act like a 'cushion' to reduce the water hammer effect on the hydraulic machinery and the waterways and also ensure the stability of the hydraulic system[64].

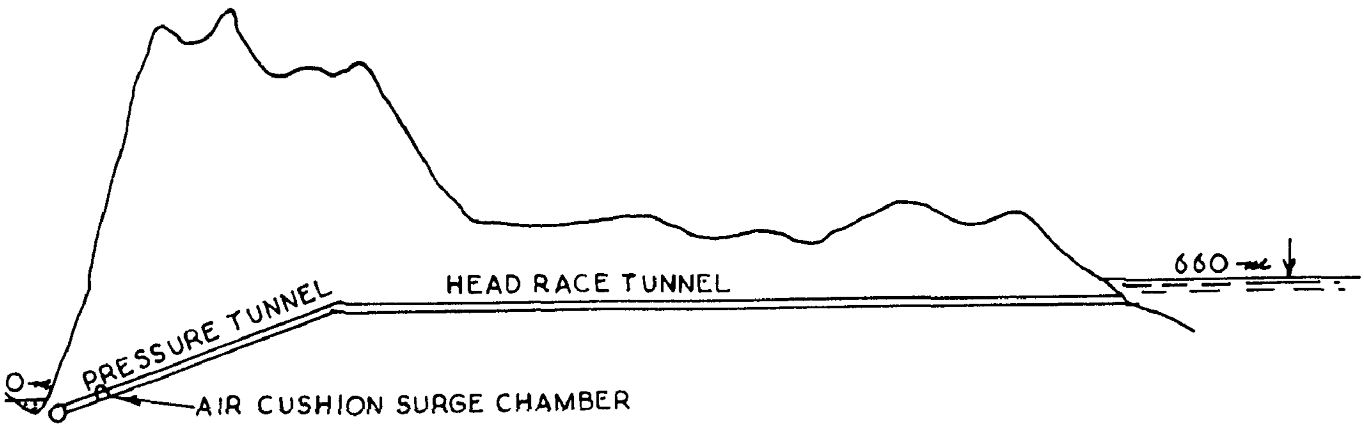
As stated earlier, such chambers were frequently used to suppress resonance in pipe lines, but Driva is the first Power Station, where it has been used for an Hydro Electric development.



6.2a WITH 150 m VERTICAL SURGE SHAFT



6.2b WITH 900 m INCLAINED SURGE SHAFT



6.2c WITH AIR CUSHION SURGE CHAMBER

FIG. 6.2 HEAD RACE WATER WAYS FOR DRIVA POWER PLANT DIFFERENT PROPOSALS.

The preliminary designs for the air cushion surge chamber were done by step by step numerical calculation, assuming that the air in the chamber behaves isothermally. The results so obtained were checked by theoretical studies in the River and Harbour Laboratory at the Technical University of Norway, Trondheim, and proved the soundness of the design. The final design of surge facilities provided at Driva Power Station is shown in Fig.3.6.

6.4. PERFORMANCE

i) The 6,000 cubic metre air chamber of Driva Power Station, filled with 5000 m^3 of compressed air gives a 13.6% rise in the static head on the turbine (H_{max}= 736.4 metres) on the instantaneous shut down of the Power Plant from full to zero load. The turbine manufacturers (Kvaerner Brug A/S Oslo) generally allow for 16% rise in maximum static head.

By way of comparison, a 20 m^2 Surge Shaft as shown in Fig.6.2(b) would give a rise of 7.9% in the maximum static head (H_{max} = 705 metres).

ii) The harmonic resonance tests were carried out during June 1973 in the air chamber at Driva and the results were found to be coinciding. The test was conducted while the air volume in the chamber was only 3000 m^3 (14). This is compared with a surge shaft alternative and found that a 600 metre Surge Shaft will cause an unstable regulating system and hence the stability computation of Driva showed that the air accumulator system was the best alternative.

iii) The air accumulator in Drive which is completely unlined has so far had no appreciable leakages of air and showed no failure in service.

iv) Though R.H.T. system gives a better solution for the value of n, when the transient behaviour commenced with an initial expansion as is typical of an air chamber installation shut down, the excellent agreement between theory and laboratory results was unfortunately not repeated.

The booster energy caused by latent heat may be the reason for the discrepancy in the laboratory value and that got in manipulating the theory.

v) Due to the aforesaid reasons, the air chamber installation can be expected to behave differently during the day when the air in the chamber can become hot than at night when the air is relatively cold, assuming that all the other parameters remain the same.

6.5. PRACTICAL PROBLEMS

Although, theoretically provision of an air chamber seems to be an economical alternative to conventional surge tanks, there may be some practical problems associated with it which are briefly discussed below.

6.5.1. Air Leakage :

In Driva, the problem of air leakage was tested at length during the construction period itself and no trace of air leakage was found, the over burden thickness being about 100 metres.

Three numbers 40 metres deep bore holes were taken for the purpose of testing and a test zone 20 to 40 metres zone was taken. This is due to the likely influence of the unfavourable stress conditions around the chamber. No leakage of air or water was measurable from the bore holes upto a pressure of 60 Kp/cm^2.

In case the rock is not of good quality, air leakage from the chamber to the pressure tunnel and even out to the valley through small fissures of rock may pose a serious problem. In that case, it can be attacked by providing lining of the chamber roof and sides with a special duo-component epoxy coating or by steel lining.

