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AERODYNAMIC EXPERIMENT PLANTS FOR HYDRAULIC MACHINES

by

Dr. C. Keller, Chief Engineer, Escher Wyss, Zurich

Aerodynamische Versuchsanlagen für hydraulische Maschinen

von

Obering. Dr. C. Keller, Escher Wyss, Zürich

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In practical machine construction water and air are treated as two entirely different working mediums. The incompressibility of water permitted a special theoretical handling of the hydraulic problems offered by water turbines and pumps, while in the construction of steam and gas turbines and compressors, the compressibility of the medium must be considered as a deciding factor. Accordingly hydrodynamics and aerodynamics have developed into two large independent research fields which have little in common from the viewpoint of the technician. Until recently hydraulic and heat engineers applied generally different methods in the calculation, construction, and experimental research relative to their machines.

Under superficial consideration an essential difference seems to exist between a liquid and a gas with finite compressibility. The more physical aspects of flow show that in both cases, because of the attainable degree of accuracy, the differences in the mechanical structure are often not of the magnitude generally expected. When the velocity of flow of gaseous mediums is small compared with the velocity of sound in that particular fluid (i.e., with low values of the Mach number $M = c/a$) and pressure and temperature changes are small, the compressibility is of relatively small importance.

In all aerodynamic and aeronautical flow problems we have to deal largely with this case. In this modern field of knowledge the air is accordingly treated as incompressible, and countless experimental studies in this field have established the qualification of this theoretically established view. From the standpoint of the modern flow theories, air and water are considered very similar fluids in several relations. They are alike in the state of rest in that both yield to change of shape.

Little application has really been made of this knowledge of the relation of gases and liquids in hydraulics and in hydraulic machine construction. Stimulated by the aerodynamic viewpoint, certain experiments were started a few years ago in the use of air as experimental medium instead of water for pressure measurements on individual parts of machines, pipe lines, draft tubes of water turbines, throttle valves, etc.[1] In the numerous hydraulic laboratories of colleges and factories that conduct research in hydrokinetics, water has hitherto been used for the study of all hydraulic machines or models of such, under operating conditions in which energy is exchanged between working fluid and impeller. In the following paragraphs is described a modern experiment plant of the Escher Wyss firm, in which it is possible, using air as working fluid, to

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[1] See, e.g., such experiments by: J. Ackeret, "Theory of the Kaplan Turbine," Escher Wyss-Mitteilung, vol. IV, 1931, p. 61; Werner C. Keller, "Air Model Experiments on Throttle Valves for Pressure Lines of Hydraulic Plants," Schweizerische Bauzeitung, vol. 107, p. 133 (March 28, 1936).

advantageously perform in several relationships all quantitative and qualitative measurements that have been made by known water experiment plants.

Since the calculation with compressible mediums and with it the possibilities of the representation of water currents by air currents are evidently but little known to the hydraulic expert, the most important theoretical relations between gas and water movements shall first be explained[1]. Our goal is to drive a model of a complete water turbine plant with air, for instance a modern Kaplan turbine plant. The relations shall be chosen in such a way as to enable the performance of numerically accurate experiments with model turbines such as are possible in a hydraulic experiment station, as: efficiency tests, measurements of discharge, characteristics, etc. Evidently, it must first of all be clearly known how large the velocity of flow of air must be with respect to the compressibility, and whether one can expect a reliable representation of the turbine or pump flows over a practical working range with accurately measurable pressure differences and velocities.

In the general aerodynamic equations[2], which underlie the nature of the flow through the pneumatic machine under consideration, the

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[1] An analogous study of stream flows was handled prior to this publication in the works of Ackeret, Keller, Salzmann: Die Verwendung von Luft als Untersuchungsmittel für Probleme des Dampfturbinenbaues, (1934, vol. 104, pp. 259*, 275*, 292*).

[2] Handbuch der Physik vol. VII, p. 289: Ackeret, Aerodynamics.

densities ($\rho = \gamma/g$) of the gases and their variations appear determinative. In contrast to water, ρ is here not constant. A direct measure for the compressibility is the relation $\frac{\Delta\rho}{\Delta p}$, the ratio of change in density ρ of a gas to a corresponding pressure change Δp. For relatively small pressure changes, which are assumed in the case under consideration, we assume as equivalent the reciprocal value of this relation $\frac{dp}{d\rho}$. The expression for the velocity of sound is $a = \sqrt{\frac{dp}{d\rho}}$, and thus our characteristic measure for compressibility is on first approach simply

$$\frac{\Delta\rho}{\Delta p} = \frac{1}{a^2} \qquad (1)$$

We can express the pressure change Δp as corresponding to the dynamic pressure $\rho\, c^2/2$ at a velocity c. It follows by inserting in (1) that

$$\frac{\Delta\rho}{\rho} = \frac{1}{2}\,\frac{c^2}{a^2} = \frac{M^2}{2} \qquad (2)$$

(M = Mach number = ratio of flow velocity to sound velocity).

At an air velocity of $c = 100$ m./sec. ($M = 0.29$) the density change with respect to still air is 4.2%. Fig. 1 shows the approximate percentage of density change as the velocity is increased.

