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EUROPEAN EXPERIENCE WITH THE THERMODYNAMIC METHOD

by
PROF. HANS GERBER*

INTRODUCTION

It is, as far as I know, likely the first time, that this new method for efficiency measurement on hydraulic machines is presented in public in this country. I will try therefore to produce in this paper some more detailed information than was possible to include in my lecture.

All those who wish to deal in an extensive manner with this new testing method, its theoretical background and the experiences recollected up-to-date, will find at the end a selection of publications especially chosen for this purpose. The author feels that with these publications the whole field is covered in a reasonable manner, at least for the first start.

FIELD TESTING

When a hydraulic machine is ordered, three main things generally are fixed in the contract, namely:

Price
Delivery time
Technical guarantees

In the future we shall deal with the third item only. In the case of a turbine these guarantee values usually fix, for a given rated speed and at constant head, the efficiencies at full or partial gate openings, that is the ratio between turbine output on the shaft and the hydraulic power input. For low head river plants, and for storage plants where the head may vary substantially, such guarantee curves are claimed for several heads covering the whole range of future operation.

For completeness may I mention that in former times we had to state guarantee values even for different rated speeds, for example in Italy before World War II, for frequencies of 42, 45 and 50 Hertz in the same plant. For storage pumps these guarantees generally include the values of flow and efficiency for given pump heads,

sometimes for speeds a few percent above or below rated speed, in order to take into account extreme network conditions.

The control of these guarantee values is subject to inaccuracies in the measuring techniques involved in the tests, and the definitions of the values to be measured are, fortunately may I say, now fixed in national and international codes. In our field of hydraulic machinery the chair of the international committee dealing with this code work is occupied in an exemplary manner by the Director of the Alden Research Laboratory, Professor Leslie J. Hooper of WPI.

IMPORTANCE OF FIELD TESTS

It must be emphasized that the importance of field tests, i.e., the checking of the guarantee values on the prototypes, is growing more and more as the machines are increasing in power, dimensions, head and flow. The costs of the hydraulic equipment may vary between 4% only in high head plants up to perhaps 15% in a low head river plant, referring to the overall costs of the whole plant. But if the hydraulic machine shows bad performance, the productiveness of the whole investment, possibly as much as several hundred million dollars, is badly affected. Actually tests on prototypes in power plants may be executed for three main and very distinct purposes: a) Checking of guarantee figures given in the contract; this is a commercial matter. b) Comparison with the results of model tests, which is of highly scientific interest. c) Watching over the influence of wear occurring by sand erosion or cavitation corrosion, thus proving to be of highly economical importance for the long time running of the plant. It depends mostly on local conditions, and sometimes on the character and quality of the people involved in the management of the plant, which one of these three items is considered to be the most important one. Based on almost 40 years of experience it is my feeling that the third item becomes more and more important.

Field tests, properly executed, are expensive, need important equipment with the corresponding well trained personnel, and are often disturbing the plant service in a considerable manner. With the improvement of model testing techniques today, field tests are often replaced by acceptance tests on a model, by using the experiences of item b) of my list. With this development in mind plant superintendents have for a long time been interested in simple methods to deal with item c) to check more or less regularly the decrease of efficiency of their machines over a long time range. For this purpose many types of so-called index tests have been developed in the past,

quite none of them really satisfactory in all circumstances.

It may easily be that in the future, the Thermodynamic Method is called to cover this area in an improved manner, but it is a delicate method and must be properly used.

PRINCIPLE

The principle of this new method proves to be very simple: almost all losses occurring in hydraulic machines are transferred in heat, therefore increasing the temperature of the water going through the machine. Measuring the temperature rise between inlet and outlet of the machine and knowing the specific heat of water, the losses can be evaluated in HP or kW and, as the input of a pump, or output of a turbine are known or measured, the flow can be calculated easily.

HISTORICAL SURVEY

Since the beginning in 1914 the method was especially investigated in France. First Poirson, alone, later together with Barbillon, tried to measure these very small temperature differences of about 0.3°C as a mean value. Professor Piccard too, when teaching physics at our school at Zurich, dealt with the problem. But the thermometers available at that time proved to be not adequate; the results were absolutely unsatisfactory and the method was considered a failure and was abandoned for many years.

It was therefore a great step ahead when after World War II in 1954 the two French research men, Willm and Campmas, together with other engineers of Electricité de France, replaced the direct *thermometric* method by the so-called *Thermodynamic Method*. The main difference consists, as we will see later on, that no temperatures are measured directly, but compared only, and pressures are measured which, when handled in a special expander, are producing the same temperature rise as in the machine to be tested. From this beginning and with increasing success a great number of field tests, far more than 1,000, have been executed on turbines and, since 1958 on pumps also. Many publications show this progress (see Bibliography, annexed).

The Swiss Committee for Field Test Codes was at the beginning rather reserved. In the 1957 third edition of the Swiss Test Code we mentioned the method, but it was considered that the method was not yet developed in such a manner to be incorporated in a code as a standard method, but both manufacturers and power plant people

were encouraged to deal with it in arranging as many comparative tests as possible. This manner of approach happened to be very successful so that in the 4th edition just now underway, we will find a complete chapter on the method, which is now accepted as a standard for field tests, within the limits fixed within the up-to-date experiences for turbines at 330 feet, for pumps at 500 feet, approximately.

ADVANTAGES AND DISADVANTAGES

The advantages of the new method in comparison to the classical ones can be presented as follows:

a) No flow measurement necessary
b) No precise electric power measurement necessary
c) Few personnel (1 to 2 men) necessary to operate the instruments, depending on their design
d) Relative short time necessary for installation of instruments, practically without disturbing normal service, and prompt availability of the results.

The difficulties in the use of this method prove to be rather frequently underestimated, not by the specialists themselves, but by others in manufacturer companies and power plant staffs wishing to work with it. Therefore it is convenient and necessary to mention the main disadvantages.

a) For successful work with this method a well and especially trained personnel with clear knowledge in physical and in general engineering measuring technique is required.
b) A small part of the data necessary for the computation of the final efficiency values can only be measured in an approximate manner, or has even to be estimated.
c) The immediate efficiency result is not the overall guarantee efficiency value of turbine or pump, but the hydraulic efficiency, as indirectly mentioned before.

Therefore all other supplementary losses, especially the mechanical losses in the bearings, have to be measured separately or estimated by computing.

BASIC RELATIONS AND EQUATIONS

The main symbols and definitions to be used are recollected in Table (I). They correspond as closely as possible to those of the International (IEC) Code, and therefore to the new chapter on the

method in the Swiss Field Code, already mentioned before. With this it was given to use the metric system, with the exception of heads, in feet, for some power plants, referred to in chapter 14.

Q	[m^3/sec lit/sec]	rate of flow (discharge)
d D	[m]	diameter of circular pipe
a A	[m^2]	sectional area
$v = \frac{Q}{a} \frac{Q}{A}$	[m/sec]	Mean velocity
$v^2/2g$	[m]	kinetic or velocity head
p	[kp/cm^2]	pressure
z	[m]	elevation above zero-level
H	[m]	head
Θ	[°C]	temperature
P	[BHP kW]	Power
g	[m/sec^2]	gravity value
γ	[kp/m^3]	specific weight
d		scale value of galvanometer
η		efficiency

TABLE I: Symbols, Dimensions and Definition

As basic equations we are using:

$H = \frac{p}{\gamma} + \frac{v^2}{2g} + z$ = Hydraulic energy per 1 kp (force), expressed in (m) of water column

= Bernoulli's total energy

U = Internal energy per 1 kp (force)

$(H + U) = \underline{H}$ = Total energy per 1 kp (force or weight)

H_u = Total energy per 1 kp, furnished by the turbine impeller to the shaft

H_u = Total energy per 1 kp, furnished to the pump runner from the shaft

Referring to Fig (1), we have the locations:

Subscript I : Low pressure side of the machine

Subscript II : High pressure side of the machine

Subscript m : Mean values

Subscript eT, eP : Entrance in the range of responsibility of the Turbine or the Pump

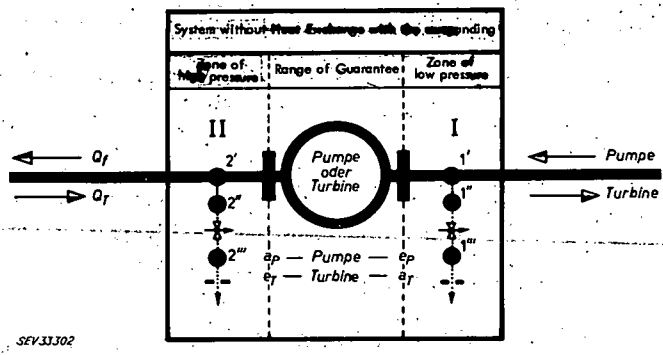
Subscript aT, aP : Leaving the range of responsibility of the Turbine or the Pump

Following the principle of the conservation of the total energy, for a complete system without heat exchange with the ambient room, and in using Fig. (1) we get:

Total energy on the side of the high pressure	=	Total energy on the side of low pressure	+	Energy on the shaft

or

$$\{H+U\}_{II} = \{H+U\}_{I} + \{H_u\}$$



Basic Scheme for Measurement of Energy - and Efficiency - Conditions with Double - Expanders in both measureing Zones.

● Measuring Locations for pressure and Temperature

Small size Butterfly - valves with adjustable positions

-- Orifices

a_P a_T Outlet Sections — e_P e_T Inlet Sections

Q_P Pump discharge — Q_T Turbine discharge

FIG. 1: Basic Diagram for the location of the Measuring Sections.

As long as the measuring points 1 and 2 remain within the whole region with no heat exchange with the ambient room, they may be located at any suitable section of flow, that means where $H_U = 0$.

Referring once more to Fig. (1), showing the fact that location of measuring points within a closed area is of no importance, we state:

$$(H+U)_{II} = (H+U)_2' = (H+U)_2'' = (H+U)_2'''$$

and

$$(H+U)_{I} = (H+U)_1' = (H+U)_1'' = (H+U)_1'''$$

H_U as the difference of the two (H + U) values may therefore be measured between any freely chosen location within the Zones I and II.

Following the laws of thermodynamics we have:

$$H_U = \left[\frac{p_{II}-p_I}{\gamma_0}\right]\cdot\left(1-\alpha_m-\beta_m\right)+J\cdot\frac{c_{p_I}}{g}\cdot\left(\theta_{II}-\theta_I\right)+\left(\frac{v_{II}^2-v_I^2}{g}\right)+\left(z_{II}-z_I\right)$$

or finally, as all is measured in [m] of water column:

$$H_U = 10\left(p_{II}-p_I\right)\cdot\left(1-\alpha_m-\beta_m\right)+427\cdot\left(\theta_{II}-\theta_I\right)+\left(\frac{v_{II}^2-v_I^2}{g}\right)+\left(z_{II}-z_I\right)$$

p = pressure in kp/cm²

θ = Temperature of water in °C

α_m = Correction coefficient # 1

$\alpha_m\cdot\frac{g}{J\cdot c_{p_I}}\cdot\left(\frac{\Delta p}{\gamma}\right)=\Delta\theta_{ad}$ = adiabatic change of temperature at efficiency = 1

β_m = Correction coefficient # 2, referring to the differences between γ_0 and γ_m what is generally expressed in the manner of $\gamma_0/\gamma = (1-\beta_m)$ see Fig. (10)

Mechanical heat equivalent:

$$J = 427\left(\text{actually} = 426.9\cdot c_p\cdot\frac{g_n}{g}\right) \text{ in m/1 degC of } \theta$$

Which means: At g = g_N and c_{PI} = 1 kcal/kg we get an expansion of 427 m at a temperature change of 1 °C. For a machine efficiency assumed to be 100%, we have $(\theta_{II}-\theta_I) = \Delta\theta_{ad}$ and $\gamma_m = 0$ or, now expressed in the main equation:

$$(H_U)_{100\%} = 10\left(p_{II}-p_I\right)\cdot\left(1-\beta_m\right)+\left(\frac{v_{II}^2-v_I^2}{2g}\right)+\left(z_{II}-z_I\right)$$

From this we get for the net (manometric) head of a turbine or a pump:

Turbine Test $H_n = 10\left(p_{eT}-p_{aT}\right)\cdot\left(1-\beta_m\right)+\left(\frac{v_{eT}^2-v_{aT}^2}{2g}\right)+\left(z_{eT}-z_{aT}\right)$

Hn = 10

Pump Test $H_n = 10\left(p_{aP}-p_{eP}\right)\cdot\left(1-\beta_m\right)+\left(\frac{v_{aP}^2-v_{eP}^2}{2g}\right)+\left(z_{aP}-z_{eP}\right)$

Hn = 10

From these measured values and computed heads the corresponding efficiencies are received as the following ratios:

Turbine: $\eta_{nT} = \frac{H_U}{H_n}$

Pump: $\eta_{nP} = \frac{H_n}{H_U}$

MEASURING TECHNIQUE

Two main values have to be measured with the necessary precision, namely:

$(H_{II} - H_I) = \Delta H$ at a precision of 0.4 m of water column

$(\theta_{II} - \theta_I) = \Delta\theta$ at a precision of 1/1,000 °C

These two differences $(H_{II} - H_I)$ and $(\theta_{II} - \theta_I)$ must be measured in a definite and reproducible manner.

For doing this four main techniques, or methods, have been developed to date:

1) Direct temperature measurement — DM
2) Partial expanding (Zero Method) — ZM
3) Auxiliary expanding — AE
4) Total expanding — TE

With the instruments available today, and considering the experiences recollected up to date, the methods 2 and 3 are considered to be the most suitable ones in many cases. In order to simplify these explanations and, as they will be included in the new chapter of the Swiss Field Test Code, I shall deal in the future with these two methods only. Furthermore we are admitting a turbine test has to be executed.

A. ZM or Zero Method

This method is especially suitable for tests on Pelton wheels, on Francis turbines, and on storage pumps with relatively high back pressure.

Principle: The pressure p_{II} in the expander is changed until $\theta_{II} = \theta_I$ which means $(\theta_{II} - \theta_I) = 0$ Then we have: $p_{II} = (p_{II})_0$

Procedure: At the tapping 2′ (see Fig. 1) a probe for total pressure $\left(\frac{p}{\gamma} + \frac{V^2}{2g}\right)$ is introduced in the adduction pipe, ahead of the spiral casing entrance, for example, and a quantity of water q_{II} will be derivated. This flow of an amount of 0.2 to 0.5 lit/sec is flowing through a pressure valve behind which, at location 2‴, pressure p_{II} and the temperature θ_{II} are measured.