6.6.2. Dissolved air

The second problem may be about the dissolved air in the turbulent water mass and the question of whether this dissolved air can cause cavitation damage to turbine parts. This problem was investigated thoroughly at the River and Harbour Laboratory, Trondheim for another Norwegian Hydel Plant at Jukla for which an air chamber has been proposed. The results indicated a maximum air loss of 0.6 to 1.6 per thousand of the water volume needed for Power production. (The air volume refers at normal atmospheric conditions). It is possible that the air loss is even less than this, as the results were based on very unfavourable assumptions. It was, however noticed that the air loss increased with increased turbulence in the chamber. This problem can be solved by providing the distance between the head race tunnel and the chamber at least 5 to 6 times the diameter of the linking tunnel.

6.5.3. Cavitation

No cavitation was found to develop on the turbines, in fact it was discovered that the presence of dissolved air reduced cavitation risks.

6.5.4. Poisonous gases

The air in the air chamber could become slowly poisonous (low oxygen content, possible content of Hydrogen Sulphide from the deposits of organic material). This will in no way affect the machines. But care should be taken that these gases (or air) from the chamber should not be released through the machine hall.

CHAPTER - VII

ECONOMIC EVALUATION OF AIR CUSHION SURGE CHAMBER

7.1 GENERAL

In the previous Chapter, various technical aspects of the air cushion surge chamber have been discussed. In this Chapter, an economic evaluation is projected. The data for this study is taken from the mammoth Idukki Hydro-Electric Project of Kerala State, where a restricted orifice type surge shaft with upper and lower expansion chambers is provided.

7.2 PROJECT IN BRIEF

The Idukki Hydro-Electric Project is located in Idukki District of Kerala State and is about 80 kilometres South-East of Ernakulam (Cochin) and 80 kms north-east of Kottayam. The entire area of this project is in Western Ghats.

The inflow of the River Periyar and River Cheruthoni will be impounded to a reservoir of 2000 Mm^3 capacity created by the construction of the following dams :-

(1) A concrete double curvature, parabolic thin Arch Dam (fully instrumented) of height 169 m at Idukki Gorge.

(2) A concrete straight gravity dam of height 138.4 m at Cheruthoni, adjacent to Idukki Gorge; and

(3) A composite (Masonry cum-concrete) dam of height 100 m at Kulamavu.

The live storage of Idukki Reservoir is diverted through a 7 m diameter horse shoe conduit from Kulamavu to an Underground Power Station located in the adjacent valley at Moolamattom. The system is designed for development in two Stages for ultimate operation at 30% load factor with a total installed capacity of 780 MW at 0.9 Power Factor. The Schematic profile of the Project is shown in Fig.7.1.

The regulation of inflow (estimated long term average flow) of 40.8 m^3/sec will be ensured by a live storage of 1460 million m^3 between reservoir elevations 694.94 Mtr and 732.62 Mtr. Flood control will ensure a maximum water level at El.734.71 Mtr.

7.3 WATER CONDUCTOR SYSTEM

Water Conductor system comprises a Morning glory intake tower positioned at about 600 metres upstream of Kulamavu dam. The sill level of the intake tower is at El.684.3 m. This is connected to the head race tunnel by means of a 7 m diameter circular conduit of 85 metres length. The head race tunnel is 2028 m long 7 m dia. modified horse shoe section and is designed to carry a peak discharge of 163 m^3/sec. A restricted orifice type inclined surge shaft with upper and lower expansion chambers has been provided at the end of the head race tunnel. From the surge shaft point, the power tunnel bifurcates into two steel lined steeply inclined pressure shafts of 3.81 m dia. with horizontal limbs at top. The butterfly valves were provided at the horizontal portion of the pressure shaft.

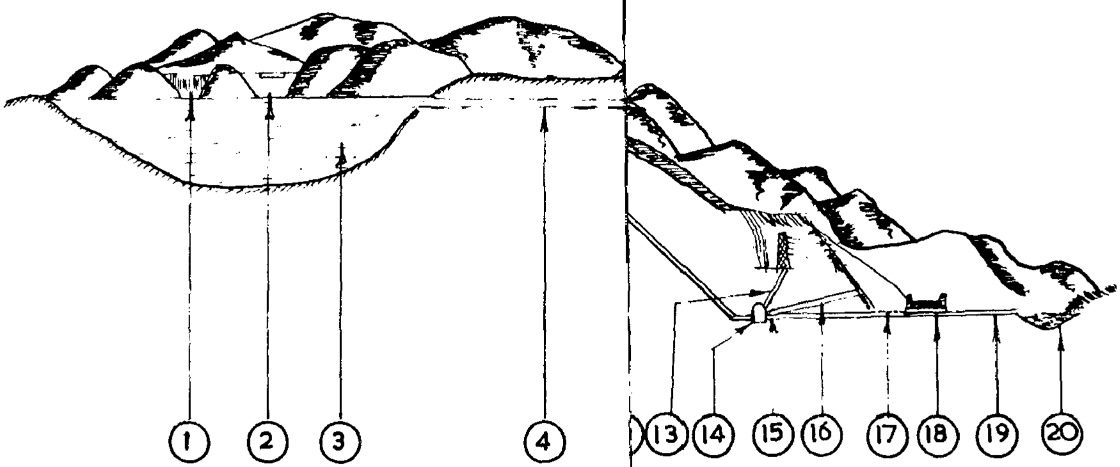


FIG. 7.1. 1D PROFILE

1. IDUKKI AMBER
2. CHERUTH
3. IDUKKI R
4. KULAMAV
5. KULAMAV
6. INTAKE TH.
7. KULAMA
8. GATE SH
9. POWER T
10 SURGE S

The pressure shaft No.1 is 993.34 m long and inclined at 51°.02'.32.3" to the horizontal whereas shaft No.2 is 955.85 m long and inclined at an angle of 52°.37'.24.2" to the horizontal. The power house is equipped with 6 Nos. 180,000 H.P.6 jets 375 rpm. Pelton turbines, operating under a maximum gross head of 679.25 m. The water will be discharged to a nearby stream through a 1220 m free flowing tail race tunnel.

