MODERN HYDRAULIC PRIME MOVERS

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Although a considerable number of publications covering the field of hydraulic prime movers are available, the demand for one covering the subject in general and in concentrated form still seems apparent.

In this paper an attempt is made to combine theory and practice in such a manner that they can be readily applied to the problems which are involved in everyday practice.

The paper is divided into five parts as follows:

The first four parts of this paper were originally published in different form in Allis-Chalmers POWER REVIEW (a confidential Allis-Chalmers Mfg. Co. publication): I — July, 1940; II — December, 1941; III — August, 1942; IV — December, 1942.

Hydraulic Turbine Handbook

PRINCIPLES OF DESIGN
•
PRACTICAL PROBLEMS OF SPEED AND PRESSURE REGULATION

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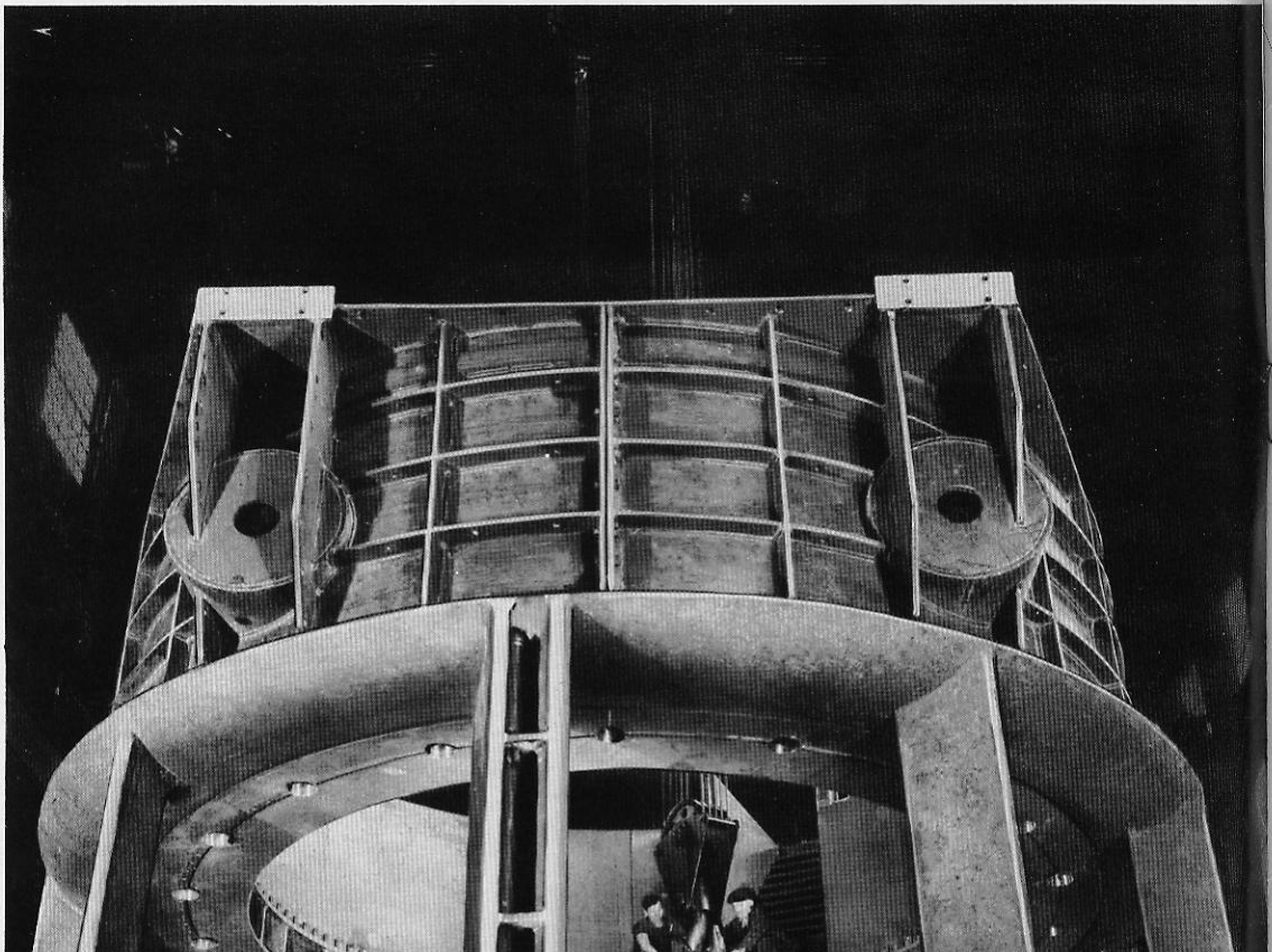


All hydraulic turbines are shop-assembled prior to being sectionalized for shipment. Shown is a 40,000 hp, 120 rpm unit.

Part I

GENERAL CHARACTERISTICS AND TYPES

The three fundamental requirements determining size and type of hydraulic prime mover are:

1. The available head
2. The maximum capacity in horsepower required, or the available flow of water in cubic feet per second
3. The revolutions per minute

To this is to be added as a fourth item the kind of service to be rendered by the unit.

1. THE HEAD

The head is determined by the topographical conditions and is, therefore, a civil engineering matter.

We must distinguish between net head and the so-called *gross head*, which is the difference in elevations of the headwater level at the intake and the tailwater level at the discharge side of the power plant. It is evident that this gross head will vary with changes in the headwater and tailwater levels. Both levels may vary in the same direction so that the gross head is not materially affected, or one may rise faster than the other in which case the gross head changes. Sometimes the power plant is located in a gorge so that during times of flood the tailwater level rises considerably, thereby reducing the available gross head materially.

To determine the best characteristics of the turbine, it is essential to know the maximum gross head (in feet), the normal gross head (normal indicating that gross head for which the turbine operates most of the time), and the minimum gross head (and, if possible, an indication as to how long it lasts) — in other words, the percentages of time of maximum, normal and minimum gross heads. It is especially necessary to know the maximum flood level at the power plant because on it depends the location of the generator floor. Equally important is the minimum tailwater level as this fixes the setting of the turbine which in turn determines the amount of excavation required in the tailrace, or discharge from the turbine.

Net Head

If the turbine were placed directly between two lakes, the upper being the headwater and the lower the tailwater, there would be no losses because of hydraulic friction. The gross head would then be identical to the net head.

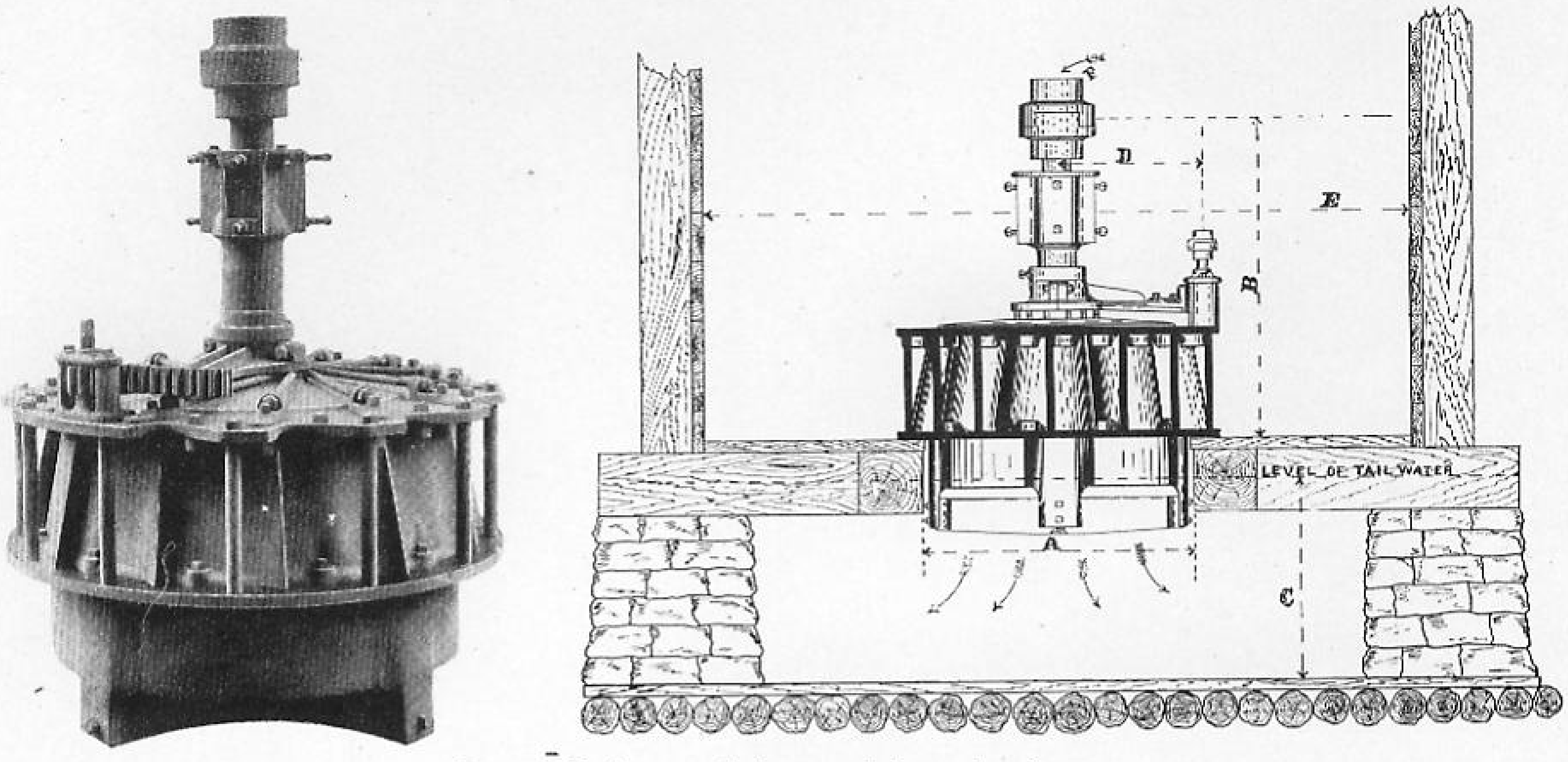
When water is carried in canals and closed conduits, however, a certain amount of the available gross head is lost by friction. This loss depends upon the length of the conduit and the velocity at which the water flows in the conduit. It is the *net head* which determines size and characteristic of the turbine; and it is, therefore, necessary to have given either the net head (maximum, minimum and normal), or else the length and diameter or area of the conduit, so that the friction losses can be computed and deducted from the gross head. Naturally, it is also essential to know how many turbines are to be (1) initially and (2) ultimately supplied by this conduit, since this determines the ultimate velocity of water and resultant friction losses.

These conduits involve problems of pressure variations also because of the kinetic energy contained in the moving column of water; and they thus influence the speed control of a turbine, or the cycle control of an hydro-electric unit, when sudden load changes alter the flow of water through the unit.

Elevation Above Sea Level

In the setting of a turbine, the elevation of the unit above sea level is also important, especially at high altitudes which involve low pressures in the discharge and particularly in units provided with a draft tube.

Many good designs of turbines proper prove defective on account of pitting (cavitation) and vibrations present only because the turbine is placed too high above the tailwater level. The information should, therefore, also indicate this elevation above sea level.



Some old-timers of the stock-in-trade class.

2. MAXIMUM CAPACITY IN HORSEPOWER

With the head determined, the horsepower capacity at once fixes the discharge quantity or cubic feet of water per second. If this quantity is not available, it is useless to install a unit of such capacity.

Unless the unit receives its water supply from a lake, or a spring of constant flow, it is hardly to be expected that the quantity available is constant. It is, therefore, advisable also to know the normal (present the greatest part of the time) flow in cfs, the minimum and the maximum flow. The minimum flow may be considerably less than the normal flow; and, unless more than one unit is installed to absorb the total flow, the one unit may then have to operate at a small so-called part-gate opening. The efficiency and life of the turbine are dependent upon the length of time these conditions prevail.

Since water levels (heads) and water quantities are also topographically linked, it is advisable to know each in relation to the other. Usually when scarcity of water prevails, the available head is highest; and during flood conditions the available head may be decidedly less than normal. This again has a bearing on the selection of the characteristics of the turbine, particularly with low head developments when a relatively small reduction in head results in a considerable loss of output, which, as will be seen later, can be partly recovered.

3. THE REVOLUTIONS PER MINUTE

Since most of the hydro developments serve to produce electrical energy, it is important to know the frequency of that system — 60, 50, 40 or 25 cycles — and also whether the plant may have to serve for two frequencies, as is the case in Southern California, where in certain plants a unit operates on 60 cycles for some hours or days, 50 cycles for others.

With head and capacity, or quantity of water, fixed, the selection of the type of unit is practically determined also and, consequently, the revolutions. It is realized that the cost of a generator varies with the capacity and the revolutions, slow speed machines generally being more expensive than moderately higher speed generators.

A manufacturing concern that designs and builds hydraulic turbines and not generators is inclined to keep the lion's share of investment cost of generating equipment on its own side and naturally tends to offer high speeds on the ground that so much more cost is saved in the generator. In many cases such an argument is fallacious.

The most economical investment or development of a hydro-electric plant is that which produces the highest revenue in kilowatt hours from the available water power, with the greatest revenue per dollar invested. This applies, not only initially, but also over a period of operating years, including the reliability and upkeep costs of the plant. It also applies to the selection of the type of turbine. It would be wrong to consider only one type of unit, for example, if such type would involve additional construction costs in the way of excavation, crane capacity, etc.

The selection of proper revolutions of a turbine should not be taken out of the jurisdiction of the turbine designer; and, furthermore, he should be in the position to give all other factors involved — such as the cost of generator, of auxiliaries, and of foundations not excluding excavation — unbiased consideration.

THE KIND OF SERVICE TO BE RENDERED

This particular point is not always given due consideration. There are two distinctly different kinds of service:

(a) Independent operation.

(b) Operation in conjunction with a large electric power distribution system.

Independent Operation

In so-called isolated plants, whether they are for direct factory supply of energy or for a very small network, the duty imposed upon the unit is often a severe one because any momentary load variations must be fully taken care of by the hydro-electric prime mover; and this throws the duty at once upon the turbine itself and its means of speed control.

Such an isolated unit in itself has a limited flywheel effect of revolving parts; and, if supplying power exclusively to electrical motors, etc. (or more so, if the current is used for non-rotating equipment, such as lights, furnaces, etc.), the total flywheel effect available in all rotating parts may be rather limited. Naturally, a sudden load change of any magnitude will constitute a substantial percentage of the total available energy so that, in the absence of inherent flywheel effect of the generator, the momentary speed drop on sudden increases of load and the momentary speed rise on sudden load rejections may become so excessive that they cause disturbing changes in the speed of the industrial machines driven by the generator.

The writer had occasion about 38 years ago to examine some linen tablecloth woven in a mill driven by a turbine which was speed-controlled by a man operating the butterfly valve that supplied water to the turbine. The tablecloth distinctly showed stretches where the weave was very tight and again others which were very loose, caused entirely by excessive speed variation. The installation of a governor so improved the product of the linen mill that the cost of the governor was off-set in less than two years by the higher price obtained with the improved quality of the tablecloth.

The most severe test that can be imposed upon a hydro-electric unit and its governor is the so-called water rheostat load. Here the load can be changed as suddenly as may ever be the case in actual commercial or industrial service. At the same time the flywheel effect of rotating parts is restricted entirely to that inherent in the unit itself.

Basis for Guarantees

Since the test conditions at a plant under commercial load are not definite, we, as a rule, give our speed regulation guarantees based on water rheostat load, remarking that better results may be expected in actual service because of the following factors:

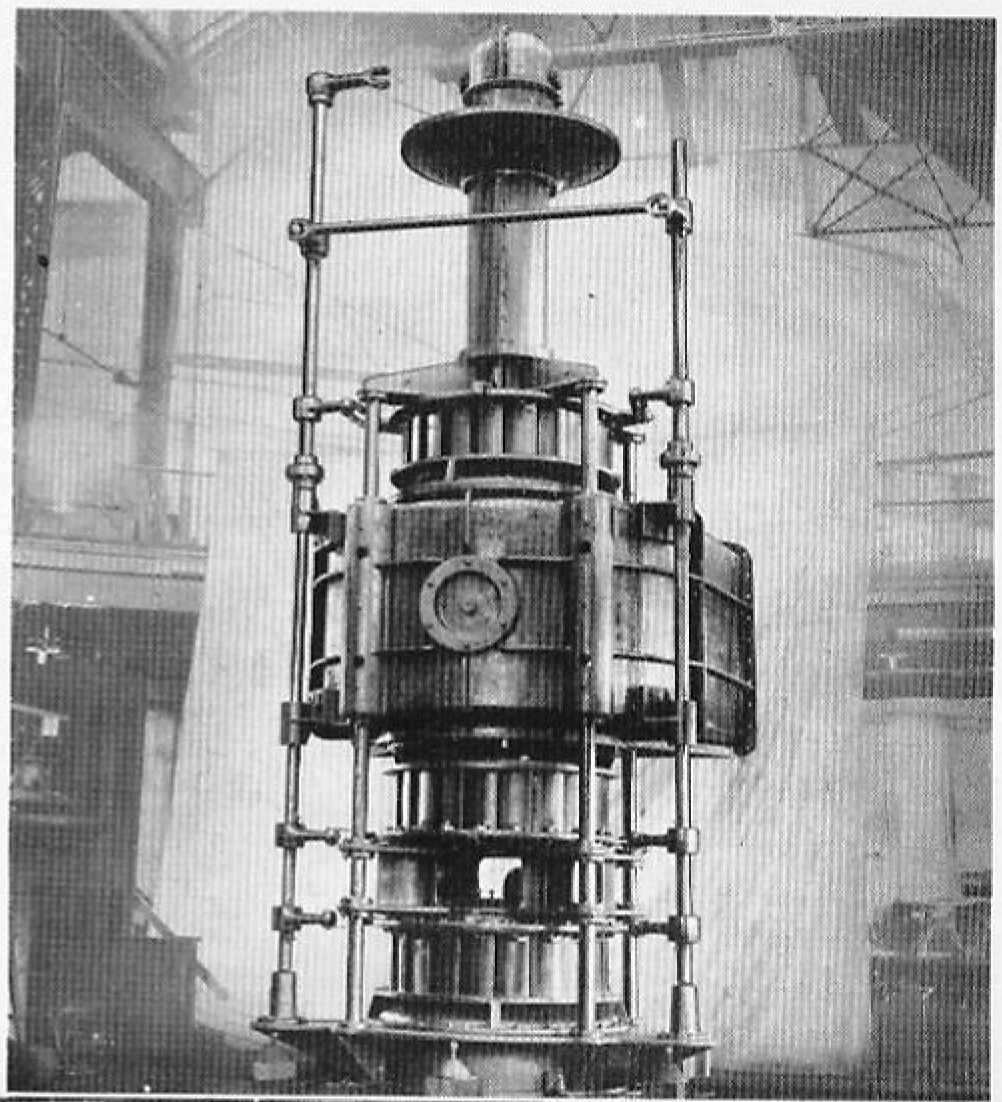
1. There may be additional flywheel effect (above that of the unit proper) available because of rotating masses operating in synchronism.
2. The load change may not be as sudden as those produced with water rheostat load.

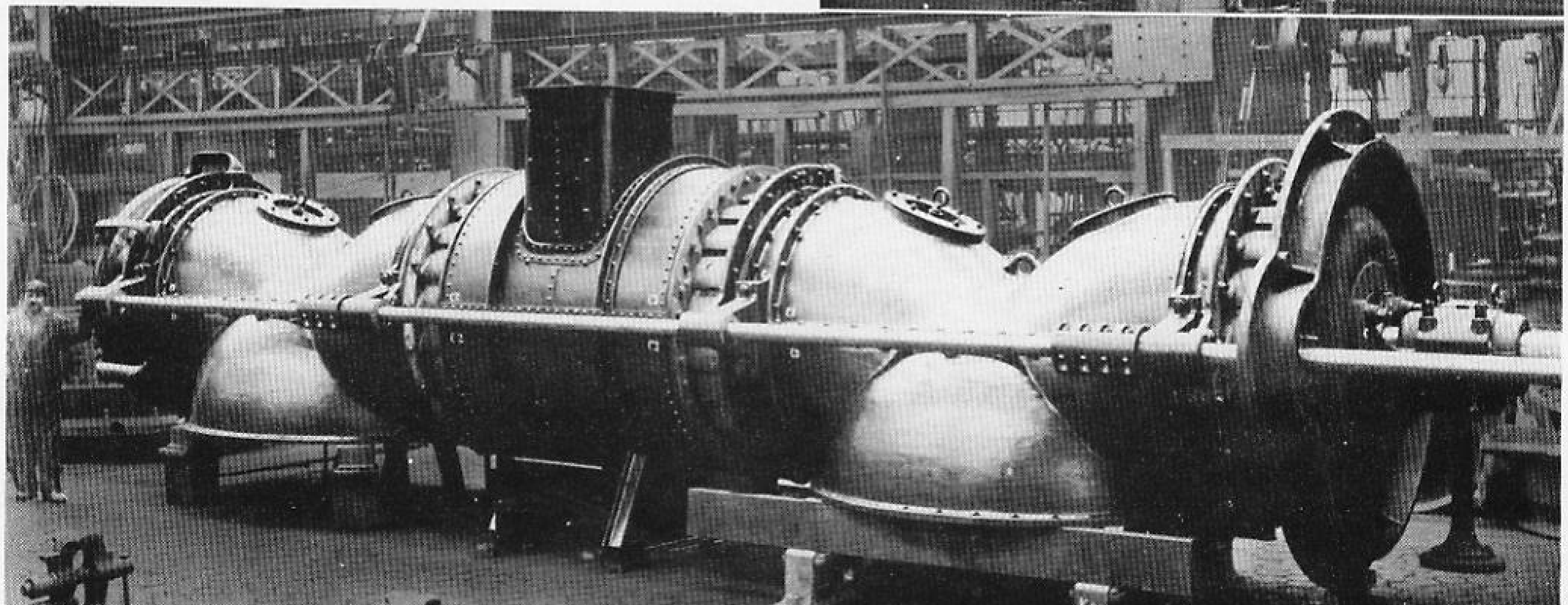
The severest plant test would be that of pulling the switch when the unit operates at full load. This would be equivalent to a full load water rheostat rejection. However, it would hardly be permissible to do this in practice, since all industrial equipment depending on the electric energy supply from the unit would be stopped.

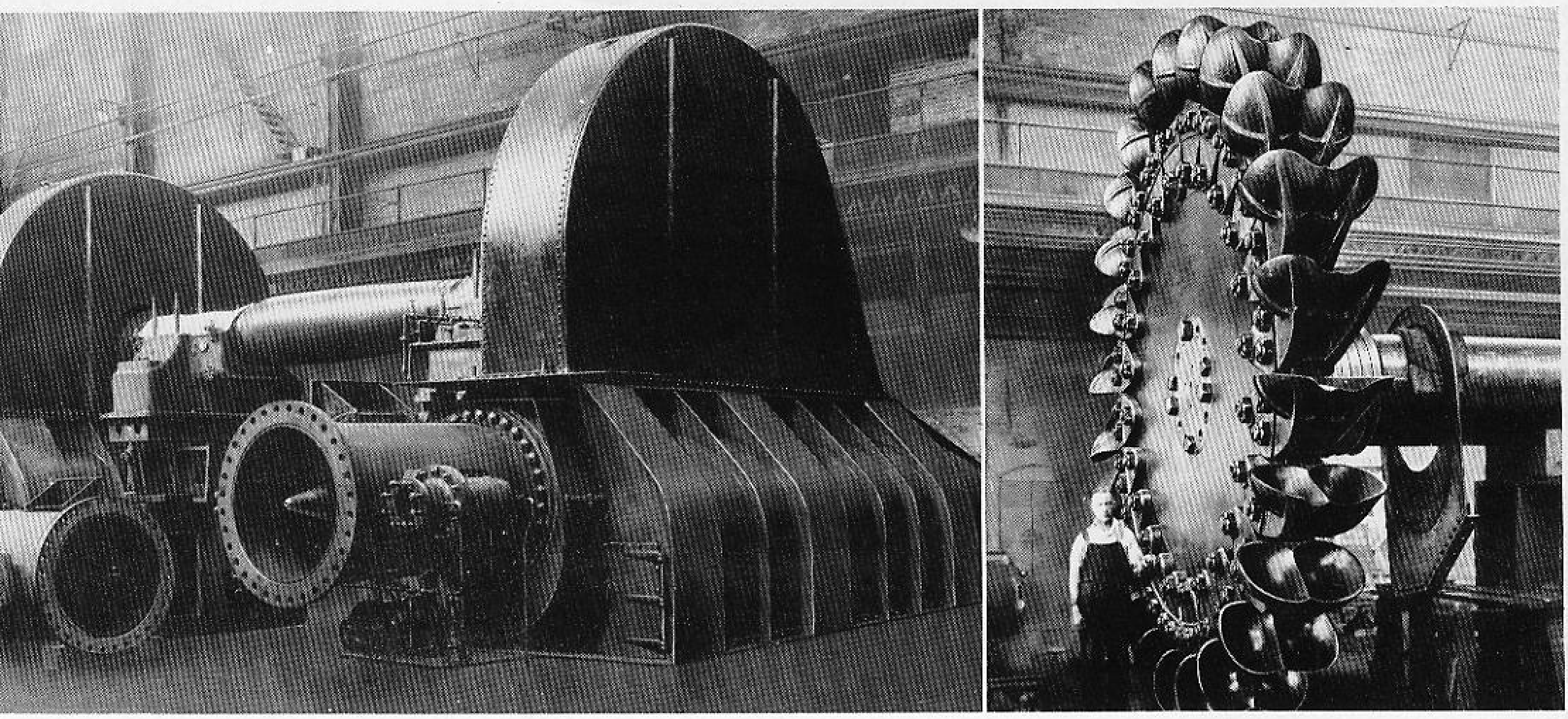
It is advisable to prescribe in the inquiry for a turbine what "maximum momentary" speed rise and speed drop on sudden "full" load rejections and load increases, respectively, are acceptable, bearing in mind, however, that on sudden load increases the speed drop will always be more severe than on equal sudden load rejections. This difference is due to the necessity of accelerating the water into the turbine on load increases (except in the rare case of a high head plant, with a so-called water wasting control, which does not affect the water velocity in the conduit).

It is evident that complicated conduits also enter into the problem in relation to the requisite amount of flywheel effect of the unit, often not only from the point of view of maximum tolerable momentary speed changes under rapidly fluctuating loads, but also from the point of view of stability; i.e., holding the "normal" speed within certain limits even under constant load (hunting or creeping of frequency or of power).