The measuring section for p_I and θ_I are after the turbine, in Zone I (may be tail race channel with free level, may be closed section after a draught tube).

The throttling valve at 2″ *is adjusted until the instrument to* measure $(\theta_{II} - \theta_I)$ generally a galvanometer, shows = 0. As the

piping between 2′ and 2‴ quite never can be insulated in an ideal manner to avoid any heat transfer, at least 3 measurements with 3 different q_{II} values were executed, in order to receive 3 values of $(p_{II})_0$.

At $q_{II} = \infty$ or equally at $1/q_{II} = 0$ no heat exchange would occur. The values of $(p_{II})_0$ are therefore plotted against the values of $1/q_{II}$. The straight line through these values, extrapolated to $(1/q_{II}) = 0$, will deliver the value $(p_{II})_0$ to be introduced in the final equation:

$$H_u = 10 \cdot \left[(p_{II})_{0_0} - p_I \right] \cdot (1 - \alpha_m - \beta_m) + \left(\frac{v_{II}^2 - v_I^2}{2g} \right) + (z_{II} - z_I)$$

B. AE or Auxiliary Expanding

Suitable for Francis turbines with back pressure and for storage pumps with a submergence of less than $H_n \left(\frac{1}{\eta_{0p}} - 1 \right)$ in m.

Principle: Complete (or only partial) calibration in (m) of the $\Delta\Theta$ measuring equipment (= galvanometer).

Procedure: 1st step: Direct measurement.

Θ probe # 1 in location 1′ or in 1″; pressure p_I

Θ probe # 2 in location 2′ or in 2″; pressure p_{II}

Result: d_P parts of scale on $\Delta\Theta$ meter.

With apparatuses where heat exchange between location 2′ or 1′ and the location for temperature measurements can be well determined, this direct measurement has to be carried out for various values of q_{II} and q_I; with this $(d_P)_0$ at $1/q = 0$ will be determined as for method ZM.

2nd step: Auxiliary expanding.

Θ probe # 1 in location 2″; pressure p″

Θ probe # 2 in location 2‴; pressure p‴

Result: d_P in function of (p″ - p‴)

At various (e.g. 6) positions of the throttling valve, and thus giving 6 values of (p″ - p‴), the positions of the throttling valve being chosen to give 3 values of d higher and 3 values lower than d_P.

Graphically we get $(p'' - p''')_0$ at $d = d_P$ which value is introduced in the general equation for Hu:

$$H_u = 10 \cdot \left[p_{II} - p_I \right] \cdot (1 - \alpha_m - \beta_m)_{II,I} + \left(\frac{v_{II}^2 - v_I^2}{2g} \right) + (z_{II} - z_I)$$
$$\mp 10 \cdot (p^{II} - p^{III})_0 \cdot (1 - \alpha_m - \beta_m)_{p_m}$$

wherein the α_m and β_m -values are taken for the mean pressure

$$p_m = \left(\frac{p^{II} + p^{III}}{2}\right)$$

For computing purposes the following substitution is made:
A = calibration factor =

$$(1 - \alpha_m - \beta_m)_{p_m} \cdot 10 \cdot \frac{\Delta p^{(II-III)}}{d}$$

Thus giving the final simplified equation for Hu

$$H_u = 10 \cdot \left[P_{II} - P_I\right] \cdot (1 - \alpha_m - \beta_m)_{II,I} + \left(\frac{V_{II}^2 - V_I^2}{2g}\right) + (z_{II} - z_I) \mp A \cdot dp$$

wherein (−) for turbines (+) for pumps

It is of high importance to check the balance of the galvanometer for the measurement of $\Delta\theta$ very carefully immediately before and after the tests.

By these rather few remarks one may realize that this thermodynamic method is requiring extensive precaution and well experienced personnel.

MEASURING MEANS

It is impossible to describe in this paper in detail the apparatuses, especially also because these equipments and instruments are in full development. The vital parts are the following:

1) *Pressure and flow probes,* ½″ to ⅝″, resistant against vibrations, with openings entering at least 1.5″ in the clear section of the flow. As the velocity head (or kinetic head) is contained in numerator and in denominator of the efficiency ratio, the influence on efficiency by any error in this measurement of head will generally for turbines never exceed an amount of 0.01%. On the other hand energy distribution at the spiral casing outlets of storage pumps proves to be often very irregular. For this reason a doctor thesis work is under way now in the Hydraulic Machine Laboratory at Zurich to investigate in detail such energy distributions (= velocity and temperature distributions) in storage pumps. These probes are inserted in the main pipe through simple stuffing boxes of 1″ to 2″ diameter.

2) *Expander equipment* with provisions to measure pressures and temperatures, with a regulating throttle valve in between and a throttle valve or orifice at the outlet (Fig. 2).

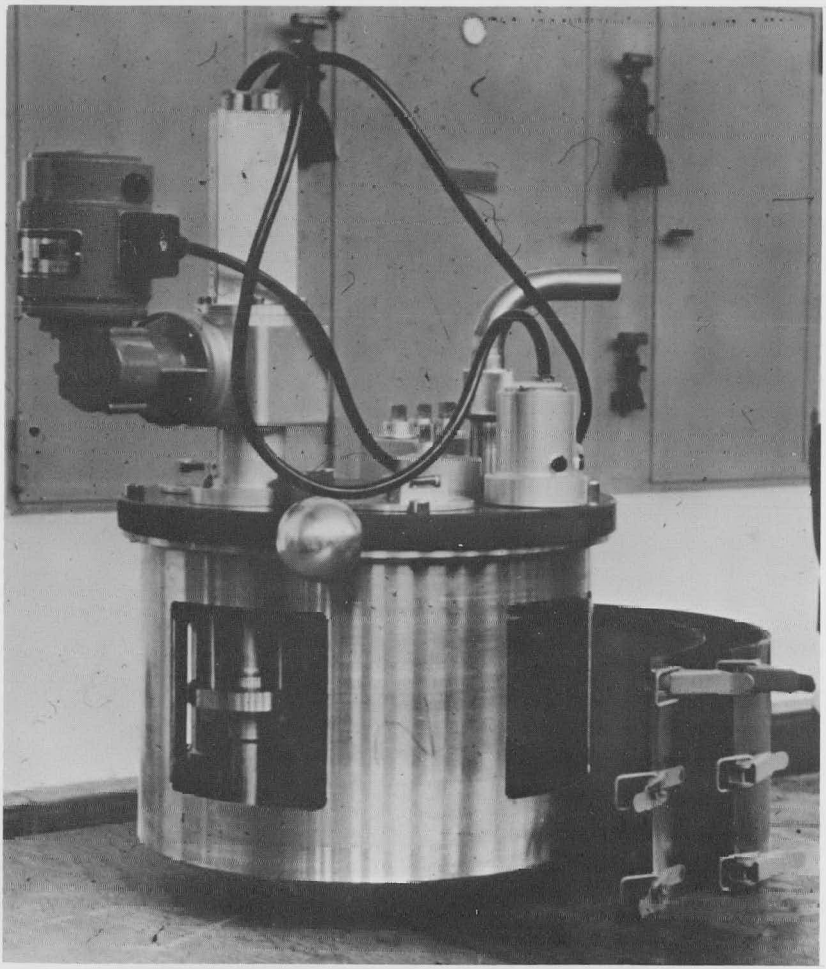


FIG. 2: Double Expander of ETH — Equipment.

3) *Temperature feelers,* generally resistance elements of about 100 ohms, made from platinum wire, located in protecting tube.

4) *Indicator for temperature differences,* mostly a galvanometer with sensibility of approximately 10^{-9} amp/1 deg. of scale section, and inserted in the diagonal of a doubly adjustable wheatstone bridge, with max. 4 mA in the bridge (Fig. 3 & 4).

5) *Adjusting and thermo-feeler balancing container,* generally thermos bottles, with agitator.

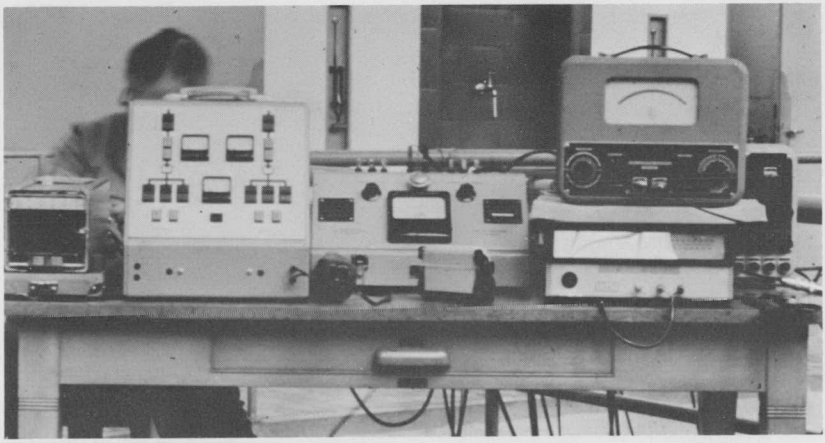


FIG. 3: Recording Instruments for Temperatures and Pressures.

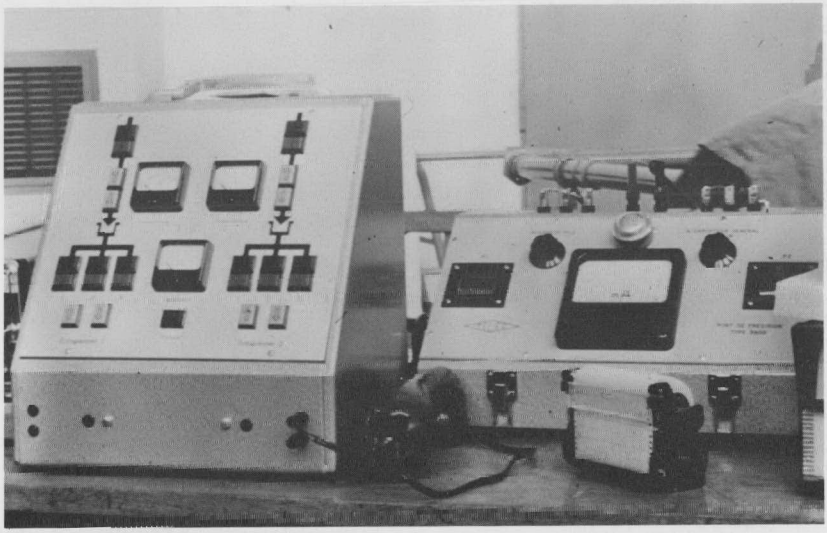


FIG. 4: AC — Bridge and Galvanometer.

6) *Precision manometers,* today mostly dead weight piston manometers completed with additional indicator for small differences (Fig. 5 & 6).

The way was long since the first standard equipment appeared on the market, at that time a remarkable design and realisation by the EdF and Neyrpic Engineers at Grenoble.

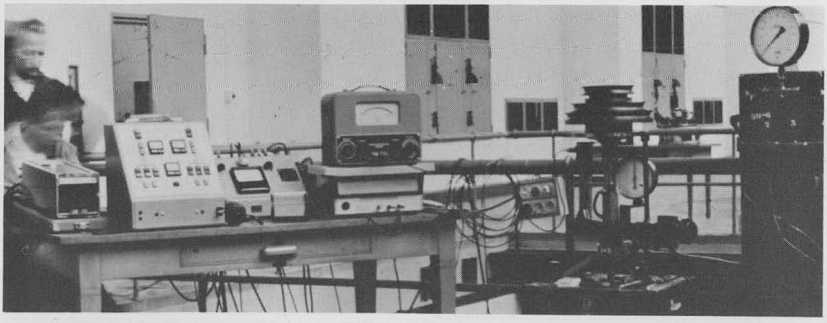


FIG. 5: View of the complete new ETH — Set.

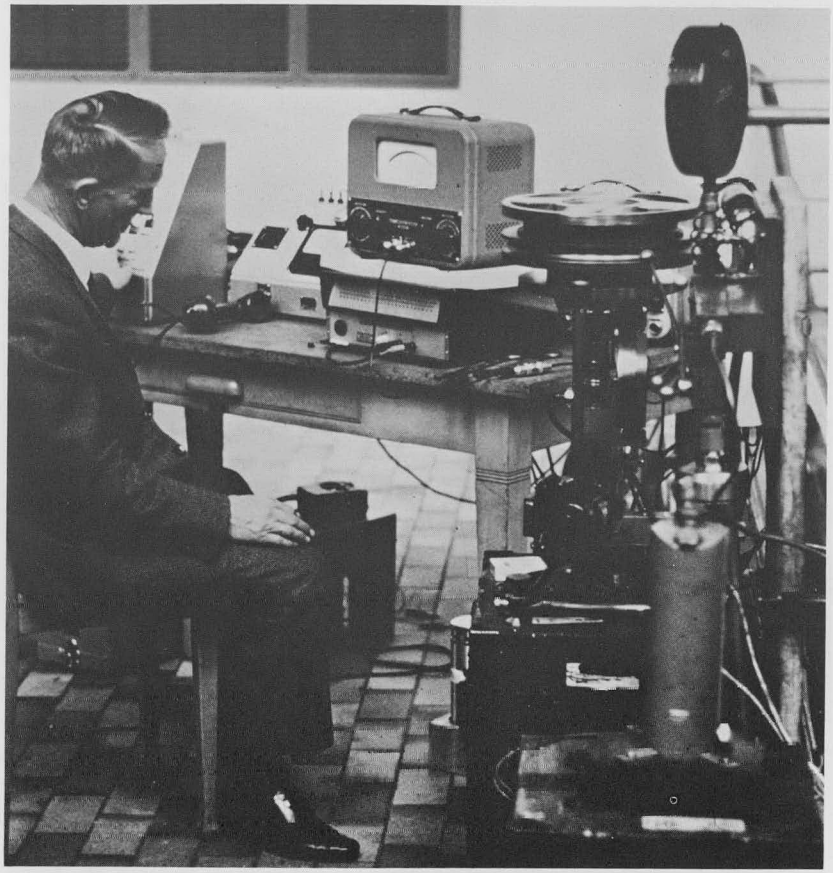


FIG. 6: ETH — Set installed in the Etzel Power Plant, for demonstrations at the 1966 meeting of the "Groupe international des Practiciens de la méthode thermodynamique".

BASIC PROPERTIES OF WATER

The properties of physically and chemically clean water are well known although some published values still show clear differences. The international tables produced as a result of corresponding meetings are adjusted to what thermal people need at high temperatures. Extrapolated to low temperatures as we have in hydraulic machines, between 0°C and 30°C, they show maximum specific weight γ (or density) at + 5°C, values for atmospheric pressure. Everybody knows that this maximum is to be found at + 4°C.