The general equation of continuity for steady flow,

$$\frac{\partial\,(\rho u)}{\partial x} + \frac{\partial\,(\rho v)}{\partial y} + \frac{\partial\,(\rho w)}{\partial z} = 0 \qquad (3)$$

is expanded to

$$\rho\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)+u\frac{\partial \rho}{\partial x}+v\frac{\partial \rho}{\partial y}+w\frac{\partial \rho}{\partial z}=0 \quad (4)$$

(x, y, z = space coordinates; u, v, w = velocity components in x, y, z directions). The continuity equation for liquids (ρ = constant) thus becomes

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \quad (5)$$

For simple unidimensional flow ($v = 0$, $w = 0$) the continuity equation for the variable hydraulic cross-section f is

$$\rho f \cdot u = \text{const., or}$$

$$fu = \text{const. when } \rho = \text{const.} \quad (6)$$

In the general Euler movement equations for steady flow with g_x, g_y, g_z as force components per unit mass at x, y, and z,

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=g_x-\frac{1}{\rho}\frac{\partial p}{\partial x}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=g_y-\frac{1}{\rho}\frac{\partial p}{\partial y} \quad (7)$$

$$u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}=g_z-\frac{1}{\rho}\frac{\partial p}{\partial z}$$

ρ can also be regarded as constant with little error, as in liquids. For a streamline irrotational motion, this resolves into the energy theorum (equation of Bernoulli) $p + \frac{\rho}{2}c^2 + \rho g z = \text{const.}$

The influence of the compressibility of the air on the pressure equation is thus less than on the continuity equation. The increase of the pressure with great velocities is given by the relation:[1]

[1] Prandtl-Tietjens: Hydro- und Aeromechanik, vol. 1, p. 212.

$$\Delta p_{st} = \frac{\rho}{2} c^2 \left(1 + \frac{M^2}{4} + \frac{2 - k}{24} M^2 + \ldots\right) \qquad (8)$$

Fig. 2 shows as a function of the air velocity, the amount of error involved when the pressure of the air is calculated in the same manner as for an incompressible fluid. At a velocity of $c = 100$ m./sec., it is around 2%. Thus if a pressure of 600 mm. water, for example, is measured by means of a manometer at normal air density, $\rho = 0.12$, the corresponding actual air velocity is not $c = \sqrt{\frac{\Delta p_{st}}{1/2\ \rho}} = \sqrt{\frac{600}{0.06}} = 100$ m./sec., but 1% smaller, or 99 m./sec. Since the corrections rapidly diminish as the velocity decreases, the error introduced is practically negligible for our purposes.

If the hydraulic model machine is operated with air, the power action on the vanes is not necessarily noticeably influenced by the compressibility. This problem is treated in an interesting theoretical investigation by Ackeret[1], one of the few existing works on problems involving great velocities. In this study is estimated the coefficient of lift c_a of a surface, for instance a blade profile, with compressible flow. It is demonstrated therein that with Mach coefficients up to relatively large values, $M < 0.8$,

$$c_a \text{ (compr.)} = c_a \text{ (incompr.)} \cdot \frac{1}{\sqrt{1 - M^2}} \text{ (approximately)} \qquad (9)$$

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[1] J. Ackeret: "Air Power at Very Great Velocities Particularly with Steady Flow." Helvetica Physica Acta 1928, p. 301.

Referring again to our experimental fluid, air, the factor $\frac{1}{\sqrt{1 - M^2}}$ varies with the velocity as shown in fig. 3. At an air velocity of 100 m./sec. the compressibility has a 4.5% increase in lifting effect. At smaller velocities this influence rapidly decreases, as in the density and pressure calculations. At 70 m./sec. the compressibility produces only 2% variation from the liquid case.

From these considerations it seems possible that with extreme velocities (relative to the blades) of perhaps 100 to 150 m./sec., the action with air should not show an essential quantitative difference from the action with water. The practical air velocities necessary for experiments are becoming smaller, as we shall see later on. Although the first considered foundations of aerodynamics were known, to our knowledge no one has investigated the possibility of conducting complete hydraulic experiments aerodynamically. In such a plant, air is drawn into the experimental machine by means of a guide apparatus, as will be explained below using the new Escher Wyss experiment plant as an example.

Figures 4 and 5 illustrate the simple layout of an aerodynamic experiment plant based on these principles. A fan (1) driven by an electric motor draws (in case A) the air from the entire system and exhausts it to the atmosphere at (2). The air enters inlet (3), goes through the guide apparatus into the turbine (4), and then flows through the draft tube into the pressure equalizing chamber (5). From here it passes through a measuring orifice (6) to the fan (1).

The entrant air drives the runner in the same manner as water. The power developed by the turbine is measured by a hydraulic or electric brake (7). The actual turbine plant comprises only the part from the inlet to the end of the draft tube (a to c). The pressure at the different points is schematically shown in fig. 4. In this case a vacuum exists in the entire system and the plant must therefore be perfectly airtight in order that only the air flowing through the guide apparatus and the runner be actually measured. With the small vacuum used (about 100 mm. water gage) the necessary density is easily obtained.