AT RIGHT: A survivor of the practice of multiple vertical shaft runner arrangement — a 900 hp, 100 rpm, 16 ft head, triplex open flume turbine. BELOW: A modernized horizontal shaft quadruplex for a Wisconsin power plant, rated 600 hp at 180 rpm.





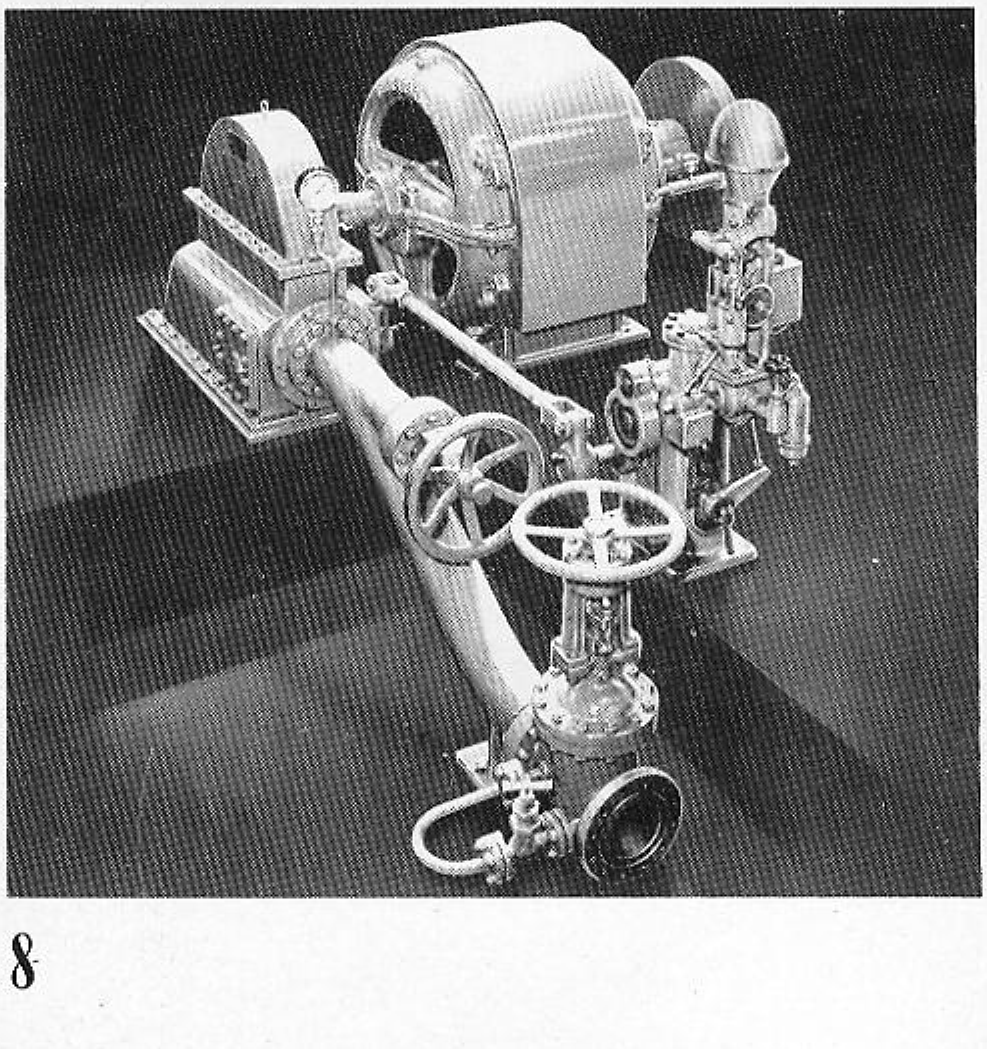


The largest (physical size) impulse type turbine in the world, installed in 1928 at the San Francisquito Plant No. 1 of the City of Los Angeles. The double overhung unit is rated 32,000 hp at 143 rpm under 800 ft head.

Operating in Large Electric Power Distribution System

In a large electric power distribution system the flywheel effect of all revolving parts is considerable so that sudden speed changes hardly can occur. At the same time the sudden load changes constitute a relatively small percentage of the total output of the operating prime movers of that system. In other words, the commercial load changes are small and gradual so that each prime mover turbine is called upon to perform only moderate speed control, except when suddenly detached from the system. In that case, however, it is only a matter of mechanical safety at overspeed; and, since a hydro-electric unit is now built to stand full runaway speed (maximum speed at no-load with turbine gates wide open), there is no problem involved, provided that excitation attributable to this over-speed does not produce excessive primary voltages in the generator. It is, therefore, not necessary here to provide a liberal flywheel effect. The unit can be synchronized either by hand or by the load limiting device on the governor, which prevents hunting of the unit before it is tied into the power system.

One of the smaller high head wheels — a 350 hp, 312 kva unit for the hydro plant of the Champion Sillimanite Corp.



PRINCIPAL TYPES OF HYDRAULIC PRIME MOVERS

When Allis-Chalmers entered the field or hydro-electrical prime mover construction 36 years ago, it found that in our country this art was developed entirely along the line of "cut-and-try" and on a "stock-in-trade" selling basis. Each manufacturer had its own type of turbine advertised by a trade name, such as Hercules, Giant, Samson, etc., and catalogued from a model tested under most favorable conditions of setting at the then well known hydraulic testing flume at Holyoke, Massachusetts. See page 18.

Revolutions per minute, cubic feet discharge, and horsepower were tabulated for various standard sizes under various heads; and the units were sold according to catalogue rating. Naturally, when the actual setting at the plant deviated from that of the model test, the plant results also deviated materially and often disappointingly; but no redress was possible for the purchaser.

This Company's policy was from the start to design and build turbines to best suit existing operating conditions.

Today the stock-in-trade practice has practically disappeared. Even the Holyoke Testing Flume has ceased its activity, owing to the fact that the few surviving and leading turbine manufacturers now have their own testing laboratories in which models homologous to the required size are tested in a setting exactly duplicating that under which the final unit will have to operate. Thus results as to field performance can be reliably determined in advance.

Today the entire field of hydro development can be covered adequately with three fundamental types:

1. For high heads, the impulse wheel usually called Pelton wheel, in honor of its originator, Lester A. Pelton, of California.

2. For medium high to medium low heads, the reaction turbine, called Francis type, in honor of its inventor, James B. Francis, the originator of the Holyoke Testing Flume.
3. The propeller type, called Nagler and Kaplan types, in honor of the originators, Forrest Nagler, associated with Allis-Chalmers from 1908-1929, Chief Engineer of Canadian Allis-Chalmers, and since 1942 Chief Mechanical Engineer of Allis-Chalmers, and the late Dr. Victor Kaplan, of Bruenn, Austria-Moravia.

There is, of course, no pronounced line of demarcation which separates the three types as to application with reference to the head.

The Pelton Type

The Pelton type can be applied for heads as high as are practicable to utilize commercially. The highest head so far developed in this country is around 2500 feet (in California). For the lower limits of head the horsepower capacity determines the speed, and it can be readily seen that this may become so low that it would involve prohibitive costs of the generator. For example, a 250 hp wheel at 400 ft head could be coupled directly to a 600 rpm generator; whereas a 5,000 hp wheel would permit only 133 rpm, or about 180 rpm if two instead of one jet per unit were used.

The application of multiple jets complicates the equipment, requiring greater maintenance and repair costs. About 25 years ago, when the Francis type turbine entered the field of higher heads and thus penetrated the field of application held by the Pelton type, attempts were made to match Francis turbine speeds with multiple jet Pelton (or Doble) wheels. For the larger unit we recommend the Francis type with about 500 rpm, provided that the water does not contain an abnormal quantity of silt or mineral acids.

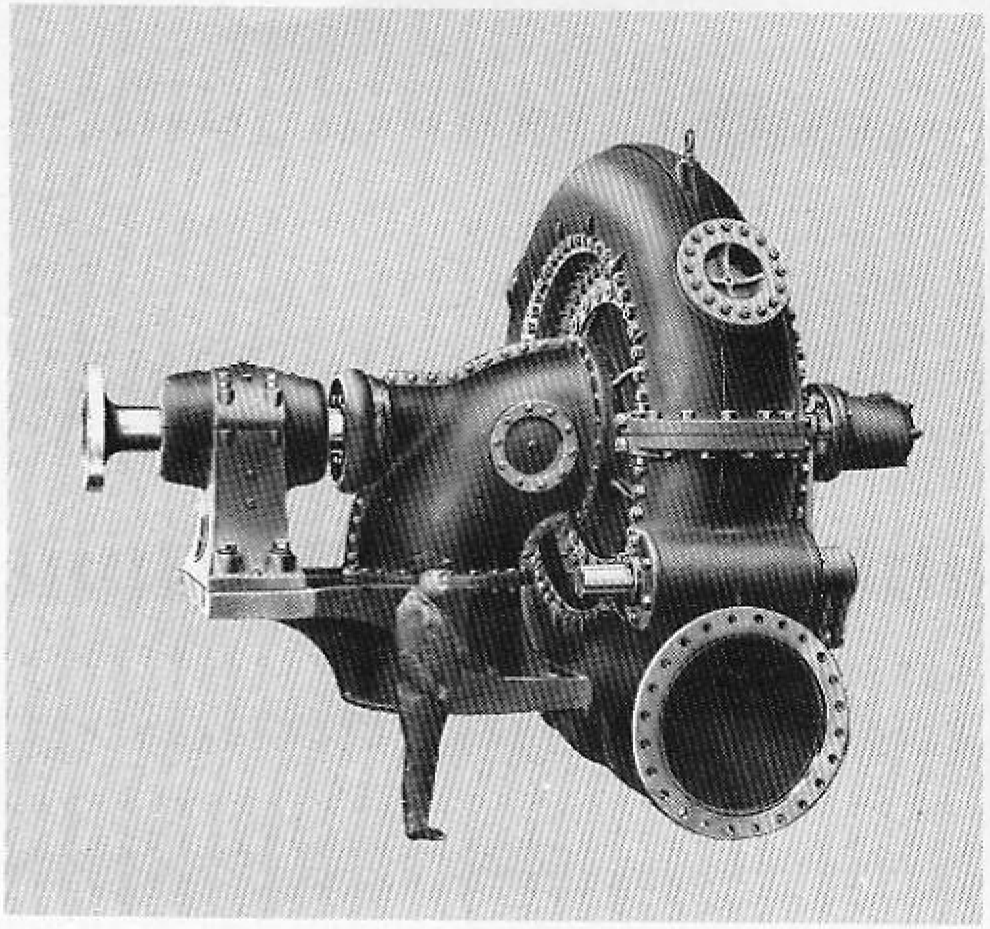
A sad case remains an historical fact, where a six-jet wheel of 2500 kw normal rating not only failed to deliver any kw output with all six jets discharging full, but even required several hundred kw from the power system to keep in parallel. This occured because so much water was crowded into the wheel housing that it filled up completely, as would be the case if the tailwater level rose about the shaft of the unit.

With the Pelton type of wheel no draft tube is used. The head acting upon the wheel is, therefore, measured directly at the centerline of the jet, and any difference between that elevation and the lowest tailwater level is not utilized. It can be seen at once that this type is not economical when the water level rises materially at times of flood because in that case the entire wheel must be placed safely above this highest water level to prevent the buckets' dipping into the tailwater, causing the wheel housing to partially fill up with the results stated above.

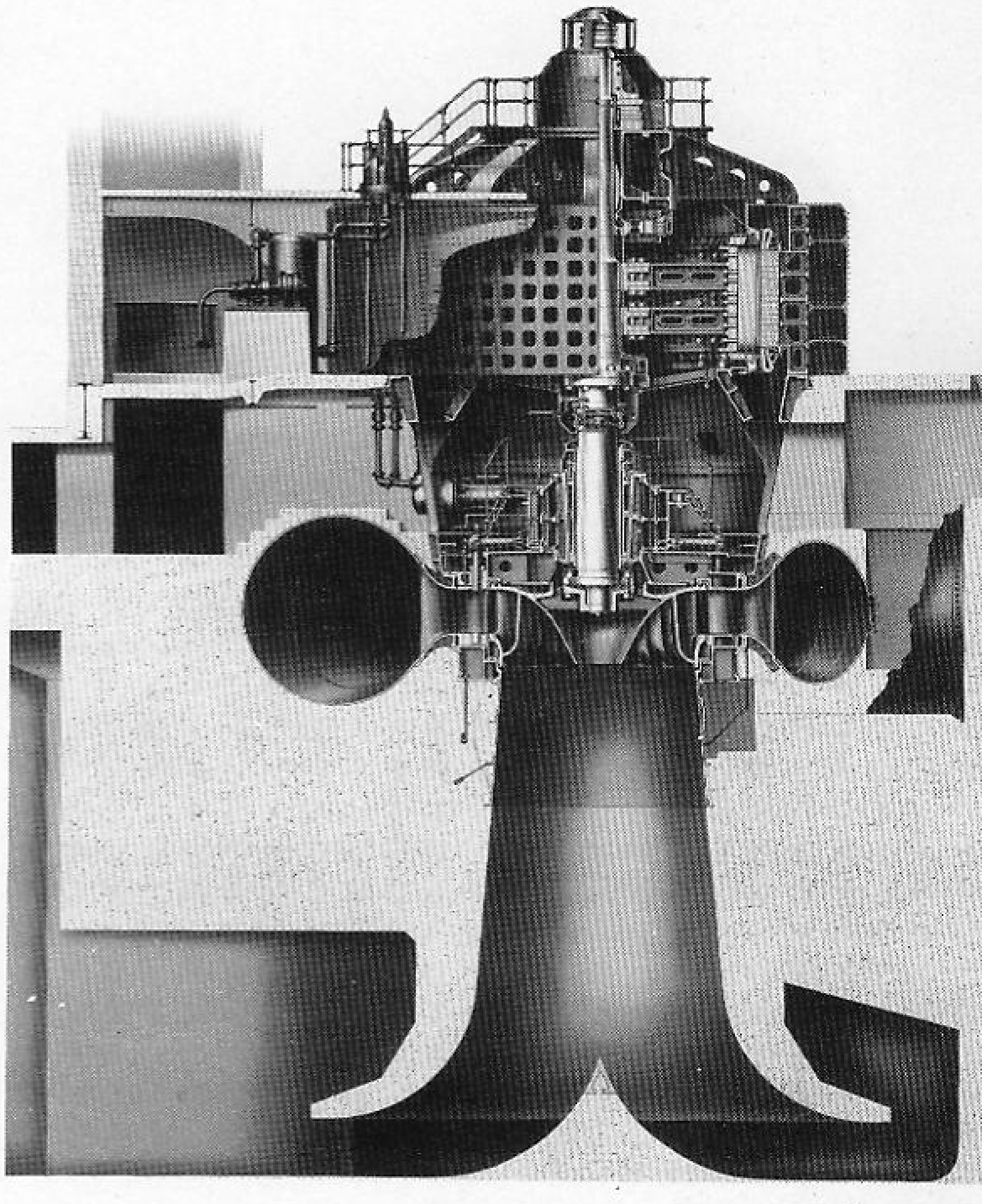
The Francis Type

As to practical applicability, the Francis type covers the widest range of head. Units at 1100 ft head are in commercial operation, and for a speed of 720 rpm such a unit would have to have a capacity of about 23,000 hp or about 15,000 hp for 900 rpm. It can be readily seen that for smaller horsepower capacities the Francis type would require rather high generator speeds so that the Pelton type would again become more adaptable with about 300 rpm for 15,000 hp using one jet only, or about 450 rpm if two jets per unit were provided.

While the upper range of head for the Francis type involves higher speed limits of the generator, the lower range in turn involves the lower practical limits of generator speed. It is hardly practical to allow generator speeds lower than 60 rpm, and this only when large kw capacities are involved. It would require a head of about 36 ft for a capacity of 16,000 hp. This is a head which is more economically developed by the third, or propeller, type as will be seen later.



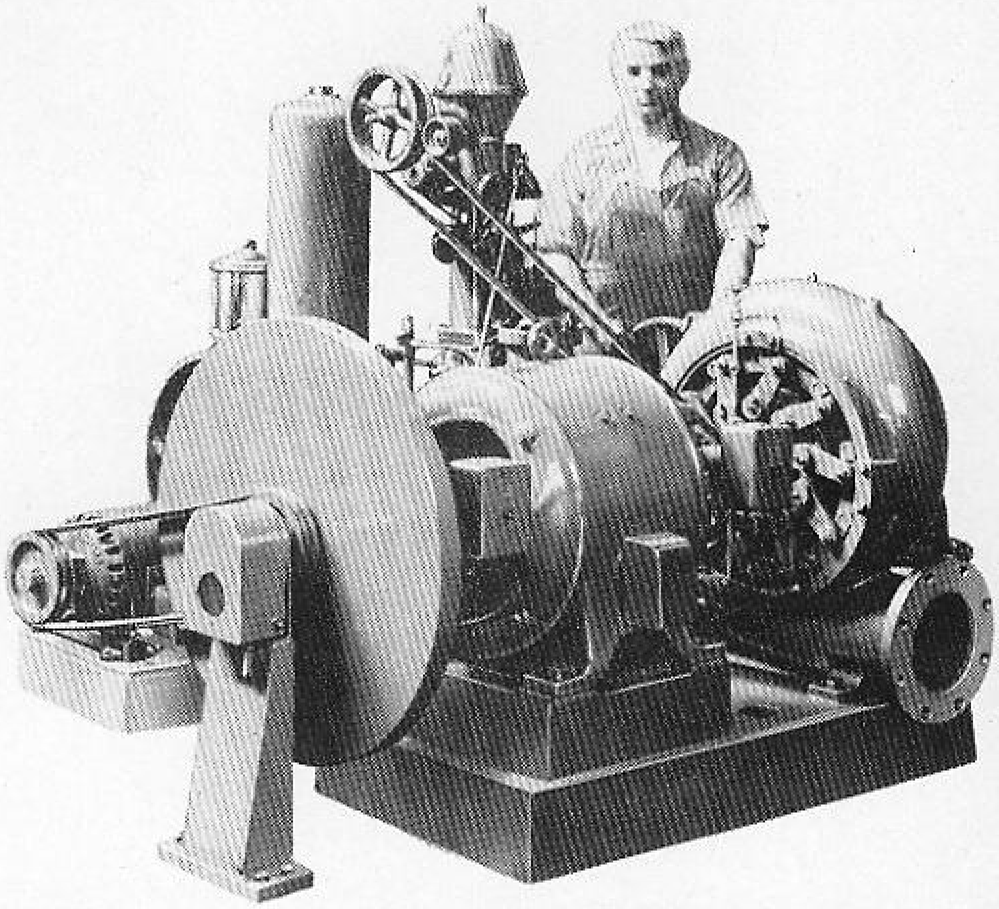
ABOVE: A pioneer of the Francis type, this 10,000 hp, 550 ft head, 400 rpm turbine is still running at the Centerville plant of the California Gas and Electric Corp. When installed in 1907, it almost doubled the head of any previous installation. BELOW: A world's record for modern, medium head, Francis turbines — the 70,000 hp, 213 ft head unit for Niagara Falls Power Co.



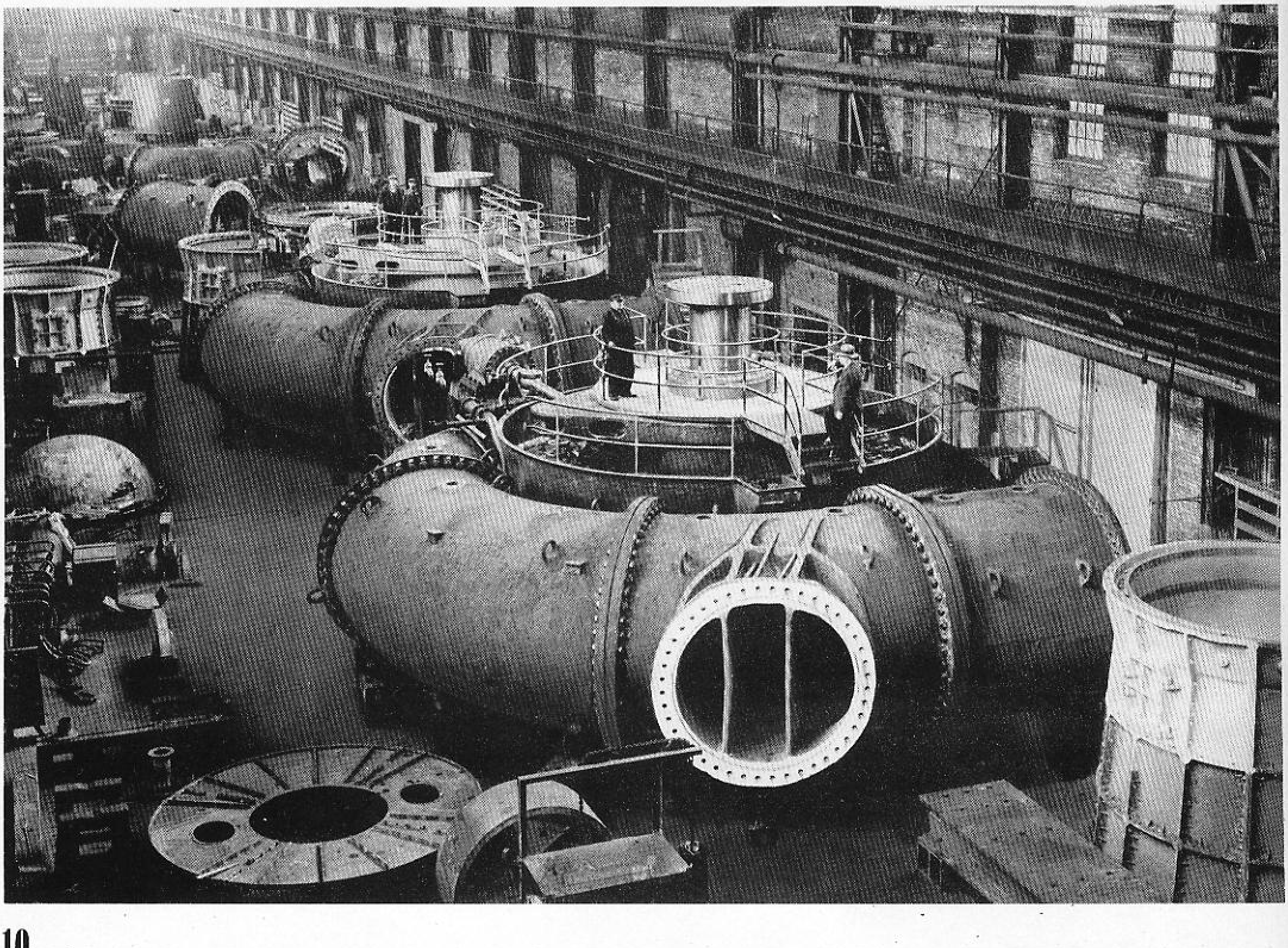
With the Francis type the water does not discharge free from an orifice, as is the case with the Pelton type, but uses a draft tube submerged in the lowest tailwater level. Thus a portion of the head, otherwise lost with a Pelton wheel, is utilized here to some extent. The water discharges from a Francis turbine in a full column, filling the entire cross area of the draft tube. Since the outlet area of the draft tube exceeds that below the immediate runner discharge, the water is decelerated so that a part of its kinetic energy is regained, thereby adding to the head acting on the turbine; and the distance between the runner and the tailwater level is also added to the head in the nature of a suction effect.

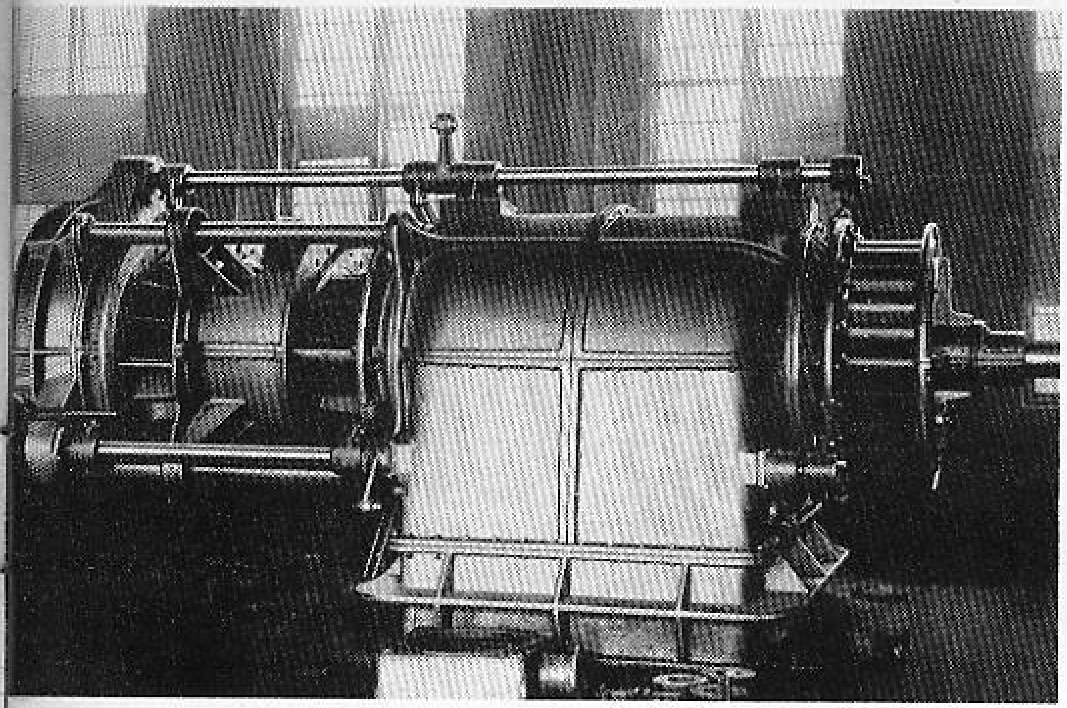
Thus three items — suction head, regain in kinetic energy, and velocity head at the discharge side of the runner — produce a combined negative pressure, or total suction head. This must not exceed a certain amount; otherwise cavitation, commonly called pitting, will appear in some portion of the discharge of the turbine, especially the runner. The elevation above sea level is also of importance because the distance between lowest tailwater level and turbine runner must be reduced as the elevation above sea level increases.