Based on these facts, taking into account the range of temperature we are interested in, and knowing about the influence on efficiency of the values finally chosen for computing, Mr. Paul Weber, first Assistant in the Hydraulic Machine Laboratory at ETH, has worked out proposals for these important values (Figs. 10 & 11).

Fig. (7) shows the values of compressibility of water, whilst Fig. (8) contains the values of specific heat of water. But as the use of the specific heat c_P is not appropriate and therefore not recommendable, the computation today is generally worked out with the coefficient of expansion, α, of water, as to be seen in Fig. (9). As a result of these proposals, or assumptions, he established a final diagram, wherein curves are given for pressures, p_M, in function of

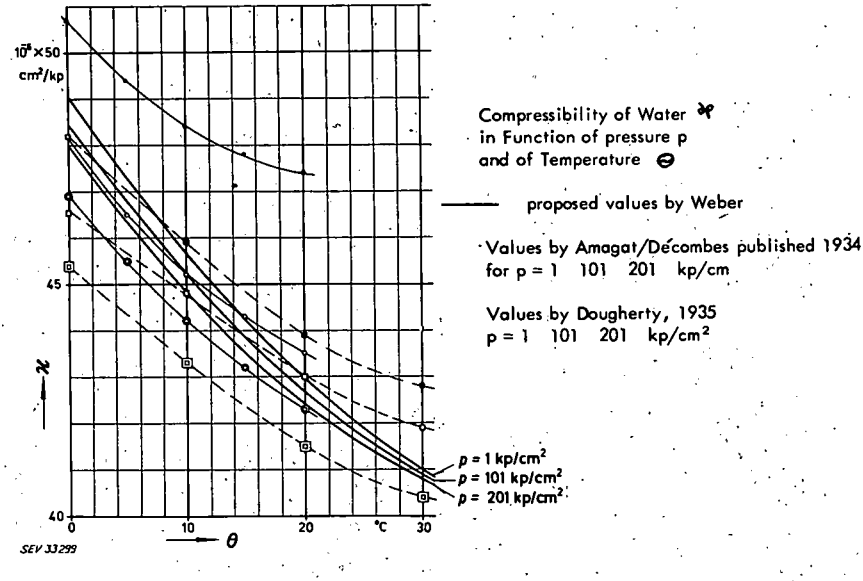
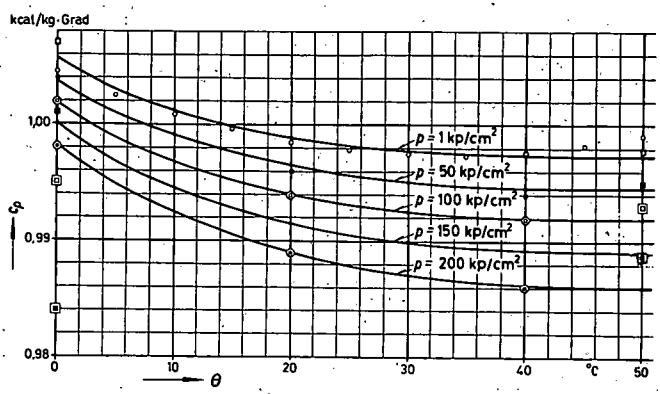


FIG. 7: Compressibility of Water, values by Weber.



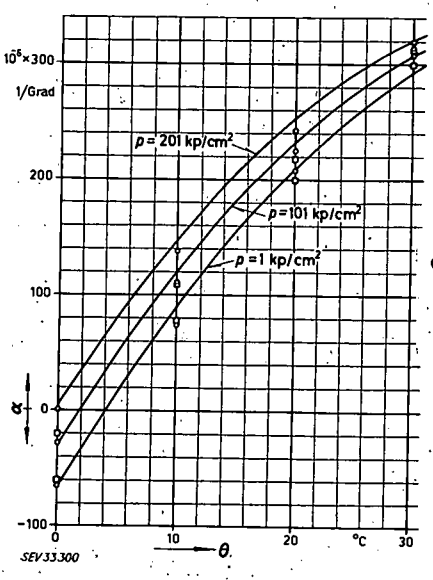
Specific Heat of Water c_p
in Function of pressure p
and of Temperature Θ

Curves by L. Prüger, contained in Pocketbook for Chemists and Physicists, ed. by d'Ans and Lax. 1943

Values from "HUtte" I Pocketbook, 27th Ed.

Values by Schmidt VDI, 1963

FIG. 8: Specific Heat of Water, values by Weber.



COEFFICIENT OF EXPANSION α OF WATER
for given pressure p and temperature Θ
▫ values of Amagat/Decombes 1935
o values of Wukalowitsch 1958

FIG. 9: Coefficient of Expansion α of Water.

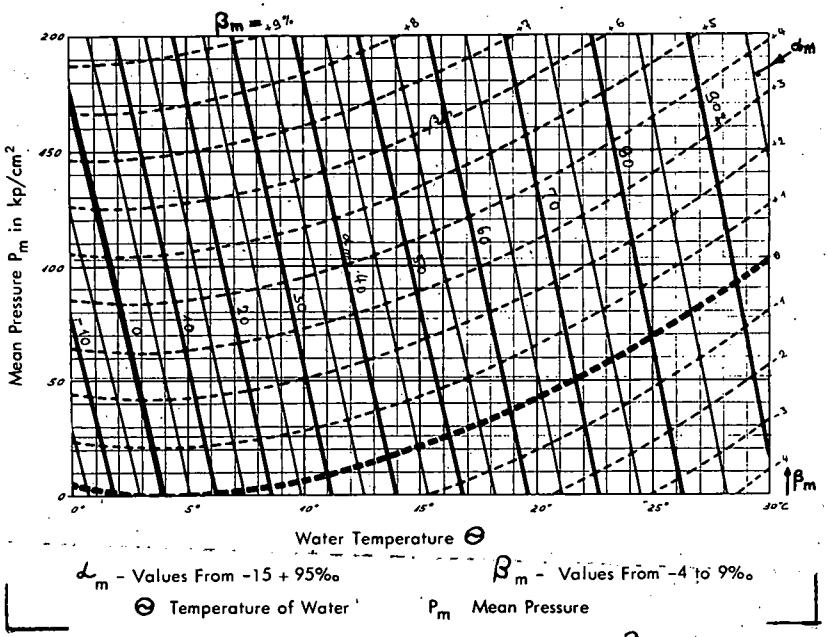


FIG. 10: Values by Weber: Correction Factors α_m and β_m for given mean pressures and temperatures.

temperature, Θ, for given values of α_m and β_m. These values will be adopted for the chapter on thermodynamic measurements in the 4th edition of the Swiss Field Test Code. In a French publication Mr. Gabaudan, a specialist in this field, stated that these values seem to be a reasonable compromise and that he is working with them at least for the moment. One thing is sure: the results of the International Steam Table Conference are not covering our require-

REQUIREMENTS FOR GOOD TESTS

As I said earlier the difficulties have been somewhat underestimated. There are quite a number of conditions to be fulfilled for making successful and reliable tests with the Thermodynamic Method. The following list therefore cannot be considered to be complete and is especially depending on local conditions.

a. Steady conditions on the hydraulic side.
b. Steady conditions on the thermic side, which often proves to be much more difficult than to be expected and may lead to measurements made during the night or even during winter time only.

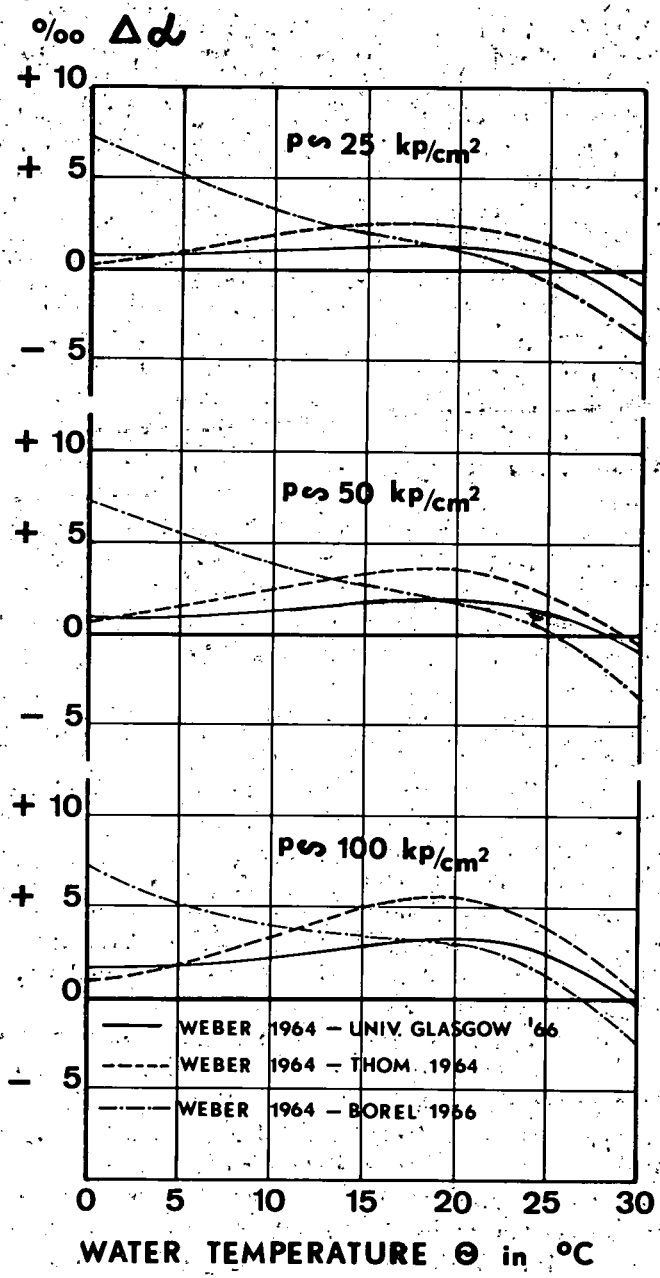


FIG. 11: Comparison of Weber's α – values with other published values.

c. No change in chemical structure of the water during its flow through the apparatuses; special attention when marshy ground is present.

d. If there is a temperature gradient during the measureing time, a special correction on the $\underline{Hu}$ values will be necessary by approximation.

e. If as usual to a certain amount, a heat exchange takes place between the surrounding and the water leading parts of the equipment, a similar correction on $\underline{Hu}$ has to be made.

f. Pay attention to the hot air channels of the electric machines, as heat exchange may occur to the water in the tailrace, between the machine and the section of the temperature to be measured.

g. Within the measuring system as defined at the beginning no cooling water may be introduced, or cooling pipes be in operation. This may call for special provisions at the time of design of the plant.

h. Cooling water for seal-rings and similar devices must be diverted and its quantity and temperature has to be measured, which will permit a further correction of Hu.

i. Sufficient space must be provided in sections I and II to introduce the probes and, eventually, to locate the expanders close to the main pipes.

k. In free sections of the tailrace special consideration must be given to the selection of the location of measuring section I. If this section is not accessible to explore the whole section for temperature distribution, then special means with remote control have to be provided.

l. If there exist doubts about the constancy of conditions with varaiable loads or pressures two or more measuring sections have to be used, especially on the high pressure side.

m. Sufficient distance between the galvanometer and any electric machine has to be provided.

n. If possible all manometers should be calibrated in site, with the plant at a standstill, using the geodetically measured water column for comparison. The result of this calibration has to be compared with the ρm values, thus producing valuable information on the properties of the local plant water.

o. If possible all pressures (of one side of the machine) should be measured with the same instrument.

p. The zero check of the temperature probes has to be made immediately before and after each series of tests.

This selection of items shows that, exactly as for any other method, quite precise conditions must be fulfilled to run thermodynamic tests successfully.

ACCURACY AND LIMITS OF APPLICATION

The highest accuracy will generally be reached with the Zero Method, because with it practically only the following items will be affected by errors:

1) pressure difference measurement
2) α_m values
3) additional losses (or corrections)

The accuracy will be somewhat reduced by using the method of auxiliary expanding because the error in calibration has to be added, and because the direct measurement and the calibration itself cannot be executed at the same time.

The *basic error,* computed from the 3 individual errors on:

Net Head	φ_H
Total Energy Head	φ_{Hu}
α_m - value	$\varphi(1-\alpha_m)$

is given by the equation:

$$[\varphi_0]_\eta = \pm\sqrt{\varphi_H^2 + \varphi_{H_u}^2 + \varphi(1-\alpha_m)^2}$$

In order to keep all auxiliary losses (or corrections) below a certain total percentage of the whole efficiency, the following limits should be respected:

If additional losses are measured $\Delta\eta < 5\%$

If additional losses are calculated $\Delta\eta < 1\%$

Provided that these limits are respected, as well as the conditions cited in the section on requirements for good tests, the *total measuring error* which will be present may be computed from the basic error by multiplying it with the following factors, established statistically from the results available today from a big number of tests.

Multiplying factor for	ZM	AE
Pelton wheels	$\sqrt{2}$	—
Francis turbines	$\sqrt{3}$	$\sqrt{4}$
Storage pumps	$\sqrt{5}$	$\sqrt{6}$

COMPUTATION OF FLOW

If, in addition to the thermodynamic measurements of efficiency, the shaft power (electric power + losses) has been measured, and if the mechanical losses in the bearings and so on are known, thus changing the hydraulic efficiencies into overall efficiencies of the respective hydraulic machine, the flow value may be computed from the following equations:

Turbine: $102 \cdot P_T = \eta_T \cdot \gamma \cdot Q \cdot H$ in kW

Pump: $102 \cdot \eta_P \quad P_P = \gamma \cdot Q \cdot H$ in kW

TEST RESULTS

It will be interesting finally to deal with some typical results with the Thermodynamic Method.

Bodio Pelton Wheel. The Bodio turbine was so small (900kW) that with local conditions proving to be very exceptional, we proposed the new method. Fig. (12) shows the general measuring arrangements. The turbine pressure pipe is branched to the main penstock, some 180 feet above the main power plant. It is a double nozzle, single wheel unit with horizontal shaft. Fig. (13) contains the ($\alpha_m + \beta_m$) and the β_m values used for computations.

In Fig. (14) the readings on the dead weight manometer versus the galvanometer readings are shown. Experience shows that these curves can be replaced by straight lines. The results of these field acceptance tests are shown in Fig. (15).

