# EUROPEAN EXPERIENCE WITH THE THERMODYNAMIC METHOD

by
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## 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 (1). 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 |
| $\Theta$ | [°C] | temperature |
| P | [BHP kW] | Power |
| g | [m/$sec^2$] | gravity value |
| $\gamma$ | [kp/$m^3$] | specific weight |
| d | | scale value of galvanometer |
| $\eta$ | | 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

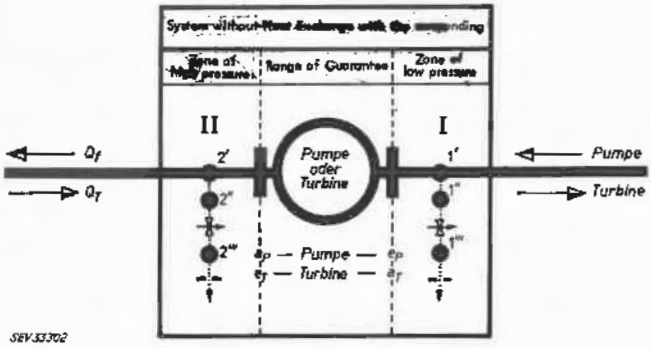
Subscript aT, aP : Leaving the range of responsibility of the Turbine or the Pump

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

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

$\underline{H}u$ 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:

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

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

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

p = pressure in kp/cm²

$\theta$ = Temperature of water in °C

$\alpha_m$ = Correction coefficient # 1

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

$\beta_m$ = Correction coefficient # 2, referring to the differences between $\gamma_0$ and $\gamma_m$ 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 = gN and cpI = 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 $\alpha_m=0$ or, now expressed in the main equation:

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

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: $\eta_{nT} = \frac{\underline{H}u}{H_n}$

Pump: $\eta_{nP} = \frac{H_n}{\underline{H}u}$

## 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 $\theta_{II}$ are measured.

The measuring section for $p_I$ and $\theta_I$ 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.

$\Theta$ probe # 1 in location 1′ or in 1″; pressure $p_I$

$\Theta$ 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.

$\Theta$ probe # 1 in location 2″; pressure p″

$\Theta$ 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 $\alpha_m$ and $\beta_m$ -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 =

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

Thus giving the final simplified equation for Hu

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

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).

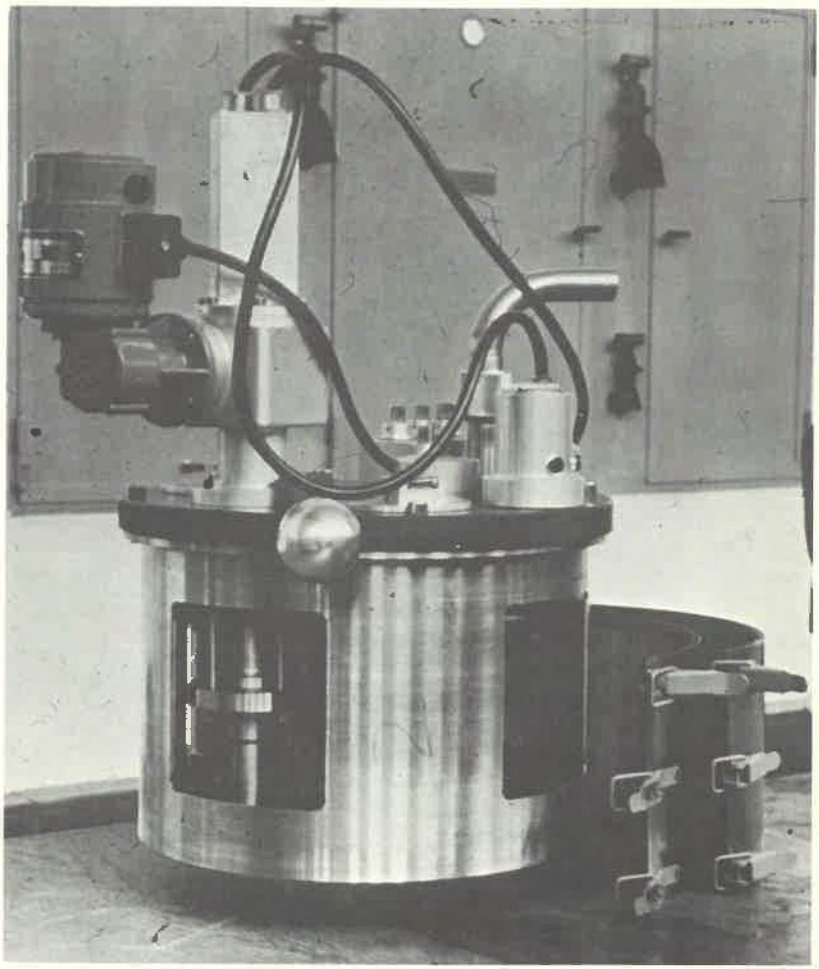


FIG. 2: Double Expander of ETH — Equipment.