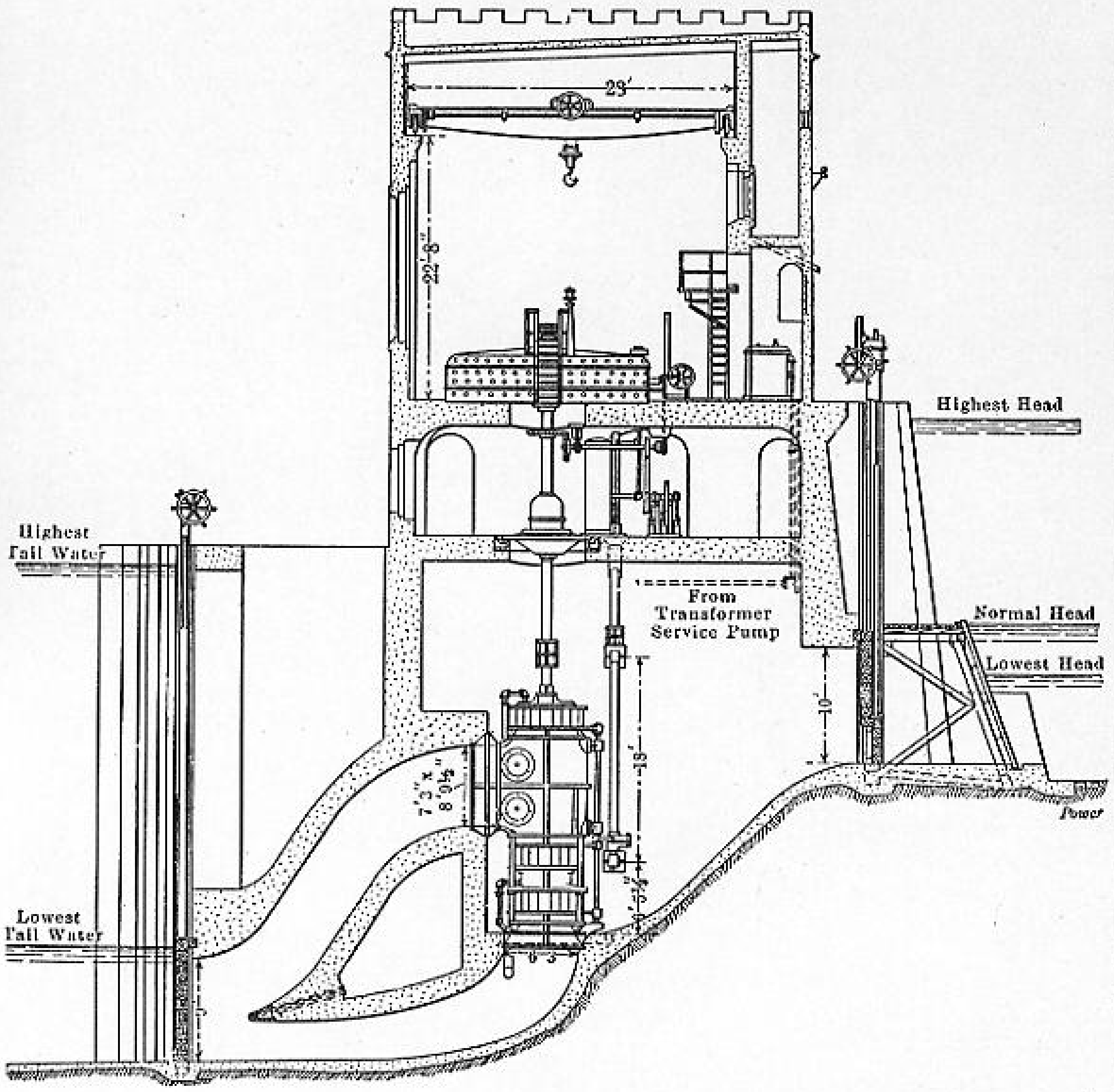
In the older development of the art the speeds of Francis turbines were boosted for economical generator cost by applying one or more runners per turbine. With a vertical shaft setting, as many as three runners were placed on one shaft; and, in horizontal setting, even so-called Octuplex (eight-runner) units were used. It is needless to remark that this involved complexities of machinery; and the efficiency, aside from excessive mechanical friction losses, was naturally rather poor.



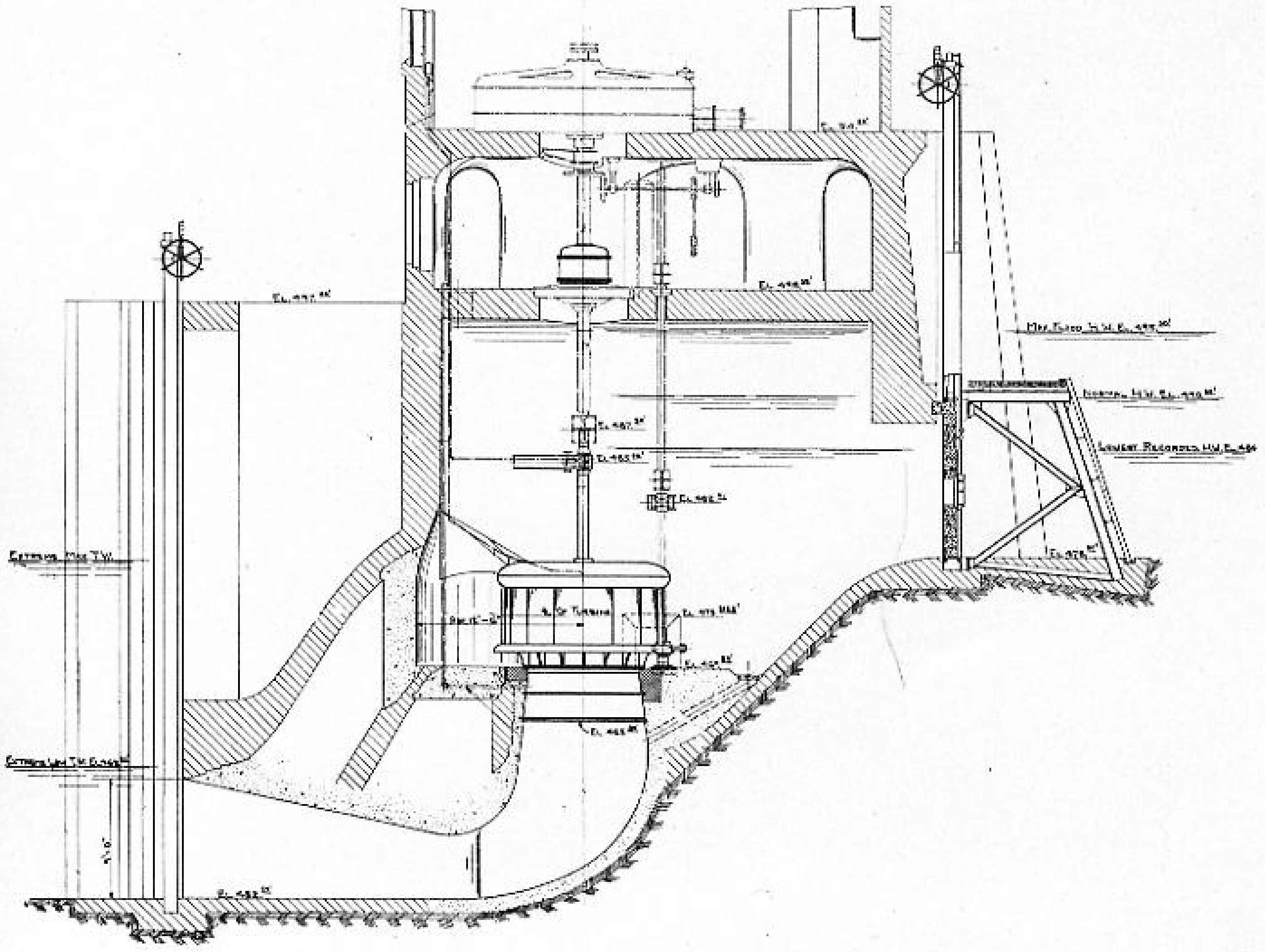
ABOVE: One of the smallest Francis type units—32 hp at 1800 rpm under 68 ft head. In contrast (BELOW), two of the six 115,000 hp, 180 rpm, 475 ft head, Allis-Chalmers turbines for Boulder Dam — a world's power record for high head Francis type.

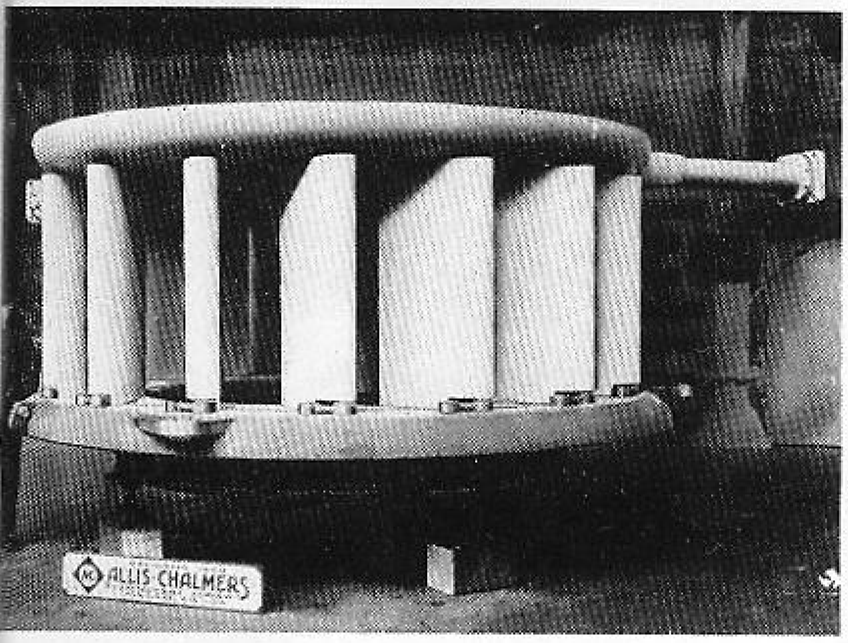






An illustration of what can be accomplished with the propeller (Kaplan) type by modernizing existing plants. The vertical triplex turbine (ABOVE), built in 1909, was replaced by the single runner propeller type (BELOW). The single propeller delivers more power under 14 ft head than did the three Francis runners under 18 ft head at the same speed.





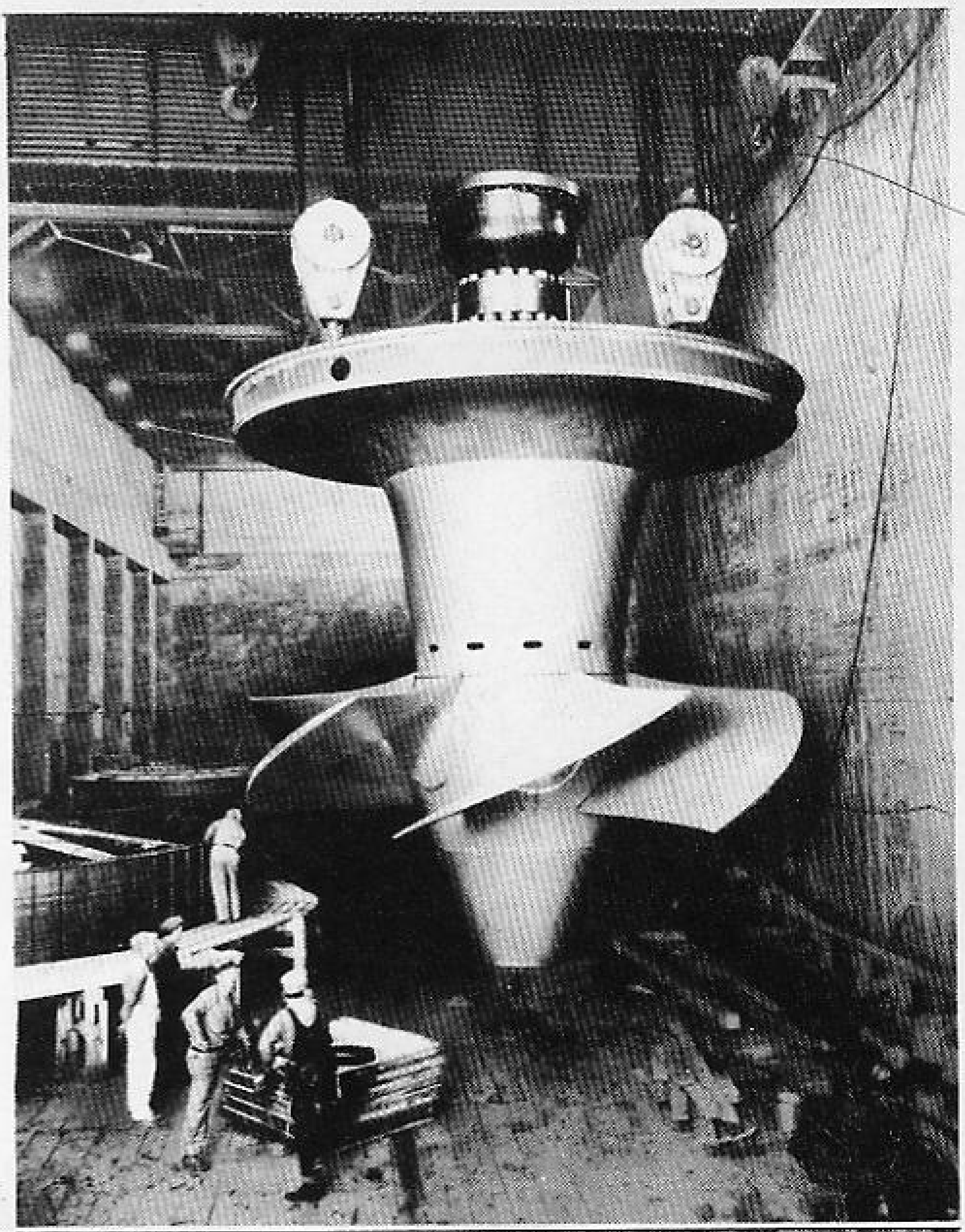
The Nagler and Kaplan Types

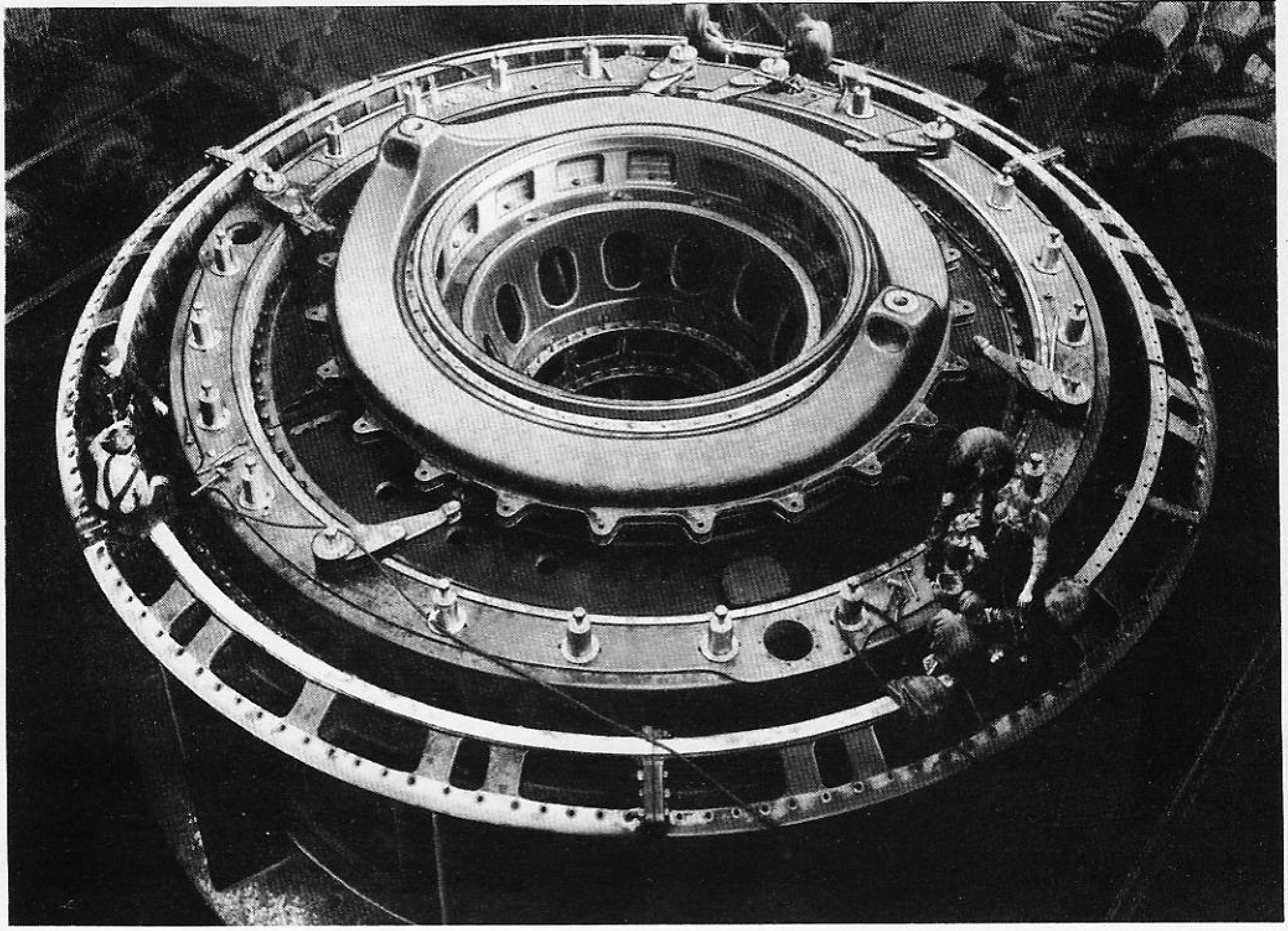
The Nagler and Kaplan types permit speeds not attainable with the Francis type and, particularly for very low heads, practically double the speed can be obtained, or almost four times the horsepower capacity for the speed obtainable with the Francis type. This type is the latest addition to the art and, because of its hydraulic peculiarities, must not be applied unless all factors entering into the development are carefully investigated.

At the outset it must be realized that a propeller runner involves high velocities of water. Any disturbance in the flow, not only through the propeller itself, but also leading to it and away from it, has an even more detrimental effect than is the case with the Francis type; and the setting above lowest tailwater level is even more important here than with the Francis type. Many failures are on record which are entirely chargeable to disregard, or lack of knowledge, of these requirements.

Referring to the example cited under the previous heading, of a 16,000 hp, 36 ft head, 60 rpm Francis type unit, the speed of a propeller type unit under the same head and horsepower capacity and at an elevation above sea level not exceeding about 500 feet could be about 100 instead of 60 rpm.

The largest Kaplan type turbines in this country — the 55,000 hp units for the Pickwick Landing Station of the Tennessee Valley Authority. A shop assembly view of one of the first two turbines is shown below, while at right is a field photograph of the assembled runner and cover plate. A third unit for this plant is now under way in the shops.





Part II

THEORETICAL CHARACTERISTICS

To understand clearly the fundamental nature of the three types of water turbines — Pelton, Francis, and Propeller (Nagler and Kaplan) — it is necessary to know the theoretical principle of operation of each type.

Common with all three types is the process of utilizing the potential energy available in water due to pressure (p), i.e., difference (H) between head and tail-water levels. This is accomplished by transforming this potential energy either entirely or partly into kinetic energy — moving water — which is transmitted to the runner of the turbine in the form of a torque on the shaft of the unit.

THE PELTON TYPE

By referring to Figs. 1 and 1a of this article, it can be seen that the water brought down under the head (H_1) is discharged through an orifice formed by a stationary nozzle tip (2) and a movable needle (3) for controlling the flow of water. At the orifice of the nozzle tip (X) practically all of the potential energy of the water is transformed into kinetic energy. When water drops through a head or distance (H), it reaches a theoretical velocity $c = \sqrt{2gH}$, called the spouting velocity, (g) being the acceleration due to gravity (32.2), (H) the head in feet, and (c) in feet per second. The water forming the jet has a velocity of slightly less than (c), varying from (0.98 to 0.92) x (c), depending on the quality of design of the orifice.

Since all the potential energy or pressure (p) is transformed into velocity, the jet itself has no internal over-pressure since otherwise it would spread instead of holding together solidly for a considerable distance as is the case with good designs. If a piezometer (P) is inserted into the jet at (X), it indicates practically atmospheric pressure. The velocity (V_1) of the jet can be determined with a Pitot tube inserted into the jet at (X) (Fig. 1b).

This jet is now directed upon the buckets of the wheel and is deflected inwardly, sideways, and outwardly, thereby producing a force or impact upon the bucket surface resulting in a torque moment on the shaft. The exertion of the water jet upon the surfaces of the bowl of the buckets causes the water to be decelerated from its velocity (V_1) down to an exit velocity (V_2), which, of course, varies over the discharge edge of each half of bowl, being highest at the inner portion and less at the two sides and at the outer portion. The flow of water sideways is shown on Fig. 1a.

Naturally when the water leaves the bucket, it has no more "driving effect" upon it. Therefore, whatever kinetic energy is still contained in the discharging water by reason of its exit velocity (V_2) is irretrievably lost. Hence, it is essential in the design of a good bucket to hold the exit velocities as low as possible without causing too small exit areas since this would hinder the water from its undisturbed exit and would result in reactions which counteract the forward torque moment. It is also important to produce as high a jet velocity $V_1 = V_1^* \text{ x } c$ as possible (where V_1^* = Velocity Coefficient), because the bucket cannot give up more energy to the shaft than it receives from the jet.

This can be clearly seen by the nature of the formula for efficiency.

The available potential energy is:

$\frac{Mc^2}{2}$; $c = \sqrt{2gH}$, $M = \frac{W}{g} = \frac{Q\gamma}{g}$ where W is its weight in lb, Q is the quantity in cu ft per sec, γ is the weight of one cubic foot of water (62.4 lb) and g is 32.2 gravity. Thus we have: $\frac{Mc^2}{2} = \frac{Q\gamma 2gH}{2g} = Q\gamma H$ in ft lb per sec; or:

$\frac{Q\gamma H}{550}$ in hp (N) represents the available imput. The actual output is:

Imput x efficiency, so that $N = \frac{QHe}{8.8}$ as a general formula for calculating horsepower output for a given quantity (Q) under a net head (H), and efficiency (e).

Hydraulic Efficiency

The hydraulic efficiency (e_h) of a hydraulic prime mover of any kind is the ratio of energy delivered to the drive shaft and energy received. Thus we have: $e_h = \frac{(M/2)\ (c^2 - V_3^2)}{(M/2)\ c^2} = 1 - \left(\frac{V_3}{c}\right)^2$ where $\frac{V_3}{c}$ or V_3^* represents the coefficient of discharge velocity of the prime mover.

A number of factors affect this hydraulic efficiency because of losses in the various parts of the prime mover. For the Impulse or Pelton type we thus have:

Fig. 1

(GIRARD) PELTON TYPE

ACTION

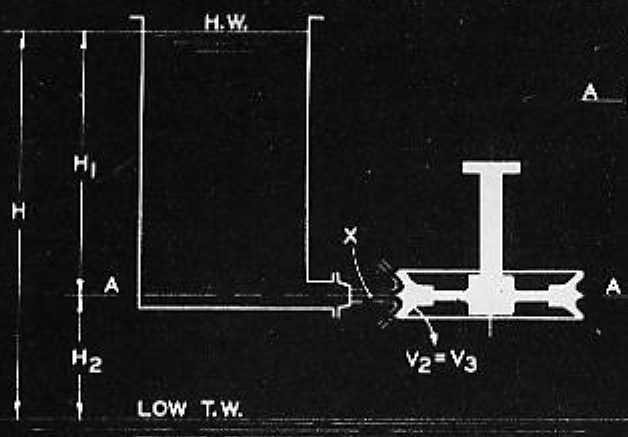


Fig. 1b

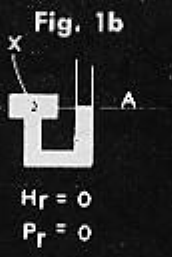


Fig. 2

(JONVAL) FRANCIS TYPE

REACTION

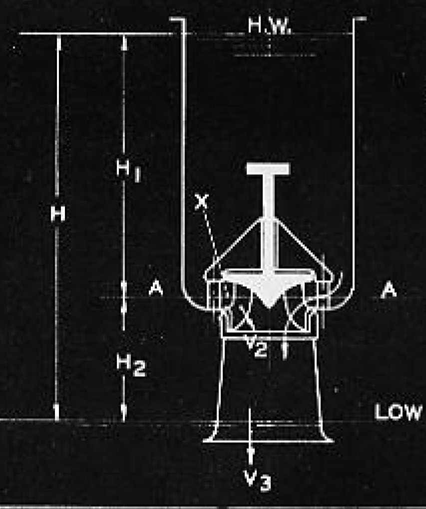


Fig. 2c

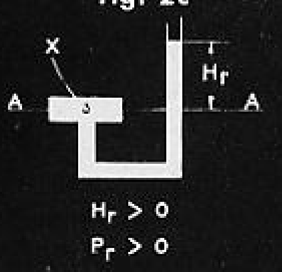


Fig. 3

PROPELLER TYPE (NAGLER KAPLAN)