The air can also be forced through the entire plant as in case B, fig. 5. In this case a fan (1) discharges air into the turbine inlet (3), from whence it flows through the turbine (4), then through the draft tube into the equalizing chamber (5), and finally to the measuring orifice (6). We now have a small pressure above atmospheric in the entire system except for a small vacuum in the neighborhood of the runner.

Beside these two fundamentally different solutions, variables for the given conditions can be taken into consideration. For instance the measuring orifice can be placed anywhere in the inlet according to the requirements. Also, the draft tube can lead directly into the atmosphere. It is always essential that the model turbine correspond to the prototype as closely as possible and that the measuring orifice indicate an even flow so that no measuring errors occur.

The head H of an actual water turbine corresponds in our case to the difference in air pressure Δp between turbine inlet and equalizing chamber. In case A (fig. 4), the turbine head is directly represented by the vacuum in the chamber (5), as shall be explained in greater detail here. The water discharge corresponds to the volume of air drawn through and measured at (6) as p_d.

As is known, conditions of similarity between model and prototype must be obtained for the achievement of accurate model results. This applies to water experiments as well as to air experiments.

The first condition is geometric similarity, which must apply not only to the runner, but to all accessories, such as inlet, guide apparatus, draft tube, etc. The second important condition is that the Reynolds numbers R_e for model and full-scale machine be very nearly equal. In the average present-day experiment plant operated with water, this requirement is never completely attained. Due to the small dimensions of the model and the low peripheral velocity compared to prototype, the Reynolds number of the model, R_{e_m}, is only about 1/10 to 1/20 of that of the prototype. Long experience with hydraulic experiment stations shows, however, that these deviations do not essentially affect the model experiment results, and that they can be used practically with sufficient accuracy when correcting coefficients are introduced.

It appears that with the same size model, similar Reynolds numbers are obtained in air experiments as in experiments using water. The Reynolds numbers bear the relation

$$\frac{R_e\text{ (air)}}{R_e\text{ (water)}} = \frac{u_A}{u_w} \cdot \frac{\nu_w}{\nu_A}$$

where u = peripheral velocity and ν = kinematic viscosity. At 15° C, 1 atm., $\nu_A = 15 \cdot 10^{-6}$ m.^{2}/sec. $\nu_w = 1.15 \cdot 10^{-6}$ m.^{2}/sec. With a value of u_A of about 120 m./sec. and a u_w of about 10 m./sec. (the latter value corresponds to average normal conditions in hydraulic experiments) we have $\frac{R_{e_A}}{R_{e_w}} = \frac{120 \cdot 1.15}{10 \cdot 15} = 0.92$, which for our purposes is close enough to unity.

An aerodynamic turbine experiment plant is therefore no more disadvantageous in regard to relative Reynolds numbers than existing water experiment plants.

It shall now be estimated what power to expect with air as working fluid, with the known limit values of the peripheral velocities, and with average model diameters.

With a discharge of Q in m.^{3}/sec. and a head of H in meters, the power of a water turbine is

$$N_w = \frac{\gamma Q H \eta}{75} \text{ (H.P.)} \tag{10}$$

The corresponding power of an air turbine with Q in m.^{3}/sec. and an available pressure difference Δp (mm. water - kg./m.^{2}) amounts to

$$N_A = \frac{Q \Delta p \eta}{75} \text{ (H.P.)} \tag{11}$$

In order to be able to compare similar points of observation of water and air (with similar velocity planes) the reduced velocities and also the usual dimensionless coefficients used in turbine design must be determined from the air data.

With u as peripheral velocity of the runner, a coefficient for high speed is expressed by

$$K_{u_w} = \frac{u}{\sqrt{2\,gH}} \tag{12}$$

Between H and Δp we have the relation

$$\frac{\Delta p}{\gamma} = H; \quad \frac{\Delta p}{\rho} = gH \tag{13}$$

For the air turbine,

$$K_{u_A} = \frac{u}{\sqrt{\frac{2\,\Delta p}{\rho}}} \tag{14}$$

It follows that

$$\Delta p = \frac{\rho}{2}\,\frac{u^2}{K_u^{\,2}} \tag{15}$$

The axial velocity c_m through the runner's cross-sectional area F of a high speed turbine is given by

$$K_{c_m} = \frac{c_m}{\sqrt{2\,gH}} \tag{16}$$

For air, as above,

$$K_{c_{m_A}} = \frac{c_m}{\sqrt{\frac{2\,\Delta p}{\rho}}} \tag{17}$$

The volume of flow $(Q = Fc_m)$ becomes

$$Q_A = FK_{c_m}\sqrt{\frac{2\,\Delta p}{\rho}} \tag{18}$$

The reduced quantity of flow upon the unit diameter D_1 is

$$Q11_A = \frac{Q_A}{D_1^{\,2}\sqrt{\frac{\Delta p}{\rho g}}} \tag{19}$$

The following relation exists between the specific speed n_{s_W} of a water turbine and the specific speed n_{s_A} of an air turbine. An air machine is equivalent to a given hydraulic machine if at the same speed it has the same volume of flow. The utilized heads are proportional to the specific gravities.