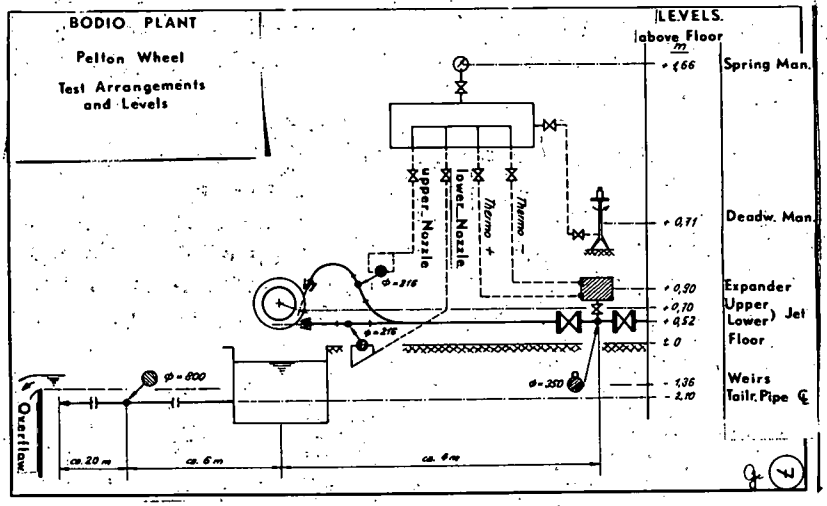


FIG. 12: Bodio Plant, General Measuring Arrangement.

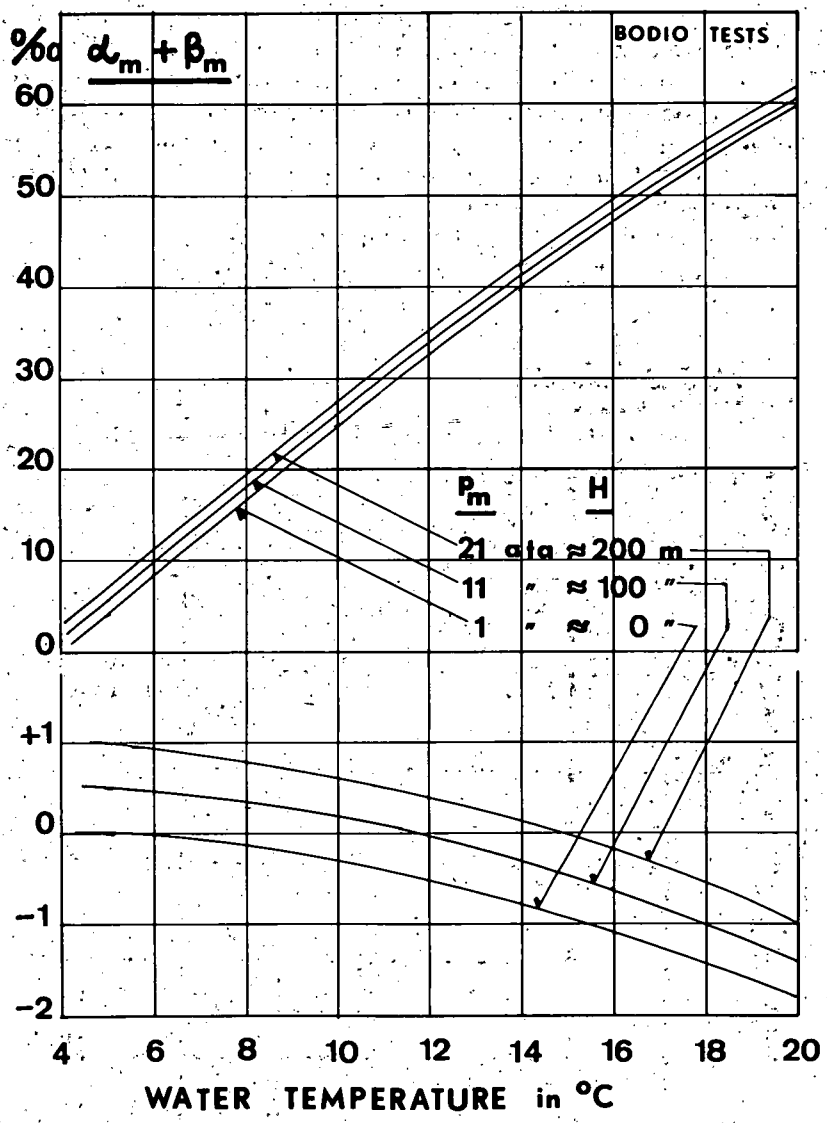


FIG. 13: Bodio, values of α_m and β_m used for computation.

Vianden Pumped Storage Plant. These extremely conclusive and reliable field tests have been described by Vaucher in the Escher Wyss News. These test were performed on the turbine and the Pump of Unit # 2. Fig. (16) contains the whole measuring arrangements.

The results found on the Francis turbine are represented in Fig. (17), those of the Storage pump in Fig. (18).

In his paper on a 4-year experience with the Thermodynamic Method, Mr. Vaucher of the Escher Wyss Manufacturing Company has published a number of test results, from whom we have chosen the following four typical examples.

Fig (19) Schwarzenbach, Pelton wheel, H = 1160 ft
Fig. (20) Hemsil I, Francis turbine, H = 1670 ft
Fig. (21) Loebbia, Pelton wheel H = 2420 ft
Fig. (22) Lünersee, Pelton wheel H = 3050 ft

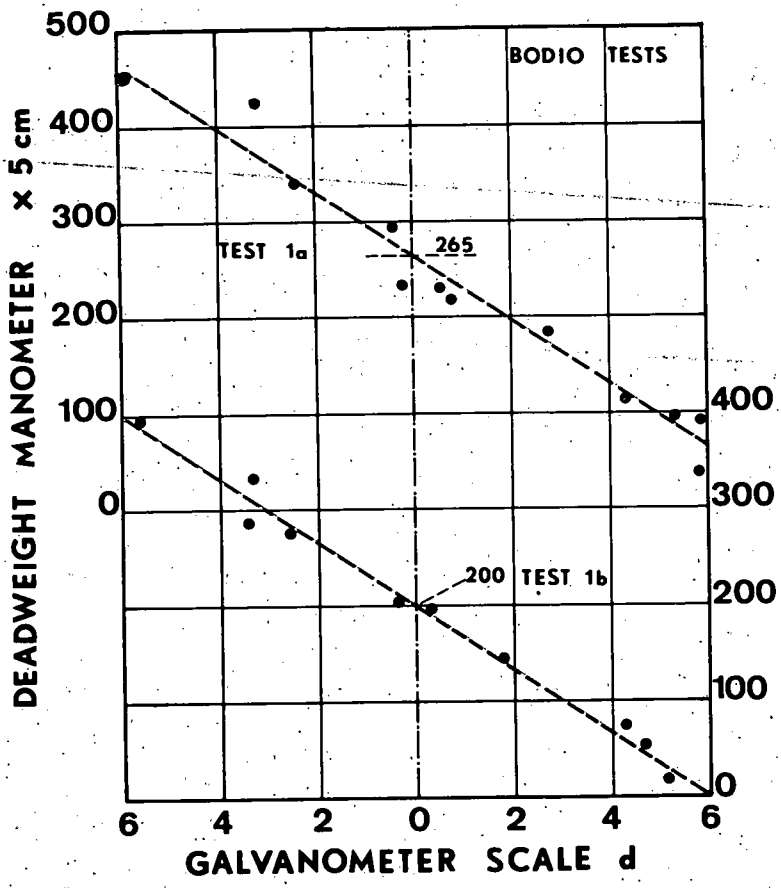


FIG. 14: Bodio, Measured pressure values, plotted versus Galvanometer (= Temperature) — Readings.

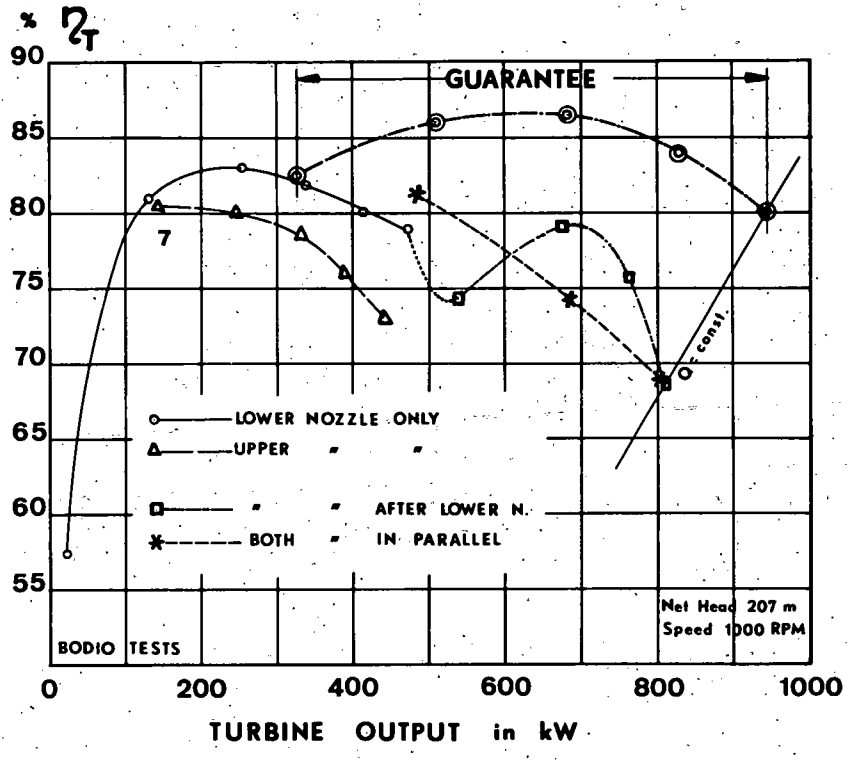


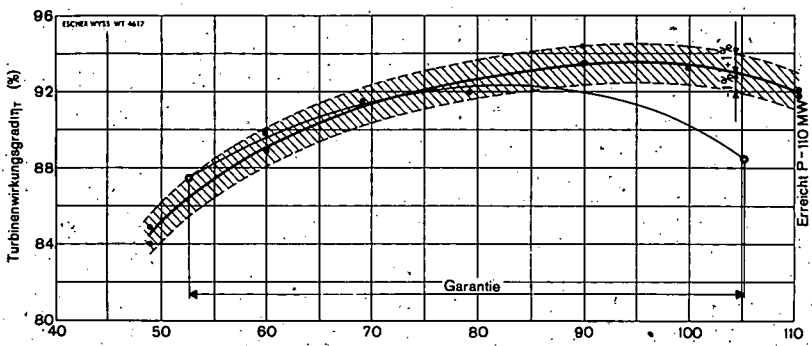
FIG. 15: Bodio, Pelton wheel, Test Results.

The agreement with the results from conventional tests, generally flow measurements made with current meters, proves to be remarkably good.

Skiok Power Plant. This test result, communicated by Prof. Knut Alming from the Technical University at Trondheim, Norway, is very interesting because of the fact that it has been obtained with *direct temperature* measurement, using modern quartz thermometers (Fig. 23).

FUTURE DEVELOPMENT

The Thermodynamic Method, by measuring pressures and comparing temperatures, is well developed and established in Europe so that nowadays it is admitted for Standard Acceptance Tests to be executed. It is replacing conventional methods in a very favourable manner in making tests easier or even possible. Nobody will go to contest it

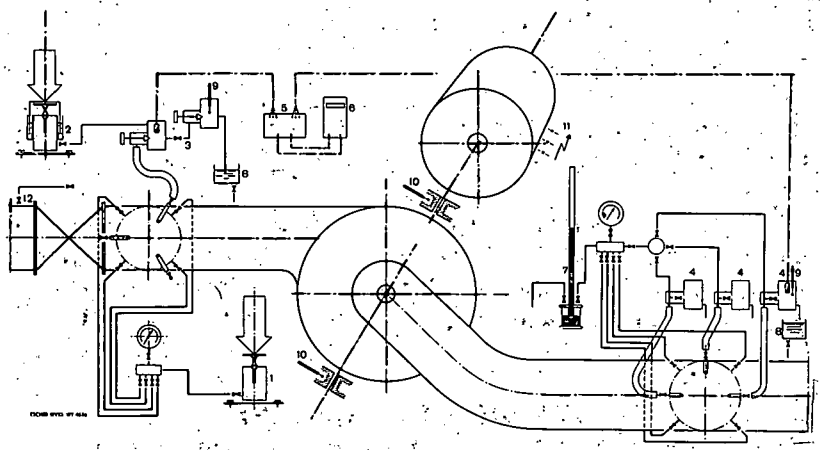


Vianden Pump Storage Plant, Luxemburg. Field Test Results, obtained with the Thermodynamic Method on Pump Nr. 2.
Pump Heads H = 292-268 m (958-880 Ft.)
Pumped Flow Q = 20.75 - 22.75 m^3/s
Speed N = 428.6 RPM

FIG. 16: Vianden Power Plant, General Measuring Arrangement.

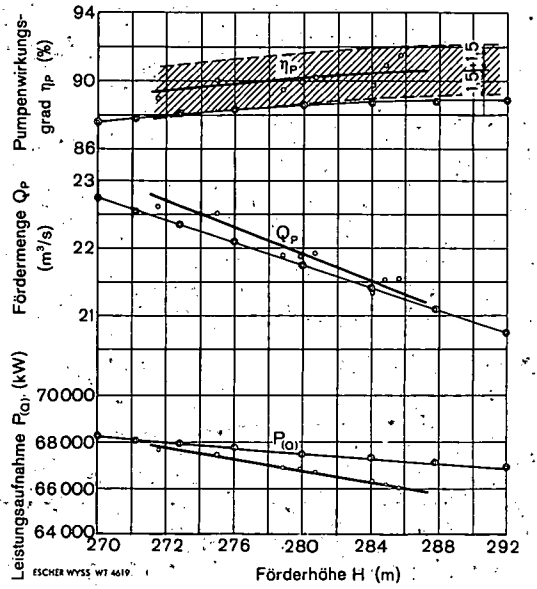
more or even to doubt as it was at the beginning. It will be improved and developed in a rapid manner in the near future. As far as we can see today this development will be directed in the following most important fields.

a. Improvement of pressure measurement. Today the lower level is fixed at 330 ft for turbines and at 500 ft for pumps. Perhaps



VIANDEN PUMP STORAGE PLANT
Tests with the Thermodynamic Method on Tubines and Pumps
Arrangment on Instruments, Unit Nr. 2

FIG. 17: Vianden, Results of Turbine Tests.