SUCTION JET

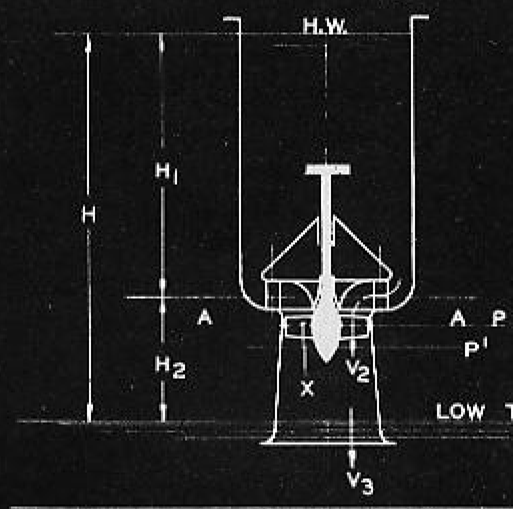


Fig. 3c

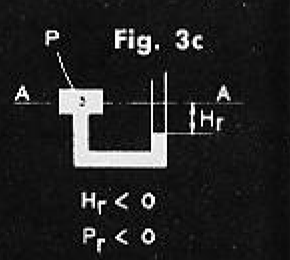


Fig. 2a

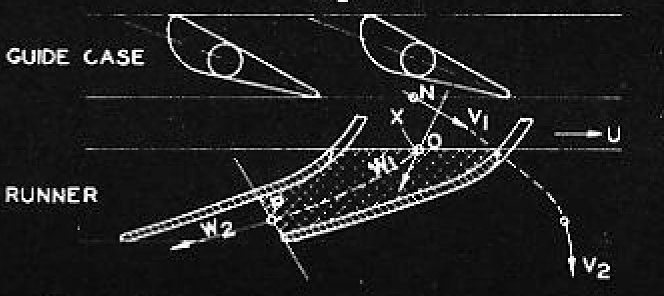


Fig. 3a

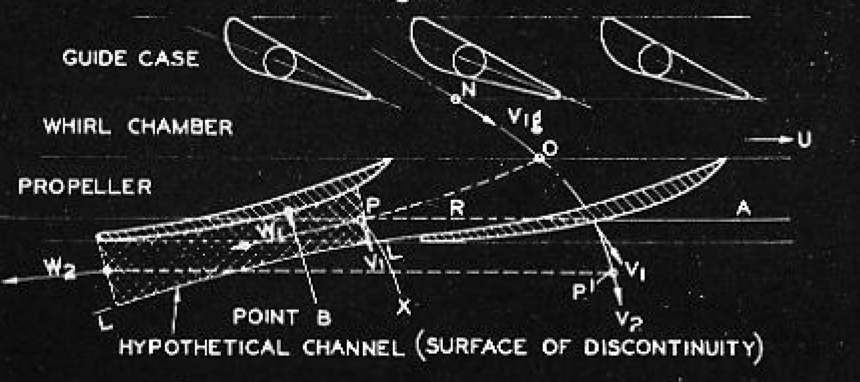
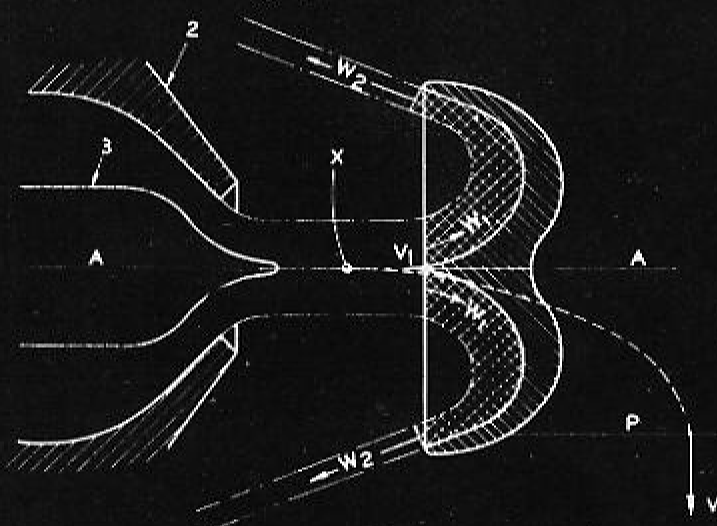


Fig. 1a



NO REGAIN; V2 = V3

Fig. 2b

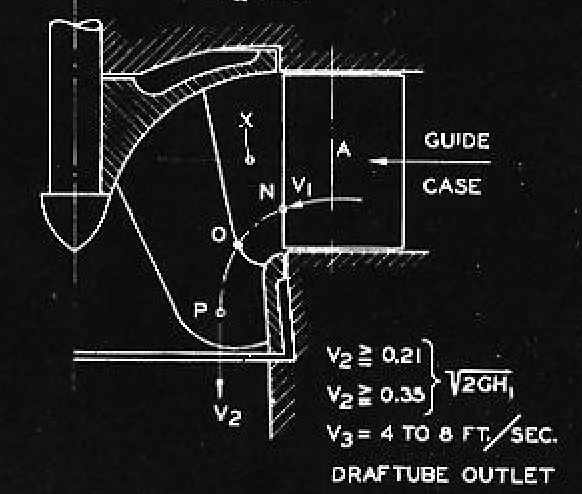
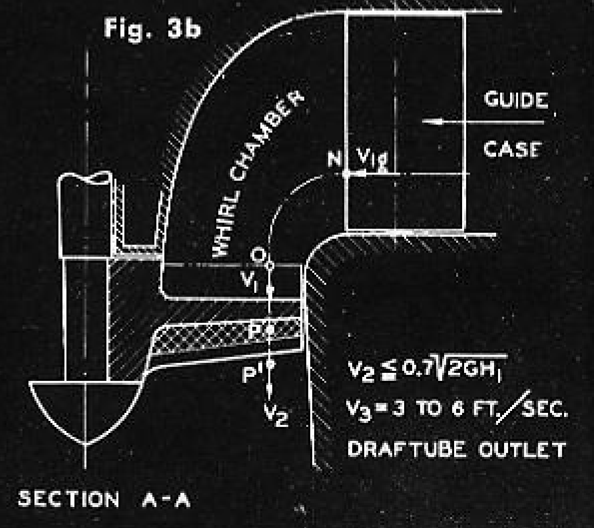


Fig. 3b



SECTION A-A

(1) Efficiency up to the nozzle pipe.
(2) Efficiency of the jet.
(3) Efficiency of the bucket itself.

Combining items (1) and (2) we have: $e_j = \frac{(M/2)\,V_1^2}{(M/2)\,c^2} = \left(\frac{V_1}{c}\right)^2$ or $(V_1^*)^2$, the square of the coefficient V_1^* of the velocity of the jet.

(3) *The efficiency of the bucket.* The hydraulic efficiency of the bucket proper (e_b) is the ratio of input energy to the bucket — outlet from the bucket, to total imput to the bucket. The jet delivers an energy of $\frac{MV_1^2}{2}$ to the bucket.

$e_b = \frac{(M/2)\,(V_1^2 - V_2^2)}{(M/2)\,V_1^2}$ and since $V_1 = V_1^*c$ and $V_2 = V_2^*c$ we have:

$e_b = \frac{(V_1^*)^2c^2 - (V_2^*)^2c^2}{(V_1^*)^2c^2} = 1 - \left(\frac{V_2^*}{V_1^*}\right)^2$. Here $(V_1^*)^2$ represents directly the efficiency of the jet, and $(V_2^*)^2$ the square of the coefficient of exit velocity of water from the buckets. It can at once be seen that a poor jet with a low coefficient $(V_1^*)^2$ is a primary cause of a low wheel efficiency. Likewise high exit velocity coefficients (V_2^*) result in lower efficiencies. A value of V_1^* of 0.98 can be considered a very high quality nozzle. Thus the bucket cannot have more than $(0.98)^2$ or 96.04% efficiency. A minimum over-all value of V_2^* is 0.10 to 0.15; thus e_h, the hydraulic efficiency of the Pelton wheel equals: $e_h = 96.04 - (1 \text{ to } 2.25\%)$ or 95.04 to 93.79%. From this must be deducted windage and friction of the rotating parts in the amount of at least 2% and also losses due to hydraulic disturbances, amounting to at least 1%.

It can be seen that the best possible efficiency attainable with a Pelton wheel can never exceed 92 to 90%; and these values become still lower when, because of a relatively high speed or high ratio of wheel diameter to jet diameter, the values of V_2^* become materially higher than 0.15. *Any guarantees exceeding these values are to be branded as spurious.*

Since no kinetic exit energy can here be further regained, it can be seen at once that the use of a draft tube to regain head is of no value, except that it slightly increases the jet velocity $V_1 = V_1^* \sqrt{2g(H_1 + H_2)}$, where H_2 (Fig. 1) represents the draft head. Usually H_2 is a negligible portion of H_1, and, therefore, the gain is negligible; whereas with a lower value of V_2^* the gain could be more appreciable. It is, however, not possible to catch the water discharging directly from the buckets in a closed conduit for the purpose of further extracting its kinetic energy by deceleration.

Effect of Elevation Negligible

Since we are dealing here with atmospheric pressure at the jet and in the wheel housing surrounding the buckets, the elevation above sea level on such a wheel has no effect, except that all the velocities (V) are also a function of gravity (g) because $c = \sqrt{2gH_1}$. It is obvious, however, that g does not vary materially. The hydraulic process of transforming the potential energy into kinetic energy, therefore, ends at the orifice (x). Therefore, the actual static head of such a unit is measured as the difference from the head water level at the intake to the pressure pipe, to the centerline of the orifice of the jet, and not to the water level in the wheel chamber. $H = H_1 + (H_2 = 0)$.

Any variation in the tailwater level, which level must by all means remain a certain safe distance below the lowest point of the bucket, does not affect the output of the unit. From this it can be realized that in cases where the tailwater level varies materially the Impulse type is not economical because it sacrifices this entire variable head below the buckets. Our Company has valid United States Patents disclosing how variable tailwater levels, even to an extent that would submerge the bucket wheel, could be utilized. However, only in cases where this flood condition occurs very frequently would the additional costs involved be justified.

At present we are shipping a 30,000 hp unit to the new Glenville plant in North Carolina of the Nantahala Power and Light Company (Aluminum Company of America). At rare occasions extreme floods occur when the tailwater level rises as much as 15 feet above the power house floor. Instead of sacrificing this head all year around, the wheel is set for normal tailwater level conditions, and provision is made with flood gates to close off the outlet and keep the unit shut down during the short periods of exceptional floods.

THE FRANCIS TYPE

By referring to Fig. 2 of this article, it can be seen that the water brought down to the turbine under the head (H) is first discharged through a series of stationary orifices, commonly called the guide case (or distributor). In this case not all the potential pressure or head (H_1) is transformed into kinetic energy, but only a portion of it depending on the characteristic of the turbine, as will be seen later. That part not transformed into kinetic energy remains as an over-pressure at x between the guide case and the runner and is called the reaction pressure (p_r) (or over-pressure), which also gives the Francis type the theoretical name of reaction or (over-pressure) turbine.

It follows at once that both the guide case and the runner must be housed-in; i.e., not exposed to atmospheric pressure as is necessary with the Pelton type. This at once necessitates close running clearances between stationary and rotating parts of the turbine, involving a mechanically more delicate mechanism than is the case with a Pelton wheel. Since the water discharges from the runner not atmospherically but into a draft tube, the entire flow through the turbine takes place in a closed conduit and presents pressures changing from positive to negative value.

Partial Efficiencies

This latter characteristic, then, at once puts this type under the influence of barometric pressure, being affected to some extent by the setting above sea level and primarily by the setting of the runner above lowest prevailing tailwater level. Thus a number of additional factors not involved in a Pelton wheel enter the problem, viz:

1. Efficiency up to the guide case.
2. Efficiency of guide case and of reaction process between guide case and runner.
3. Efficiency of runner proper.
4. Efficiency of draft tube.

While each individual value cannot be fixed theoretically in numerical value, a picture can at least be obtained by representative equations of the process. To obtain a simple picture of the process of flow through a Francis turbine, assume that it is of a purely axial discharge so that the diameter along a streamline of flowing water remains constant.

The available head (H) is equivalent again to the spouting velocity $c = \sqrt{2gH}$ at the entrance to the turbine. Within the turbine leading to the guide case certain hydraulic losses take place so that the "net" velocity (c_n) at the guide case can be related to the available velocity (c) by the equation:

$c_n^2 = c^2 - c_w^2$, where c_w represent lost velocity.

(1). *Thus the efficiency of the turbine up to the guide case proper can be written again as the ratio of output energy to imput energy.*

$e_h = \frac{(M/2)\, c_n^2}{(M/2)\, c^2} = \frac{c^2 - c_w^2}{c^2} = 1 - \left(\frac{c_w{}^2}{c}\right)$; and, since again $c_w = c_w^* \text{ x } c$, we have $e_h = 1 - (c_w^*)^2$, similar to the efficiency up to the nozzle of the Impulse wheel. The relative loss (c_w) to the guide case may be made up of friction loss and losses due to eddy currents, called turbulent flow. It points toward the necessity of having smooth surfaces exposed to the flow of water and keeping changes in direction of flow and abrupt changes in velocity to a minimum. It shows at once the importance of careful design to lead the water properly through racks, head gates, penstocks, casings or flumes to the turbine proper. Many sad failures to deliver the contracted output of a plant in the older days were attributable to neglect of this requirement. To use a popular expression, "It is just as important to deliver water properly f.o.b. turbine runner and tailrace outlet as it is to design the turbine itself correctly."

(2). *Efficiency of the guide case and of reaction process between guide case and runner.* The water passes through the guide case leaving it at a velocity (V_1) and under a reaction pressure (p_r) between guide case and runner. This pressure (p_r) can be represented by a reaction velocity (c_r), produced by a reaction head (a piezometer value (H_r) at point (x) — Fig. 2, to 2c): $H_r = \frac{p_r}{0.433}$, so that $c_r = \sqrt{2gH_r}$. Thus: $c_n^2 = V_1^2 + c_r^2$.

In the case of the Pelton wheel, $c_r = 0$ (no reaction pressure), or $c_n = V_1$ of the jet issuing from the orifice.

This reaction pressure (p_r) causes an acceleration of the relative water velocity (w_1) at the entrance into the runner to a value (w_2) at the discharge, or $c_r^2 = w_2^2 - w_1^2$, indicating that the water discharges from the runner between its vanes at a higher relative velocity than it enters. $w_2 > w_1$ (therefore, $c_r > 0$). From this it can be seen at once that the greater the reaction pressure (p_r) of the specific type of Francis turbine, the lower will be the velocity (V_1) through the outlet of the guide case, requiring a larger guide case area than in the case where $c_r = 0$. Likewise, the larger the value c_r, the greater is the increase in relative velocity from w_1 to w_2 — i.e., what may be termed the "speedity" of the Francis runner — and also the quantity of water discharged into the draft tube per diameter.

The absolute discharge velocity (V_2) is a function of the relative outlet velocity (w_2) and the peripheral speed (u) of the runner at that point. If u denotes the peripheral speed, $u = \frac{d\pi n}{60}$, where u is in ft per sec, $\pi = 3.1416$, d in ft, and n in rpm, or $w_2^2 + V_2^2 = u^2$, as per Fig. 4, where the discharge V_2 is at right angles to u; i.e., where the discharge is straight downward without any forward or backward whirl as indicated in dash-dot lines in Fig. 4.

β_2 represents the discharge angle of the runner vanes and $\alpha_2 = 90°$, the angle at which the water discharges into the draft tube. Likewise, Fig. 4a shows the inlet diagram, V_1 being the absolute velocity at which the water leaves the guide case and w_1 the relative velocity at which the water enters the runner passages, α_1 the guide case angle, β_1, the runner vane entrance angle. The values u, V_1, V_2 w_1, w_2, etc. are absolute velocities in ft per sec applying to the respective head (H) under which the turbine operates. Obviously by dividing these velocities by $c = \sqrt{2gH}$, there is obtained coefficients u^*, V_1^*, V_2^*, w_1^*, w_2^*, etc., applicable to any head and thus characteristic values of design. Diagrams drawn up on such basis are, therefore, typical for the respective type and applicable to that range of heads (H) for which this type is suited.

The efficiency of the process at the end of the guide case and of the runner entrance can be written as:

$$e_p = \frac{(M/2)\,(V_1^2 + c_r^2)}{(M/2)\,c_n^2} \text{ or } \frac{V_1^2}{c_n^2} + \frac{c_r^2}{c_n^2}, \text{ and, since } c_r^2 = 2gH_r,$$

where H_r is the piezometer head at (x) on line A-A in Fig. 2c:

$$e_r = \left(\frac{V_1}{c_n}\right)^2 + \frac{2gH_r}{c_n^2}.$$

This efficiency, it can be seen, is composed of two parts: first, that involving velocity $\left(\frac{V_1}{c_n}\right)^2$, and, second, that involving the overpressure between the outlet of guide case and the entrance to the runner. The second factor involves an overpressure and shows that leakage in the clearance space between stationary parts of guide case and revolving parts of runner directly reduces the pressure (p_r) and thus affects the efficiency. It is this factor which is an addition over those involved with an Impulse wheel because in that case $p_r = 0$. (See Fig. 1).

Reaction Process in Runner

From relation $c_r^2 = w_2^2 - w_1^2$ it can be seen that the value c_r is dependent on the conditions prevailing in the runner, by the flow of the water from runner entrance (w_1) to runner outlet (w_2), again introducing another complication over that prevailing with an Impulse wheel, where $c_r = 0$ and $w_2 = w_1$. Again we have the efficiency of this particular portion of process within the runner:

$$e_w = \frac{(M/2)\,(w_2^2 - w_1^2)}{(M/2)\,c_r^2} \text{ or } \left(\frac{w^2}{c_r}\right)^2 - \left(\frac{w_1}{c_r}\right)^2,$$ which again embraces two values:

1. Friction loss reducing w_1 and w_2.
2. Correct relation between w_1 and w_2 by reason of incorrect runner channel area through which the water flows at the relative velocities changing from w_1 to w_2.

Draft Tube Effect

The absolute velocity of water at the runner discharge is V_2, assuring discharge parallel to the axis of the turbine, as previously illustrated in Fig. 4. The kinetic exit energy still contained in the discharging water is $\frac{M}{2} \times V_2^2$. If the water were discharging free from the runner, all this exit energy would be lost just as it is with the Impulse wheel. By applying a draft tube, the flow of water in an inclosed conduit is extended until it leaves the tube at the velocity (V_3). Thus a regain of head (H_d) is attained:

$$H_d = \frac{V_2^2 - V_3^2}{2g} e_d,$$ in which e_d represents the efficiency of the draft tube. Naturally the deceleration from V_2 to V_3 cannot be sudden in a flaring but short draft tube but requires a certain length of it; otherwise the water separates from the outer walls of the draft tube and discharges at high velocity using only a part section of the outlet of the draft tube and, therefore, causing eddies which naturally lower the efficiency (e_d) of the draft tube. If the water did discharge free from the runner at the velocity (V_2), there would, of course, be no suction or negative pressure below the runner.

Factors Affecting Design

The factors affecting pitting, aside from a faulty design of runner itself are:

1. Elevation above sea level — B (barometric pressure).
2. Vapor pressure, which increases with higher and higher temperatures of water.
3. Distance (H_s) of runner above tailwater level.
4. Outlet velocity (V_2) of water from runner.
5. Regain in draft tube due to deceleration of velocity from V_2 at runner discharge to V_3 at draft tube outlet.

If these factors are not properly considered, cavitation or pitting of the runner, and possibly portions of the upper part of draft tube, are inevitable; and usually pitting is also accompanied by vibrations of rotating parts. It is, therefore, essential as a basis for proper design-selection to know all the various factors involved, such as:

1. Elevation above sea level.
2. Temperature and nature of water (whether it contains gases or mineral acids, etc.)
3. Variations of tailwater levels and of corresponding head water levels.
4. Character of load to be carried (whether unit is to operate at fractional heads for any length of time.)

It can be readily seen that the nature of these various factors involved, not fully realized by the early art, precludes the use of standard stock-in-trade turbines and fully justifies the policy adopted by this Department at its inception in 1904 to build turbines to suit the operating conditions in each individual case.

THE PROPELLER (NAGLER AND KAPLAN) TYPE

This type is the latest of the three and, because of its hydraulic peculiarities, requires the most care in its commercial application.

Figure 3 shows that the Propeller type also operates in an entirely closed conduit from intake to tailrace. As is the case with the Francis type, it has a guide case in which the potential pressure or head (H_1) is also only partly transformed into a kinetic energy, the remaining portion being an overpressure between guide case and runner. The first fundamental difference between the Francis and the Propeller type is that the propeller runner is an axial discharge runner, within which the water is not turned around as is the case with a Francis type where the water enters almost radially inwardly and is turned downward within the runner.

The "Whirl-Chamber"

A considerable space is provided between the radial (centripetal-flow) guide case of a propeller turbine and the axial flow propeller. In this space, commonly termed the "whirl-chamber", two processes of flow and pressure conditions characteristic of this type take place under normal operating conditions:

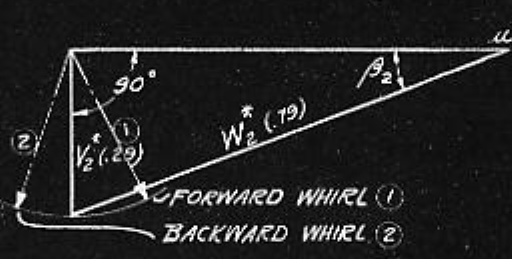


Fig. 4 (Typical values only)

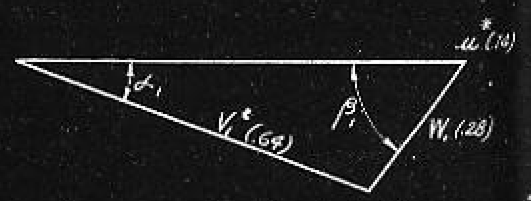


Fig. 4a (Typical values only)

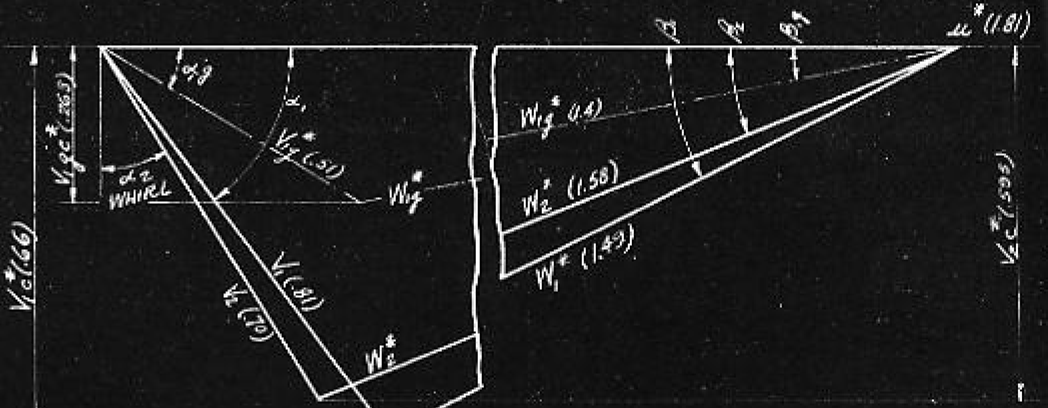


Fig. 5 (Typical values only)

1. The water velocity is greatly increased from the discharge end of the guide case (guide vane tips) to the entrance (runner vane tips) of the propeller, indicating that per diameter the propeller can handle greater quantities of water than the Francis runner.

2. As the water issues radially inwardly (centripetally) from the guide case, wherein it forms a vortex, this vortex is somewhat reduced, the water flowing to the runner at a steeper angle than is the case with the Francis type. This is shown in Fig. 5, where V_{1g} indicates the water velocity through the guide vane openings, V_1 the whirl velocity at entrance to the propeller, producing the relative entrance velocity (w_1) into propeller as against (w_{1g}) which would be the relative entrance velocity into a Francis runner. α_1 and β_1 are steep angles in case of the propeller compared to α_{1g} and β_{1g} of the Francis type.

In all these types, Figs. 1, 2, and 3, the cross-hatched portion in the respective bucket, runner and propeller indicates the flowing water. In the Francis type, Fig. 2, the water is confined to closed channels formed by four walls, the front and rear surfaces of the runner vanes of the runner band and of the runner crown. With the propeller, Figs. 3a and 3b, the channel is not actually closed, especially if the discharge end of the blade does not overlap the preceding blade. The cross-hatched part is located underneath the propeller blade so that the channel is formed only by three material walls — the rear surface of the propeller blade, the surface of the hub and the stationary discharge ring. The fourth wall is open, similar to that in Fig. 1a, where the water discharges along the curvature of the bucket bowl.

"Surface of Discontinuity"

Stroboscopic pictures of water flowing as indicated in Fig. 3b show a distinct dividing line L-L, which indicates a pronounced change from high water velocity to moderate velocity; and this dividing line (really a surface) is termed "surface of discontinuity." Its shape depends upon the flow conditions brought about by the relation of rear propeller vane surface and front surface at end of preceding vane and also by the speed of propeller and operating head. When this surface retains a steady condition, it indicates an orderly flow; and, when it curls and flutters, it indicates a disturbed flow. It is noted that this stream of water shows a contraction downstream of its upper part, followed by gradual enlargement. This is an indication of a diffuser effect, resulting in a lower relative velocity (w_2) at the end of the stream than at the contraction where w_1 is a maximum. This diffuser action is similar to that taking place in a draft tube resulting in a regain.

$h_d = \frac{V_2^2 - V_3^2}{2g}$ x e_d. (See Fig. 3c.)

In this case of the process of flow between each propeller vane, we have $h_{dw} = \frac{w_2^2 - w_1^2}{2g}$ x e_{dw}, producing what may be termed an additional or "hydro-dynamic" regain taking place immediately below propeller vanes proper. Since $w_1 > w_2$, the value $h_{dw} < 0$ (negative), producing at B on the propeller blade surface an additional negative pressure and locating the maximum danger of cavitation directly at that point on the propeller blade, similar to the conventionally termed "Point B" in the design of airplane propellers.

This additional phenomena, not present with the Francis type, shifts the danger point of cavitation even closer and narrows the limits of setting of a propeller turbine. Since all velocities are direct functions of the available head (H), it follows that the negative value $\frac{w_2^2 - w_1^2}{2g}$ or $\sqrt{2gH}\left(\frac{(w_2^*)^2 - (w_1^*)^2}{2g}\right)$ can become a prohibitively negative amount in itself. This can be offset only by placing the runner at correspondingly lower and lower elevations relative to the tailwater level; i.e., by making the suction head (h_s) negative. It is evident that the cost-saving of a propeller type unit, brought about by the higher generator speeds than those of a Francis type unit, is offset by the increased cost of excavation for casing, draft tube, and tailrace.

It is, therefore, advisable to investigate each individual case from the point of view not only of the cost of turbine but also of the total cost, including generator and powerhouse construction. Our Company, designing and building the complete unit, is in a unique position to recommend the most economical solution.