7.4 LAYOUT CONSIDERATIONS

7.4.1. General

The major portion of the planning of Idukki Project was carried out during the early years of last decade. During that period the idea of providing Air Cushion Surge Chamber was not fully developed and so the designers and engineers could think only of a conventional type development with a mildly sloping head race tunnel and a steeply sloping pressure shafts (or penstocks). Hence this project was also outlined in this fashion at the early stage and refinements were made later on.

7.4.2. Head Race Tunnel

Various alternatives for the head race tunnel alignments were studied in detail the present alignment ($A_1V_2S_1$)(Fig.7.2) was considered to be the best alternative due to the following reasons :-

(a) A dam intake taking off from Kulamavu dam, though found technically and economically sound, a separate adit may be needed at tunnel inlet either from upstream or from downstream of dam for tunnelling. This may lead a

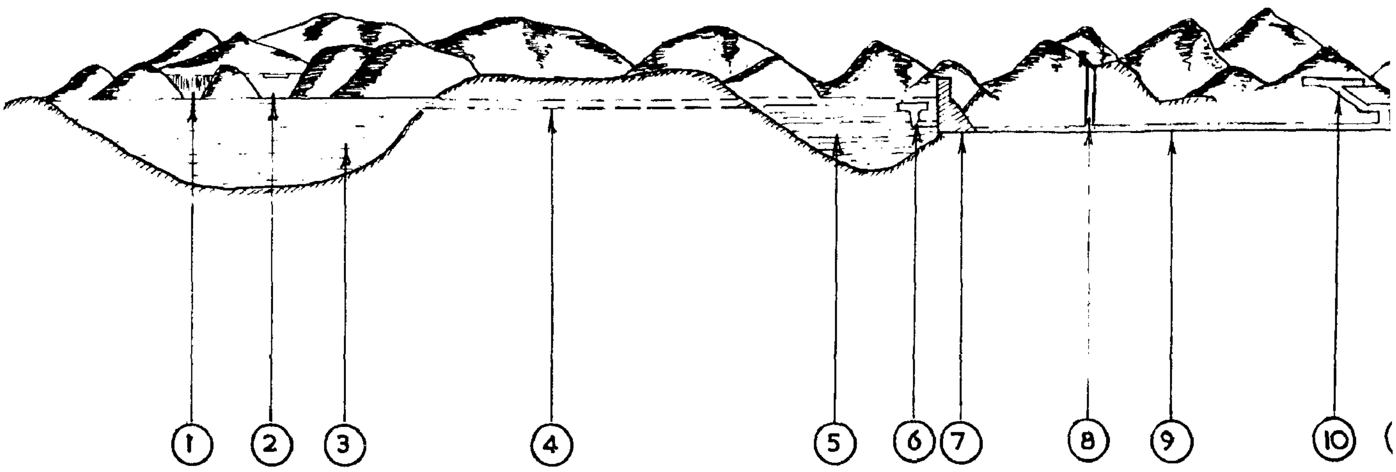


FIG. 7.1. IDUKKI HYDRO ELECTRIC PROJECT-SCHE

1. IDUKKI ARCH DAM
2. CHERUTHONI DAM
3. IDUKKI RESERVOIR
4. KULAMAVU CHANNEL
5. KULAMAVU BASIN
6. INTAKE TOWER
7. KULAMAVU DAM
8. GATE SHAFT
9. POWER TUNNEL
10 SURGE SHAFT
11. BUTTERFLY VAL
12. PRESSURE SHAF
13. CABLE TUNNEL
14. POWER HOUSE C.
15. TAILRACE TUNNI
16. ACCESS TUNNEL
17. TAIL RACE CHAI
18. SUPER PASSAGE
19. BYE PASS CHANNI
20 VALIAR