Vianden Pump Storage Plant, Luxemburg. Field Test Results, obtained with the Thermodynamic Method on Pump Nr. 2.
Pump Heads H = 292-268 m (958-880 Ft.)
Pumped Flow Q = 20.75 - 22.75 m^3/s
Speed N = 428.6 RPM

FIG. 18: Vianden, Results of Storage Pump Tests.

in the near future the use of the method for heads even lower than 300 ft will be possible, as successful test results, realized under especially good test conditions, are already available.

b. Use for detail research on prototypes and in the laboratory. Partial efficiencies have been measured showing the temperature distribution and thus the energy distribution in the housings of Pelton wheels. Many other fields of application have been detected quite recently.

c. One very interesting and important field of application will be the use in measuring efficiencies of big boiler feeding pumps. It is well known that these pumps generally cannot be measured under true working conditions in the manufacturer's workshop, and that field tests in the steam power plant prove to be very difficult. Today results of comparative tests with conventional methods are available from plants where the boiler feeding pump was driven by an electric motor, duly calibrated before

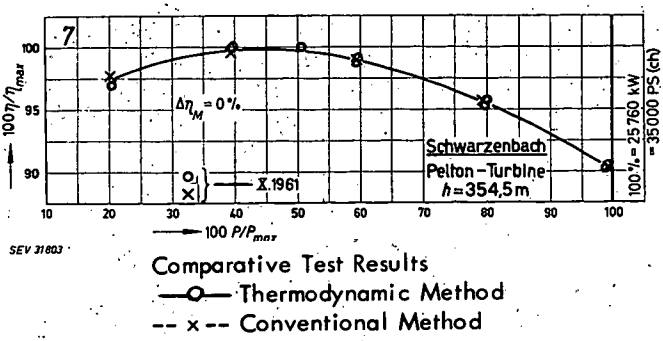


FIG. 19: Comparative Tests at Schwarzenbach Plant; H = 1160 ft.

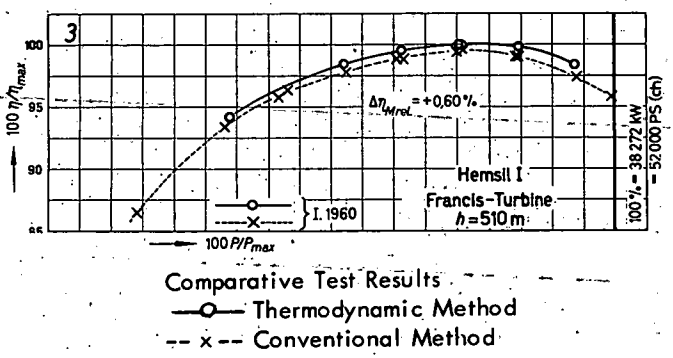


FIG. 20: Comparative Tests at Hemsil I Plant; H = 1670 ft.

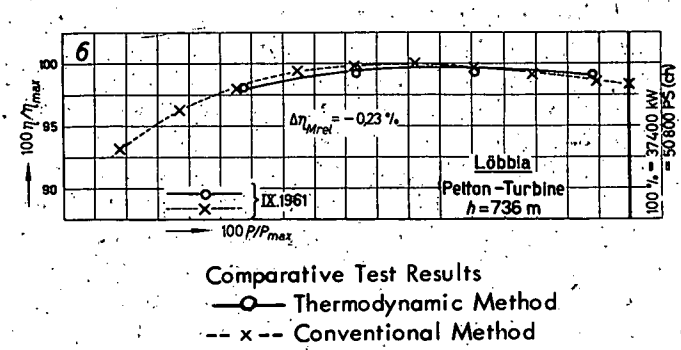


FIG. 21: Comparative Tests at Loebbia Plant; H = 2420 ft.

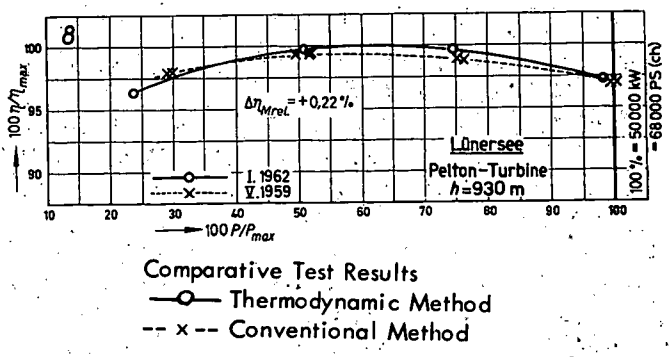


FIG. 22: Comparative Tests at Luenersee Plant; H = 3050 ft.

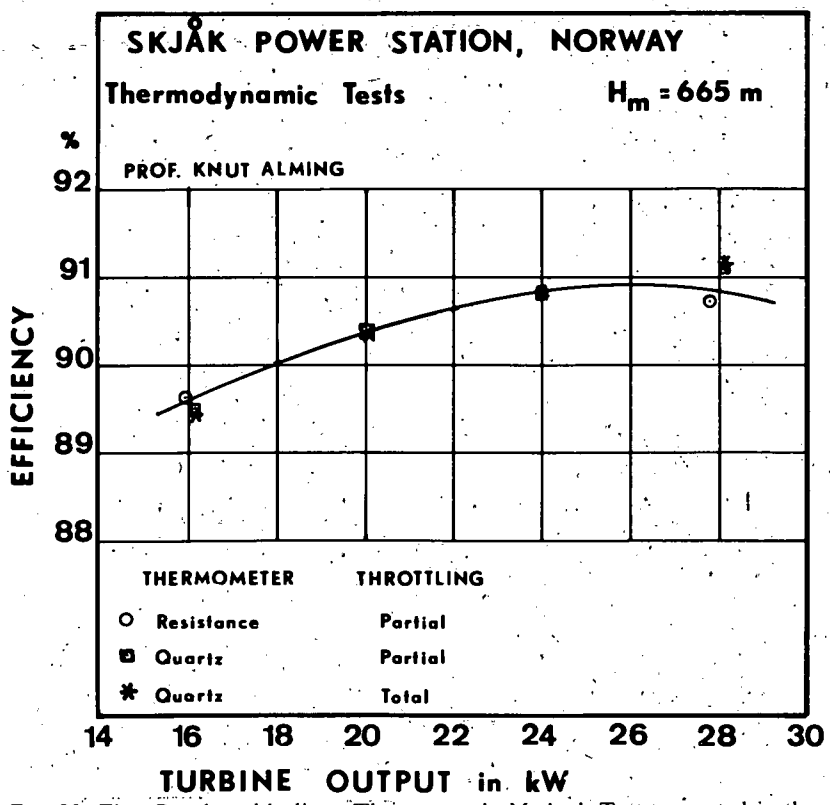


FIG. 23: First Results with direct Thermometric Method; Tests executed in the Skjåk Plant by Prof. Knut Alming, Trondheim, Norway.

the tests. In their paper, presented in 1963 to the VDI meeting at Freiburg, Western Germany, Roegener and Arens-Fischer express their firm opinion that the Thermodynamic Method is called to become the most reliable one for measuring efficiencies of boiler feeding pumps, especially when they are driven by condensating steam turbines.

d. Finally, and going back to the first idea of Poirson, the direct thermometric measurement will be developed, within the limits of accuracy of the modern thermometers of the quartz type and others, now available. Whether it will replace or not the indirectly working Thermodynamic Method as it stands today, is fully open.

BIBLIOGRAPHY

1. G. Willm et P. Campmas. Mesure du rendement des Turbines Hydrauliques par la Méthode Thermodynamique Poirson. La Houille Blanche, Nr. 4/5, Grenoble, 1954.
2. G. Willm. La précision de la Méthode Thermodynamique a la lumière de deux années d'application industrielle. Communication à la Société Hydrothechnique de France. 1955.
3. J. Chappuis. Première application en Suisse de la Methode Thermodynamique à la mesure du rendement de Turbines Hydrauliques. Bull. Technique de la Suisse Romande, Lausanne, Nr. 11, 1956.
4. S. Ferry, G. Willm et J. Thouvenin. L'influence de l'usure sur le rendement des Turbines Hydrauliques. Vemes Jounees de l' Hydraulique; Aix-en-Provence, Question IV, Rapport A Nr. 1. 1958.
5. R. Vaucher. Thermodynamische Wirkungsgradmessungen an hydraulischen Maschinen. Escher Wyss News, Zurich, Switzerland, Heft 2/3. 1959.
6. Gaylon S. Ross and Herbert D. Dixon. Automatic Precise Recording of Temperature. Journal of Research of the National Bureau of Standards. C, Engrg. and Instrumentation, Vol. 64c, Nr. 4. 1960.
7. F. L. Brand. Die Messung des Wirkungsgrades von Pumpen nach einem Nullverfahren. Paper to the "Réunion des Practiciens de la Méthode Thermodynamique, Grenoble, France (not publ.) 1961.
8. A. S. Thom and S. Y. Ahmad. Thermodynamic Method of Measuring Turbine Efficiency, including a special technique for low Heads. Nat. Engineering Laboratory, East Kilbride/Glasgow, report Nr. 17. 1962.
9. T. R. Foord, R. C. Langlands and A. J. Binnie. A Transformer Ratio Bridge Network with Precise Lead Compensation for the Measurement of Temperature and of Temperature difference. Nat. Eng. Laboratory, East Kilbride/Glasgow. Rep. 67, 1962.
10. R. Vaucher. Rueckblick auf eine vierjaehrige Anwendung der Thermodynamischen Methode bei Wirkungsgradmessungen hydraulischer Maschinen. Bulletin A.S.E., Zurich, Switzerland, Nr. 3. 1963.
11. A. S. Thom. Thermodynamic Method of Measuring the Efficiency of Hy-

draulic Machines. I.A.H.R. congress, London, Nr. 4.13. 1963.
12. H. ROEGENER AND F. ARENS-FISCHER. Erfahrungen mit dem thermodynamischen Messverfahren bei Kesselspeisepumpen. Paper (not yet published) presented at the VDI hydraulic Meeting, Freiburg/Breisgau, W. Germany. 1964.
13. ROSEMOUNT ENGINEERING COMPANY. Pamphlet on Platinum Resistance Temperature Sensors. Minneapolis. 1963.
14. MINCO PRODUCTS, INC. Pamphlet on Minco Thermal Devices (Probes), Detectors. 1964.
15. KNUT ALMING, PROF. Thermodynamic Method of Measuring Turbine Efficiency. Correction for Heat Transfer in a Probe. Paper (not published) presented at the 2nd Meeting of the "Groupe des Practiciens de la Methode Thermodynamique", Heidenheim, W. Germany. 1964.
16. P. WEBER AND R. VAUCHER. Thermodynamische Methode zur Wirkungsgradbestimmung von Wasserturbinen und Speicher-Pumpen. Proposal for a new Chapter in the Swiss Rules for Field Acceptance Tests (not yet published). 1964.
17. P. U. WEBER. Bemerkungen zur thermodynamischen Methode der Wirkungsgradbestimmung von Wasserturbinen und Speicherpumpen. Bulletin A.S.E., Zurich, Switzerland. Nr. 24. 1964
18. A. A. R. EL AGIB. The pair ratio method of measuring small temperature difference. Journal of Scientific Instruments, Vol. 41, 1964.
19. A. A. R. EL AGIB. A hydraulic bridge for measuring small temperature difference. Journal of Scientific Instruments, Vo. 41, 1964.
20. R. VAUCHER. Abnahmeversuche an den hydraulischen Maschinen von zwei Gruppen des ersten Ausbaues im Kraftwerk Vianden. Escher Wyss News, Zurich, Switzerland. Nr. 1. 1965.
21. A. S. THOM. Thermodynamic Testing of Turbines and Pumps. Journal of Mechanical Engineering Science, Vol 7, Nr. 3 1965.
22. HEWLETT-PACKARD JOURNAL. Pamphlet on: The linear Quartz Thermometer — a new Tool for measuring Absolute and Difference Temperatures. 1965.
23. L. BOREL. Methode thermodynamique pour la détermination du rendement des machines hydrauliques. Bulletin Technique Vevey, Suisse. 1966.
24. HAMYUKI SUZUKI. Thermodynamic method exercise for water turbines in Japan. Paper (not published) presented at the 1966 meeting of "Groupe des Practiciens de la méthode thermodynamique," Zurich, Switzerland. 1966.
25. C. CINNAMOND. Thermometric Method for the Determination of the Efficiency of Large Boiler Feeding Pumps. Letter (not published) to Dr. Thom, from G. & J. Weir Limited, Glasgow. 1966.
26. KNUT ALMING. Thermodynamic Measurements (with Quartz Thermometer) in the Skjoek Power Station (not Publ.). 1966.
27. J. GABAUDAN, P. M. THOMAS ET J. GALLIAN. Evolution de la Méthode Thermodynamique de détermination du rendement des Machines Hydrauliques. Publication EdF in "La Houille Blanche", Grenoble. 1966.
28. L. BOREL. Mesure du rendement par la méthode thermodynamique. Projet de rédaction d'un nouveau chapitre pour les "Regles Suisses pour Machines Hydrauliques". (not yet published). 1966.
29. J. W. DZIALLAS. Measurements of small Temperature Differences in an incompressible Flow. Thesis for M.D., presented at the Worcester Polytechnic Institute, Worcester, Mass. (not yet published). 1966.
30. F. L. BRAND. Die Entwicklung des Thermodynamischen Messverfahrens in den vergangenen fuenfzig Jahren. Voith Forschung und Konstruktion, Heidenheim, Western Germany, Heft 15 (to appear). 1967.

UNUSUAL ASPECTS OF HYDRAULIC TRANSIENTS IN PUMPING PLANTS*

By
JOHN PARMAKIAN**

SYNOPSIS

The computational procedures for the analysis of hydraulic transients in pump discharge lines with electric motor driven pumps has been known for many years. It started with the basic waterhammer contributions for valve operation by Joukousky and Allievi over 60 years ago. This was followed by many others in later years with the application of numerical, graphical, and high speed computer techniques. Although the mechanics of computation of the hydraulic transients in pump discharge lines has advanced rapidly in recent years, there are a number of unusual aspects which are still troublesome. It is the purpose of this paper to bring some of these to the reader's attention.

GENERAL CONSIDERATIONS

SCOPE OF PAPER: The aim of this paper is threefold. The first and major portion of the paper contains a discussion of some practical and unusual aspects of hydraulic transients in pumping plants which are sometimes overlooked. The second is to call attention to the available waterhammer solutions in the engineering literature which provide a ready solution of the hydraulic transients in pumping plants for a variety of waterhammer control devices. Finally, there is a discussion on the observed and computed transients in pumping installations with various types of pressure control devices.