From the definition,

$$n_{s_W} = \frac{n}{H}\,\frac{N_W}{\sqrt[4]{H}} \tag{20}$$

It follows that

$$n_{s_W} = n\,Q^{1/2} \cdot 75^{-1/2} \cdot \gamma_W^{\,5/4} \cdot \Delta p^{-3/4} \cdot \eta^{1/2} \tag{21}$$

For air

$$n_{s_A} = n\,Q^{1/2} \cdot 75^{-1/2} \cdot \gamma_A^{\,5/4} \quad (\Delta p\,\gamma_A/\gamma_W)^{-3/4} \cdot \eta^{1/2} \tag{22}$$

The ratio of the specific speeds is therewith

$$\frac{n_{s_W}}{n_{s_A}} = \sqrt{\frac{\gamma_W}{\gamma_A}} \tag{23}$$

When $t = 15^\circ$ C, and $b = 736$ mm. Hg.,

$$n_{s_W} = 29.1\, n_{s_A}$$

Furthermore, in working with air, it may be advantageous to use the usual coefficients in the fan design; the pressure coefficient $\Psi = \frac{\Delta p}{\frac{\rho u^2}{2}}$, and the flow coefficient $\phi = \frac{c_m}{u}$, as well as the dimensionless specific speed $\sigma = 2.105\, Q^{1/2}\, (\Delta p/\rho)^{-3/4} \cdot n_{sec}$, rather than n_s. The following relations hold:[1]

$$n_{s_W} = 578\, \sigma \sqrt{\eta}, \quad \Psi = \frac{2\, Hg}{u^2} = \frac{1}{K_u^{\,2}}, \quad \phi = \frac{K_{c_m}}{K_u},$$

$$\sigma = K_u\, (K_{c_m}\, (1 - \nu^2))^{1/2} \qquad (25)$$

The power of a model runner driven by air, for instance of a modern propeller turbine, can now be simply estimated. The model runner diameter shall be $D = 500$ mm., the hub ratio $\nu = d : D = 0.4$, and $F = \frac{\pi}{4}(D^2 - d^2) = 0.165\ m^2$; it shall be given further that $K_u = 1.6$, $K_{c_m} = 0.6$. With $u = 100$ m./sec., equation (15) gives $\Delta p = \frac{10^4 \cdot 0.12}{2 \cdot 2.56} = 235$ kg./m.^{2} = 235 mm. water. From equation (17) $c_m = 45.9$ m./sec. Also, $Q = Fc_m = 7.58$ m.^{3}/sec. Thus we derive the theoretical air turbine power, $N = \frac{Q \Delta p}{75} = 23.3$ H.P. The speed n is $\frac{30\, u}{\pi R} = 3800$ revolutions per minute, $\sigma = 1.14$; i.e., $n_{s_W} = 608$ when $\eta = 0.85$.

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[1] C. Keller: "Axial Flow from the Standpoint of the Wing Theory." Dissertation E. T. H., 1934, page 29 and following pages.

As one can see from these calculations, in low and average pressure plants, with favorable dimensions and technically favorable pressures, model air turbines deliver powers which can also be fairly easily measured.

The first experiment plant of the Escher Wyss laboratories for water turbines was built as in fig. 4, with a suction system. The possibility of having an undisturbed turbine inlet is thus attained, and both the turbine and the draft tube can be annexed to the equalization chamber. Photograph 1 is a view of the plant from the inlet side. The turbine shaft and the draft tube axis, as well as the entire pipe system are arranged in a horizontal plane over the floor. In contrast to plants using water, where the machine axis must be vertical to utilize the natural head, any convenient arrangement can be used for an air plant. Ready assemblage of individual units is permitted by the horizontal layout. The free accessibility to all points in the turbine plant is a great advantage; in hydraulic experiment plants the entire plant is inaccessible under water. The new air experiment plant can be easily adapted to any locations. As the photographs show it is mounted on light and easily moved framework, so that different turbine arrangements can be experimented with after alterations that can be quickly made. Since only air serves as motive power, many plant parts can be made of wood which is lighter and cheaper.

Hydraulic experiment plants require long inlet and discharge-measuring channels for stabilization. This is not the case in the aerodynamic plant and therefore its floor space requirements are considerably smaller, as shown in fig. 6, where the old hydraulic and the new aerodynamic experiment plants of the Escher Wyss laboratories (built for same model sizes) are compared in the same scale.

In photograph 1 is seen in the right foreground an inlet guide, the turbine housing, and the adjoining draft tube in the rear. Farther back is the equalization chamber. The subsequent measuring channel is given three right-angled bands to save space, and the fan is contained in the last horizontal part to the left. This drives the air through a vertical flue into the atmosphere. The hydraulic brake seen in the middle foreground is coupled to the turbine shaft.