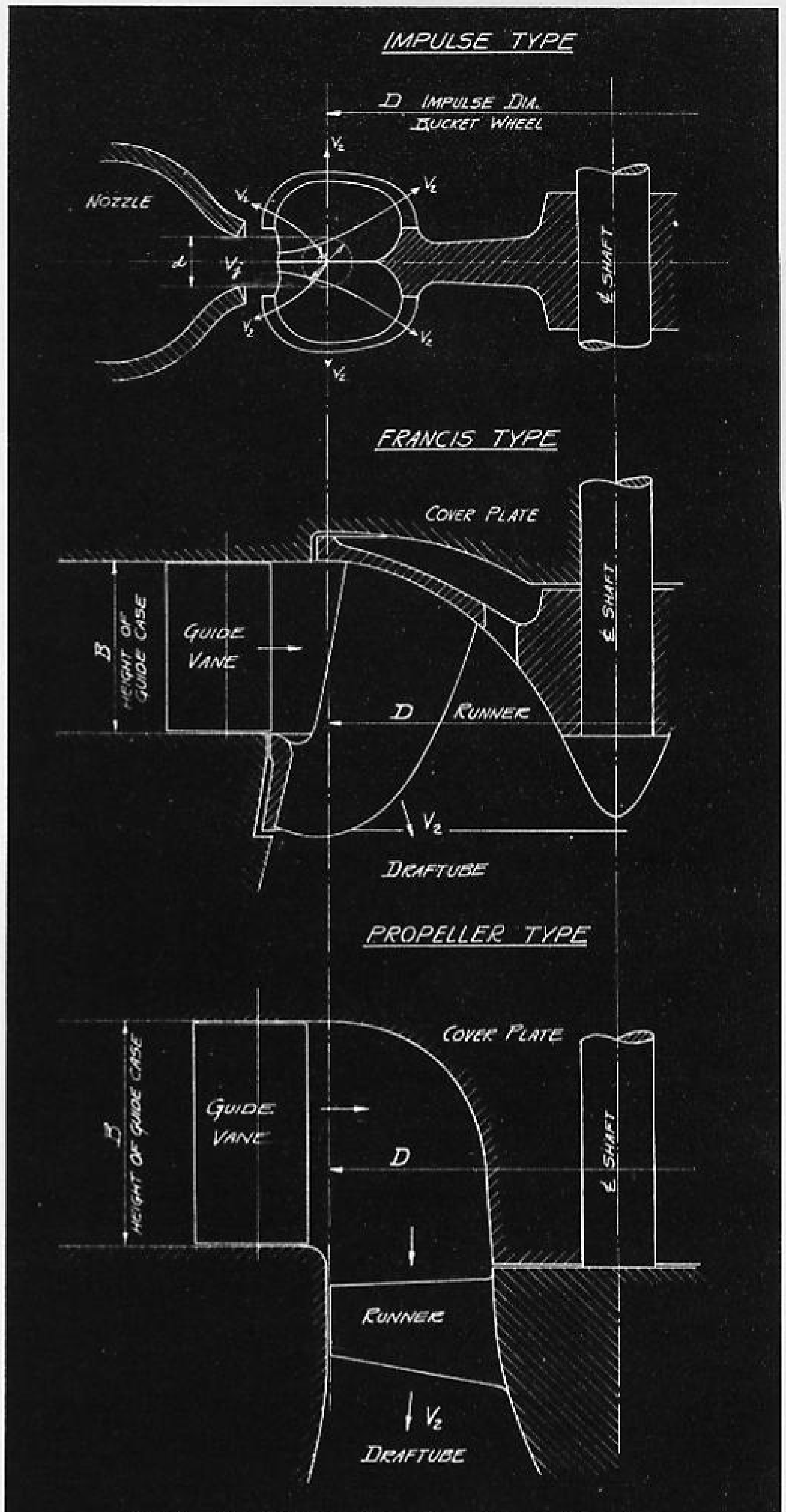


Fig. 6 — Coefficients of the three types of turbines.

Classification by Specific Speed

A review of the types of turbines discussed may be presented by classifying their runners according to the values of their speed characteristics (n_s), conventionally termed specific speed. It is the speed, or actual rpm, of a model of such physical size as will develop one horsepower at one foot head.

$n_s = \frac{n\sqrt{N}}{(H)5/4}$ or $n_1\ \sqrt{N_1}$ where

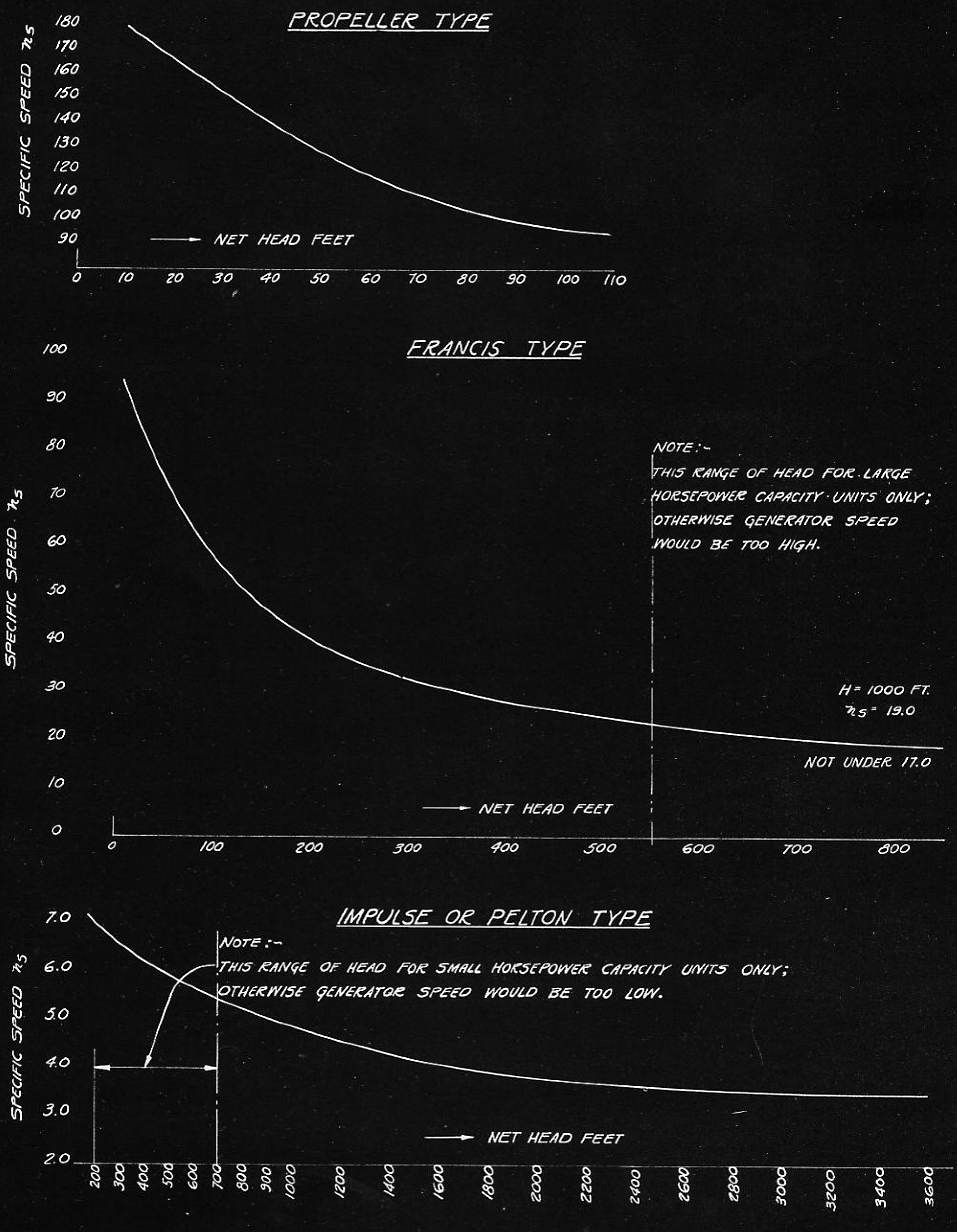
n = revolutions per minute (rpm)

N = horsepower output (hp)

H = head in ft

n_1 = rpm at 1 ft head; $n_1 = \frac{n}{\sqrt{H}}$

N_1 = hp at 1 ft head; $N_1 = \frac{N}{H\sqrt{H}}$



NOTE:- THIS RANGE OF HEAD FOR LARGE HORSEPOWER CAPACITY UNITS ONLY; OTHERWISE GENERATOR SPEED WOULD BE TOO HIGH.

NOTE:- THIS RANGE OF HEAD FOR SMALL HORSEPOWER CAPACITY UNITS ONLY; OTHERWISE GENERATOR SPEED WOULD BE TOO LOW.

Fig. 7 — Recommended upper limit values of n_s based on one runner and one jet for impulse type.

This formula is applicable to all three types as can be seen from Fig. 6, in which D denotes the reference diam of the runner in ft, d the jet diam of an Impulse wheel in ft, and B the height of the guide case of a Francis or of a Propeller type turbine in ft, and u* the coefficient of pheripheral speed in ft per sec based on D. The velocity of water discharging through the area of D is (V) in ft per sec. Or

$$u^* = \frac{u}{\sqrt{2gH}} = \frac{\pi Dn}{60\sqrt{2gH}} = \frac{a_1 Dn}{\sqrt{H}},$$

$$a_1 \text{ being } \frac{\pi}{60\sqrt{2g}} \text{ or } n = \frac{u^*\sqrt{H}}{a_1 D}$$

$$\text{and } N = \frac{QH\gamma e}{550},$$

where Q = discharge in cu ft per sec, γ = 62.4 lb, weight of 1 cu ft of water, 550 = ft lb per sec for 1 hp, and e = efficiency:

$$\text{Thus: } Q = \frac{\pi D^2}{4} V = \frac{\pi D^2 V^*}{4} \sqrt{2gH}$$

$$\text{and, } N = \frac{\pi D^2}{4} \text{ x } \frac{V^*\sqrt{2gH} \text{ x } He\gamma}{550}$$

$$\text{or } \frac{a_2 D^2 V^* \sqrt{H} \text{ x } He\gamma}{550}$$

where $a_2 = \frac{\pi\sqrt{2g \text{ x } \gamma}}{4 \text{ x } 550}$. Substituting the explicit values of N and n in the general formula $n_s = \frac{n\sqrt{N}}{(H)5/4}$,

$$n_s = \frac{u^*\sqrt{H}}{a_1 D} \text{ x } \sqrt{a_2} \text{ x } D\sqrt{V^*} \text{ x } \sqrt[4]{H^3} \text{ x } \sqrt{e} \text{ x } \frac{1}{(H)5/4}$$

$$\text{or if } \frac{\sqrt{a_2}}{a_1} = a_3,$$

$n_s = a_3 u^* \sqrt{V^*} \text{ x } \sqrt{e}$. Here, as previously stated, the values u* represent the coefficient of peripheral speed at diameter (D), V* the coefficient of velocity of water discharged through the area of diameter (D) and e the efficiency. Applied to each of the three types, the value of n_s can be written as follows:

Impulse Wheels

If there be substituted for V* the coefficient of velocity (V_j*) of water of the jet (d) instead of the wheel diameter (D),

$$Q = V_j^* \sqrt{2gH} \text{ x } \frac{\pi d^2}{4} = V^* \sqrt{2gH} \text{ x } \frac{\pi D^2}{4}$$

$$\text{or } \sqrt{V^*} = \frac{d}{D}\sqrt{V_j^*} \text{ and}$$

$n_s = a_3 u \frac{d}{D} \sqrt{V_j^*} \text{ x } \sqrt{e}$; for impulse wheel u* varies in design between the values of 0.45 to 0.50, V* between 0.97 to 0.92 depending on the quality of nozzle design, and e from 0.89 to 0.9 as a maximum to whatever the design quality affords. Hence,

$$n_s = c \text{ x } \frac{d}{D}.$$

The specific speed of an Impulse wheel is directly proportional to the ratio of jet diameter to wheel diameter (or of twice the distance from centerline of wheel to centerline of jet.)

Francis and Propeller Types

If for the value V* the radial velocity of water through the guide case is substituted instead of that through the runner area,

$$V_g^* = \frac{Q}{\pi DB\sqrt{2gH}}, \text{ as against } V^* = \frac{4Q_2}{\pi D^2 \sqrt{2gH}},$$

$$\text{so that } \sqrt{V^*} = \sqrt{V_g^*} \text{ x } 2\sqrt{\frac{B}{D}} \text{ and,}$$

$n_s = \frac{a_3}{2}\sqrt{\frac{B}{D}} \text{ x } \sqrt{V_g^*} \text{ x } \sqrt{e}$, where $\frac{B}{D}$ represents the ratio of guide case height (B) to runner diameter (D), and V_g* the coefficient of velocity of water through the guide case cylinder at diameter (D).

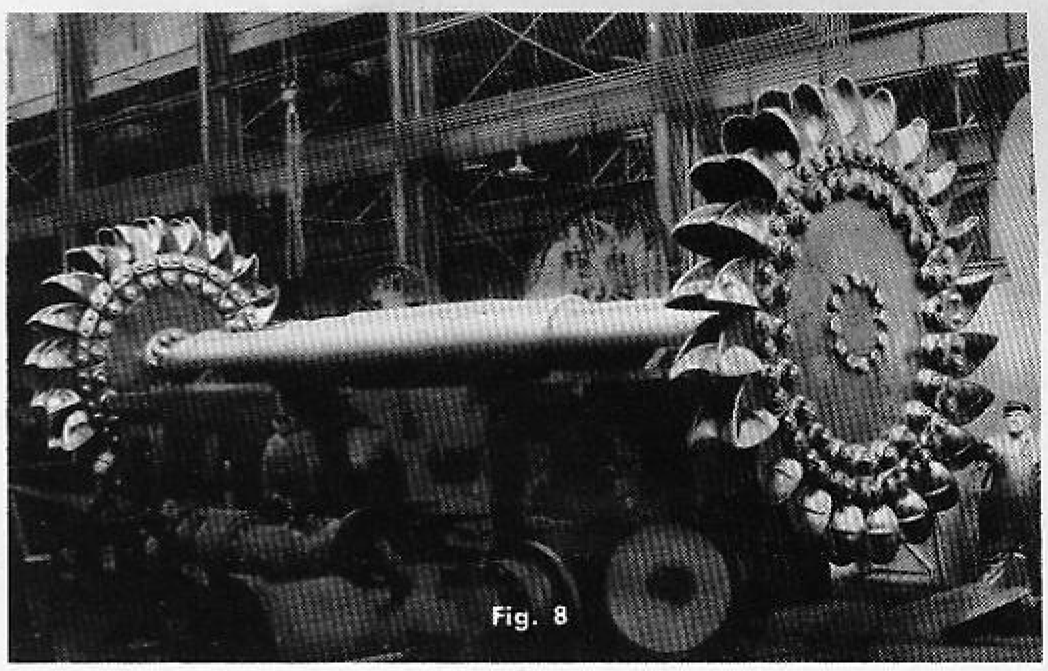


Fig. 8

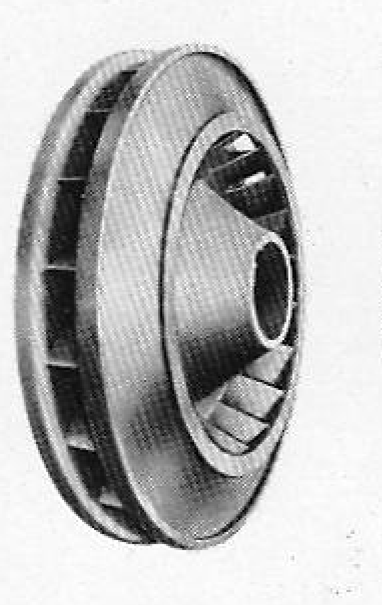


Fig. 9

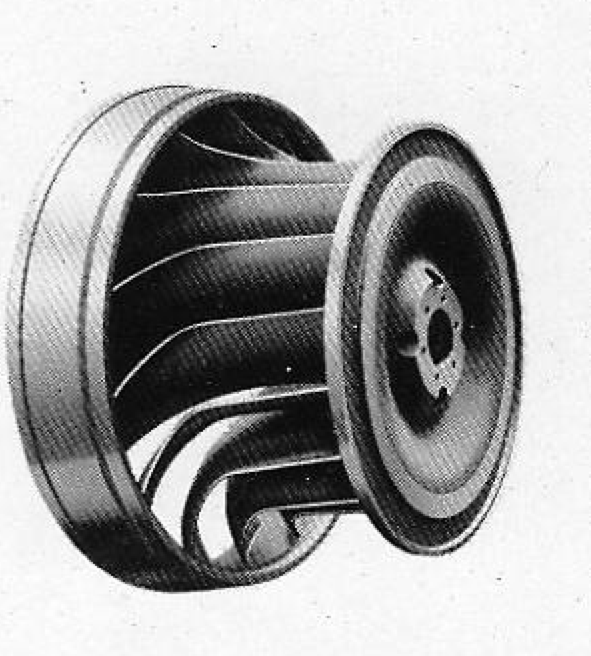


Fig. 10

Figure 7 shows recommended upper limits of practical values of n_s for the three characteristic types and for head ranges applicable thereto.

The value n_s of the Impulse type is based on the use of one jet per wheel. If z is the total number of jets per unit, the speed or revolutions per minute of the unit is increased by $\sqrt{z}$. A similar rule also applies to all three types in relation to the number of runners. The modern art refrains from using more than one Francis or Propeller runner per unit. Impulse or Pelton wheels are often provided with two bucket wheels (runners), one placed overhung on each generator shaft. A cheap design (as offered by foreign manufacturers for export) employs two jets per wheel so that, with a double wheel arrangement and a total of four jets, the revolutions will be double those of a single jet, single wheel unit.

Pictorial Review

A review of representative runners according to specific speed may be pictorially given by the following pictures of respective runners.

IMPULSE OR PELTON TYPE

Figure 8 — Buckets of a high head Impulse wheel. Head 2243 ft, capacity per wheel (one jet) 20,000 hp, speed 360 rpm. n_s = 3.2. Ratio D/d = 115/7.5 or about 15.

FRANCIS TYPE

Figure 9 — A high head, low specific speed runner. n_s = 17. The photo was taken after the runner had been in use daily for 18 years. (Note its excellent condition).

Fig. 10 — A high specific speed Francis type runner operating under 13 ft head. n_s = 90.

Figure 11 — One of the first four cast steel runners for Boulder Dam, operating under 490 ft normal head at 180 rpm, developing 115,000 hp. n_s = 27.0. These units were the first installed at Boulder Plant and were started at an exceptionally low head at the time Lake Mead was being filled. Under the ultimate maximum head these units will operate under as high as 593 ft head, under which condition they would develop at full discharge in excess of 150,000 hp but must, of course, operate only at part discharge so as not to overload the generators. It can be seen from the above that these units operate under a wide range above and below the normal specific speed value of 27.

Figure 12 — The runner of a 56,000 hp turbine operating under 213 ft head at 150 rpm. n_s = 44.

Figure 13 — One of four runners for Wilson Dam, Tennessee Valley Authority. Head 92 ft, capacity 35,000 hp, speed 100 rpm. n_s = 66. This runner is a solid steel casting weighing about 100,000 lb.

PROPELLER

Figure 14 — A high speed, fixed-blade propeller operating under 37 ft, developing 13,500 hp at 106 rpm. n_s = 135. Four units like this are located on the Ohio River at Louisville, Kentucky, and are subjected to flood conditions such that there

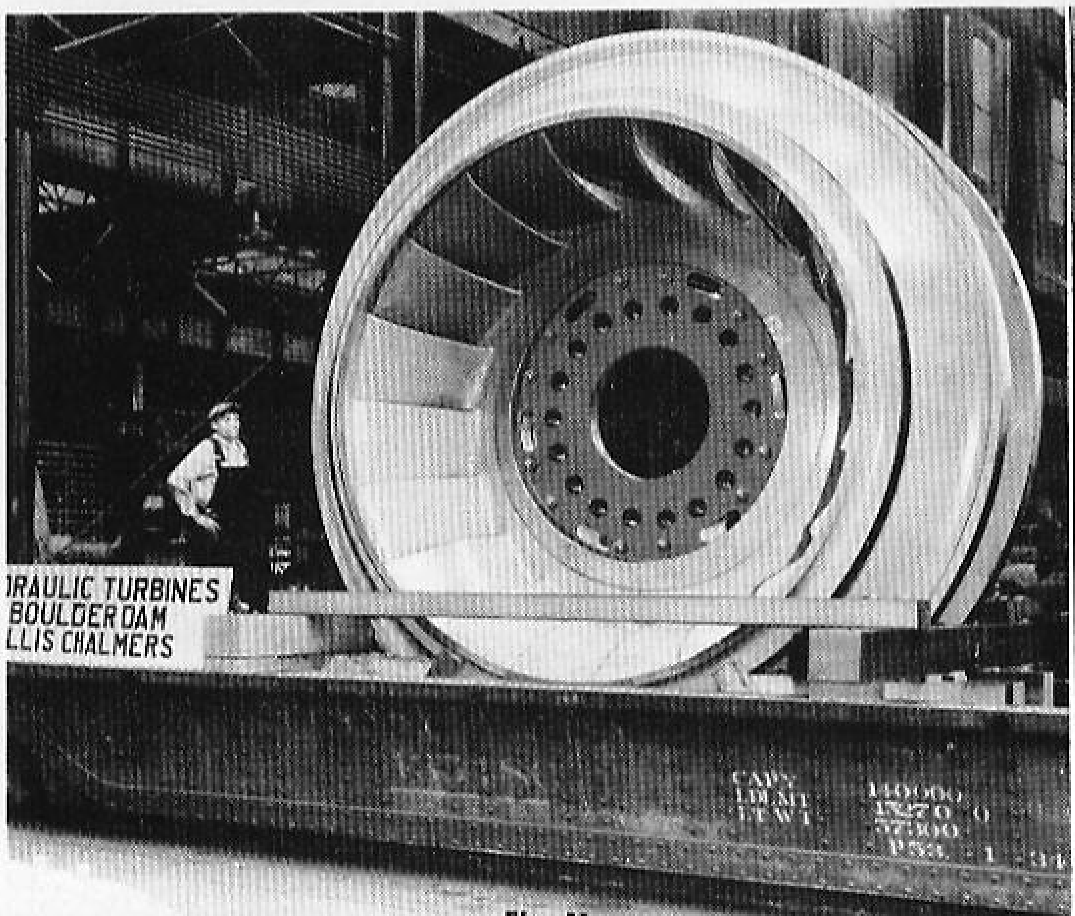


Fig. 11

Fig. 12

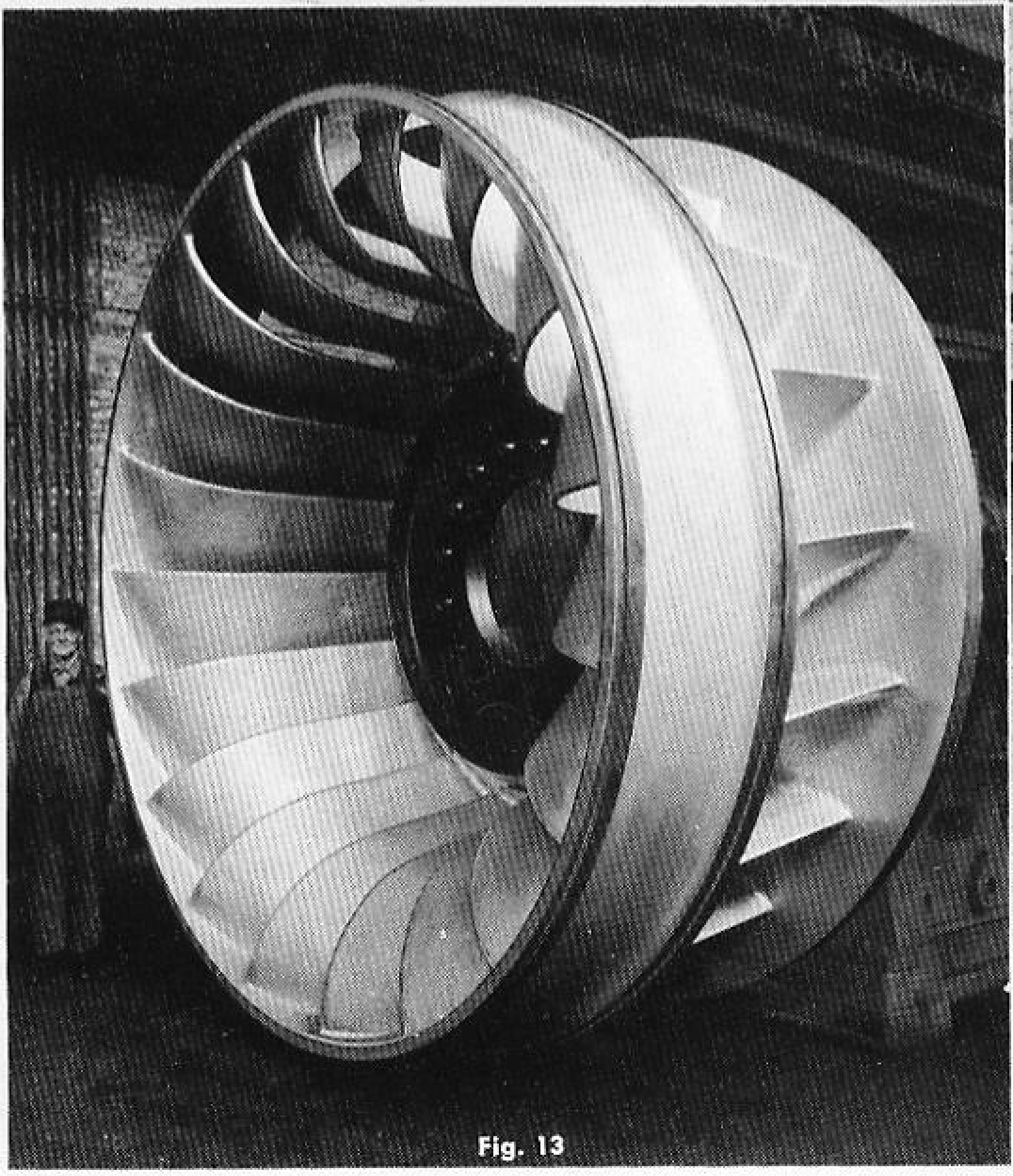


Fig. 13

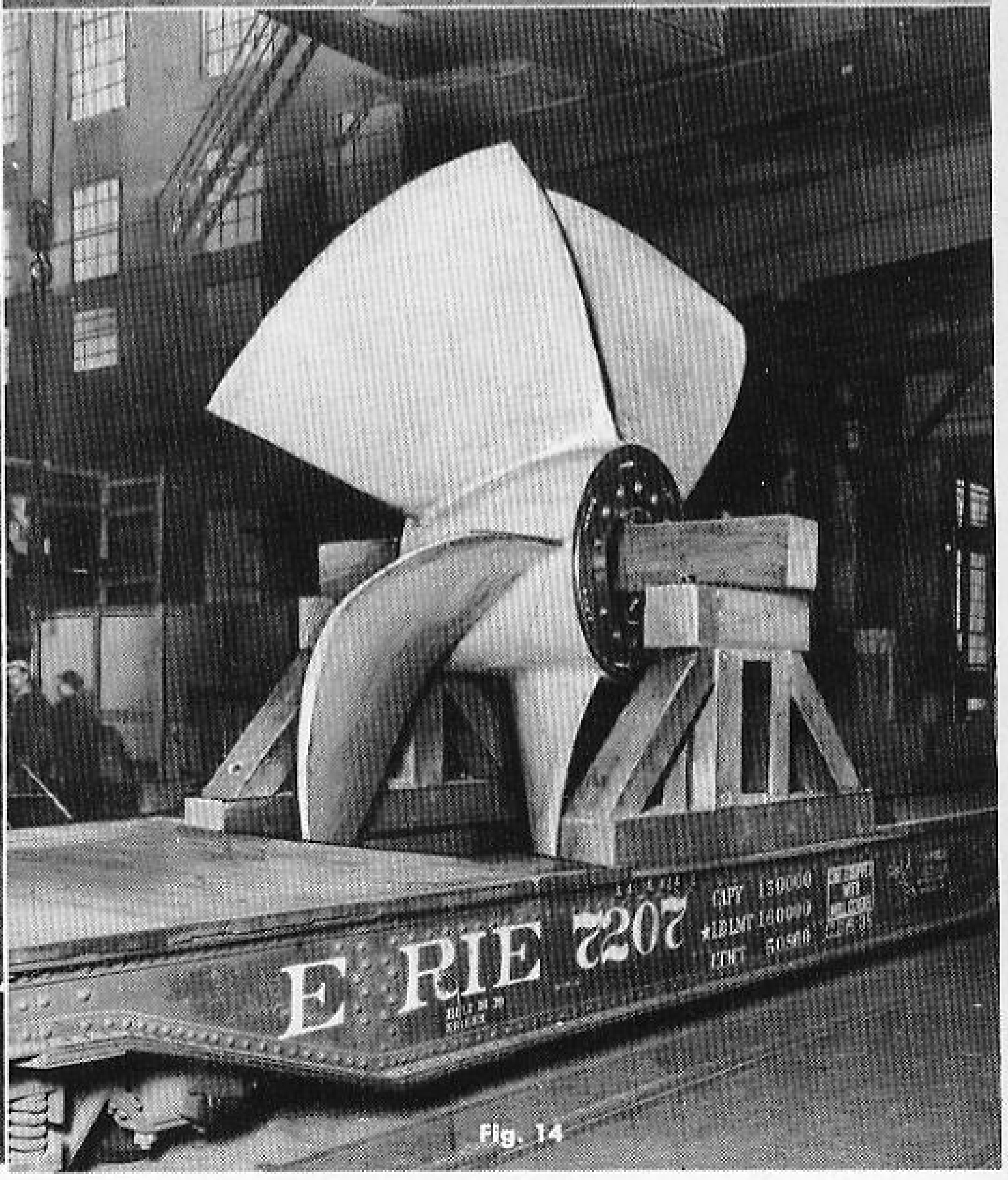


Fig. 14

is no difference between head and tailwater levels. Each develops some output at normal speed until the head has become as low as six feet.