BASIC ASSUMPTIONS: A number of assumptions were made in the derivation of the fundamental waterhammer equations and in the solution of the hydraulic transients in pumping systems. These assumptions involve the physical properties of the fluid and pipeline, the kinematics of the flow, and the transient response of the pump as follows:

(1) The fluid in the pipe system is elastic, of homogeneous density, and is always in the liquid state.

(2) The pipe wall material or conduit is homogeneous, isotropic, and elastic.

(3) The velocities and pressures in the pipeline, which is always flowing full, are uniformly distributed over any transverse cross section.

(4) The velocity head in the pipeline is negligible when compared to the pressure changes.

(5) At any time during the pump transient, when operating in the zones of pump operation, energy dissipation, and turbine operation, there is an instantaneous agreement at the pump, as defined by the steady state complete pump characteristics, of the pump speed and torque corresponding to the transient head and flow which exist at that moment at the pump.

(6) The length between the inlet and outlet of the pump is so short that waterhammer waves propogate between them instantly.

(7) Windage effects of the rotating elements of the pump and motor during the transients are negligible.

(8) Water levels at the intake and discharge reservoirs do not change during the transient period.

HIGH AND LOW HEAD PUMPING SYSTEMS: Waterhammer is of greater importance at low head pumping systems than at high head systems. The normal steady water velocities in both high head and low head pumping systems are usually of about the same order of magnitude. However, the pressure changes are proportional to the rate of change in the velocity of the water in the line. Then, for a given rate of change in the velocity, the pressure changes in the high and low head pumping systems are of about the same order of magnitude. Therefore, a head rise of a given amount would be a larger proportion of the pumping head at a low head pumping system than at a high head system.

DISCHARGE LINE PROFILE: The pump discharge line profile is usually based on economic, topographic and land right-of-way considerations. However, in selecting the alignment along which a pump discharge line is to be located, there are other considerations which often make one pipeline profile and alignment more favorable than another. For example, upon a power failure at the pump motors, the envelope of the maximum downsurge gradient along the length of the pipeline is a concave curve. Therefore, it may be possible to avoid the use of expensive pressure control devices at a pumping plant if the pipeline profile is also concave and is not located too far above the downsurge gradient curve. In some cases it may even be economical to lower the profile of the discharge line at the critical locations by deeper excavation. This was done at several large pumping plant installations and as a result some large expensive surge tanks were eliminated. If a surge tank at the pumping plant is definitely required, the most favorable pipeline profile is one with

high ground near the pumping plant where the surge tank structure above the natural ground line would be much shorter in height.

RIGID WATER COLUMN THEORY: The question is often raised as to whether the rigid water column theory of waterhammer is sufficiently accurate for the computation of the hydraulic transients in pump discharge lines. In the rigid water column theory the water is assumed to be incompressible and the pipe walls rigid. In the author's experience, the accuracy of the rigid water column theory is often questionable for most of the waterhammer problems that occur in pump discharge lines. However, it can be used with sufficient accuracy in the analysis of the transients in pipelines with such devices as surge tanks and air chambers. With these devices the velocity changes in the pump discharge line are substantially reduced.

WATERHAMMER WAVE VELOCITY: A number of articles have appeared in the technical literature during the past decade on the analysis of the wave velocities in steel pipes with various types of fixity. However, the differences in the numerical value of the wave velocities as computed by these various analyses and those given in Reference 1 are insignificant. It is also difficult to determine accurately the waterhammer wave velocity in rock tunnels because the elastic properties of the rock throughout the length of the tunnel are seldom known. As a practical matter a difference of 10 to 20 per cent in the magnitude of the waterhammer wave velocity usually has very little effect on the critical hydraulic transients in most pump discharge lines. The effect on the hydraulic transients of a possible error in the wave velocity can be verified by first computing the wave velocity as accurately as possible, and then recomputing the transients for the critical cases with a wave velocity which is about 20 percent different. At installations where alternative materials for the pipeline such as steel or concrete are being investigated, one waterhammer wave velocity and solution for the hydraulic transients for either alternative will usually suffice regardless of the pipe material finally selected.

PIPELINE SIZE: The diameter of the pipeline is usually determined from economic considerations based upon steady state pumping conditions. However, the waterhammer effects in a pump discharge line can be reduced by increasing the size of the discharge line since the velocity changes in the larger pipeline will be less. This is usually an expensive method for reducing waterhammer in pump discharge lines, but there are sometimes occasions, such as in small pipelines, where an increase in pipe size may be justified to avoid

the use of more expensive waterhammer control devices.

NUMBER OF PUMPS: The number of pumps connected to each pump discharge line is usually determined from the operational requirements of the installation, availability of pumps, and other economic considerations. However, the number and size of pumps connected to each discharge line has some effect on the hydraulic transients. For pump start-up the more the number of pumps on each discharge line the smaller the pressure rise. Moreover, if there is a malfunction at one of the pumps or control valves, a multiple pump installation on each discharge line would be preferable to a single pump installation because the flow changes in the discharge line due to such a malfunction would be less. When a simultaneous power failure occurs at all of the pump motors, the fewer the number of pumps on a discharge line, the smaller are the hydraulic transients. For a given total flow in the discharge line, a large number of smaller pumps and motors will have considerably less total kinetic energy in the rotating parts than a fewer number of pumps. Consequently, for the same total flow, the waterhammer effects due to a power failure are a minimum when there is only one pump connected to each discharge line.

FLYWHEEL EFFECT (WR^2): Another method for reducing the waterhammer effects in pump discharge lines is to provide additional flywheel effect (WR^2) in the rotating element of the motor. As an average the motor usually provides about 90 per cent of the combined flywheel effect of the rotating elements of the pump and motor. Upon a power failure at the motor, the increased kinetic energy of the rotating parts of the motor and pump will reduce the rate of change in the flow of water in the discharge line. As a rule of thumb, an increase of 100 per cent in the WR^2 of large motors can usually be obtained at an increased cost of about 20 per cent of the original cost of the motor. Ordinarily, an increase in WR^2 is not an economical method for reducing waterhammer, but it might be possible in marginal cases to eliminate other more expensive pressure control devices.

SPECIFIC SPEED OF PUMPS: For a given pipeline and steady flow conditions, the maximum head rise which can occur in a discharge line subsequent to a power failure, where the reverse flow passes through the pump depends first on the magnitude of the maximum reverse flow which can pass through the pump during the energy dissipation and turbine operation zones, and then upon the reduced flow which can pass through the pump at runaway speed in reverse.

Upon a power failure the radial flow (low specific speed) pump will produce slightly more downsurge than the axial flow (high specific speed) pump (see Reference 5). The radial flow pump will also produce the highest head rise upon a power failure if the reverse flow is permitted to pass through the pump. There is practically no head rise at mixed flow and axial flow pumps when a power failure occurs. Hence, the results obtained for radial flow pumps for the hydraulic transients due to a power failure will give conservative results of the downsurge and upsurge for any type of pump. During a power failure with no valves, the highest reverse speed is reached by the axial flow pump and the lowest by the radial flow pump. Care must therefore be taken to prevent damage to the motors with the higher specific speed pumps because of these higher reverse speeds.

Upon pump start-up against an initially closed check valve, the axial flow pump will produce the highest head rise in the discharge line since it also has the highest shut-off head. On pump start-up a radial flow pump will produce a nominal head rise but an axial flow pump can produce a head rise of several times the static head.

COMPLETE PUMP CHARACTERISTICS: In order to determine the transient conditions due to a power failure at the pump motors, the waterhammer wave phenomena in the pipeline, the rotating inertia of the pump and motor, and the complete pump characteristics, as shown in References 1 and 5, as well as other boundary conditions and head losses must be known. In the solution of waterhammer problems with computers, the complete pump characteristics are usually approximated by a polynomial expression in which the coefficients of the polynomial are obtained by fitting a representative curve through 3 adjacent points at specific locations on the pump characteristics diagram. Pump manufacturers sometimes provide limited information to determine these coefficients. However, a comparison between the polynomial values and the complete pump characteristics diagram indicates serious discrepancies in many cases especially in the zone of energy dissipation. Care must therefore be exercised in the use of an approximate polynomial expression as a substitute for the correct complete pump characteristics, to insure that a serious error does not result in the computation of the hydraulic transients.

COMPLEX PIPING SYSTEMS: As noted above in the basic assumptions, the waterhammer theory is strictly applicable for a pipeline of uniform characteristics. However, for waterhammer purposes a complex piping system can be reduced to a satisfactory equiv-

alent uniform pipe system. The approximations are made by neglecting the wave transmission effects at the junctions and points of dicontinuity and by utilizing the rigid water column theory. The pertinent waterhammer equations are then found to be analogous to those used in electrical circuits. In practice the waterhammer analysis with these approximations will usually be found to give more conservative results than those experimentally obtained for the actual pipe system. For a pipeline with a step-wise change in the diameter the equivalent length of uniform pipe is given in Reference 10.

AVAILABLE WATERHAMMER SOLUTIONS: The waterhammer solutions for various types of hydraulic transient problems in pumping plants are given in convenient chart form in Reference 1. These include the following:

(1) Hydraulic transients at the pump and midlength of the pump discharge lines for radial flow pumps with reverse flow passing through the pumps.

(2) Surge tanks.

(3) Air chambers.

In addition to the solutions noted above, References 2, 3, 4 and 5 also provide some additional waterhammer solutions in convenient chart form. These waterhammer solutions will be described in more detail below.

POWER FAILURE AT PUMP MOTORS

WITH NO VALVES AT THE PUMP: When the power supply to the pump motor is suddenly cut off, the only energy that is left to drive the pump in the forward direction is the kinetic energy of the rotating elements of the pump and motor. Since this energy is small when compared to that required to maintain the flow of water against the discharge head, the reduction in the pump speed is very rapid. As the pump speed reduces, the flow of water in the discharge line adjacent to the pump is also reduced. As a result of these rapid flow changes, waterhammer waves of increasing subnormal pressure are formed in the discharge line at the pump. These subnormal pressure waves move rapidly up the discharge line to the discharge outlet where complete wave reflections occur. Soon the speed of the pump is reduced to a point where no water can be delivered against the existing head. If there is no control valve at the pump, the flow through the pump reverses, although the pump may still be rotating in the forward direction. The speed of the pump now drops more rapidly and passes through zero speed. Soon the maximum reverse flow passes through the pump. A short time later the pump, acting as a turbine, reaches runaway speed in reverse. As the pump

approaches runaway speed, the reverse flow through the pump is reduced. For radial flow pumps this rapid reduction in the reverse flow produces a pressure rise at the pump and along the length of the discharge line.

For a given set of radial flow (low specific speed) pump characteristics, the results of a large number of waterhammer solutions are given in chart form in Reference 1. These charts furnish a convenient method for obtaining the hydraulic transients at the pump and mid-length of the discharge line when no control valves are present at the pump. Although the charts are theoretically applicable to one particular set of radial flow pump characteristics, they are useful for estimating the waterhammer effects in any pump discharge line which is equipped with radial flow pumps. If the friction head in the discharge line during normal pumping operation is more than 20 per cent of the total pumping head, and provided that water column separation does not occur, the maximum head at the pump with reverse flow passing through the pumps will not exceed the initial pumping head. (See Reference 2.)

PUMPS EQUIPPED WITH CHECK VALVES: There are a number of problems asociated with the use of check valves in pump discharge lines. Under steady flow conditions the pump discharge keeps the check valve open. However, when the flow through the pump reverses subsequent to a power failure, the check valve closes very rapidly under the action of the reverse flow and the resulting dynamic forces on the check valve disc. Under these conditions the head rise in the discharge line at the check valve is about equal to the head drop which existed at the moment of flow reversal. However, as shown in Reference 1, in the event that the check valve closure upon flow reversal is momentarily delayed due to hinge friction, malfunction, or inertial characteristics of the check valve, the maximum head rise in the discharge line at the check valve is considerably higher. On the other hand, if the check valve closure can be accomplished slightly in advance of the time of flow reversal, the head rise in the pump discharge line at the check valve is lower than that obtained with the check valve which closes at the moment of flow reversal. This feature is utilized by a number of check valve manufacturers by providing spring loaded or lever arm weighted devices on the check valve hinge pins to assist in closing the valve discs before the flow reverses. With these devices the hydraulic forces on the valve disc under normal flow conditions must be sufficient to overcome the sping or lever arm weight forces in order to keep the

check valve disc wide open so that the head losses at the valve will be a minimum.

CHECK VALVE SLAM: In addition to the waterhammer phenomena noted above, there is another difficulty which often occurs with check valves in pumping plant installations. This involves the objectionable slamming of the check valve discs against their seats upon closing. Upon a power failure, for the same pumping head conditions, short discharge lines will produce a much more severe check valve disc slam than long discharge lines for the same check valve. For long discharge lines the water in the pipeline takes a longer time to slow down and to reverse its direction. This permits the check valve disc more time to adjust its closing movement to the flow changes before an appreciable reverse flow has been established at the valve. In short pump discharge lines the water column reverses in much less time, and unless the valve disc can close very quickly, the water in the discharge line will attain a high reverse velocity. This will accelerate the closing movement of the disc and cause it to seat with a considerable slam. For check valve slam purposes, pump discharge lines which are equipped with surge tanks or air chambers at the pumping plant perform as very short discharge lines. Upon a power failure, with such pressure control devices, there is only a slight reduction in the head at the discharge side of the pump and consequently the flow reversal at the check valve occurs very rapidly.

Check valves in pump discharge lines may be grouped into two general classes, namely, rapid-closing check valves or slow-closing check valves. From the considerations noted above, the primary requirement for a check valve upon a power failure is that it should close quickly before a substantial reverse flow has been established. When this primary requirement for a fast closing check valve cannot be met due to the flow characteristics of the system as noted above and the design of the check valve, another alternative is to provide a device such as a dash-pot which will slow down or cushion the last portion of the check valve closure. This feature has been utilized by a number of check valve manufacturers.

CONTROLLED VALVE CLOSURE: A method of analysis is given in Reference 1 for determining the hydraulic transients in a pump discharge line subsequent to a power failure at the pump motors, when there is a controlled closing of a valve on the discharge side of the pump. In this method of analysis it is assumed that either the pump or the controlled closing valve controls the flow, whichever permits the least flow to pass through the system. Experimental

evidence at several large pumping plants indicates that this method of analysis is reasonably accurate (see References 6 and 7).

Another method of analysis which is sometimes used is to include the effect of the closing valve as a variable concentrated head loss at the discharge side of the pump. The use of such a head loss concept is somewhat tedious. However, it does provide a method for computing the changes in the pump speed throughout the entire transient, as well as the head and flow changes in the discharge line. For practical purposes the pressure changes obtained by both methods are comparable.