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.

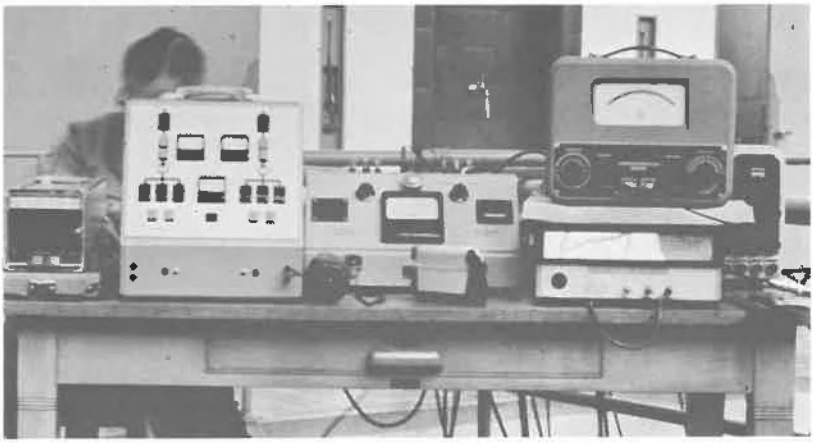


FIG. 3: Recording Instruments for Temperatures and Pressures.

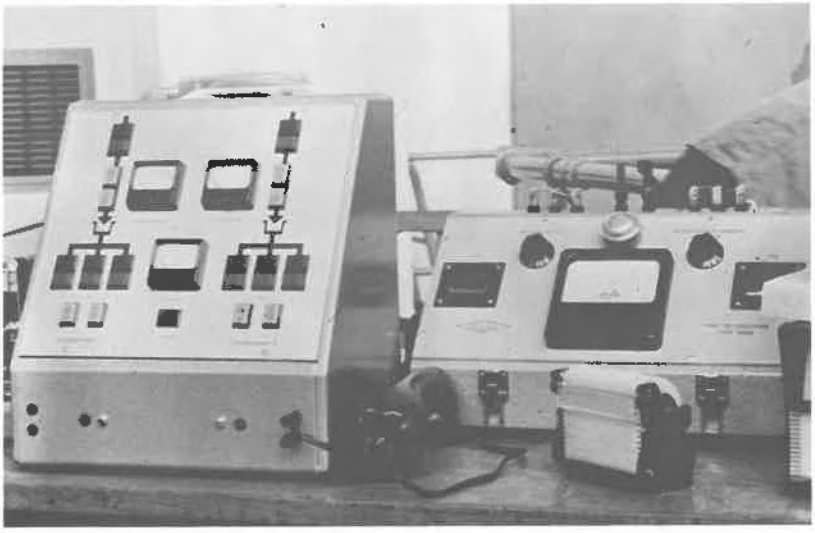


FIG. 4: AC — Bridge and Galvanometer.

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.

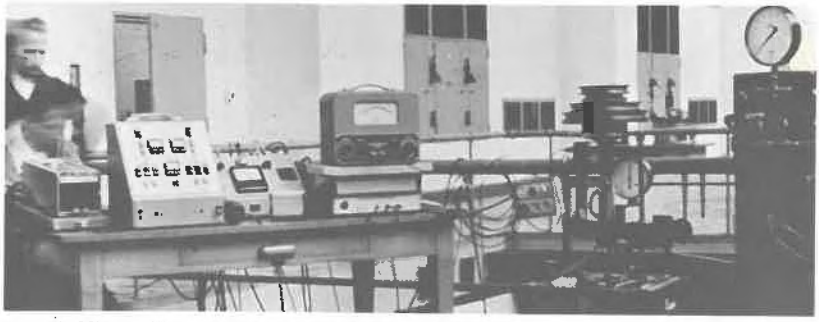


FIG. 5: View of the complete new ETH — Set.