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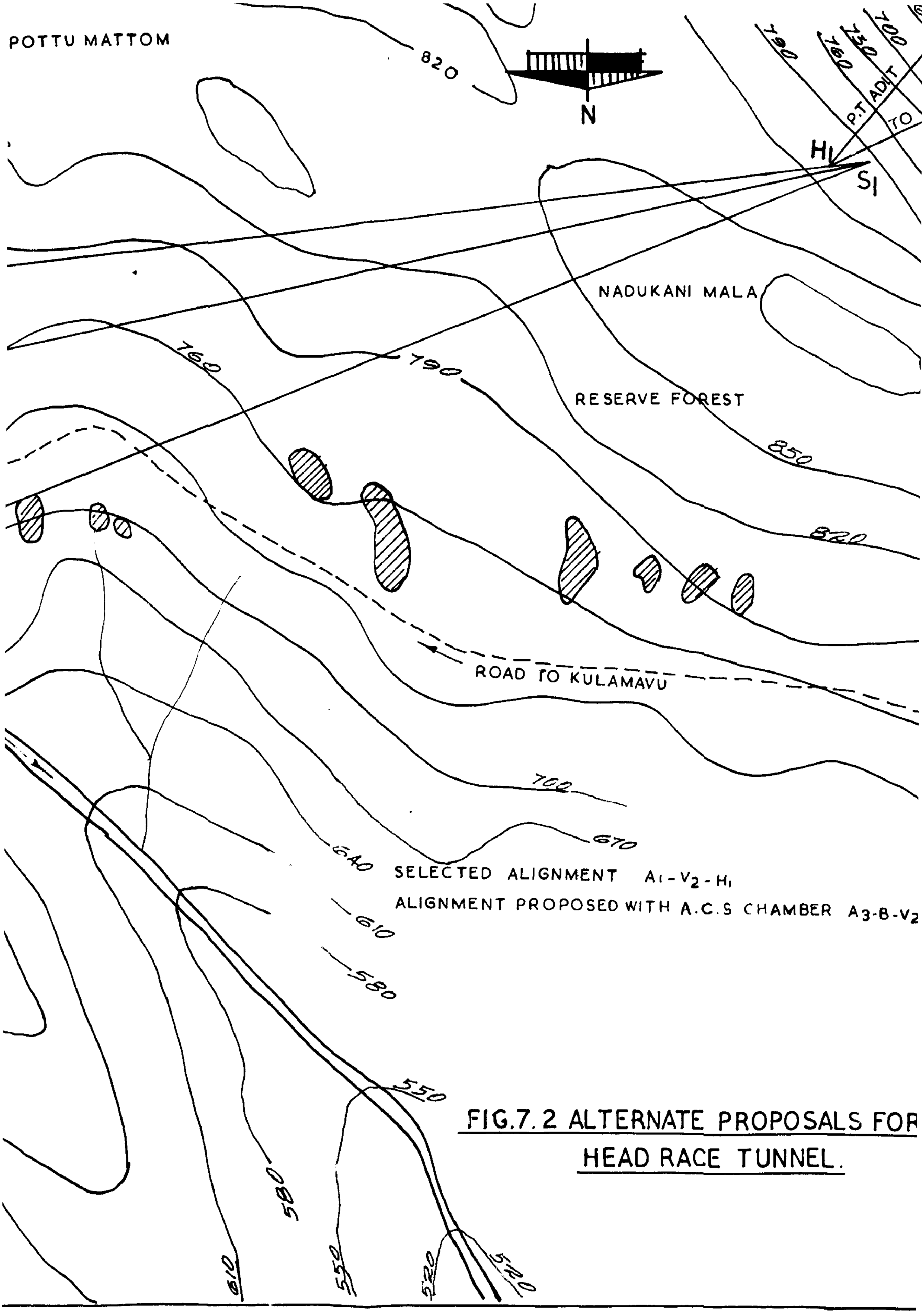


FIG.7.2 ALTERNATE PROPOSALS FOR HEAD RACE TUNNEL.

bit congestion at Kulamavu dam site as well as increase in length of the head race tunnel.

(b) Sufficient rock cover was not available at V_0 for the straight alignment $A_1V_0S_1$.

(c) The alignment $A_1V_1S_1$ was not considered suitable as it had more length than $A_1V_2S_1$ alignment.

The tunnel passes through hard granite rock and hence lining was not structurally necessary. Economic studies proved that the difference in cost was also marginal. Hence it was decided to give a nominal lining of 460 mm minimum thickness to reduce friction factor.