At most pumping plants the use of a single speed valve closure upon a power failure will limit the head rise in the discharge line to an acceptable value. However, it will usually be found that with the optimum single speed closure some reverse rotation below the maximum runaway speed of the unit in reverse will occur. If it is desired from other considerations to prevent or to limit the reverse speed of the unit, a two speed valve closure can be used. In such cases the discharge valve should close the major portion of its stroke very rapidly up to about the moment that the flow reverses. It should then complete the remainder of its stroke at a slower rate in order to limit the pressure rise in the discharge line to an acceptable value. At pumping plants where there are more than one pump on the same discharge line, a compromise must be obtained on the optimum single speed and two speed closure rates for the various combination of pumps which might be in operation at the time of a power failure.

SURGE SUPPRESSORS: Surge suppressors are sometimes used in pumping plants to control the pressure rise that occurs in pump discharge lines subsequent to power interruptions. A typical surge suppressor consists of a pilot-operated valve which opens quickly after a power interruption through the loss of power to a solenoid, of by a sudden pressure reduction at the surge suppressor. This valve provides an opening for releasing the reverse flow of water. The valve is later closed at a slower rate by the action of a dash-pot to control the pressure rise as the reverse flow of water is shut off. A properly sized and field adjusted surge suppressor can reduce the pressure rise in the discharge line to any desired value including no pressure rise. The charts given in Reference 3 can be used to determine the required flow capacity of the surge suppressor.

The proper field adjustment of a surge suppressor is very important. If the surge suppressor opens too rapidly subsequent to a power

failure, the downsurge at the pump and along the discharge line profile would be more than if no surge suppressor was present. As a result a water column separation condition may actually be produced in the discharge line by a faulty opening action of the surge suppressor. If the surge suppressor closes too rapidly after the maximum reverse flow has been established, a large pressure rise will occur.

WATER COLUMN SEPARATION: Water column separation in a pump discharge line subsequent to a power failure at the pump motors occurs whenever the momentary hydraulic gradient reduces the pressure in the discharge line to the vapor pressure of water. Whenever this condition occurs, the normal waterhammer solution is no longer valid. If this subatmospheric pressure condition inside the pipe persists for a sufficient period, the liquid water in the discharge line parts and is separated by a section of water and vapor. Whenever possible, water column separation should be avoided because of the potentially high pressure rise which often results when the two liquid water columns rejoin.

An accurate solution of the hydraulic transients in pump discharge lines where water column separation occurs, which includes all of the pertinent factors, is quite tedious. As an example, a detailed graphical waterhammer solution utilizing the elastic water column theory, and including the inertial of the pump and motor, is given in Reference 8. A subsequent paper (see Reference 9) provides a simplified approximate waterhammer solution for the water column separation phenomena in pump discharge lines. In this paper the following assuptions are made:

(1) The rigid water column theory is utilized until the instant that the separated water columns rejoin.

(2) The effect of the pump and motor rotating inertia on the deceleration of the water columns upstream or downstream of the separation are neglected.

(3) The cushioning effect of the small amount of air, if any, which enters the pipeline at the point of separation is neglected.

The referenced paper gives the necessary equations for computing the total interface separation of the water columns, the velocity of the columns at the moment the columns rejoin, and the maximum head rise due to the rejoining of the water columns. The results obtained by this latter method are conservative when compared to field observations and the more detailed method of analysis noted above.

QUICK-OPENING SLOW-CLOSING VALVES: A quick-opening slow-closing valve can be used to limit the pressure rise at the high points in the discharge line where water column separation occurs. When the pressure in the pipeline at the point of water column separation drops below a predetermined value for which the valve is set, the valve opens quickly and a small amount of air is admitted into the pipeline. After the upper water column in the pipeline stops, reverses, and returns to the point of separation near the valve, the valve is now wide open. At first the air and water mixture, and then clear water discharges through the valve. The open valve provides a nearby point of relief and reduces the pressure rise due to the rejoining of the water columns. The valve is later closed slowly under the action of a dash-pot so that the head rise in the discharge line at the valve location due to shutting off the reverse flow is not objectionable. Whenever these valves are used, precautions should be taken to insure that they are properly sized, field adjusted to the proper opening and closing times, and adequately protected against freezing.

ONE-WAY SURGE TANKS: The one-way surge tank which was introduced by the writer about 10 years ago (see Refernce 8) is an effective and economical pressure control device for use at locations where water column separation occurs. Numerous pipelines are now in service with one or more one-way surge tanks in the discharge line system. A one-way surge tank is a small tank filled with water to a level far below the hydraulic gradient. It is connected to the main pipeline with check valves which are held closed by the discharge line pressure. Upon a power failure, when the head in the discharge line at the one-way surge tank drops below the head corresponding to the water level in the tank, the check valve opens quickly and the tank starts to drain, thus filling the void formed by the separation of the water columns. When the flow in the upper column starts to reverse, the check valves at the one-way tank close before any appreciable reverse flow is established in the discharge line. Thus, the pressure rise due to the rejoining of the water columns is avoided. The connecting pipes to the one-way surge tanks which have been built and field tested to date have been equipped with more than one non-slam type check valves. The initial level of water in the one-way surge tank is usually maintained automatically with float control valves. It should be noted that the one-way surge tank does not act during the starting up cycle of the pump discharge line, and that it must also be protected against freezing.

AIR CHAMBERS: An effective device for controlling the pressure surges in a long pump discharge line is a hydro-pneumatic tank or air chamber. The air chamber is usually located at the pumping plant. The air chamber can be of any desired configuration and may be placed in a vertical, horizontal or sloping position. The lower portion of the chamber contains water, while the upper portion contains compressed air. The desired air-water levels are maintained with float level controls and an air compressor. When a power failure occurs at the pump motor, the head and flow developed by the pump decreases rapidly. The compressed air in the air chamber then expands and forces water out of the bottom of the chamber into the discharge line, thus minimizing the velocity changes and waterhammer effects in the discharge line. When the pump speed is reduced to the point where it cannot deliver water against the existing head, which is usually a fraction of a second after power failure, the check valve at the discharge side of the pump closes rapidly, and the pump then slows down to a stop. A short time later the water in the discharge line slows down to a stop, reverses, and flows back into the air chamber. As the reverse flow enters the chamber, usually through a throttling orifice, the air volume in the chamber decreases and a head rise above the pumping head occurs in the discharge line. The magnitude of this head rise depends on the volume of air which has been provided in the tank. For estimating purposes the total volume of the air chamber can usually be taken as about 5 per cent of the total volume of the discharge line.

The results of a large number of graphical waterhammer air chamber solutions using the elastic water column theory is given in convenient chart form in Reference 1. These charts are based on the following assumptions:

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

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

(3) The rotational inertia effects of the pump and motor are neglected.

(4) The pressure volume relation for the compressed air in the air chamber is taken as PV^{1-2} = constant.

(5) The differential throttle at the entrance to the air chamber is proportioned so that the ratio of the total head loss for the same flow into and out of the air chamber is about 2.5 to 1.

(6) The hydraulic losses in the pump discharge line are small when

compared to the head loss at the orifice for the same flow passing into the air chamber.

Another study of the action of air chambers at pumping plants subsequent to a power failure, using the rigid water column theory and the isothermal pressure volume relation PV = constant for the compressed air in the air chamber, is given in Reference 4. Although the isothermal relation for the pressure volume changes in not attained in actual practice, the results of this study are useful for estimating purposes because they include separately the orifice head losses and the hydraulic losses in the pipeline.

A characteristic of air chambers at pumping plants which is often overlooked is that when a power failure occurs at the pump motors, there is also a sudden temperature drop associated with the rapid expansion of the air in the chamber. At a recent installation the computed minimum air temperature in the air chamber associated with the initial pressure drop and an assumed pressure volume relation PV^{1-2} = constant was about −20°F. The inside of the air chamber was therefore provided with suitable protective coatings to minimize the effect of the sudden temperature drop on the air chamber shell.

SURGE TANKS: A surge tank is one of the most dependable devices that can be used at a pumping plant to control the pressure changes resulting from rapid changes of flow in the discharge line subsequent to a power failure at the pump motor. Following a power failure the water in the surge tank provides a nearby source of potential energy which will effectively reduce the rate of change of flow and waterhammer in the discharge line. The analysis of the surges in surge tanks has been worked out by many investigators. One presentation and a set of charts are given in Reference 1 from which the surges due to the sudden starting or stopping of a pump can be readily obtined.

One of the disadvantages of a conventional surge tank is that since the top of the tank must extend above the normal hydraulic gradient to avoid spilling, the tank could be quite tall and expensive at high head pumping installations. At booster pumping plant installations a lower level surge tank on the suction side of the booster plant is an effective and economical method for controlling the pressure surges in the combined discharge line system. In order to obtain the most economical surge tank design, care should be given to the proper sizing of the throttling device at the base of the tank.

NON-REVERSE RATCHETS: Another device for reducing the

pressure rise in a pump discharge line upon power failure which is occasionally used is a non-reverse ratchet on the pump and motor shaft which prevents the reverse rotation of the pump. This device is effective for controlling the pressure rise upon a power failure because of the large reverse flow which can pass through the stationary impeller. The author's experience to date with the non-reverse ratchet mechanisms has been very disappointing. At a number of small pump installations where these devices were used, the shock to the pump and motor shaft system due to the sudden shaft stoppage created other serious mechanical difficulties.

AUTOMATIC RESTART OF MOTORS: At small unattended pumping plants it is often desirable after a power failure ot automatically return the pumps to service as soon as the power is restored. However, at one project it was found that occasionally, subsequent to a very short power outage, a number of the induction motors at various pumping units could restart and come quickly up to speed while a large reverse flow was still passing through the pump. Under these conditions the pressure rise in the discharge line was very objectionable. If a pump motor has the capability of restarting under such transient conditions, a time delay or similar device should be installed at the motor controls so that the pump motor can be restarted only when it is safe to do so.

LONG SUCTION LINES: The detailed analysis of the hydraulic transients upon power failure for a pumping plant with a long suction line is given in Reference 1. The selection of the most appropriate waterhammer control device for a particular installation with a long suction line is a matter of judgment and economics. Surge tanks and air chambers on the suction side of the pumps often provide an effective method for controlling the pressure surges in long suction lines. At a recently built installation, the control of the pressure rise in a 7 mile long suction line consisted of a number of bypass valves on the suction side of the pumps. When a power failure occurs, these bypass valves open quickly and discharge water from the suction line into the atmosphere thus relieving the pressure rise due to the rapid deceleration of the flow in the suction line. The bypass valves are later closed at a slow rate to minimize any further waterhammer effect in the suction line.

SIPHON OUTLETS: A siphon structure is sometimes used at the outlet end of a pump discharge line to prevent backflow from the receiving reservoir or canal during a pump outage. During normal pumping operation the siphon flows full and a negative pressure is

created at the siphon either naturally from the flow of water or with the aid of a vacuum pump. When a power failure occurs at the pump motor, the siphon breaker valve opens quickly and admits air into the siphon, thus disrupting the flow and preventing backflow from the canal or reservoir into the discharge line. This operation generates a positive pressure wave at the siphon outlet which moves opposite to the direction of the flow toward the pump. This positive pressure wave assists in reducing the amount of the downsurge in the discharge line and in some cases may actually prevent a water column separation which might otherwise occur. The magnitude of the generated positive pressure wave is equal to the difference between atmospheric pressure and the negative pressure that existed at the siphon during steady state flow conditions.

NORMAL PUMP START UP

WITH CONTROLLED VALVE OPENING: At some pumping plants the pump is brought up to speed against a closed valve on the discharge side of the pump. The valve is then opened slowly and there is very little waterhammer in the discharge line. However, it will be found that nearly all of the pump flow in the discharge line is established with only a relatively small valve opening since the head loss across the valve decreases very rapidly during the opening stroke. For long discharge lines the head loss and flow characteristics of the valve during the opening stroke must be considered in determining the optimum rate of opening.

WITH CHECK VALVES: At pumping plants where the pipeline is held full with pump check valves, the pressure rise in the discharge line due to a pump start up can be objectionable in some cases. If the motor comes up to speed very rapidly, the pump will develop a pressure rise in the discharge line as the sudden increase in flow moves into the line. As noted above this pressure rise is lower for radial flow (low specific speed) pumps than for axial flow (high specific speed) pumps.

WITH CASING UNWATERED: At pumping plants which are equipped with large pumps or pump-turbine units, normal starting as a pump is often performed with the pump casing unwatered. This is accomplished by depressing the water level below the impeller by means of compressed air, which is admitted into the pump casing with the pump discharge valve closed and the discharge line full. After the motor is synchronized on the line, the compressed air in the pump casing is bled off allowing water to re-enter the pump from the suction elbow, after which the discharge valve is slowly opened.

This type of operation has been satisfactory with most large pumping units and there are normally no significant waterhammer effects in the discharge line. However, there have been some difficulties with this type of operation at large pump-turbine units. In the latter case, when the rising water level in the suction elbow first reaches the pump impeller, a very fast pumping action occurs within a few seconds resulting in a sudden large power demand on the electrical system. If the discharge valve is still closed or nearly closed when this fast pumping action occurs, there is usually very little if any waterhammer effect in the discharge line.

WITH SURGE TANKS OF AIR CHAMBERS: With a surge tank or air chamber at the pumping plant, it makes very little difference whether the increased pump flow is sudden or graduâl, inasmuch as the major portion of the sudden increased flow will enter the surge tank or air chamber. With these devices the steep front of the pressure rise in the discharge line is transformed into a much smaller pressure rise and a subsequent slow oscillating movement in the surge tank or air chamber.

EMPTY DISCHARGE LINE: A large pressure rise may occur in a pump discharge line during the rapid filling of an empty pipe. At one installation a very large pressure surge and a pipe failure occurred when the fire fighting pumps at the power plant were started up to fill a long empty pipeline. When the water in the discharge line reached the air release valves and spray nozzles at the upper end in the switchyard, the air release valves closed abruptly and a very large pressure rise occurred. To avoid this condition empty pipes must be filled slowly and furthermore the air from the high points in the pipeline must also be released slowly.

NORMAL PUMP SHUT DOWN

The pumping installation which produces the least waterhammer effect in a pump discharge line during a normal pump shut down is one in which the control valve on the discharge side of the pump is first closed, and then the power to the pump motor is shut off. If check valves only are in operation on the discharge side of the pumps, and the power to one of several pump motors which are connected to the same discharge line is cut off, the flow at the pump which has been shut down will reverse very rapidly, and the check valve will also close very rapidly. At several nuclear installations very severe check valve slam and damage occurred due to such sudden flow reversals. Consequently special anti-slam features were incorporated in the replacement check valves.