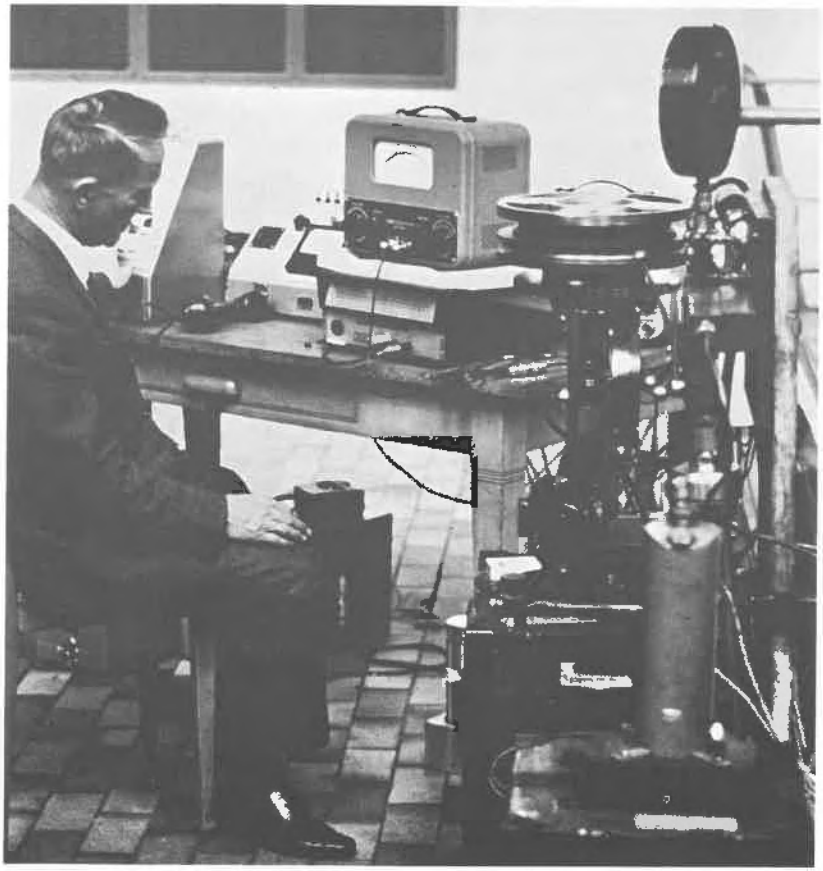


FIG. 6: ETH — Set installed in the Etzel Power Plant, for demonstrations at the 1966 meeting of the "Groupe international des Practiciens de la méthode thermodynamique".

## 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 $\gamma$ (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, $\alpha$, 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

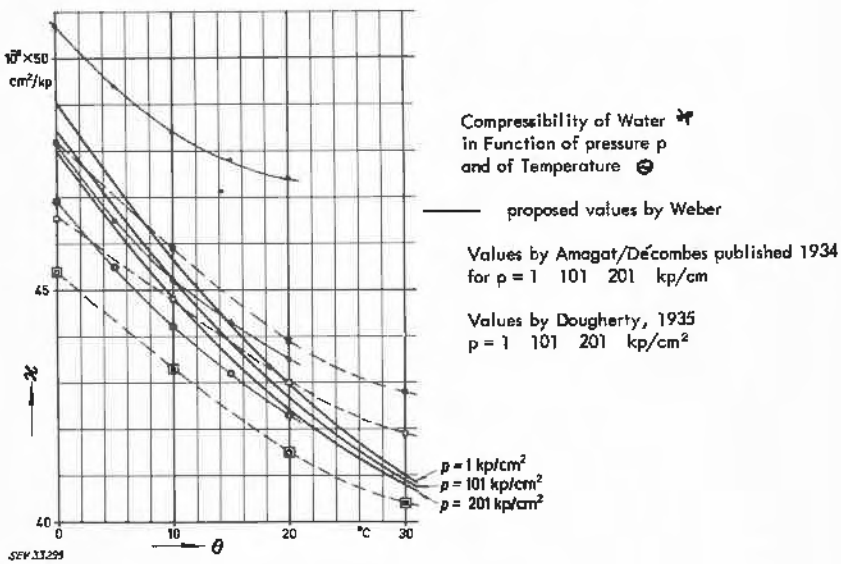
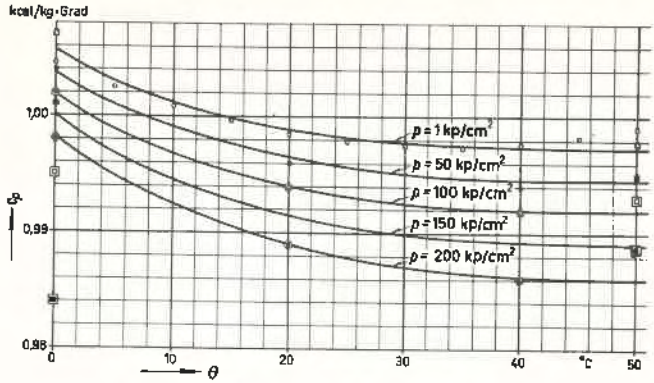



FIG. 7: Compressibility of Water, values by Weber.