Photograph 2 is a view of the air plant from the fan side. To the left is the equalization chamber with the adjoining measuring orifice, and the fan is in the pipe turning at right angles to the back. This is a two-stage axial fan with adjustable blades for the regulation of the conveyed volume Q and the head H (or Δp). The fan is operated by a belt drive from a controllable electric motor. The adjustment of the speed of the turbine and fan, the volume regulation by throttling apparatus, the hydraulic brake operation, as well as the reading of all required pressures by means of a sensitive manometer, can be handled from a central point. On account of the ease of adjustment to a stable condition, measurements can be performed much quicker than with water operation; even the unavoidable oscillations of water level due to the great inertia of water columns are avoided. The discharge at any moment is ascertained at least as accurately by means of orifices as with a weir.

The dimensions of the new patented experiment plant of Escher Wyss were chosen in such a way that the individual elements of the ND Experiment Plant[1] could be used extensively in the air plant. The possibilities of experimental investigation were considerably enlarged by means of this interchangeable fabrication. Direct comparative tests can be performed on the same turbine plant at the same time with both water and air to check the validity of the theoretical results.

Fig. 7 shows comparative efficiency tests on the same runner with similar geometric construction and directly measured values on the old ND water plant without any supplementary corrections of the new air plant. The model wheel was a four-bladed Kaplan runner with an n_{s_W} of approximately 650. The curves show the efficiency (+ for water values, O for air values) as a function of the speed in r.p.m. with fixed positions of guide vanes and runner blades. The correspondence between water and air values is very good, even to details. The flattening at the crest, and in particular the very high speeds, are accurately reproduced in both cases.

Fig. 8 compares measured values of the discharge Q_1. The small difference in the discharge is caused by small variations of the guide wheel opening which could not be very accurately reproduced by the

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[1] See: "The Hydraulic Experiment Plants of the Escher Wyss Firm." Special Edition EW-Mittlg., 1934.

comparative water measurements of a few years ago. Nevertheless it is observed that these experiments correspond very well in other respects.

Beside the previously mentioned advantages of the air plant, it is above all important for research purposes that accurate pressure measurements on different parts of the plant are now possible. In the work with water, air locks occurring in the lines and the great pressures exerted by the water on the Pitot tube precluded accurate results. On the other hand the necessary measurements of small positive or negative pressures of a few cm. water are very accurately obtainable with sensitive manometers in the air plant without the introduction of errors. To the hydraulician the measurement of such small pressure differences is probably unusual at first. One must consider, however, that such measurements in flow studies, especially in aerodynamics, belong to the ordinary problems and offer no fundamental difficulties.

This fact encouraged us to search more carefully to see if a measurement of the static pressures on the rotating runner blade might be possible. If the pressure distribution on the runner is known, the power effect on the blades can be accurately followed in detail, which naturally is an important forward step compared to previous observation methods. For the purpose of such measurements of the pressure upon the runner blade of a Kaplan turbine, measuring ducts were made in the model blades, opening into small holes on the surface. The hollow turbine shaft contained an axial plunger movable from outside, with successive airtight connections with different measuring ducts to the hub seat. By means of the hollow shaft the pressure is transmitted to the coupling end of the hydraulic brake and thence through an oil-sealed stuffing box with ground packing rings, from the rotating part to a stationary pressure storage chamber. It is thereby possible to measure during the rotation of the runner a total pressure consisting of two components, one being the static pressure on the surface of the runner blade and the other the pressure in the measuring ducts set up by the centrifugal force. To ascertain the desired static pressure, the dynamic component must be deducted, the correction depending on the distance from the axis of rotation. The difficult problem of measuring accurately the pressure on a runner revolving at 1,000 to 3,000 r.p.m. could readily be solved by this method.

Fig. 9 shows the result of such measurements on the middle section of a Kaplan blade at different heads and guide vane openings β_o. The location of the fourteen measuring points in the cross-section is given above. The range of pressure distribution on each blade surface in normal working fields $K_u = 1.75$ and $K_u = 2.0$, especially on the suction side, is a consequence of the screening effect of neighboring vanes. In the case of $K_u = 1.2$, the low depression point on the front edge is a result of local high velocities at a large angle of incidence to the surface. The location of the resulting force and the torque of the blade may be determined from the pressure distribution. This is important to the designer in the location, for example, of the axle with respect to the governing forces. In the future such measurements should also considerably assist in the explanation of the great unsolved question as to the screening effect of wing profiles in turbomachines.

The now measurable pressure on the rotating blade under different operating conditions offers us furthermore a means of approaching along a new line of thought the important question of cavitation effects. In the customary hydraulic ND model experiment plants where one works chiefly with a head of only a few meters, no cavitation takes place because the velocities on the blade are small. The hydraulic ND experiment plant does not offer anything more in this regard than the air station. For the cavitation investigations, model runners were changed in a manner customary for increased heads and tested with correspondingly high velocities.