Figure 15 — A hand adjustable propeller runner operating normally under 32 ft head, developing 21,000 hp at 100 rpm. $n_s = 192$. This is an abnormally high specific speed and was adopted because of the special condition that the ultimate head of the plant will be 48 ft so that, with the speed and capacity of present generator unchanged, the propeller will then operate under a specific speed of only 114. Four of these units have been in commercial operation since 1931 at Rock Island, Washington, on the Columbia River.

Figure 16 — A completely shop-assembled rotating element of a Kaplan type turbine for operating under 70 ft head at 120 rpm, developing 40,000 hp. Specific speed 119.

From the above specific speeds it can be seen that tolerance above and below the values of n_s, as given in Fig. 7, is allowed, in due consideration of special requirements as the case may be.

In conclusion of this article it may again be emphasized how necessary it is to analyze carefully all underlying conditions on which proper selection of type and characteristic of the hydraulic prime mover is to be based.

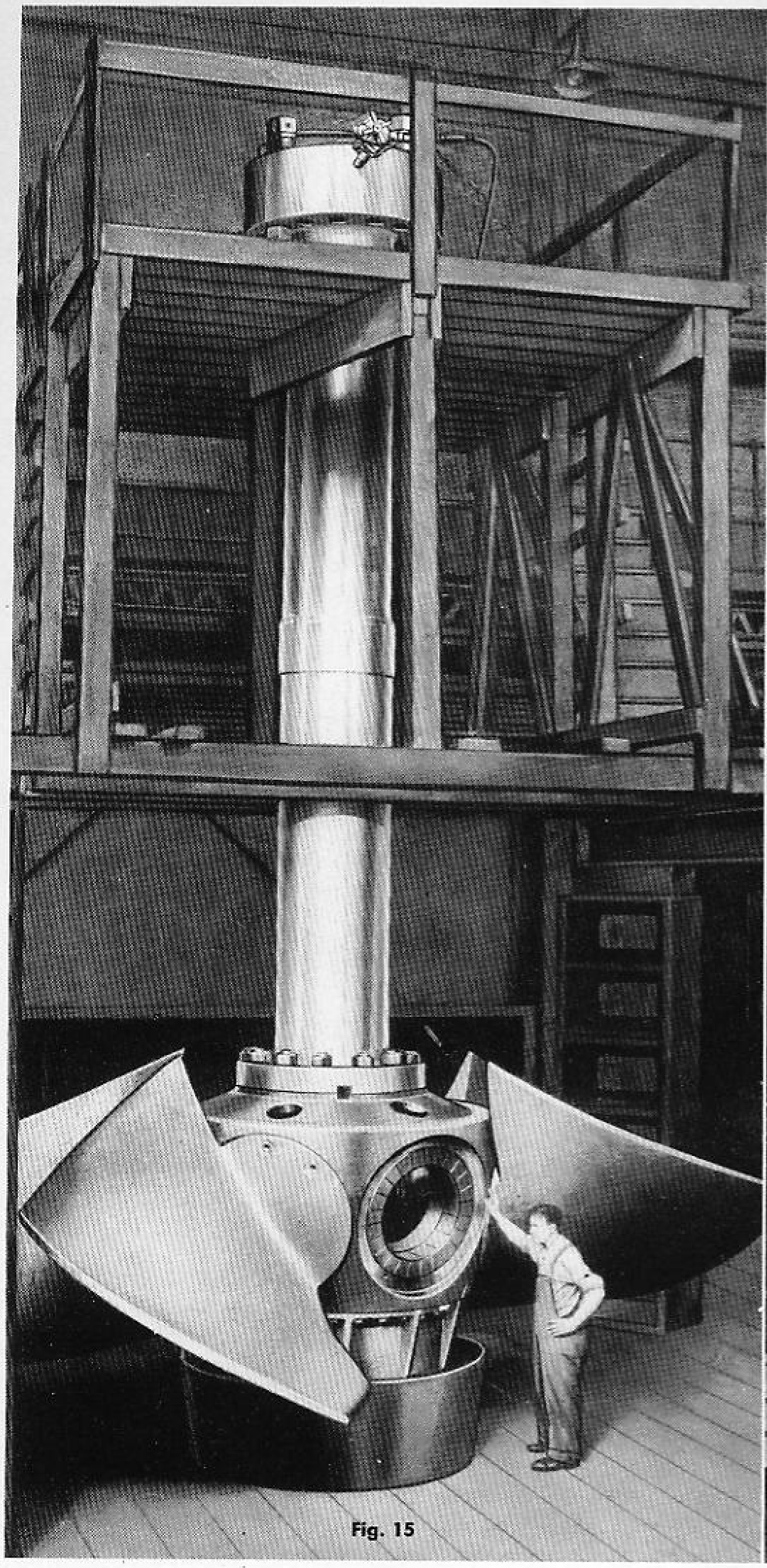


Fig. 15

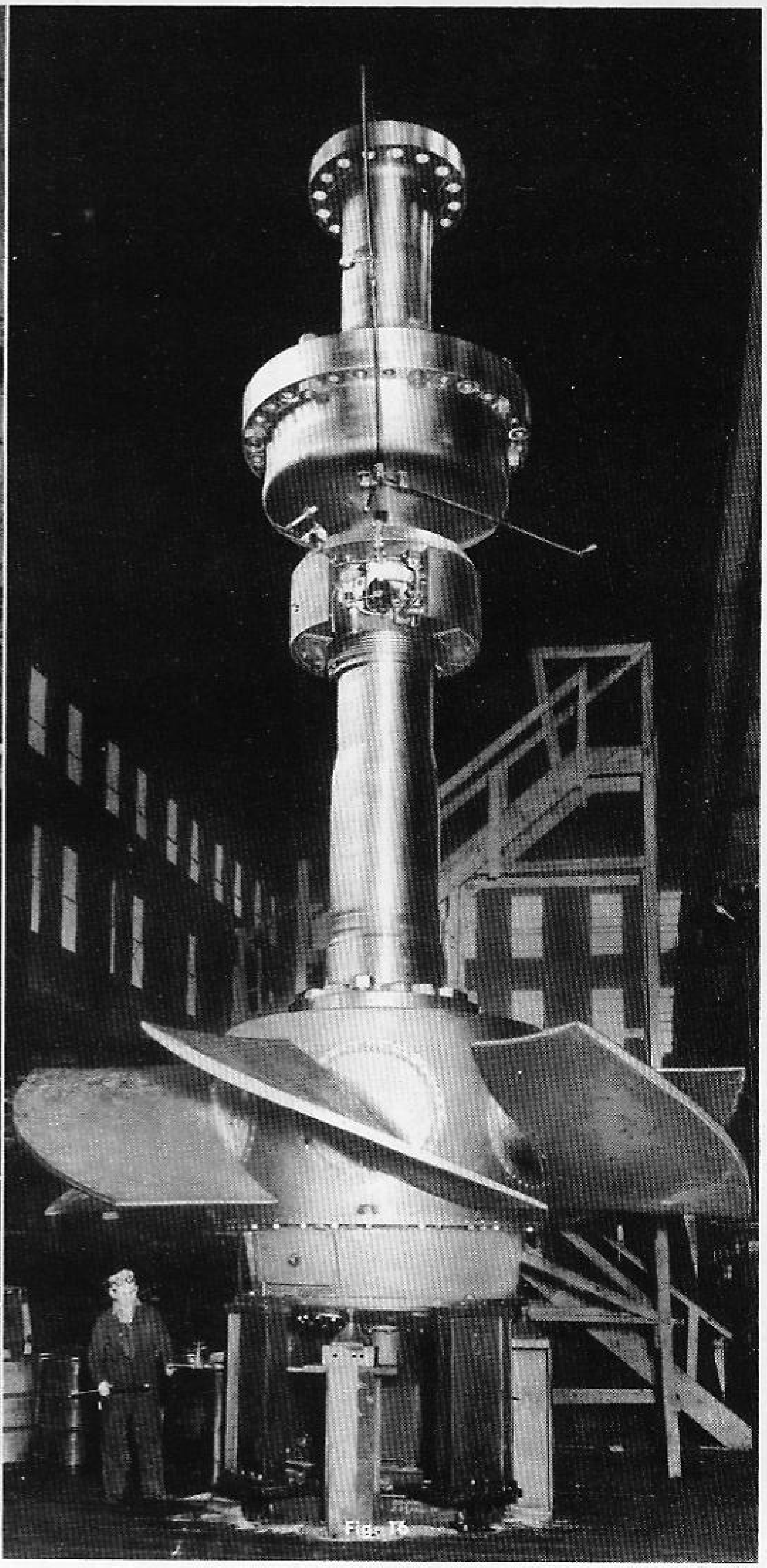
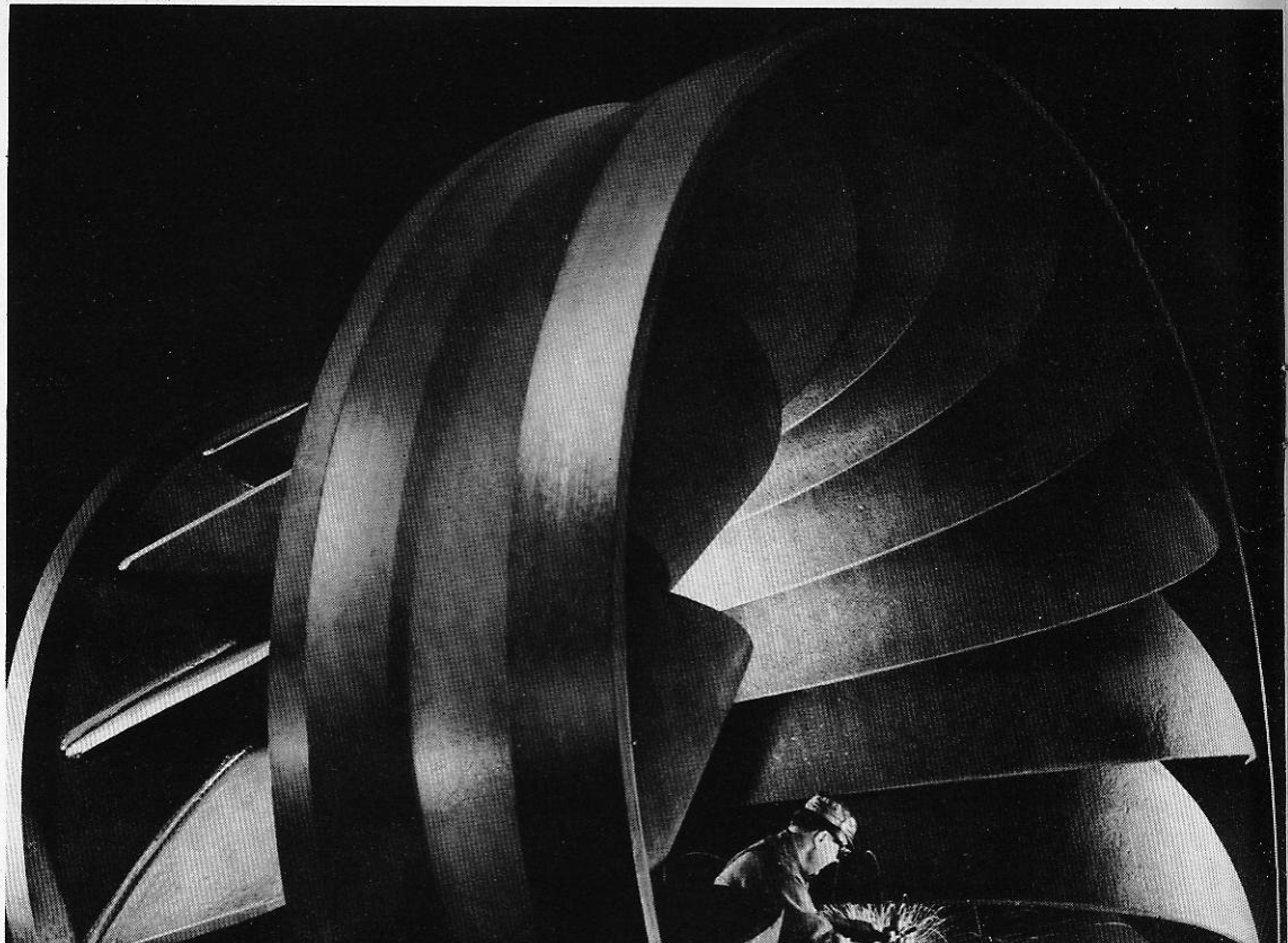


Fig. 16



For one of TVA's new power projects, this 95,000 lb cast steel runner for an hydraulic turbine will revolve at 100 rpm.

Part III

RELIABILITY OF SERVICE

●

SPEED AND PRESSURE CONTROL

In the previous parts of this article the fundamental nature of the three types of water turbines has been set forth. This article shall deal to some extent with the accessories. As was pointed out, it is important that the water be led to and from the turbine with the least possible disturbance because the losses produced affect the efficiency and the life of the unit and, consequently, the return on the invested capital.

With equal emphasis it can be stated that the lack of necessary accessories and of their proper selection can seriously affect both the revenue and safety of a plant.

The item affecting the revenue is composed of a number of individual factors, such as:

(1) Efficiency, already pointed out and discussed.

(2) Reliability — absence of outage caused by wear and tear involving repairs.

RELIABILITY UNDER MECHANICAL WEAR

Surfaces exposed to the flow of water are subjected to natural wear. It is obvious that, whenever such surfaces form an integral portion of a main part of the turbine, its repair or replacement is expensive and involves an outage which may seriously affect the revenue. Such surfaces should be made of material readily replaceable, or integrally of such material as permits welding in place.

Consider an example in connection with a large unit that, under continuous service, earns $2,400 every 24 hours, or $100 per hour. If the design can save one day of outage per year it is worth, on the basis of a 10-year amortization of capital investment, a saving of $24,000, which goes a long way in providing for ready replacements.

Take, for instance, the nozzle tip of an impulse wheel, the downstream end of which forms the throat or orifice from which the water jet issues at high velocity. This part is subject to natural wear. For a 60,000 hp unit such as the one placed in commercial operation in 1928 at Big Creek Plant No. 2-A in California under about 2200 ft head, the complete nozzle tip of cast steel weighs about 3,000 lb and costs about $1,500. Its replacement at the plant would require at least six hours outage of the unit, or about 216,000 kwh, involving a revenue loss based on 0.25 cents per kwh of $540, so that a total loss of $2,040 results.

Allis-Chalmers design provides for a renewable throat ring of wear-resisting material, weighing about 95 lb and costing about $75. This ring can be readily replaced in one hour, thereby reducing the revenue loss to $90.00 and involving a total expenditure of only $165.00, about 8% of the other item.

Reliable Accessories

Under the same heading of reliability-outage belongs an item not given due consideration. Before access is possible to a turbine for inspection or repair, the water must be shut off completely from both ends. Much time is often lost on account of inadequate, or entire absence of, such facilities, and this applies almost equally to all three types of turbines. Especially in connection with low head installations, provision of head and tail-water gates is sometimes lacking completely. Often even stop logs are not provided, nor equipment such as motor-driven centrifugal pumps for unwatering the turbine flumes.

Plants with penstocks or pipe lines of considerable lengths, such as are involved with high head turbines and impulse wheels, should have means at the power house for shutting the water off *completely and individually from each unit.* They should be so designed that they are sufficiently tight to permit safe access to the interior of the turbine at full pressure in the pipe line. It is essential that such valves be designed and built not only for normal operation, but also for emergency conditions when it becomes necessary to close them against full flow of water in case the control mechanism of the turbine refuses to function. Such a shut-off valve should constitute a "gold bond," so to speak, for the protection of the plant equipment. Many serious accidents resulting in loss of human life and property are directly attributable to the inadequacy of such protective equipment.

Safety in Operation

One of the main requirements for reliability is that a unit must be capable of withstanding without injury the full runaway speed which it may attain under any conditions of operating head, and under full discharge opening, without any load on its generator (or resistance on its shaft if mechanical drive is involved). Such tests should be made initially in every instance to assure the operators of the absolute safety of the equipment, which will permit them to take care of other more serious disturbances in the plant first. This involves coordination of designs and cooperation between generator and turbine designers, from the latter of the two to come the data as to possible maximum overspeed. Many serious wrecks were caused in the earlier days because generator specifications were issued and equipment purchased in complete disregard of this important point.

RELIABILITY IN QUALITY OF SERVICE

Hydro-electric units operate mainly in parallel with a network of plants often tied together over long transmission lines. The stability of operation of each individual prime mover is, therefore, of importance because the network must maintain as nearly a constant speed (or frequency) as possible. For this reason a turbine operating in parallel with such a network must in itself maintain a constant speed and output under constant load and operating head; otherwise so-called power swings will occur which, especially in connection with large networks and long transmission lines, will increase in amplitude at the station where the various transmission systems are tied together.

Where the load of such a system varies, be it gradually or momentarily, the prime movers operating in parallel in the system must be capable of adjusting the output over the full range of load change; and it is evident that the control equipment of each unit must be so adjusted that it takes only its proportionate share of the change. If the control is not so adjusted, another prime mover will have to correct any corresponding surplus or shortage; and, if this has to come in over a transmission line, surges are likely to be set up which again may disturb the stability of the network.

SPEED CONTROL

It can be seen that the speed control equipment of such a prime mover is of paramount importance. This control is accomplished in any prime mover by adjusting the output producing medium according to the demand at the time. The output producing medium may be fuel in a diesel engine, steam in a steam-turbine, water in an hydraulic prime mover. In the last cited case the problem of kinetic energy control is by far more serious than in any other type of prime mover.

This may be illustrated by the comparison of kinetic energy of a water turbine and of a steam turbine of about equal horsepower outputs.

The kinetic energy is:

$\frac{Mv^2}{2}$ or $\frac{Wv^2}{2g}$ or $\frac{Q\gamma v^2}{2g}$ as explained in previous article (POWER REVIEW, Dec., 1941).

Here $Q\gamma$ is the quantity in pounds flowing through the prime mover and v the velocity in ft per sec at which the quantity passes. This quantity must be regulated in accordance with the output demand.

Example

(a) A 55,000 hp, propeller type hydraulic turbine under 48 ft head at 81.8 rpm.

There are 12,000 cu ft of water passing through the propeller at a velocity of about 31.2 ft/sec. Thus:

$$\frac{Q\gamma v^2}{2g} = \frac{12{,}000\times 62.4\times (31.2)^2}{2\times 32.2} = 11.35 \text{ million ft lb approx.}$$

The full capacity of this unit is about 38,500 kw.

(b) A 35,000 kw steam turbine unit, rated 850 lb G pressure, 29 in. vacuum, consumes about 74 lb of steam per sec at a velocity of the steam passing through the inlet of 125 ft/sec.

The kinetic energy is:

$$\frac{Mv^2}{2} = \frac{Wv^2}{2g} = \frac{74\times 125\times 125}{2\times 32.2} = 17{,}900 \text{ ft lb.}$$

Thus: $\frac{17{,}900}{11{,}350{,}000} = 0.00158$, from which it can be seen that the kinetic energy in the steam turbine of about equal capacity of that of the propeller turbine is only 0.158%. Therefore, the control of the water turbine requires governing equipment of much greater ft lb capacity than is required for a steam turbine.

The problem of speed control of a water turbine is of a complex nature.

If the hp output of the turbine could be changed at the same rate at which the hp load absorbing this output changes, then there would be no problem of speed control because for each existing load the output would be the same so that neither a surplus nor a shortage between the two would exist. However, it is practically impossible to attain this since the output of the turbine is affected by many factors independent of the load change.

The revolving masses of the unit (and of the rotating masses revolving in synchronism with it if the unit is paralleled to an electric power system) represent a fixed kinetic energy at a fixed speed, or frequency of the unit. Therefore, a surplus of energy produced in the prime mover or turbine over that absorbed by the load of the system will cause an increase or inflow of kinetic energy into the revolving masses, thus producing an increase in rpm of the turbine. Similarly, a shortage produces a decrease in the speed. It is this change which causes the governor to change the output of the turbine.

Sensitiveness of Governing Equipment

If the governor remained inactive this speed would continue changing. It is therefore important that the governor respond with the least possible delay. As stated before, the control mechanism of the turbine, i. e., the mechanism which controls the output of the turbine, requires a total capacity which with large turbines runs into hundreds of thousands of foot pounds, and this must be available during as few seconds of time as is required for obtaining commercially satisfactory speed control. This energy is made available by oil stored up under suitable pressure in a pressure tank supplied with pressed oil by an oil pump at a rate high enough to prevent

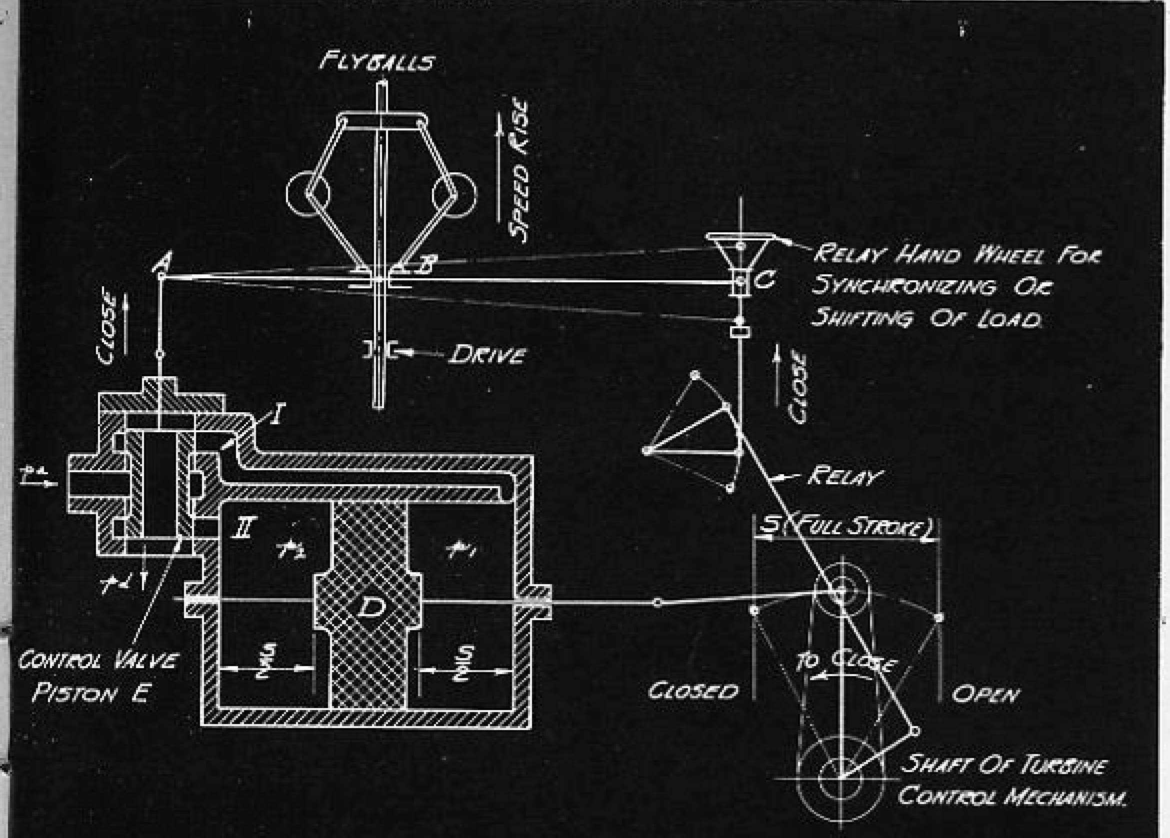


Fig. 1 — Diagramatic sketch of governor.

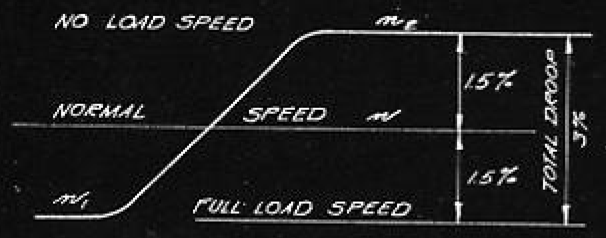


Fig. 2

serious pressure drop in the oil pressure system supplying oil to the governor.