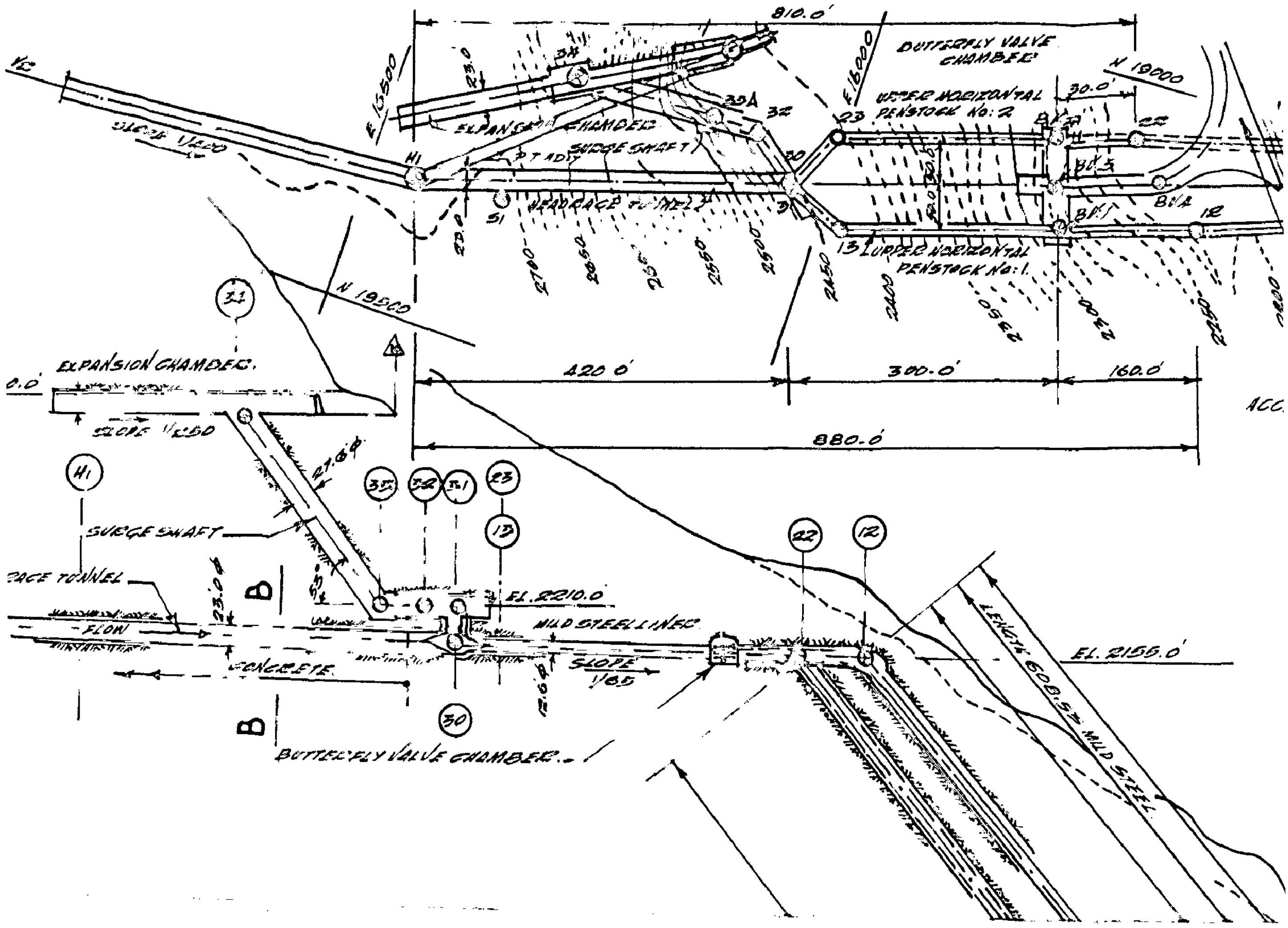
7.4.3. <u>Pressure Shafts</u>

The pressure shafts were defined in relation to the underground power station for the following considerations :

(a) Shaft inclination was selected to achieve self mucking;

(b) Turbine setting with respect to the surface of the ridge slope was selected for optimum rock cover (sufficient cover for resisting inside pressure and to minimise outside pressure from ground water. Two alternatives were considered for pressure shaft alignment. These are (i) Two pressure Shafts running parallel and branching near the Power Station (ii) Two Pressure Shafts cross near the Power Station.

The cost of alternative (ii) worked out to be cheaper and therefore adopted. Economic studies taking into account of Civil works costs, capitalised revenue losses due to head losses and capitalisation have led to two 3.81 m diameter sloping shafts which compared favourably with the largest high head power shafts elsewhere in the world. A single shaft would have to be in the range of 5.5 m diameter and welding with the required thickness of steel would have necessitated very elaborate techniques and therefore abandoned. In addition, considering the development in two stages, a single penstock required large investment too. In the alignment of pressure shaft, it is of obvious interest to go deep enough underground to ensure minimum rock cover required. By going deeper the liner would still have to be designed to the yield stress in considering internal pressure. By going deeper, the requirements of external ground water pressure will govern the design, especially in the upper sections of the penstocks. Initially the design was made with 50% design head as rock cover. Before finalising the design, an intermediate adit at about middle of the shaft was driven for detailed investigation and later to be used for penstock installation. From these studies, it was found possible to make a change in the alignment of penstocks. The inclined part was shifted downstream and it was relocated as close as possible to the manifolds in order to reduce the length of the lower horizontal part. With this arrangement, the penstocks will cross each other and the same at higher elevation will approach the ground surface so that the reduction of external pressure will reduce the thickness of steel. The saving due to reduction of external pressure and length of lower horizontal portion of penstock worked out as 10% of the total cost of penstocks. In general layout of the Idukki Pressure Shaft and Surge shaft is given in Fig.7.3.



PRESSURE SHAFT
SECOND

DEVELOPED PROFIL

100' 0 100'

7.4.4. Surge Shaft

Regarding the type of surge tank various alternatives with conventional type surge chambers were considered. After going in detail into all the alternatives, it was found that a restricted orifice type inclined surge shaft of 8.7 m finished diameter and 76.25 m long having upper and lower expansion chambers as shown in Fig.2.6 was the best solution for this water conductor system and hence adopted.

7.5 FEASIBILITY STUDY OF AIR CUSHION SURGE CHAMBER FOR IDUKKI

7.5.1. It was possible to provide an Air Cushion Surge Chamber in this project instead of the conventional type Surge tank, already provided. For providing the Air Cushion Surge tank economically, the concept of the development with a mildly sloping head race tunnel has to be abandoned and a new Water Conductor System streamlined, which will be entirely different from the conventional type.

With the help of modern tunnelling techniques it is not very much difficult to drive tunnels upto a slope of 1 in 8 efficiently. Mucking can be done with the help of tyred dumpers and loaders as was done in major tunnels of Idukki Project.

7.5.2. It is proposed to conduct only a preliminary study to get a rough idea regarding the economy behind the concept of Air Cushion Surge Chamber. For easier comparison, a new development is planned with minimum changes in the present Water Conductor system. A Schematic profile of the new Water Conductor System with an Air Cushion Surge Tank is shown in Fig.7.4.