Aerodynamic experiments with pressure measurements can now replace cavitation investigations to some extent. The pressure distribution in the full-scale turbine is similar to that of the aerodynamic model under equal operating conditions, i.e., with similar velocities. The absolute water pressure of the particular type of machine can be calculated in the air model experiments from the relation of the specific gravities of air and water. If the pressure is below the vapor pressure of water, then cavitation will occur at certain planes. Since the negative pressures are unlimited in air motion, the vapor pressure establishes a lower limit to the pressure range on the turbine vane with water action and cuts off the deepest negative pressure points in the measured air pressure curve. The pressure conditions were stated by Ackeret[1]. The minimum pressure on the suction side of a Kap_an turbine runner can be shown as:

$$H_{min.} = B - H_s - \eta_s K_{c_m}^2 H - \lambda K_{w_2}^2 H \qquad (26)$$

(B = Barometric height, H_s = vacuum head, H = total head, η_s = draft tube efficiency, λ = measure of local negative pressure on the blade in multiples of the pressure $\rho w_2^2/2$). At the worst $H_{min.}$ can reach nearly zero, whence we get the condition for the Thomas number $\sigma*$ of the cavitation:

$$\sigma* = \frac{B - H_s}{H} = \eta_s K_{c_m}^2 + \lambda K_{w_2}^2 \qquad (27)$$

η_s and λ can in our case be directly calculated for the axial runner exit, from the pressure measurements on the air turbine plant.

$$\eta_s = \frac{p_3 - \overline{p}_2}{\rho/2 \cdot (c_2^2 - c_3^2)}, \quad \lambda = \frac{p_{min} - \overline{p}_2}{\rho/2 \cdot w_2^2} \qquad (28)$$

($\overline{p}_2$ = mean static pressure on the draft tube inlet, p_3 = static pressure in the equalization chamber, p_{min} = static pressure on the blade surface, w_2 = relative emersion velocity from runner).

Let us take for example the measurement in fig. 9 with a guide wheel position β_o = 55% and K_u = 1.75. The total head was H = 57 mm. water, $\overline{p}_2$ = -64.4 mm. water. From the measured discharge

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[1] Ackeret: "The Maximum Permissible Vacuum Head of Water Turbines." Schweiz. Bauzeitung vol. 91, p. 135 (March 17, 1928).

Q, we have $c_m = 12.2$ meters/sec. It follows, according to equation (17), that $K_{c_m} = 0.4$. The draft tube efficiency was measured to be $\eta_s = 0.82$. From the emersion velocity C_2, measured with the Pitot tube, w_2 can be calculated from the velocity triangle up to a value of 42.4 m./sec. $K_{w_2} = 1.38$; $\rho w_2^2/2 = 108$ mm. water; p_{min}, according to fig. 9, was -83 mm. water; therefore $\lambda = \frac{83 - 64.4}{108} = 0.122$. Using this data $\sigma*$ is evalued from equation (27). $\sigma* = 0.82 \cdot 0.16 + 0.172 \cdot 1.90 = 0.131 + 0.327 = 0.458$. At this value of $\sigma*$ the beginning of cavitation in the examined middle section is to be expected. It is seen that the value of $\sigma*$ in this case is affected by the second member of the equation and also by the greatest local negative pressure. By the reduction of λ (i.e., by avoiding all negative pressure points), one can obtain blades safe from cavitation. The air experiment makes it possible to directly measure the effect of blade alterations caused by certain damaging vacuums, and thus to develop favorable profiles in advance without actual cavitation experiments.

For turbine design, in addition to the pressure distribution on the runner, it is also important to have information relative to the exact inlet and outlet velocity distribution in regard to magnitude and direction. By means of cylindrical Pitot tubes[1] and by

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[1] For cylindrical Pitot tube, see C. Keller: "New Experimental Devices and Studies for Turbo-compressors." Escher Wyss--Mitteilungen vol. VIII, 1935, p. 164.

direct stroboscopic observations, these values can be determined more simply and accurately than by hydraulic methods.

The absolute inlet and outlet velocities c_1, and c_2, as well as the inclination α to the turbine axis and the static pressure p, are directly measured with the cylindrical Pitot tube. The velocity triangle can then be ascertained for each radius for a known rotational speed, and the operation continued for all radii and compared with theoretical evaluations. The difference of the summated static pressures on the entrance and exit surfaces of the runner gives the axial thrust. Fig. 10 for instance gives results of such measurements in the inflow and discharge chambers of a high speed runner. The axial velocity and the tangential component are represented respectively by c_m and c_u. The measuring points . - produce the various details with remarkable accuracy.

The new experimental method is not only used for turbine experiments but also in research work on hydraulic pumps. An air plant for investigation of pump impellers, designed on the basis of previous experiments in aerodynamic research, was set up some time ago in the Escher Wyss laboratories. This plant is sketched in fig. 11. In contrast to the above described turbine plant, this plant can be operated in accordance with a proposal of Prof. Ackeret, on the principle of wind tunnels with open or closed circuits. In the closed circuit the air is circulated by pump (2) driven by motor (1). The closed circuit has the advantage that the Reynolds number can be controlled by variation of the air density, either by admitting compressed air or by drawing air out of the tunnel (at equal temperatures, the kinematic viscosity ν in the expression $R_e = \frac{cl}{\nu}$ is inversely proportional to the pressure). The pump discharges air instead of water through the diffusion ring (3) into a radiator (4), where the accumulated frictional heat is discharged. The air can be throttled at the radiator inlet. It flows below through a rectifier (5) to the changeable measuring nozzle (6), to be drawn again from the pump impeller. The pump with the drive and the measuring arrangement of this plant are shown in photograph 3.