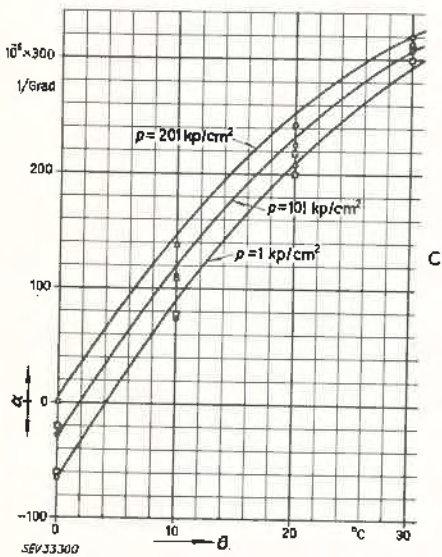

Specific Heat of Water $c_p$
in Function of pressure p
and of Temperature $\Theta$

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

Values from "Hütte" 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 $\Theta$
□ values of Amagat/Decombes 1935
○ values of Wukalowitsch 1958

FIG. 9: Coefficient of Expansion α of Water.

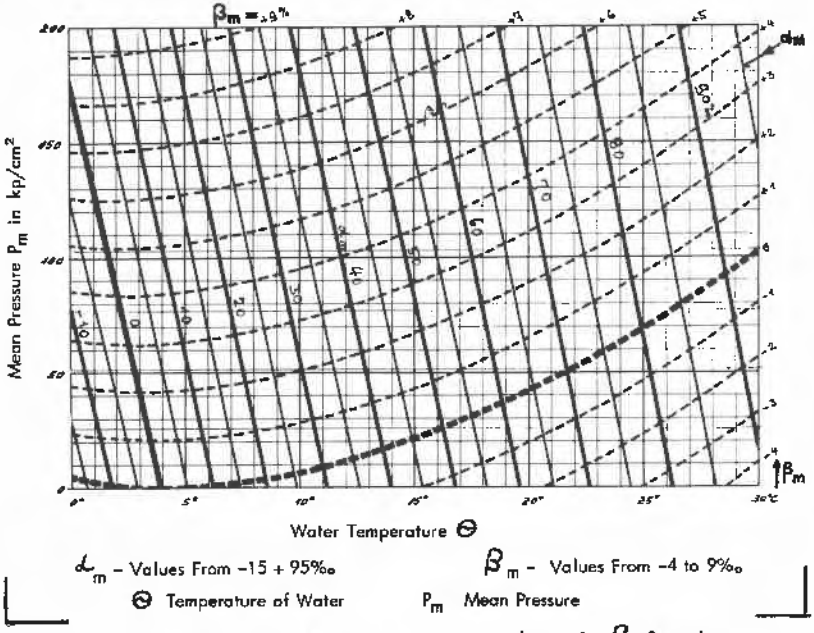



FIG. 10: Values by Weber: Correction Factors $\alpha_m$ and $\beta_m$ for given mean pressures and temperatures.

temperature, $\Theta$, for given values of $\alpha_m$ and $\beta_m$. 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.

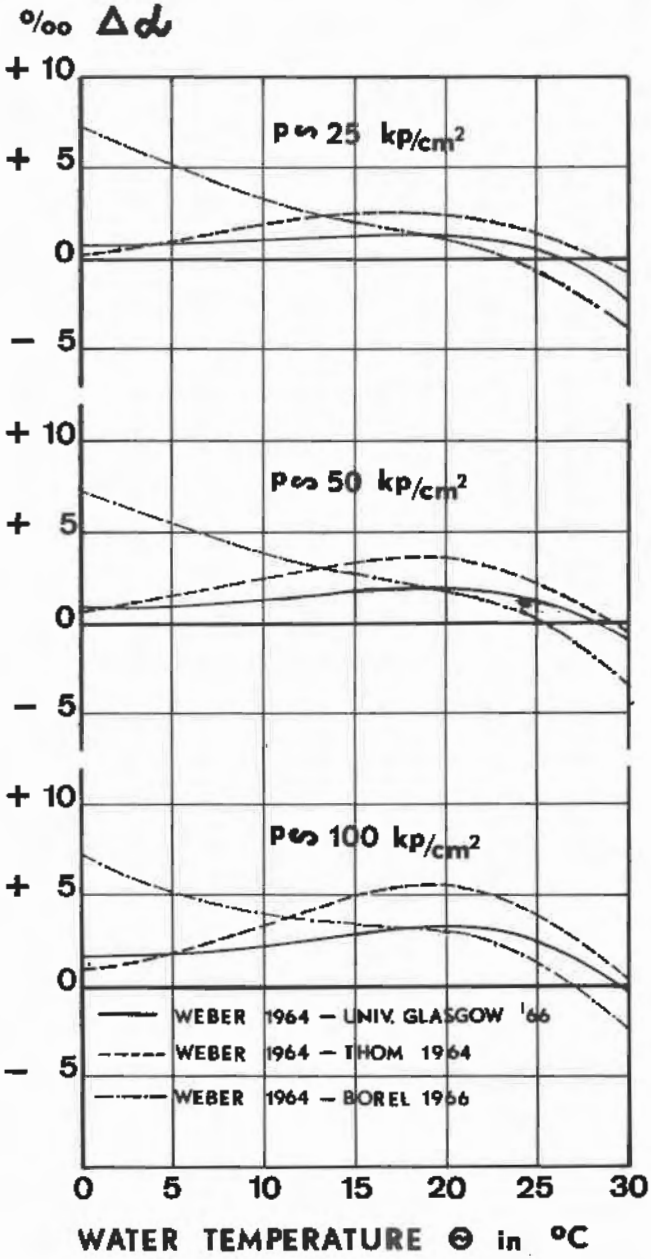



FIG. 11: Comparison of Weber's $\alpha$ – values with other published values.