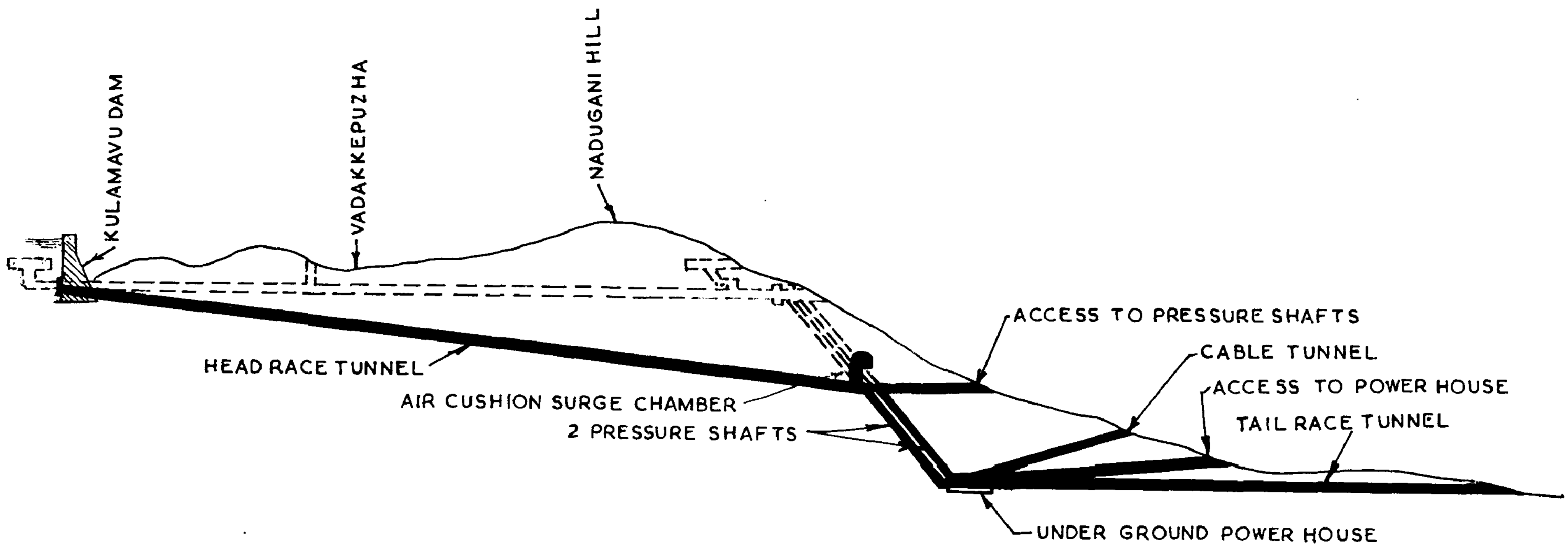


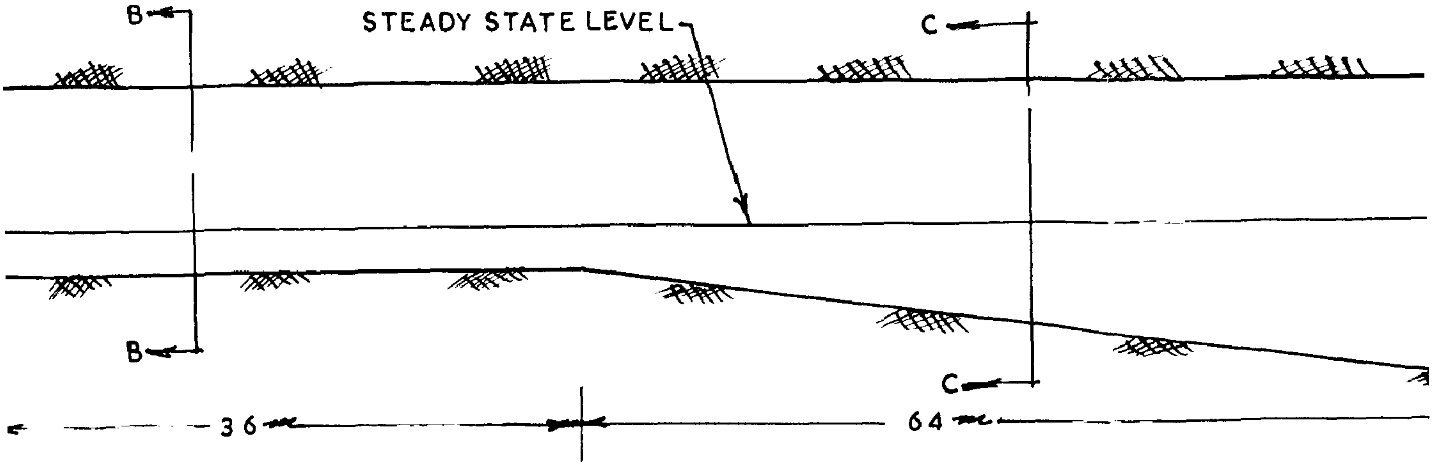
FIG. 7.4 SCHEMATIC PROFILE OF A NEW WATER CONDUCTOR SYSTEM FOR IDUKKI.

The Intake and control gate have been located near the Kulamavu Dam itself, as it is costlier to provide too long control shaft, which will become necessary when the slope of the tunnel increases in the new alignment. Moreover, the modified proposal wi will enable in getting the full length of tunnel under control. Incidently, the complexities involved in the construction of expensive morning glory intake tower and connecting conduit can be avoided.

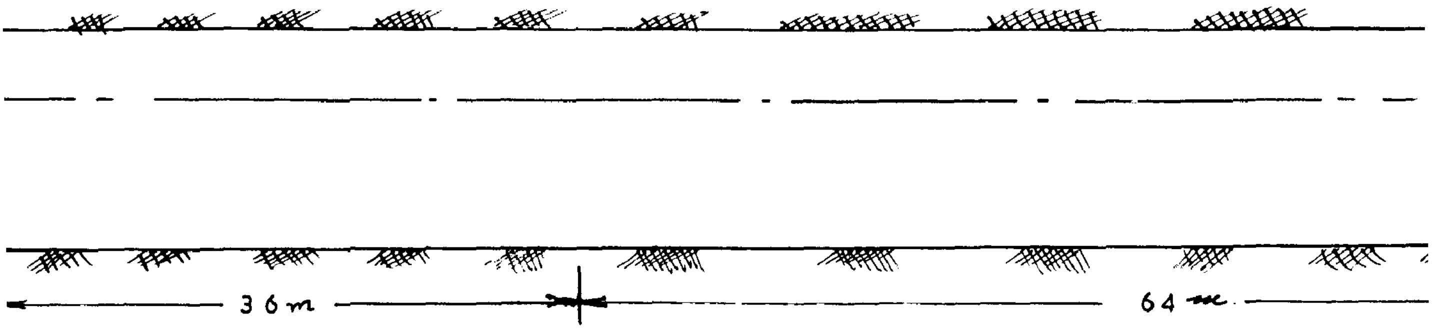
The alignment of the new head race tunnel can be A_3-B-V_2-H_1-ACS (vide Fig.7.2). The tunnel will have a length of 8000' (2438 m) with 1 in 8 bed slope. Sill level of the tunnel inlet at A_3 and exit at ACS will be 677.00 m and 366.00 m. The tunnel is proposed to be kept as unlined having a minimum diameter of 8.5 mts modified horse shoe section. The overbreakage allowance of 10 cms were provided at the outer periphery. It is proposed to drive the tunnel from both ends only.