The pressed oil is admitted to a servomotor, usually consisting of a cylinder (often two cylinders for convenient arrangement, especially with large Francis and propeller type turbines or double-overhung impulse wheel units) and piston connected directly to the control mechanism. The oil is directed to the two cylinder spaces by a control valve operated by a speed-responsive element, usually called flyballs, or centrifugal governor, which responds to the speed change of the unit and causes the control valve to distribute the oil accordingly.

To stop the movement of the servomotor piston, the control valve must be brought into neutral or "dead-beat" position, locking the flow of oil to and from the regulating cylinder. This mechanism is called the relay, or restoring mechanism, of the governor. The adjustment of this relay, either by hand or remotely, permits changing the speed of the unit for the purpose of paralleling and for shifting the load (output) of the turbine when paralleled.

Figure 1 shows a diagrammatic section of the governor (exclusive of the oil pressure system, namely, oil pump, pressure and receiver tanks). It can be seen that the control valve at A stops the flow of oil to and from the regulating system when it is in "dead-beat" or central position; i. e., when the two ports to the cylinder are closed off. If these ports had "lap" (if they overlapped the port openings), then a certain movement at A would be required to cause a flow of oil. In other words, A could then not be considered a fixed point. The opposite end C of the relay lever is connected to the regulating piston.

At point B the flyballs are located, operating in such a way that on a speed increase they raise point B; and, since the servomotor is not in motion (C stationary), the valve is raised at A, thereby admitting pressed oil to the side p_1 of the piston and causing a closing movement of the turbine control mechanism and reducing its output. When B remains stationary (no further increase in speed), then the movement of piston in closing direction, as indicated by the arrow, causes point C to rise and thus lowers A with B as fulcrum becoming stationary. The control valve is thus restored to its central position and the control mechanism becomes stationary again.

It is evident that, without the relay, C would remain a fixed point so that A would remain raised and the control mechanism would be moved in closing direction until the output of the turbine had become so reduced that a speed drop causes B to return the valve to central position. In other words, the speed governor would be so active that it would tend to hold the speed of the turbine absolutely constant, or it would cause the governor to "walk all over its entire stroke" at the slightest speed change. Such a governor would be unstable, causing what is commonly-termed hunting, or racing; and two generators, each controlled by such a governor, would not hold their share of kw load.

A relay is, therefore, indispensable, and the resultant stroke at C, producing a corresponding stroke at B to hold central position of valve at A, then causes a difference in rpm of the flyballs and of the turbine. This is commonly called the "droop."

A droop of 3%, as shown on Fig. 2, indicates a change of speed of 3% from full-load speed to no-load speed of the unit.

The total governor time T_g for a full servomotor stroke is composed of the following elements:

T_F, the time it takes the flyballs to begin to move the control valve (at A) out of central position, to build up over-pressure upon the servomotor piston.

T_0, the time required to build up the over-pressure to actuate the piston until it is moving at its required rate of speed.

T_n, the actual time to produce the full stroke of the servomotor piston.

$$T_g = T_F + T_0 + T_n$$

During the time T_F the full difference between output and load change causes a corresponding speed change. During the time T_0 a portion of the turbine output is already changed but lacks the full effect until the servomotor has attained the required speed.

During the time T_n the output of the turbine is reduced at maximum rate of speed. The time T_n is fixed by the amount of oil displaced in the servomotor and depends on the ratio of quantity of oil passed through the ports of the control valve to the total displacement of oil in the servomotor cylinder.

T_0 depends upon size and quality of performance (quickness of valve movement) of the control valve.

T_F is fixed by the sensitiveness of the flyballs and is a measure of quality of design and workmanship. The shorter T_n, the greater relatively is the impairing effect of T_F and T_0.

Calculation of Speed Change

Referring to Fig. 3, the sudden load change from full to zero is represented by line ABC. To hold the speed normal under load, the turbine output must be equal to the load. Instead of decreasing to zero suddenly, line AB, as the load decreases the output of the turbine, is reduced along line ADEC over a total governor time of T_g seconds. The surplus energy is represented by area ADECB and is composed of the energy surpluses:

ADFB caused by the delay T_F of the flyballs.

DEGF caused by the control valve not being able to produce suddenly a full rate of movement of the control mechanism during T_0 seconds.

ECG represents the output energy reduced at full rate of movement during T_n seconds.

Let ϕ be the load factor; therefore $\phi = 1.0$ for full load. During the time T_g a total kinetic energy of ϕ T_g or AHCB is cut off by the sudden load change.

The governor takes off only the energy represented by the ADECB. Therefore a surplus of energy ADECB must be taken up in the energy of all the rotating masses involved.

If the unit is disconnected from the system, then only its own rotating masses are involved. If, however, other rotating masses remain connected — e. g., all masses revolving in synchronism if the unit is not detached from the system — then a much larger rotating mass must be accelerated on load rejection or decelerated on load increase.

So long as a unit remains paralleled to the power system, the speed change will naturally be materially less than in the case when the unit is detached entirely. In making regulation guarantees it is always assumed that the unit is not paralleled to the power system so that only its own moment of inertia (WR^2) of revolving masses is available. The load is usually produced by a water rheostat, and the changes can be made practically instantaneously.

From Fig. 3 it can be seen at once that any load change not along line ABC greatly reduces the resultant speed change. If the load were rejected along line ADEC there would be no speed increase, except that caused by the droop, or position of the relay of the governor to hold the control mechanism to friction load speed n_2 (Fig. 2).

The line ADEC is affected by several factors:

(1) The nature of the governor proper, as pointed out before.

(2) During the speed change the horsepower output, even at a constant head and gate opening of turbine, varies.

A water turbine, be it an impulse wheel, Francis type, or propeller type turbine, has a fixed speed, called runaway speed (n_r) when it discharges water under wide open gates (orifice of impulse wheel). Let n be the number of normal revolutions per minute and N (hp) the corresponding output. When the turbine is held at zero speed ($n=0$), the output has become zero; and likewise, when the runaway speed n_r is reached, the output has disappeared ($N_r=0$).

In either direction, from n to 0 and from n to n_r, the turbine loses output and, therefore, acts as a brake, reducing the speed rise on load rejections and aggravating the speed drop on load increase.

For impulse wheels the runaway speed n_r may be taken as 1.85n.

For Francis turbines it varies, depending upon the specific speed characteristics n_s as outlined in previous articles. It may vary from $n_r=1.65$ to 1.9.

For propeller type units the value n_r may be as high as 2.5n, from which it can be seen at once that generators driven by propeller turbines must be figured for much higher runaway speeds than the other two types.

(3) When water is supplied to a turbine in a closed conduit, any change of discharge will cause a change of velocity in the conduit, thereby changing the kinetic energy. This produces a pressure rise on retardation of the flowing water and a drop on acceleration. This causes a change of head on the turbine, thereby changing the output in the ratio of $\left[\frac{H_2}{H_1}\right]^{3/2}$, where H_2 represents the head after and H_1 that before the change. This would thus result in raising or lowering curve ADC between A and C of Figs. 3 and 4.

Figure 3 applies to sudden load rejection. Fig. 4 indicates a sudden load increase AB. The output demand during time T_g is ABHC, and the kinetic energy supplied by the turbine is ADEHC. Therefore, the shortage of energy is again ABHED causing a retardation of the revolving masses. It is thus evident that the lines ADEC for load rejection and the lines ADEH for load increase are curves of manifold shapes. For accurate calculations, therefore, a step-by-step method will have to be employed, dividing up BC or AC abscissae in a number of increments and figuring the respective points between, from the output characteristic as they result for each interval. It is evident that this is a rather tedious task but the only one assuring closely correct results. The end result can be put in the form of the following equation:

Let n_x be the highest (or lowest) speed after the load change.

n_1 the initial speed before the load change.

N_2 the output in hp after the load change.

N_1 the output in hp before the load change.

N the normal full load output of the turbine.

n the normal rpm of the turbine.

T_g the total governor time in seconds as before.

M the revolving masses or $\frac{W}{g}$, where W=weight in pounds and g=32.2 ft/sec/sec.

The relative load change is:

$\frac{N_2-N_1}{N}$, and the maximum relative speed change $\frac{n_2-n_1}{n}$.

The change in kinetic energy of revolving masses is from $\frac{Mv_1^2}{2}$ to $\frac{Mv_2^2}{2}$ or $M\left(\frac{v_2^2-v_1^2}{2}\right) = \frac{W}{g}\left(\frac{v_2^2-v_1^2}{2}\right)$.

The value is positive on speed rise (load rejection) and negative on speed drop (load increase).

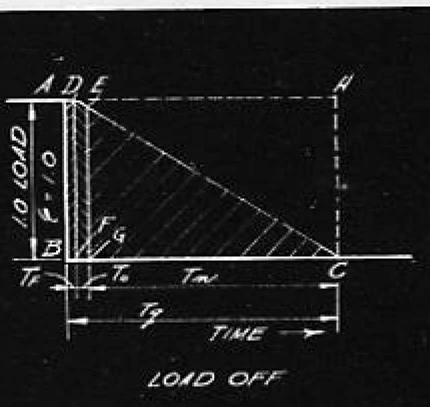


Fig. 3

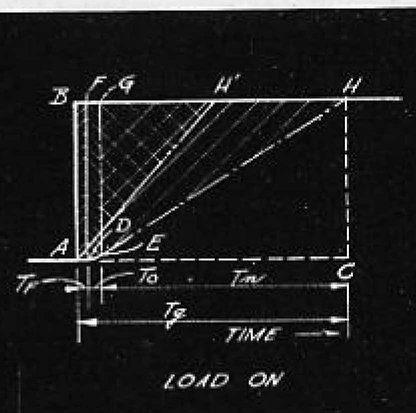


Fig. 4

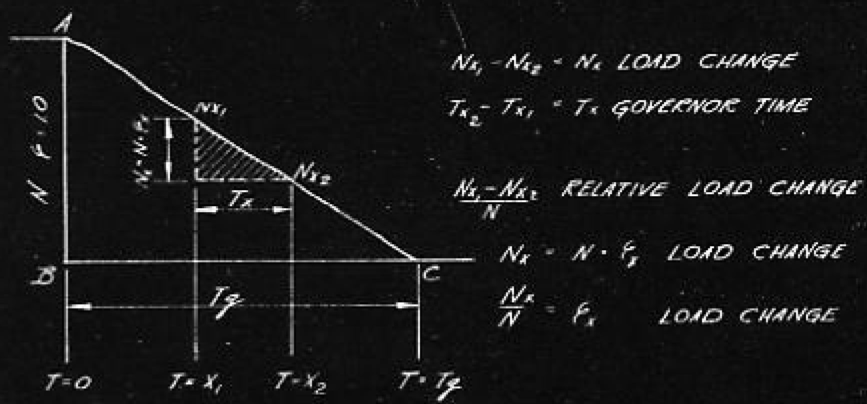


Fig. 5

The change of load is N_2-N_1 in hp or in ft lb/sec $(N_2-N_1)550$, because 1 hp=550 ft lb/sec. N_2-N_1 is represented by the area ADECB and ABHED in Figs. 3 and 4, respectively, over the time T_g. For any part-load change between full-load and no-load, the change in kinetic energy (as previously represented for full-load in Figs. 3 and 4) can be written in general form as shown in Fig. 5.

$$\Delta E=550\int_{t=x_1}^{t=x_2} N_x\,dt,\text{ expressed in ft lb/sec instead of hp.}$$

This kinetic energy change must be balanced by the kinetic energy of the revolving masses engaged.

$$\Delta E=\frac{M}{2}(v_2^2-v_1^2)\text{ or }\frac{W}{2g}(v_2^2-v_1^2)$$

where W is the weight of the revolving masses, g=32.2 ft/sec/sec gravity, and v the respective peripheral speeds at the radius of gyration R of the revolving masses.

$$v=\frac{\pi Rn}{30}\text{ in ft/sec.}$$

Thus:

$$\Delta E=550\int_{t=t_1}^{t=t_2} N_x\,dt=\frac{WR^2}{2g}\left(\frac{\pi}{30}\right)^2(n_2^2-n_1^2).$$

From this it can be seen that a change of kinetic energy due to a load change affects directly the square of the speed of the rotating masses and thus permits calculating the speed change. Multiplying both sides of the equation by $\frac{n^2}{n^2N}$ we obtain:

$$\frac{\Delta E}{N}=\frac{1}{N}\int_{t=x_1}^{t=x_2} N_x\,dt=\frac{1}{550}\,\frac{1}{2g}\left(\frac{\pi}{30}\right)^2\times\frac{WR^2n^2}{N}\times\frac{n_2^2-n_1^2}{n^2}$$

$\frac{WR^2n^2}{N}=C$ is the flywheel constant as explained in a previous article; and, writing $N_x=N\phi_x$ in which N represents the full hp output and ϕ_x the part-load factor, also $n_2-n_1=n_x$, and n_1 is approximately equal to n since n_1 on a small droop is within 1½% of normal speed n_1 at a total droop of 3%, there is obtained

$$\int_{t=0}^{t=T_g}\phi\,dt=\frac{C}{3,232,991}\left(\frac{n_2}{n}\right)^2-1$$

or the relative resultant speed change

$$\frac{\Delta n}{n}=\frac{n_2}{n}-1=\sqrt{1+\frac{3,232,991}{C}\int_{t=0}^{t=T_g}\phi\,dt}-1,$$ for speed rise due to load rejection.

$$\frac{\Delta n}{n}=1-\frac{n_2}{n}=1-\sqrt{1-\frac{3,232,991}{C}\int_{t=0}^{t=T_g}\phi\,dt},$$ for speed drop due to load increase.

If the output were changed by the governor along a straight line AC or AH, Figs. 3 and 4, the respective areas would be ACB and AHB and would for full load change be $\frac{\phi(\ =1)\,T_g}{2}$ so that, for a relative speed rise:

$$\frac{\Delta n}{n}=\frac{n_2}{n}-1=\sqrt{1+\frac{3,232,991}{C}\,\frac{T_g}{2}}-1,\text{ or}$$

$$\text{abbreviated}=\sqrt{1+\frac{1,616,495}{C}}\,T_g-1$$

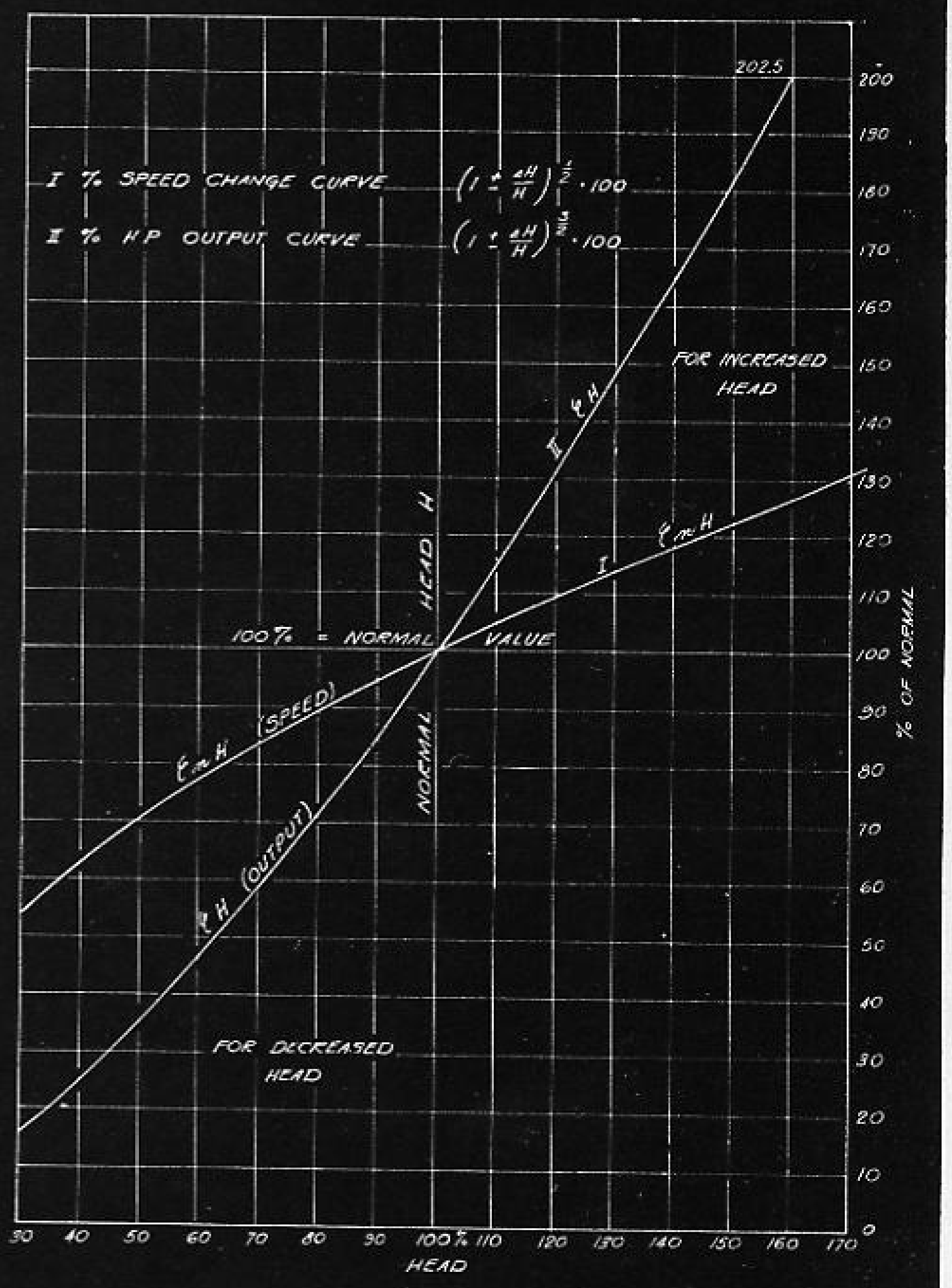


Fig. 6

This formula is correct only if during the load change the horsepower does not vary over the entire speed range and, furthermore, if the head is not changed during the process of load change. The above may demonstrate that, to obtain close results, it is necessary to analyze each case carefully, taking into consideration every factor involved, such as individual characteristics of the runner and of the conduit admitting water to the runner. It is thus practically impossible to arrive at a reliable end formula, but requires the tedious method of proceeding step by step, a detail outline of which is beyond the scope of this article.

Variable Head

In previous chapters it has been explained that, when the head is changed, both the discharge and the output of the turbine change.

The discharge Q varies as the 1/2 power of the head.

$$Q_2=Q_1\sqrt{\frac{H_2}{H_1}}\text{ or }Q_1\left(\frac{H_2}{H_1}\right)^{1/2}$$

where H_2 and Q_2 are the new values. The output N varies as the 3/2 power of the head.

$N_2=N_1\left(\frac{H_2}{H_1}\right)^{3/2}$. From above relations it follows:

$\frac{Q_2}{\sqrt{H_2}} = \frac{Q_1}{\sqrt{H_1}}$ and $\frac{N_2}{H_2\sqrt{H_2}} = \frac{N_1}{H_1\sqrt{H_1}}$, namely that during unity of head the discharge and the horsepower remain the same. With turbines this does not hold true unless the speed at unity head remains the same; i. e., if $\frac{n_2}{\sqrt{H_2}} = \frac{n_1}{\sqrt{H_1}}$. It can thus be seen at once that the problem becomes complicated if the above relation does not exist.

Example

(a) Head increase: $H+\Delta H=1.20$ (or 20% rise in head).

ϕ_{nH} speed rise factor $(1.2)^{1/2}=1.095$ or 109.5%.

ϕ_H hp output factor $(1.2)^{3/2}=1.3145$ or 131.45%.

(b) Head decrease $=H-\Delta H=0.8$ (for 20% drop in head).

$\phi_{nH}=(0.8)^{1/2}=0.8944$ or 89.44%.

$\phi_H=(0.8)^{3/2}=0.71562$ or 71.562%.

On Fig. 7 are plotted curves showing hp N, discharge Q, and percent efficiency under a fixed head at speeds from 0 to normal and to full runaway speed n_r, based on the characteristics of a typical Francis type runner. N and Q are here shown in percent of full load values, and the speed is shown 100% for normal speed. Naturally at zero speed and runaway speed, $n=0$ and $n=n_r$, the efficiency is 0, but not so with the discharge Q. It will be noted that with this particular type of runner, the horsepower suffers a material loss on overspeeds and on underspeeds.

Again it must be stated that each runner has an individual characteristic. With propellers, for instance, the Q curve rises up to a certain speed increase and drops only after a certain increase is exceeded.

The above should suffice to prove the complexity of the problem of speed control.

Factors Influencing End Result

The various factors influencing the end result can thus be expressed as follows:

ϕ actual output factor, or ratio of output in question, to full rated normal output of turbine (for full output $\phi=1$).

ϕ_{nH} corrective factor for change in output due to speed change (see Fig. 6).

ϕ_r output factor during process of reduction of output by the governor during the time T_n.

ϕ_H corrective factor for change in output during change of head. For head increase $H=\left(1+\frac{\Delta H}{H}\right)^{3/2}$

For head decrease $H=\left(1-\frac{\Delta H}{H}\right)^{3/2}$

T_F dead flyball time, during which the output is not changed by the governor.

T_0 time taken to start moving the control mechanism of the turbine until maximum rate of movement is reached.

T_n actual total time of gate movement at maximum rate

$$T_F+T_0+T_n=T_g$$

The phases of action producing the energy changes are:

(1) Area ABFD (see Figs. 3, 4, and 8)$=\Delta E_1$.

(2) Area DEGF (see Fig. 8)$=\Delta E_2$.

(3) Area ECG (Figs. 3 4, and 8)$=\Delta E_3$. In Fig. 8 these values are shown for $\phi=1.0$, and $\phi=0.5$.

For simplicity of illustration of formula it is now assumed:

(a) That no change in head H takes place during the course of closing ($\phi_H=1.0$).

(b) That the change in speed n does not affect the output N ($\phi_{nH}=1.0$) of the turbine.

(c) That the output is reduced linearly during the actual closing time T_n; i. e., that EC is a straight line, ending in N ($\phi_r=0$) at the end of the governor time T_n. In actuality these factors modify line ADMEC, but the correct value of N or ϕ can be determined correctly only by calculating step-by-step and not by an end result equation.

With special reference to the simplified conditions, represented by Fig. 8 we have:

(1) $\Delta E_1=T_F\phi$

(2) $\Delta E_2=T_0\,\phi_0$ where T_0 is an empirical value depending on the time T_n, and in turn fixing ϕ_r.

(3) $\Delta E_3=\frac{T_n\,\phi_r}{2}$

so that in very abbreviated form:

$$E=T_F\,\phi+T_0\,\phi_0+\frac{T_n\,\phi_r}{2}$$

Since T_F and T_0 are relatively short compared to T_n, it can be seen that the major cause of speed change is the value ΔE_3.

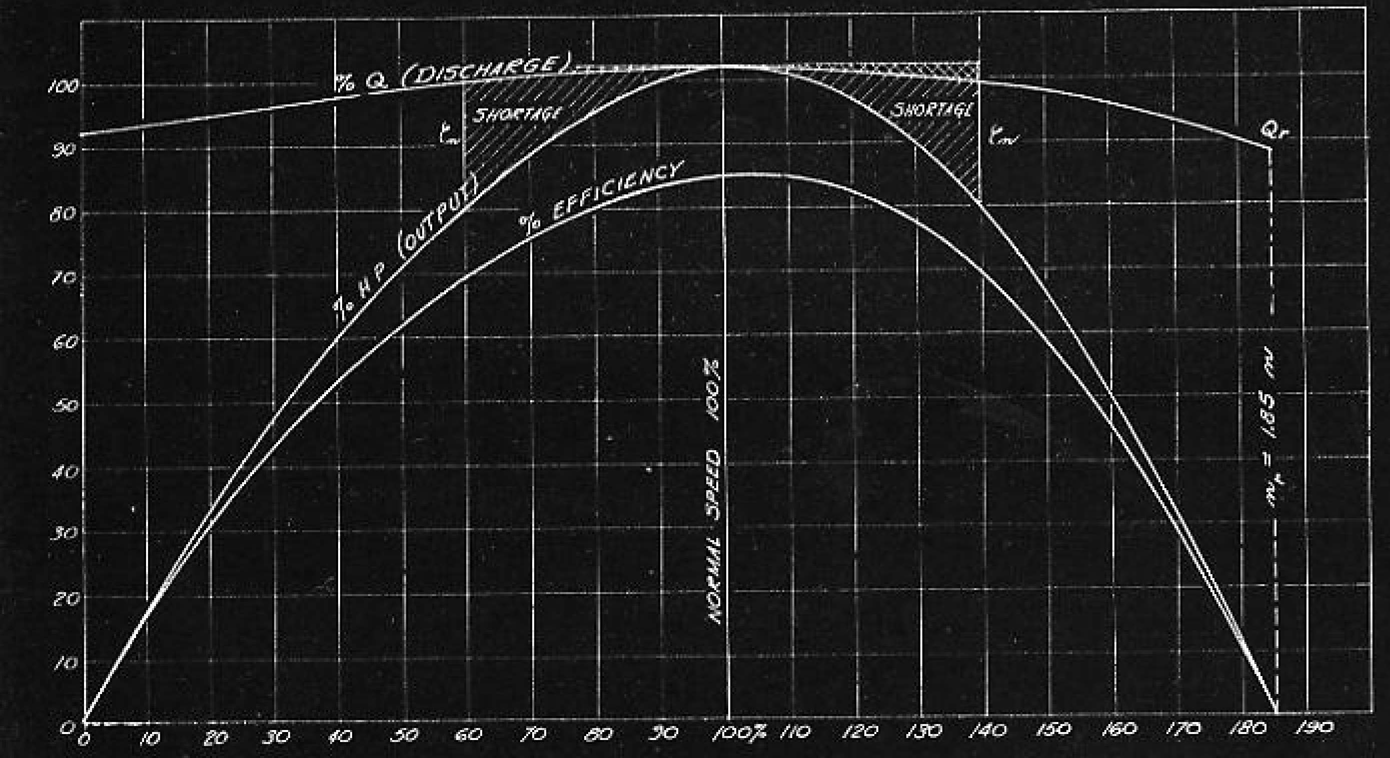


Fig. 7

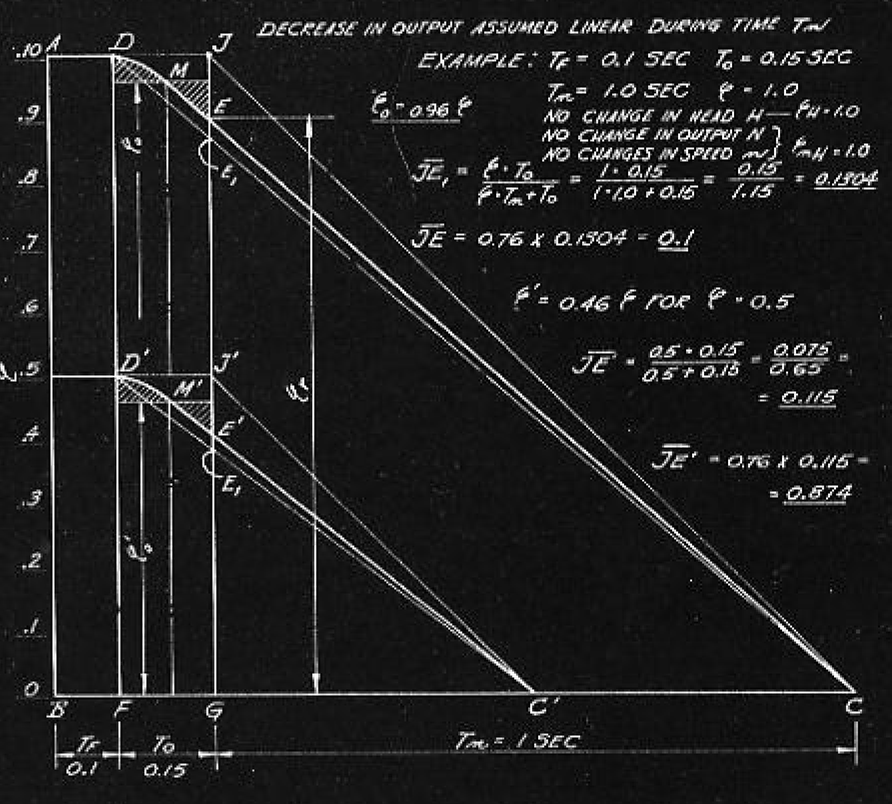


Fig. 8

The value $\frac{T_n \phi_r}{2}$ represents the triangle ECG but also a rectangle with base T_n and constant height $\frac{\phi_r}{2}$. This is illustrated in Fig. 9, where ΔE_3 is divided into four sections, III to VI.