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 $H_u$ 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 $H_u$ 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 $p_m$ 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) $\alpha_m$ 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 | $\varphi_H$ |
| Total Energy Head | $\varphi_{Hu}$ |
| $\alpha_m$ - value | $\varphi(1-\alpha_m)$ |

is given by the equation:

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

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 $\Delta\eta_o < 5\%$

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

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: $102 \cdot \eta_P \quad P_{P\cdot} = \gamma \cdot Q \cdot H$ 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 $\beta_m$ 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).

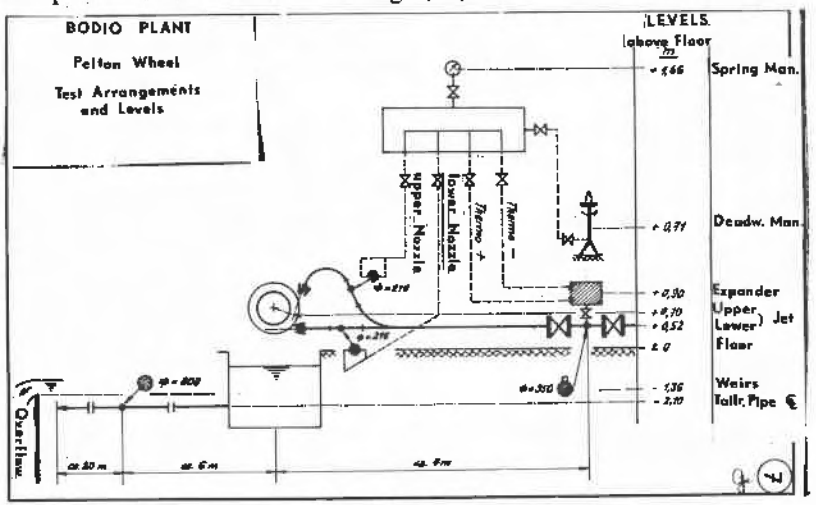



FIG. 12: Bodio Plant, General Measuring Arrangement.

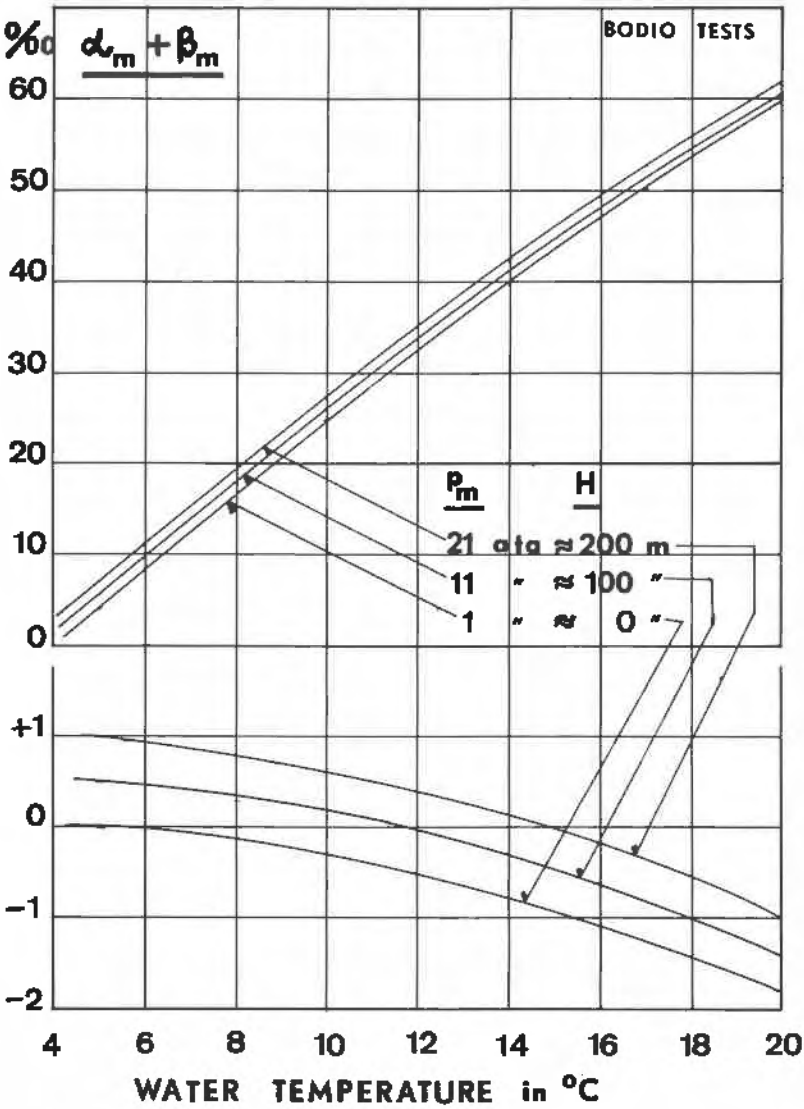



FIG. 13: Bodio, values of $\alpha_m$ and $\beta_m$ used for computation.