From the stability considerations an Air Cushion Surge Chamber of size 100.0 x 14.0 x 12.0 metres has to be provided at the exit point of the head race tunnel. The general layout of the Air Cushion Surge Chamber for Idukki is shown in Fig.7.5. The design of the air cushion surge chamber is given in Appendix 1.

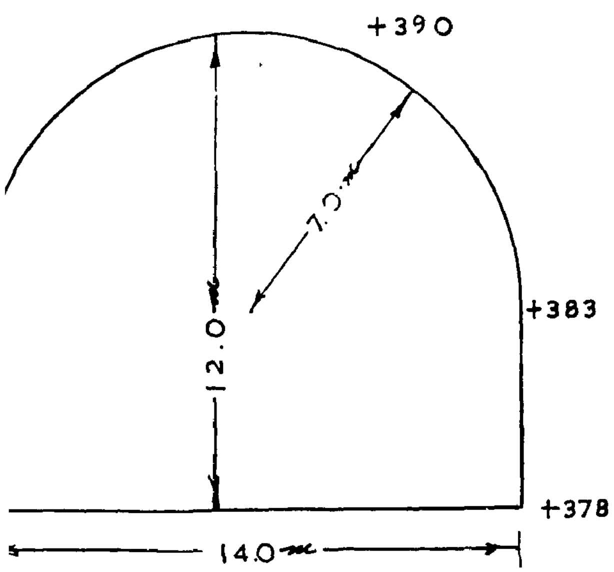
Provision of 2 Nos. Spherical valves has been made at the upper horizontal limb of the pressure shafts. The valve House can be at the existing intermediate adit area. This is in lieu of the butterfly valves already provided at the exit of Power Tunnel.

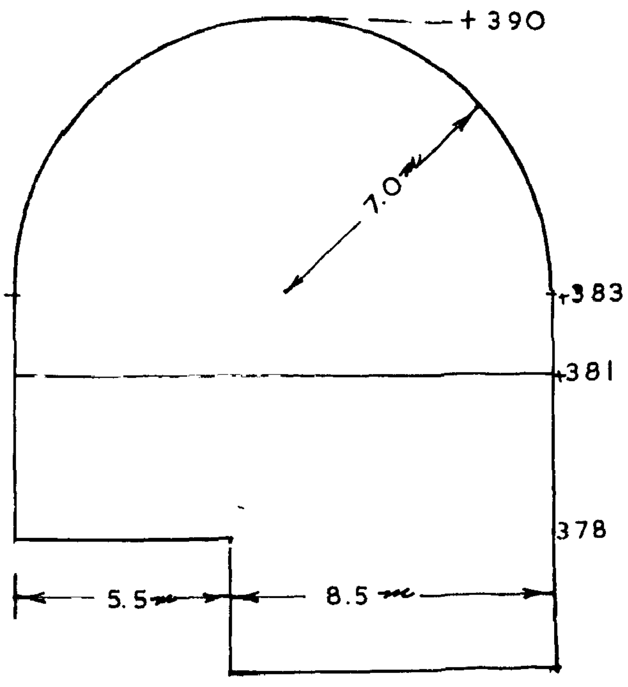


SECTION.A.A



PLAN.





SECTION.B.B. SCALE.1:200.

SECTION.C.C SCALE.1:200.

Necessary compressors for supplying air to the chamber can be installed outside the Intermediate adit tunnel as the length of the conduit is not much or even in a separate cavity near the valve house.

Precisely the Air Cushion Surge Chamber of Idukki will have 1/4th volume of the Power House cavern. The works of the cavity can be attacked through the linking tunnel. Gunite lining is proposed in the Surge Chamber as well as in head race tunnel.

The existing water conductor system below points 11 and 12 (Fig.7.3) is proposed to kept as such. The length of the two inclined pressure shafts will be reduced to 399.8 m and 391.2 m which will be just half the original length. The total savings in the new proposal work out approximately 23 percent of the cost of the Water Conductor System.

7.5.3. The maximum surges for full load rejection and for full load acceptance (although full load acceptance is not generally followed in practice), have been worked out, assuming the initial pressure at steady state condition, is the same as of that of Drive Air Chamber, for isothermal, adiabatic and polytropic (n:1.2) compression and expension and shown in Appendix 2.

However detailed calculations have to be carried out for optimizing the initial volume and pressure of the air mass in the chamber, so as to reflect the minimum water hammer pressures in to the head race tunnel. Positioning of the air chamber may also has to be studied in detail.

CHAPTER - VIII

CONCLUSIONS

1. For high head power plants, surge tank is an essential component. However, conventional type surge tanks are normally very costly. Air Cushion Surge tank can be an economic alternative for a hydro electric scheme fed by a long pressure conduit. This has been tried at Driva Power Plant, Norway and is functioning satisfactorily. Several high head plants with Air Cushion Surge tanks are now being designed and constructed in Norway.