COMPARISON BETWEEN COMPUTED AND OBSERVED TRANSIENTS

In the author's experience in the field testing of pumping plants, the agreement between the computed and observed hydraulic transients has been acceptable but not as good as that obtainable with valve operation only. In most cases the observed transients were less severe than those which were computed.

At one pump-turbine installation large periodic instantaneous type pressure shocks were observed on the oscillograph pressure traces during the pumping power failure tests. There is a possibility that these pressure shocks were due to wicket gate vibration or to the interchanges in the control of the flow between the closing wicket gates and the pump as the pumping unit was increasing its reverse speed.

On water column separation tests it has been found that axial symmetry of the water columns at the point of separation does not always exist. Instead it is found that occasionally at the point of separation there is a long void or cavity at the upper portion of the pipe cross section and liquid at the bottom portion. The observed maximum pressure rise due to the rejoining of the water columns has not been in excess of the computed values based on the assumption of axial symmetry of the separated water columns.

The convential surge tank and one-way surge tank solutions agree very well with observations. In tests with air chambers there is good agreement if the air expansion exponent K in the equation PV = constant is taken between 1.2 to 1.3.

CONCLUSIONS

A variety of waterhammer control devices for pumping plants are available to the designer. In most cases the experienced designer can narrow the choice of the most suitable device to a few practical alternatives. A prior knowledge of the available waterhammer solutions for these devices will reduce the amount of detailed computational work which must be made to determine the critical hydraulic transient effects.

REFERENCES

1. "Waterhammer Analysis" by John Parmakian, Prentice-Hall, Inc., 1955, Dover Publication, 1963.
2. "Waterhammer Charts for Centrifugal Pump Systems" by Hitoshi Kinno and John F. Kennedy, Proceedings of ASCE, Journal of the Hydraulics Division, May, 1965.
3. "Charts for Determining Size of Surge Suppressors for Pump-Discharge Lines" by Carl W. Lundgren, ASME Journal of Engineering for Power, January, 1967.

4. "New Graph for the Calculation of Air Reservoirs, Account Being Taken of the Losses of Head" by G. Combes and R. Borot, La Houille Blanche, Oct.-Nov. 1952.
5. "Complete Pump Characteristics and the Effects of Specific Speeds on Hydraulic Transients" by Benjamin Donsky, ASME Journal of Basic Engineering, December, 1961.
6. "Pressure Surges in Pump Installations" by John Parmakian, Transactions ASCE Volume 120, 1955.
7. "Pressure Surges at Large Pump Installations" by John Parmakian, Transactions ASME, August, 1953.
8. "One-Way Surge Tanks for Pumping Plants" by John Parmakian, Transactions ASME, Volume 80, 1958.
9. "Pressure Surges Following Water-Column Separation" by J.T. Kephart and Kenneth Davis, Transactions ASME, Journal of Basic Engineering, 1961.
10. "Water-Hammer in a Complex Piping System — Comparison of Theory and Experiment" by Stephen S. Jones, ASME Paper 64 - WA/FE-23.

PROCEEDINGS OF THE SOCIETY

December 6, 1967.— A joint meeting of the Boston Society of Civil Engineers with the Sanitary Section was held this evening at Patten's Restaurant, 173 Milk Street, Boston, Mass., and was called to order by President Harry L. Kinsel, at 7:45 P.M.

President Kinsel stated that the minutes of the meeting held October 25, 1967 would be published in a forthcoming issue of the Journal and that the reading of those minutes would be waived unless there was objection.

Prof. Paul A. Dunkerley, Secretary Pro-tem announced the names of applic nts for membership, and that the following had been elected to membership November 15, 1967:—

Grade of Member — Bernard B. Berger, Victor Elias, Allen F. Goulart, Richard K. Guzowski, Thomas L. Jester, Jekabs B. Vittands, James T.Y. Wong, Newton L. Worth, Walter T. Jackson, Brian J. Watt.

Grade of Junior — Bruce O. Tobiasson, Thomas H. Turton.

President Kinsel requested Prof. Dunkerley to present a recommendation of the Board of Government to the society for initial action. The president stated that this matter was before the society in accordance with the provisions of the by-laws, and that notice of such action published in the ESNE Journal dated December 4, 1967.

Prof. Dunkerley presented the following recommendation of the Board of Government to the society for action to be taken at this meeting.

MOTION — "to recommend to the Society that the Board of Government be authorized to transfer an amount not to exceed $4,000 from the principal of the Permanent Fund to the Current Fund for current expenditures".

On motion duly made and seconded it was VOTED "that the Board of Government be authorized to transfer an amount not to exceed $4,000 from the principal of the Permanent Fund to the Current Fund for current expenditures".

President Kinsel stated that final action would be taken on this matter at the January meeting of the society.

President Kinsel announced that the John R. Freeman Fund Committee is offering a John R. Freeman Scholarship this year.

President Kinsel stated that this was a joint meeting with the Sanitary Section and turned the meeting over to Walter M. Newman, chairman of that section to conduct any necessary business of that section at this time.

Chairman Newman introduced the guest speaker of the evening, Mr. Robert Davidson, Solid Waste Disposal Study Project Director of Metropolitan Area Planning Council, who gave an interesting talk on "A Solid Waste Disposal Program for Metropolitan Boston".

A question period followed the talk.

Forty-four members and guests attended the dinner and meeting.

Meeting adjourned at 8:45 P.M.

Respectfully submitted,

Paul A. Dunkerley
Secretary Pro-tem

HYDRAULICS SECTION

A meeting of the Hydraulics Section of the Boston Society of Civil Engineers was held in the society rooms. The meeting was called to order at 7:00 P.M. by Mr. Allan Grieve, Jr., chairman of the section.

Mr. Grieve introduced the speaker of the evening, Mr. Thomas F. Cheyer, of Camp, Dresser and McKee, consulting engineers. Mr. Cheyer spoke on "The Warragamba Pipe Lines Stilling Basins, Sydney, Australia." Mr. Cheyer explained the problem of the basins and the constraints that had to be satisfied in their design. It was pointed out that this design was an extension beyond the normal designs for which experience and research data were readily available.

The meeting had an attendance of 20 and was adjourned at 9:00 P.M.

Respectfully submitted,

Ronald T. McLaughlin
Clerk

STRUCTURAL AND CONSTRUCTION SECTIONS

The joint meeting of the Structural and Construction Sections was held in the society rooms on the evening of January 10, 1968. The meeting was called to order at 7:00 P.M. by Prof. Charles C. Ladd, chairman of the Structural Section.

Prof. Ladd introduced the first speaker, Mr. H. B. Mandel, Project Engineer, Parsons, Brinkerhoff, Quade & Douglas, who gave a short talk on the engineering aspects of the design of the foundations for the Newport-Jamestown, R.I. Bridge. Mr. Mandel's talk was confined generally to the ascertaining of the economical location of the bridge and the general design problems of span versus foundation costs. His talk was accompanied by slides.

Mr. William B. Wiley, chairman of the Construction Section, then introduced the next speaker, Mr. Morse Klubock, Chief Engineer of the Marine Division of Perini Corporation, substituting for Mr. C. R. Richardson, Vice President. Mr. Klubock talked about the construction aspects of the foundation for the above bridge, pointing out with great clarity the problems of acquiring the proper equipment and finding shops capable of producing the prefabricated structural section required for this project. He also mentioned the prohibitive cost of using divers at depths of 150-160 feet below water surface where men could work only for periods of twenty to forty minutes in a working day. His talk was also accompanied by slides.

Interesting question and answer periods followed both talks.

The meeting adjourned at 8:55 P.M.

There were fifty-three members present.

Respectfully submitted,

Albert B. Rich, Clerk

STRUCTURAL SECTION

A regular meeting of the Structural Section was held in the society rooms on the evening of October 10, 1967. The meeting was called to order by Chairman Charles Ladd at 7:03 P.M.

The chairman introduced the speaker of the evening, Dr. Frank J. Heger, Jr., of Simpson, Gumpertz & Heger, who spoke on "Design of Pavilion Dome at Expo 67" (illustrated).

Dr. Heger showed how the pavilion fitted in with the surroundings at Expo. He showed slides showing the general design of the structure, including the transparent cover, the procedure of erecting the steel, and another set illustrating the erection of the transparent cover. He indicated that the cost of the steel ran around $1,000 to $1,100 per ton in place, and the rough cost per square foot of dome surface was around $5.50 to $6.00.

The meeting adjourned at 8:25 P.M. after a short discussion period.

There were 24 members and guests present.

Albert B. Rich, Clerk

STRUCTURAL SECTION

A regular meeting of the Structural Section was held in the society rooms on the evening of December 13, 1967. The meeting was called to order at 7:00 P.M.

Chairman Charles C. Ladd introduced the speaker of the evening, Prof. Sepp Firnkas, Associate Professor of Civil Engineering, Northeastern University, who spoke on the subject "High-Strength Concrete".

Prof. Firnkas gave a very enlightening talk on the possible future of concrete having strengths in the vicinity of 15,000 - 20,000 p.s.i. These high-strength concretes would possibly be brought about by placing a hard aggregate in the forms and injecting an epoxy as a binder. The talk was followed by an interesting question period.

The meeting adjourned at 8:35 P.M.

There were 35 members present.

Respectfully submitted,

Albert B. Rich, Clerk

NEW MEMBERS

Bernard Berger, Amherst, Mass.
Bromwell, Lester G., Winchester, Mass.
Nestor D. Disenfeld, Brookline, Mass.
Victor Elias, Chestnut Hill, Mass.
Ernest R. Goodwin, Marblehead, Mass.
Allen F. Goulart, Raynham, Mass.
Richard K. Guzowski, Manchester, Mass.
H. Hobart Holly, Braintree, Mass.
Thomas L. Jester, Marblehead, Mass.
Walter T. Jackson, Ramsey, N.J.
Jekabs P. Vittands, Norton, Mass.
Brian J. Watt, Boxford, Mass.
James T.Y. Wong, Newton, Mass.
Newton L. Worth, Arlington, Mass.
Frank W. Stockwell, Boston, Mass.

New Junior

Marcia A. Root, Brookline, Mass.

Deaths

Arthur H. Gordon, January 10, 1968
James H. Manning, June 17, 1967

ERRATA

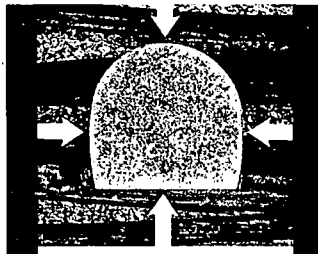
Errata for article by H.E. Mohr, Journal of the Boston Society of Civil Engineers, October, 1967, entitled "Pile and Caisson Foundations".

1. Page 153, 5th par. 2nd line, change "distributed" to "disturbed".
2. Page 157, 6th par. 6th line, omit second "and".
3. Page 158, 5th line, inset "on" between "welded" and "Plate".
4. Page 162, 5th par. 4th and 5th line, change "inside mandrel" to "outside drive pipe".
5. Page 166, 1st par, 5th line, change last "the" to "as".
6. Page 170, End of 5th par., change "steam curling" to "steam curing".
7. Page 179, 3rd par., end of 8th line, omit the word "even".
8. Page 183, 3rd par., 1st line change "constructed" to "contracted".
9. Page 183, 4th par., 1st line, change "plant" to "plan".
10. Page 188, 5th par., 3rd line, change "Caldwell" to "Calweld".

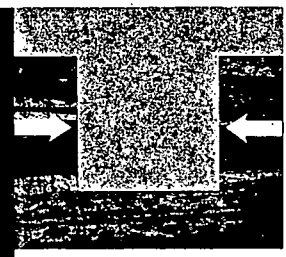
CONTROL

COST, DESIGN AND SAFETY DURING CONSTRUCTION

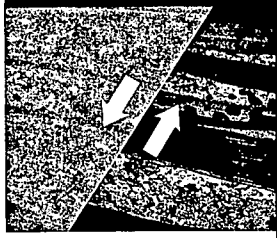
Terrametrics Instrumentation gives you precise measurements, enabling your support and structural designs to be based on known load conditions, not compromise or assumption! Terrametrics' in situ determination of rock and soil characteristics means new cost reductions, new design improvements, and new safety standards for your construction job!



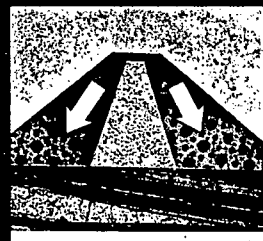
TUNNELS AND UNDERGROUND STRUCTURES—Terrametrics instruments for controlled construction accurately analyze the exact design parameters of all types of underground openings. Terrametrics Multiple Position Borehole Extensometer (MPBX) and Terrametrics Load Cells (PLC) produce accurate measurement of the stability of roof, walls, floors of tunnels, slopes—precise information concerning in situ stresses.



FOUNDATION EXCAVATIONS—Unexpected rock movement, soil displacement, or movement of adjacent foundations because of inadequate shoring or unknown conditions, can have a major effect on your structure—or those nearby. Terrametrics instrumentation techniques can accurately measure rebound or other movements and detect displacements as small as 0.001 inch during any stage of excavation or construction. Terrametrics Borehole Deflectometer, Pore Water Pressure Cells and the new Geocel Pressure Meter (System Menard) are recommended for these measurements.



SLOPE STABILITY—Slope weaknesses or slope movement often create hazards in construction. Effective preventive steps can be taken when these slope conditions are more precisely understood. Proper use of Terrametrics MPBX, Borehole Deflectometer, and/or Pore Water Pressure Cells will give the engineer data so he can effectively define the magnitude of the problem and design proper control measures. These instrumentation techniques also will allow a continuing study of the effectiveness of the corrective procedures in controlling the movement.



EARTH FILL EMBANKMENTS — In situ measurement is the only sure method of determining if your embankment is properly compacted, stable, and will perform without excessive deformation. Through use of Terrametrics Borehole Deflectometer, Soil Pressure Cells, Pore Water Pressure Cells, in combination, changes in soil behavior can be immediately known and design changes can be made without delay.

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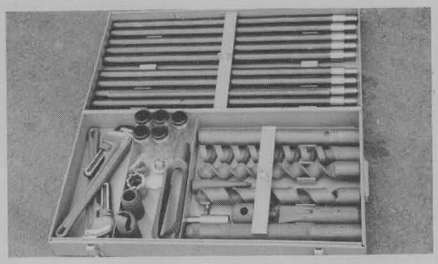

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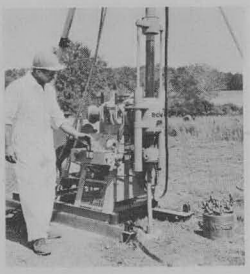

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