*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

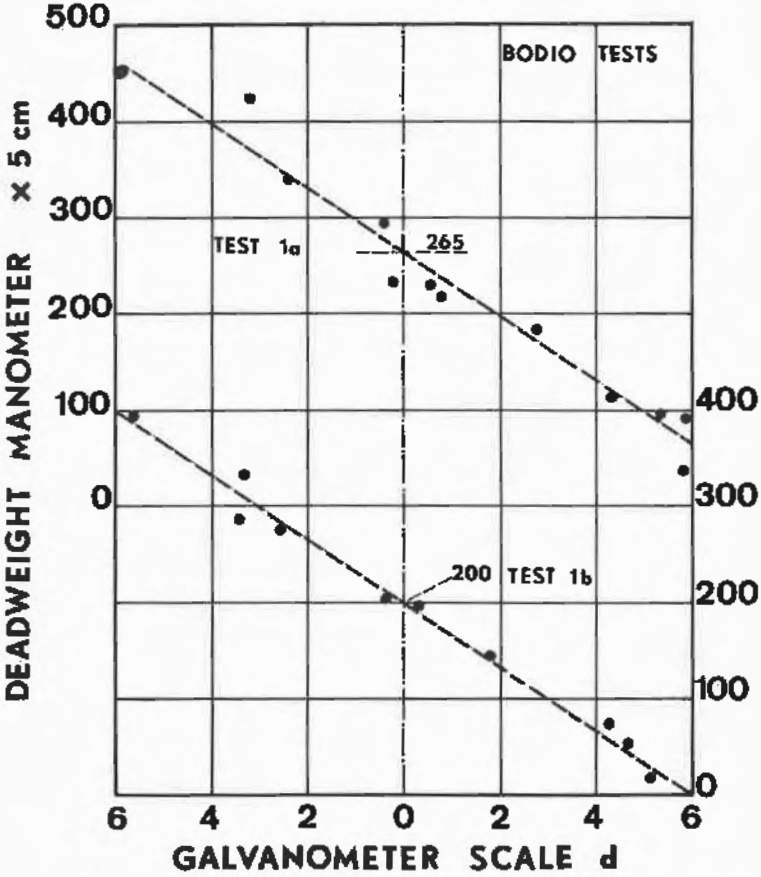



FIG. 14: Bodio, Measured pressure values, plotted versus Galvanometer (= Temperature) — Readings.