2. In an air cushion surge chamber, oscillations in the water surface due to load fluctuations are taken care by the compression and expansion of the air above the water surface. It has been found that the upper and lower surge limits are moderated considerably.

3. Air Cushion Surge tanks are specially suitable for hydro electric projects where long pressure shafts and surge shafts are necessary. In many cases, where the hydraulic gradient line is above the rock surface line, Air Cushion Surge tank may be the only economically feasible solution.

4. The new concept of Air Cushion Surge tank could be adopted successfully in this country, where rocks are of good quality, as in the case of Western Ghats. This can be provided even in Himalayan geology (where rocks are very poor comparatively) if adequate precautionary measures to prevent air leakage etc. are taken.

5. The adoption of Air Cushion Surge tank is likely to reduce the construction cost of the water conductor system considerably. A preliminary study for the comparitive cost of conventional type surge tank and Air Cushion Surge tank made for the recently commissioned Idukki Hydro-Electric Project of Kerala State, shows that the provision of Air Cushion Surge Chamber reduces the cost of the water conductor system by about 23%.

6. It has not yet been possible to define the law governing the expansion and compression of air cushion accurately. The process is neither adiabatic nor isothermal. But for all practical purposes, this pes process can be considered as polytropic ($P.V^n$ = a constant), assuming the value of n = 1.2.

7. Air Cushion Surge tanks with restricted orifice also can be developed successfully. The diameter of the orifice should be optimized to get better performance.

8. Sufficient laboratory as well as field data is yet to be collected to verify the various assumptions in the stability analysis with an air cushions surge chamber.

9. Precautions have to be taken to guard against any poisonous gases being released into the atmosphere.

10. The effect of the dissolved air on the efficient functioning of the turbines has also to be investigated in details and precautions taken to guard against any adverse effects.

CHAPTER - IX

SUGGESTIONS FOR FUTURE RESEARCH WORK

Even now the works on air cushion surge chambers are at its infancy. There is ample scope for further research work in the following fields.

1. Polytropic Constant 'n'

The exact behaviour of air mass in the air chamber is still unknown. Hence further research work has to be carried out to find out the exact value of 'n'.

2. Poisonous Gases

The air in the chamber could become slowly poisonous. Though it will not in any way affect the machines, the biological aspects involved in releasing the poisonous gases to atmosphere when it is absolutely necessary to do so have to be studied in detail.

3. Dissolved Air

Though the quantum of dissolved air in the turbulent air mass is negligible, it can cause appreciable power loss and cavitation to the machines. This has to be analysed at length in each case.

4. Air Leakage

As stated in para 6.5.1, air leakage ... from the chamber can be a big problem when the rock is not of good quality. Hence cheaper methods of air proof lining should be foundout.

5. <u>Provision of Orifice in Air Chamber</u>

The effect of providing a restricted orifice for the air cushion surge chamber will be the next stage of development.

6. <u>Verification of theoretical laws</u>

Since as stated above, whole mechanism of air cushion surge chamber is in its infancy, more and more prototype data is necessary to verify the assumptions in the theory.

APPENDIX - I

DESIGN OF AIR CUSHION SURGE TANK FOR IDUKKI

(a) Data

Length of head race tunnel	L	= 2520 m
Area of tunnel	A_t	= 60.26 m^2
Maximum discharge	Q	= 140.17 m^3/sec
Maximum velocity in the tunnel	v	= 2.33 m/sec
Net head on turbine	H_o	= 640.0 m
Total head loss upstream of surge tank using Manning's n (minimum)=0.025		$h_f = \beta v^2 = 158.56 \times 10^{-6} Q$ = 3.12 m
Specific weight of water		= 1000 kg/m^3

(b) Assumptions

The following assumptions are based on the design of Driva Air Cushion Surge Chamber:

Initial pressure in Air Chamber	P_o	= 36.0 kg/cm^2
Polytropic constant	n	= 1.2
Length of air column in Chamber at steady state condition	l_o	= 9.0 m

(c) Computations

Thoma area of open surge chamber = $A_{Th} = \dfrac{L\,A_t}{\beta . v^2 . H_o} \cdot \dfrac{v^2}{2g}$

$= 21.0\ m^2$

$= A^*_s$ (Critical area)

Critical area of air cushion surge chamber

$$= A^*_o = A^*_s \left(1 + \frac{n . P_o}{\gamma . l_o} \right) = 1029\ m^2$$

Provide an Air Cushion Surge Chamber of Size

100 m x 14 m x 12 m

Area of air Chamber provided = 100x14 = 1400 m^2

APPENDIX - II

COMPARATIVE SURGES AND PRESSURES IN AIR CHAMBER FOR IDUKKI

a) Assumptions

1. The size of the Air Chamber is as per Appendix - I
2. Computations have been made using Eqs. 3.1 to 3.5 and 3.10.
3. At maximum upsurge or downsurge condition, velocity of water in the head race tunnel, $v_2 = 0$.

b) Comparative Results

Type of Compression or Expansion	Load thrown off (100% 0%)			Load thrown on (0% 100%)		
	Maximum upsurge y	Time taken to reach y t	Maximum absolute pressure in air chamber P_2	Maximum down-surge y	Time taken to reach y t	Maximu absolu to pre ssure in air Chamb
	m	sec	kg/cm^2	m	sec	kg/cm^2
Isothermal	1.14	19.77	41.24	1.31	22.66	31.43
Adiabatic	0.96	16.65	42.20	1.10	19.10	30.60
Polytropic	1.05	18.04	41.78	1.20	20.68	30.98

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