Extensive investigations have already been conducted for two years at the new aerodynamic experiment plants of the Escher Wyss laboratories for hydraulic machines. Through these experiments, aerodynamic plants have proved valuable supplements to the existing hydraulic experiment plants in modern research studies of hydraulic machines under operating conditions. Their outstanding advantage is that with relatively low expenditures, many unfamiliar details of the energy flow of turbines and pumps can be accurately observed and measured, thus promising rapid strides in the scientific investigation of hydraulic machines.

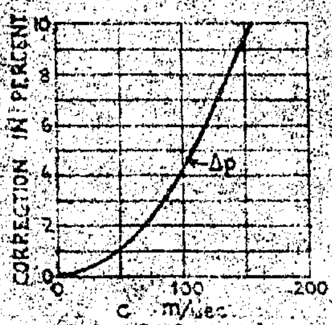


FIG. 1

Influence of the velocity of flow of air on its density. (20°C., 1 atmosphere)

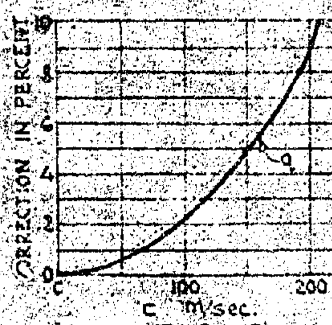


FIG. 2

Influence of the compressibility of the air upon the pressure with varying velocity of flow.

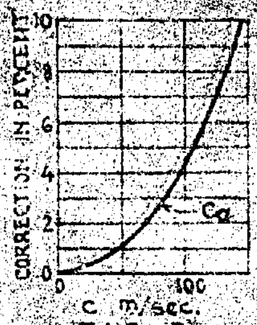


FIG. 3

Influence of the compressibility of the air upon the buoyancy coefficient C_a with varying velocity of flow.

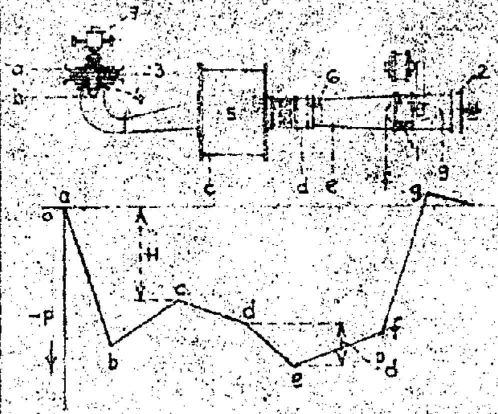


FIG. 4

Distribution of pressure (Below atmospheric)

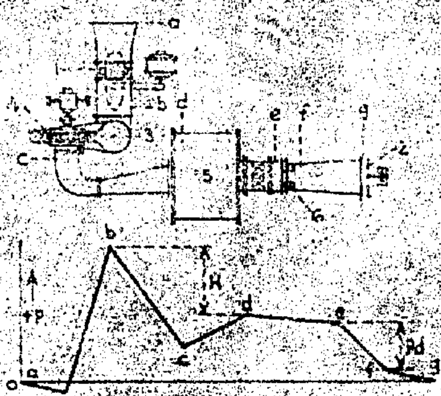


FIG. 5

Distribution of pressure (Above atmospheric)

1- Turbine with brake
2- Draft tube
3- Reservoir entrance
4- Measuring channel
5- Weir crest
6- Pump
7- Intake channel

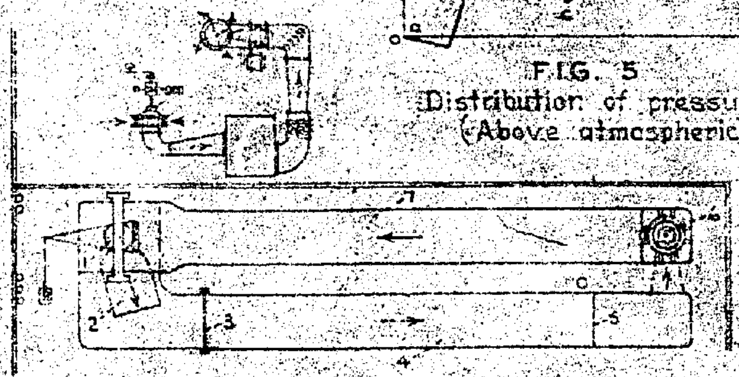


FIG. 6

Space requirements (floor plan) of an aerodynamic turbine experiment plant (above), and a common hydraulic plant (below)

Guide opening $\beta_0 = 70\%$
Runner position $\beta_2 = 0$

Guide opening $\beta_0 = 60\%$
Runner position $\beta_2 = 0$

FIG. 7

Efficiency measurement on a Kaplan turbine model with water (+) and air (o)