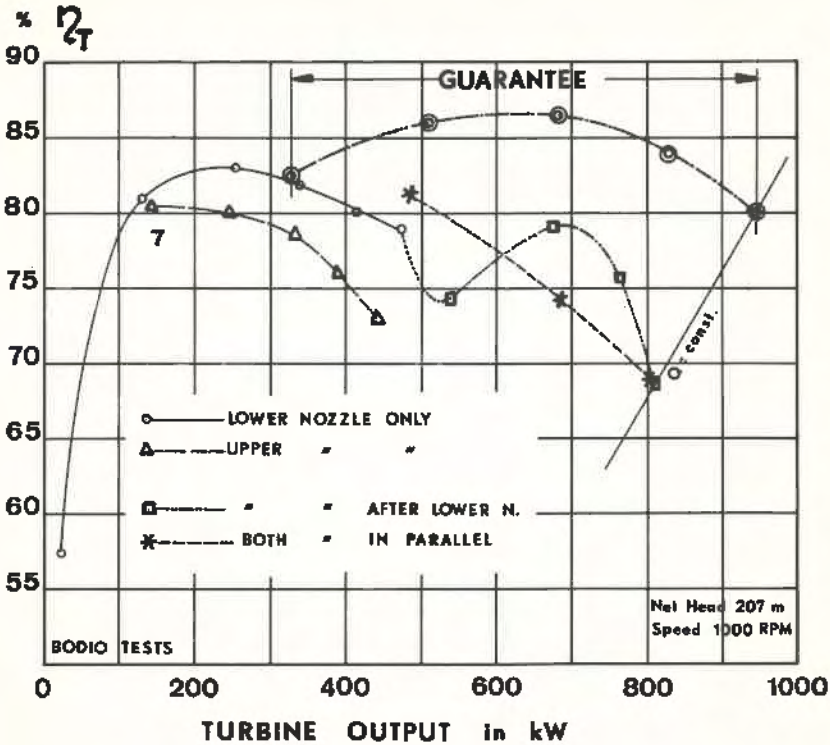


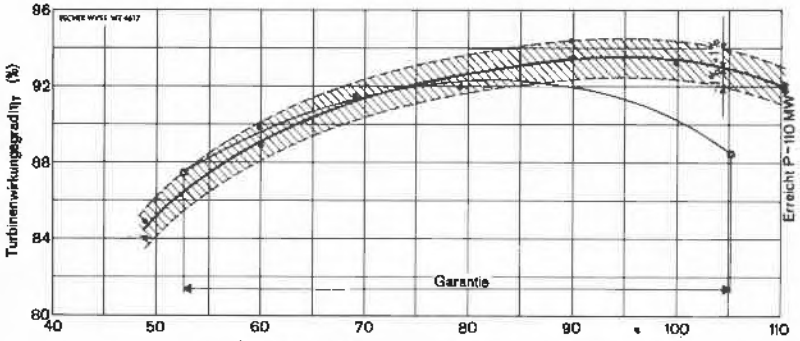

FIG. 15: Bodio, Pelton wheel, Test Results.

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

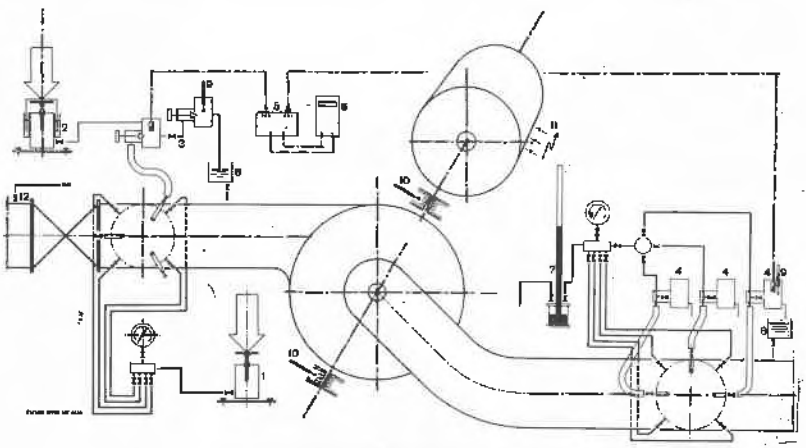



Vianden Pump Storage Plant, Luxemburg. Field Test Results, obtained with the Thermodynamic Method on Pump Nr. 2.
Pump Heads H = 292-268 m (958-880 Ft.)
Pumped Flow Q = 20.75 - 22.75 $m^3/s$
Speed N = 428.6 RPM

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

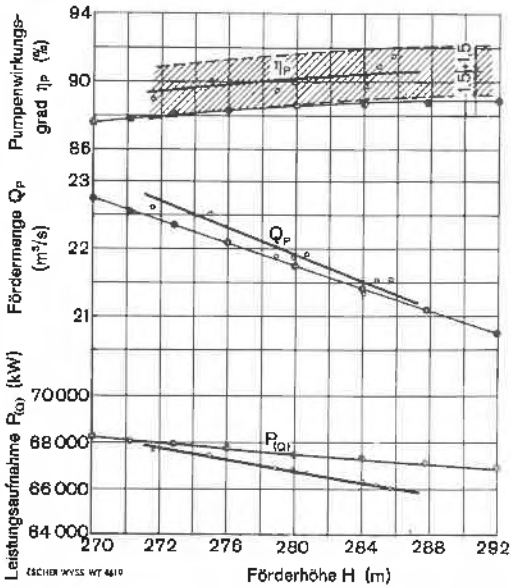
more or even to doubt as it was at the beginning. It will be improved and developed in a rapid manner in the near future. As far as we can see today this development will be directed in the following most important fields.

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

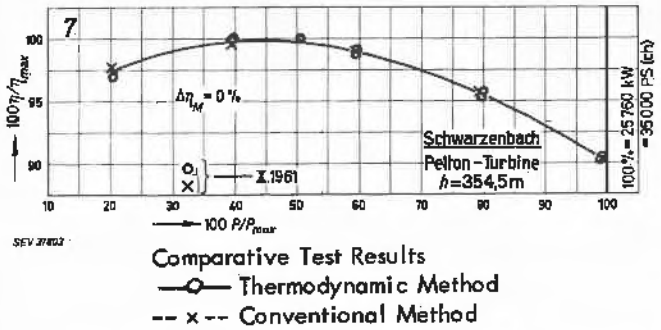



FIG. 19: Comparative Tests at Schwarzenbach Plant; H = 1160 ft.

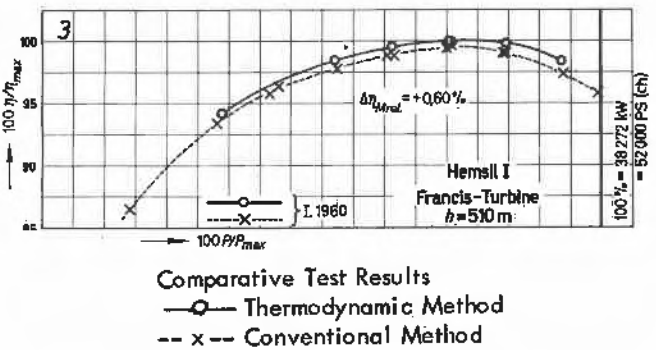



FIG. 20: Comparative Tests at Hemsil I Plant; H = 1670 ft.

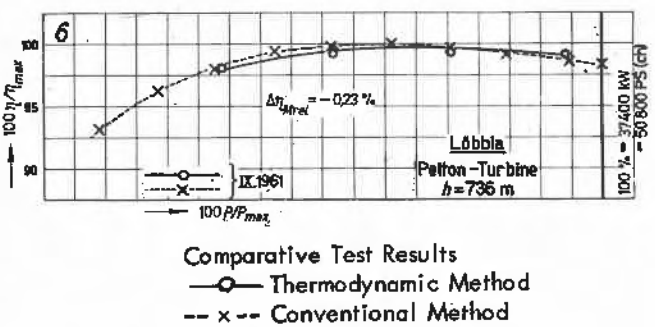



FIG. 21: Comparative Tests at Loebbia Plant; H = 2420 ft.

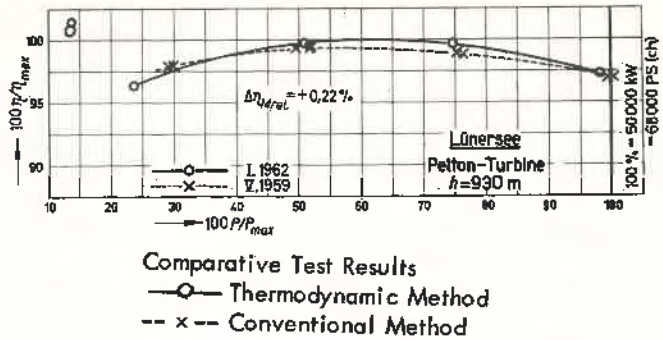



FIG. 22: Comparative Tests at Luenersee Plant; H = 3050 ft.

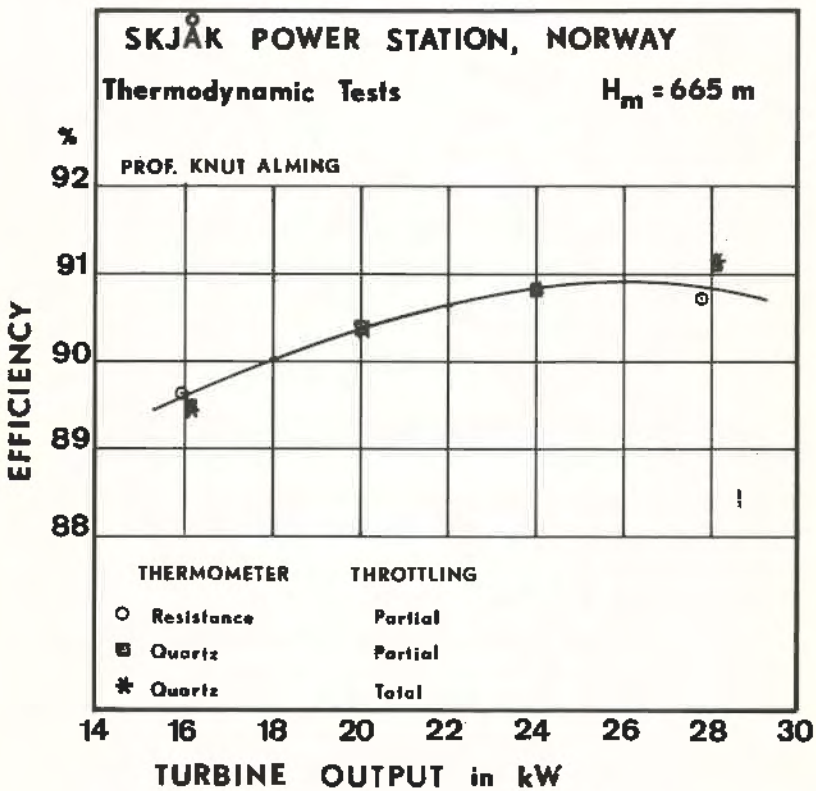



FIG. 23: First Results with direct Thermometric Method; Tests executed in the Skjåk Plant by Prof. Knut Alming, Trondheim, Norway.

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.

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