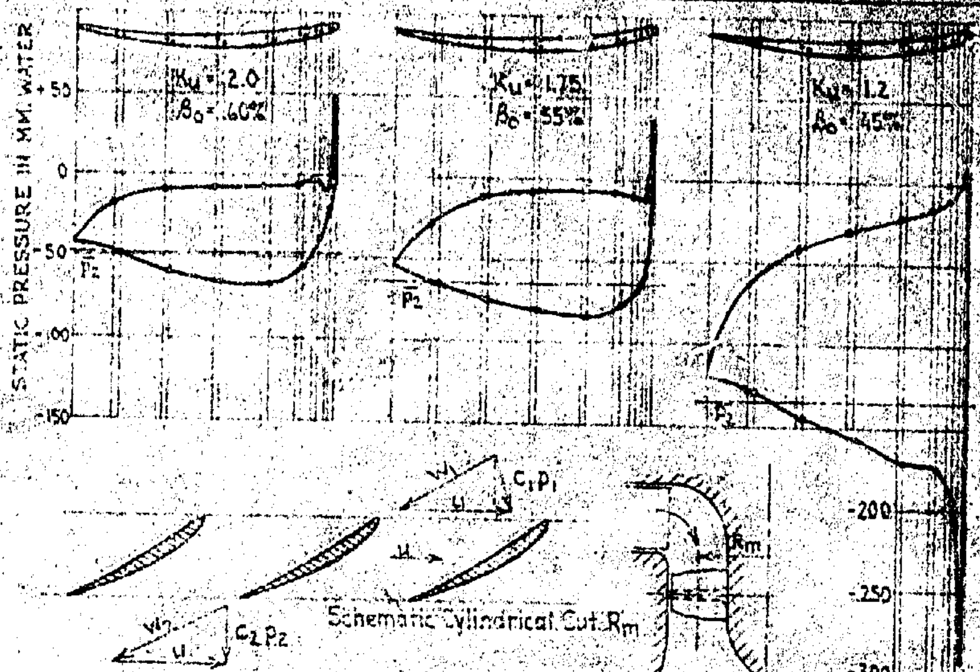


FIG. 9
Variation of pressure on the middle section R_m of a Kaplan turbine vane with change of head; $n = 1700$ r.p.m.

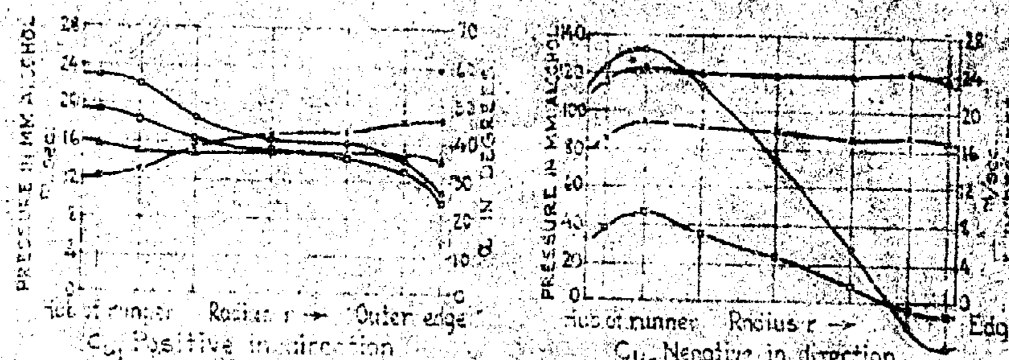
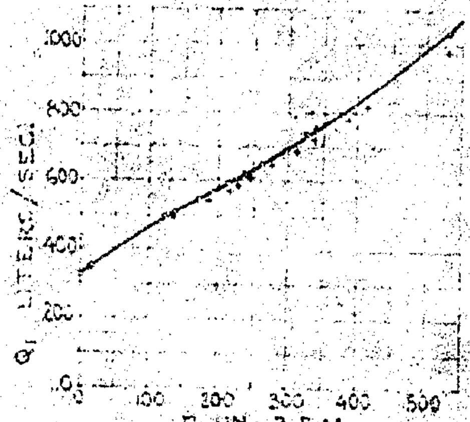


FIG. 10
Measured pressure and velocity distribution on front (left) and back (right) of the runner of a Kaplan turbine



Guide opening $\beta_0 = 70\%$
Runner position $\beta_2 = 0$
FIG. 8
Discharge measurement in water experiment (+) and in air experiment (o)

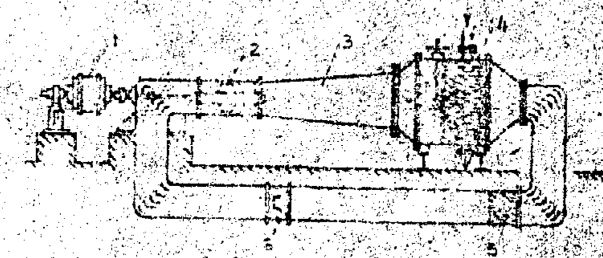


FIG. 11
Design of the Escher Wyss aerodynamic experiment plant for pump research work.