If ΔE_3 is calculated according to the end formula $E_3 = \frac{T_n \phi_r}{2}$, the end result will not be quite the same if each section is figured individually; and the individual points will be on a different curve, as shown on Fig. 9a.

For speed change due to load rejection, the formula previously developed is:

$$WR^2 (n_2^2 - n_1^2) = 3{,}232{,}991 \int_{t=0}^{t=T_g} N\,dt$$

$$n_2^2 = n_1^2 + \frac{3{,}232{,}991}{WR^2} N_r \int_{t=0}^{t=T_g} \phi_r\,dt$$

$$\frac{n_2}{n} = \sqrt{\left[\frac{n_1}{n}\right]^2 + \frac{3{,}232{,}991}{C} \phi_a T_g}$$

For speed rise

$n_2 = n + \Delta n$ and $\frac{n_2}{n} = 1 - \frac{\Delta n}{n}$ where $\frac{\Delta n}{n}$ is relative speed rise.

Setting $N = N_r \phi_r$

N_r being rated horsepower output

ϕ the output factor at times

$\frac{WR^2 n^2}{N_r} = C$, flywheel constant.

$$\int_{t=0}^{t=T_g} \phi_r\,dt = \phi_a T_g$$

ϕ_a being the average output factor

n_2 being speed after load change

n_1 being speed before load change.

Thus:

Relative speed rise

$$\frac{\Delta n}{n} = \frac{n_2}{n} - 1 = \sqrt{\left[\frac{n_1}{n}\right]^2 + \frac{3{,}232{,}991}{C} \phi_a T_g} - 1$$

Likewise for speed change due to load increase:

$$n_1^2 - n_2^2 = \frac{3{,}232{,}991}{C} \phi_a T_g$$

and

$$\frac{n_2}{n} = \sqrt{\left[\frac{n_1}{n}\right]^2 - \frac{3{,}232{,}991}{C} \phi_a T_g}$$

and since $n_2 = n - \Delta n$,

$$\frac{n_2}{n} = 1 - \frac{\Delta n}{n} \quad \text{or} \quad \frac{\Delta n}{n} = 1 - \frac{n_2}{n}$$

Relative speed drop

$$\frac{\Delta n}{n} = 1 - \frac{n_2}{n} = 1 - \sqrt{\left[\frac{n_1}{n}\right]^2 - \frac{3{,}232{,}991}{C} \phi_a T_g}$$

Applying these to a numerical example with

$n = 225$ rpm $\quad N_r = 16{,}000 \quad C = \frac{WR^2 (225)^2}{16{,}000} = 8{,}081{,}273.$

$$\frac{3{,}232{,}991}{C} = 0.4$$

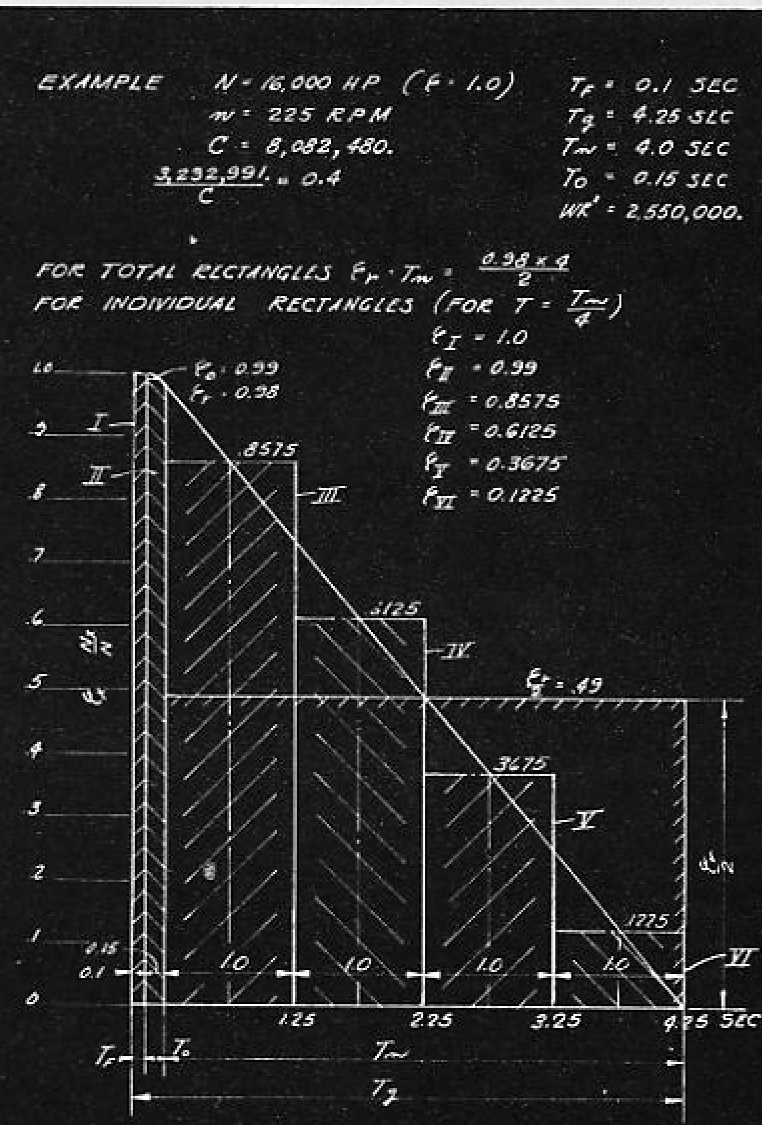


Fig. 9

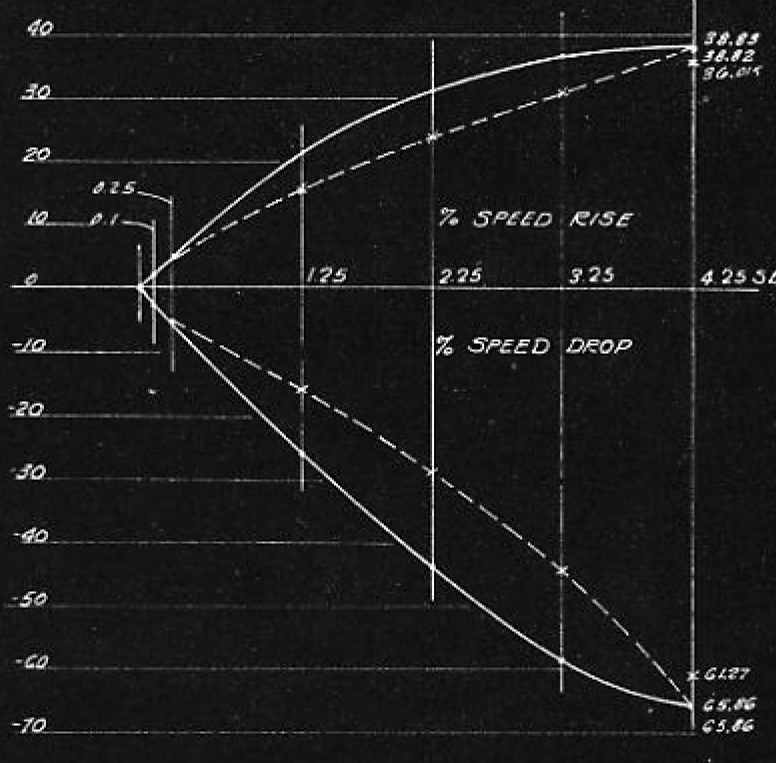


Fig. 9a

for speed rise, full load off: $\quad n_1 = n = 225$

$$\phi_a = \frac{1.0}{2} = 0.5$$

$$\frac{\Delta n_2}{n} = \sqrt{\left[\frac{n_1}{n}\right]^2 + 0.4 \times \frac{1.0}{2} \times 4.25} - 1$$

$$\frac{\Delta n_2}{n} = \sqrt{1 + 0.85} - 1 = 1.36015 - 1 = 0.36015, \text{ or}$$

36.015% relative speed rise on full load rejection of 16,000 hp.

Figuring for each kinetic change individually: ΔE_1, ΔE_2, and ΔE_3

I. $\frac{\Delta n_I}{n} = \sqrt{\underset{(\phi)\ (T_F)}{1+0.4\times1.0\times0.1}} -1= \sqrt{1.04}-1=0.0198$, or 1.98% relative speed rise. $n_1=n$; $T_F=0.1$; $\phi=1.0$ (Fig. 9)

$$n_I=1.0198\times225=229.455 \text{ rpm.}$$

II. $\frac{\Delta n_{II}}{n} = \sqrt{\left[\frac{229.455}{225}\right]^2 +0.4\times0.99\times0.15}-1=$

$$\sqrt{1.04+0.0594}-1= 0.0485, \text{ or } 4.85\%.$$

$$\phi_0=0.99 \quad T_0=0.15 \text{ sec (Fig. 9).}$$

$$n_{II}=1.0485\times229.455=240.583 \text{ rpm.}$$

VI. $\frac{n_{VI}}{n} = \sqrt{1.14319+0.784}-1= \sqrt{1.92719}-1=$

$$\phi_a= \frac{0.98}{2} \quad T_n=4 \text{ sec entire rectangle (Fig. 9).}$$

0.388, or 38.8% relative speed rise on full load rejection as against 36.015% before.

If each portion of interval of 1 sec is figured step-by-step, with $\phi_a= \frac{0.98}{2}$, $T=1$ sec.

III. $\frac{\Delta n_{III}}{n} = \sqrt{\underset{(=1.1439)}{\left[\frac{240.583}{225}\right]^2}+0.4\times \frac{0.98}{2}\times1} -1= \sqrt{1.3391}-1=$

1.1572−1=0.1572, or 15.72% relative speed rise after 1.25 sec.

$$n_{III}=1.1572\times225=260.35 \text{ rpm.}$$

IV. $\frac{\Delta n_{IV}}{n} = \sqrt{\underset{(=1.3391)}{\left[\frac{260.35}{225}\right]^2}+0.4\times \frac{0.98}{2}\times1} -1= \sqrt{1.5351}-1=$

1.239−1=0.239, or 23.9% relative speed rise after 2.25 sec.

$$n_{IV}=1.239\times225=278.775 \text{ rpm.}$$

V. $\frac{\Delta n_V}{n}= \sqrt{\underset{(=1.5351)}{\left[\frac{278.775}{225}\right]^2}+0.196} -1= \sqrt{1.7311} -1=1.30808-1=$

0.30808, or 30.808% relative speed rise after 3.25 sec.

$$n_V=1.30808\times225=294.42 \text{ rpm.}$$

Total relative speed change:

VI. $\frac{n_{VI}}{n}\sqrt{\underset{(=1.7311)}{\left[\frac{294.42}{225}\right]^2} +0.196} -1= \sqrt{1.9271} -1=1.3882-1=$

0.3882, or 38.82% as against 36.015% before.

Or, taking the respective rectangles (Fig.9):

$\phi_{III}=.8575$ $\phi_{IV}=0.6125$ $\phi_V=0.3675$ $\phi_{VI}=0.1225$

$$T=1 \text{ sec}$$

I and II same as before.

$\frac{\Delta n_I}{n}=1.98\%$ $n_I=229.455$ rpm.

$\frac{\Delta n_{II}}{n}=4.85\%$ $n_{II}=240.583$ rpm.

III. $\frac{\Delta n_{III}}{n} = \sqrt{\left[\frac{240.583}{225}\right]^2 +0.4\times0.8575\times1.0} -1= \sqrt{1.48619}$ $-1=1.2190-1$, or 0.2190 or 21.90% as against 15.72% before.

$$n_{III}=1.219\times225=274.25 \text{ rpm as against 260.35 rpm.}$$

IV. $\frac{n_{IV}}{n} = \sqrt{\left[\frac{274.25}{225}\right]^2 +0.4\times0.6125\times1} -1= \sqrt{1.731192} -1=$ $1.3157-1=0.3157$, or 31.57% relative speed rise after 2.25 sec as against 23.9% before.

$$n_{IV}=1.3157\times225=296.03 \text{ rpm.}$$

V. $\frac{n_V}{n} = \sqrt{\left[\frac{296.03}{225}\right]^2 +0.4\times0.3675\times1} -1= \sqrt{1.87819} -1=$

1.3704−1=.3704, or 37.04% relative speed rise after 3.25 sec as against 30.808%.

$$n_{VI}=1.3704\times225=308.34 \text{ rpm.}$$

VI. $\frac{n_{VI}}{n} = \sqrt{\left[\frac{308.34}{225}\right]^2 +0.4\times0.1225\times1} -1= \sqrt{1.92719} -1=$ $1.3883-1=0.3883$, or 38.83% total relative speed rise after 4.25 sec as against 38.82% before.

Likewise for speed drop

$\frac{\Delta n_2}{n} =1- \sqrt{\left[\frac{n_1}{n}\right]^2 -0.4\times \frac{1.0}{2} \times4.25}$. Here again $n_1=n=$

$1- \sqrt{1-0.85} =1- \sqrt{0.15} =1-0.3873=0.6127$, or 61.27% total relative speed drop.

If each portion of ΔE_1, ΔE_2, and ΔE_3 is figured step-by-step,

I. $\frac{\Delta n_I}{n}=1-\sqrt{\underset{(\phi)\ (T_F)}{1-0.4\times1.0\times0.1}} =1- \sqrt{1-0.04} =1- \sqrt{0.96} =$ $1-0.9798=0.0202$, or 2.02% relative speed drop due to insensitiveness of flyballs.

II. $\frac{n_{II}}{n}=1- \sqrt{\underset{(=0.96)}{\left[\frac{220.455}{225}\right]^2} -0.4\times0.99\times0.15} =1- \sqrt{0.9006} =$ $1-0.94899=0.05101$ or 5.101% relative speed drop after 0.25 sec.

$$n_{II}=0.94899\times225=213.52 \text{ rpm.}$$

VI. $\frac{n_{VI}}{n} =1- \sqrt{\underset{(=0.9006)}{\left[\frac{213.52}{225}\right]^2} -0.4\times \frac{0.98}{2}\times4} =1- \sqrt{0.1166} =$ $1-0.3414=0.6586$, or 65.86% full relative speed drop after 4.25 sec.

If each portion of interval of 1 sec is figured in steps,

I. $\frac{\Delta n_I}{n}=2.02\%$ $n_I=220.455$ rpm as before.

II. $\frac{\Delta n_{II}}{n}=5.101\%$ $n_{II}=213.52$ rpm as before.

III. $\frac{\Delta n_{III}}{n}=1- \sqrt{\left[\frac{213.52}{225}\right]^2 -0.4\times \frac{0.98}{2}\times1} =1-\sqrt{0.7046} =$ $1-0.8394=0.1606$ or 16.06% relative speed drop after 1.25 sec.

$$n_{III}=0.8394\times225=188.86 \text{ rpm.}$$

IV. $\frac{\Delta n_{IV}}{n} =1- \sqrt{\left[\frac{188.86}{225}\right]^2 -0.4\times\frac{0.98}{2} \times1} =1- \sqrt{0.5086} =$ $1-0.7131=0.2869$ or 28.69% relative speed drop after 2.25 sec.

$$n_{IV}=0.7046\times225=160.4475 \text{ rpm.}$$

V. $\frac{\Delta n_V}{n} =1- \sqrt{\left[\frac{160.4475}{225}\right]^2 -0.4\times \frac{0.98}{2}\times1} =1- \sqrt{0.3126} =$ $1-0.5591=0.4409$ or 44.09% relative speed drop after 3.25 sec.

$$n_V=0.5591\times225=125.7975 \text{ rpm.}$$

$$\text{VI.}\quad \frac{\Delta n_{VI}}{n}=1-\sqrt{\left[\frac{125.7975}{225}\right]^2-0.4\times\frac{0.98}{2}\times 1}=1-\sqrt{0.1166}=$$

1−0.3414=0.6586 or 65.86% total relative speed drop for full load rejection (same as above).

Or taking the respective rectangles (Fig. 9)

$\phi_{III}=0.8575$ $\phi_{IV}=0.6125$ $\phi_{V}=0.3675$ $\phi_{VI}=0.1225$

I and II same as before.

$\frac{\Delta n_I}{n}=2.02\%$ $n_I=220.455$ rpm.

$\frac{\Delta n_{II}}{n}=5.101\%$ $n_{II}=213.52$ rpm.

$$\text{III.}\quad \frac{\Delta n_{III}}{n}=1-\sqrt{\left[\frac{213.52}{225}\right]^2-0.4\times 0.8575\times 1}=1-\sqrt{0.5576}=$$

1−0.74688=0.25312, or 25.312% relative speed drop after 1.25 sec (as against 21.9% before).

$n_{III}=0.74688\times 225=168.048$ rpm.

$$\text{IV.}\quad \frac{\Delta n_{IV}}{n}=1-\sqrt{\left[\frac{168.048}{225}\right]^2-0.4\times 0.6125\times 1}=1-\sqrt{0.3126}=$$

1−0.5591=0.4409, or 44.09% relative speed drop after 2.25 sec.

$n_{IV}=0.5591\times 225=125.795$ rpm.

$$\text{V.}\quad \frac{\Delta n_{V}}{n}=1-\sqrt{\left[\frac{125.795}{225}\right]^2-0.4\times 0.3675\times 1}=1-\sqrt{0.1656}=$$

1−0.4069=0.5931, or 59.31% relative speed drop after 3.25 sec.

$n_V=0.4069\times 225=91.55$ rpm.

$$\text{VI.}\quad \frac{\Delta n_{VI}}{n}=1-\sqrt{\left[\frac{91.55}{225}\right]^2-0.4\times 0.1225\times 1}=\sqrt{0.1166}=$$

1−0.3414=0.6586, or 65.86% for total relative speed drop after 4.25 sec (same as before).

Analyzing the results of the preceding computations, it becomes evident that the end result formula, taking both the full governor time of T_g (4.25 seconds) and $\phi=1.0$, the relative speed rise and speed drop are short of actual value 36.015% and 61.27%, respectively. Figuring the value of ΔE_1 and ΔE_2 individually and the value of ΔE_3, in one additional end result there are obtained 38.8% for relative speed rise and 65.86% for relative speed drop. Figuring each individual section I to VI, the end results are identical whether figured as four equal rectangles of $\phi=0.98$ each, or as four rectangles of $\phi=0.585$, 0.6125, 0.3675, 0.1225 each. However, it will be noted from Figure 9a that this last computation results in an asymptotic approach of the maximum relative speed rise 38.83% and 65.86% relative speed drop. This is correct because output and load have become balanced at that point.

The results of the various calculations plotted on Fig. 9a proved that end formulae are not sufficiently accurate but that calculations in successive steps only are reliable. The step-by-step method also permits the application of corrections for relative change in head $\frac{\Delta H}{H}$ due to change in discharge Q by reason of governor action, of change in head and of change in speed, change in output (ϕ_{nII}) due to change in speed, which cannot be calculated by an equation leading directly to the end result.*

PRESSURE CONTROL—WATER HAMMER

When the flow of water through the turbine is completely stopped on a sudden full load rejection, such as takes place when the unit becomes disconnected from the network, thereby causing the governor to shut off the water, the kinetic energy of the water produces a pressure increase in the conduit admitting the water to the turbine. This pressure increase is called water hammer and is affected by a number of factors, such as length of conduit, velocity of water flowing through conduit, material of conduit and time and rate at which the water is decelerated.

The flow of water in a conduit may be compared with a column of soldiers marching at uniform speed. When the front end is stopped without notice to the entire marching column, there will be a congestion at the front end, and the crowded soldiers are apt to react and push back the still-marching neighbors. The same phenomena takes place in a conduit, and the reacting wave travels at a velocity "a" which is called the speed of the wave.

$$a=\sqrt{\frac{g}{W\left(\frac{1}{K}+\frac{D}{Eb}\right)}}\quad\text{where:}$$

a=velocity of wave in ft/sec

f=acceleration: 32.2 ft/sec/sec

W=density of water in lb/cu ft

K=modulus of elasticity of water in compression

E=modulus of elasticity of pipe line material in tension

D=diameter of conduit in ft

b=thickness in ft of metal of conduit or in abbreviated form:

$a=\frac{4660}{\sqrt{1+0.01\frac{D}{b}}}$ from which it is evident that the value of wave velocity "a" depends on the ratio of conduit diameter to thickness of material of conduit. For large diameters and light material such as are involved with conduits for moderate head installations, the wave velocity "a" is materially lower than that of a high head pipe line and one of relatively small diameter D. The time required for the wave to travel to the top inlet of the conduit and return to the point of origin, the lower end of the conduit, is:

$T_c=\frac{2L}{a}$, where L is the length of the conduit in feet. This time is called the critical time. When the flow is stopped within this time T_c, the pressure rise in feet in the conduit is a maximum and is $\Delta H=\frac{av}{g}$ where v the velocity of water in ft/sec at the beginning of closure. It is directly proportionate to the velocity v. The relative pressure rise is $\frac{\Delta H}{H}=\frac{av}{gH}$, where H is the head in feet. Naturally the lower H the higher is the relative pressure rise.

Where the pipe consists of sections of different diameters, "a" must be figured individually for each section, and the average value $a_m=\frac{L_1a_1+L_2a_2+L_na_n}{L_1+L_2+L_n}$.

Example

Assume H=50 ft

v=6 ft/sec

a=3,000

Pipe line diameter the same over entire length L.

Thus: $\frac{\Delta H}{H}=\frac{3000\times 6}{32.2\times 50}=11.2.$

The relative pressure rise $\frac{\Delta H}{H}$ will be 11.2 times the head H.

Assume a length L of pipe line of 500 ft. Then $\frac{2L}{a}=\frac{2\times 500}{3000}$ =1/3 sec. Therefore, if the complete closure takes place in

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* For explicit step-by-step calculations, refer to the excellent paper by C. R. Strowger and L. S. Kerr, Ass. M. A. S. M. E., "Speed Changes of Hydraulic Turbines for Sudden Changes of Loads," spring meeting, A. S. M. E., San Francisco, Calif., 1926.

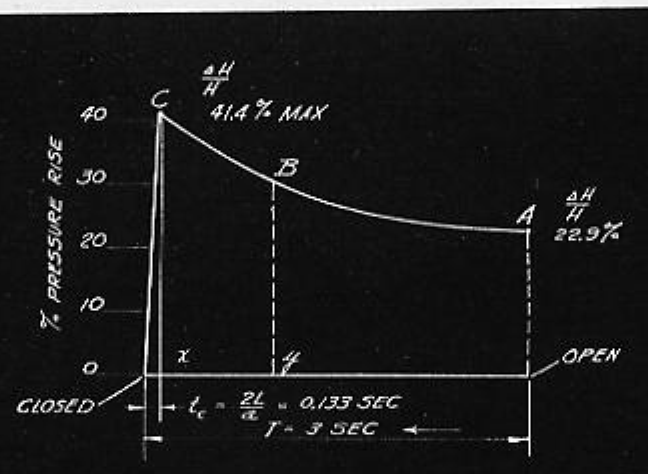


Fig. 10

1/3 sec or less, the pressure will rise to 11.2 times its original value. This is a water hammer which no normally built pipe could withstand. For a pipe of 5000 ft length the critical time T_c or $\frac{2L}{a}$ is 3 sec. To avoid such destructive pressure rises or water hammers, it is necessary to decelerate the water in the conduit at a materially slower rate. When water is decelerated in the conduit at a constant rate $T > \frac{4L}{a}$ the pressure rise takes place along a curve as illustrated in Fig. 10. At the beginning A of the closure at a steady rate T, the relative pressure rise is $\frac{\Delta H}{H}$ and can be figured according to formulae and tables set up by Professor Dr. Allievi, as follows: $\frac{\Delta H}{H} = \frac{n}{2}(n \pm \sqrt{n^2+4})$ where $n = \frac{Lv}{gHT}$.

This formula is sufficiently accurate only if $T > \frac{4L}{a}$. It deviates materially from actual results as T becomes shorter.

For example:

$L = 200$ ft

$v = 6$ ft/sec

$H = 60$ ft

$a = 3000$

$T = 3$ sec

$g = 32.2$ ft/sec/sec

$$\frac{2L}{a} = \frac{400}{3000} = 0.133 \text{ sec}$$

$$n = \frac{Lv}{gHT} = \frac{200 \times 6}{32.2 \times 60 \times 3} = 0.207$$

thus:

$$\frac{\Delta H}{H} = \frac{0.207}{2}\left(0.207 + \sqrt{(0.207)^2 + 4}\right)$$

or $\frac{\Delta H}{H} = 0.229$, or 22.9%.

As the flow is reduced at the rate of T sec over the full deceleration, the pressure rise $\frac{\Delta H}{H}$ increases and reaches a maximum value at C or $\frac{\Delta H_r}{H}$ when $T - \frac{2L}{a}$ sec are spent out of the entire T sec. The maximum pressure rise can be figured from formula:

$\frac{\Delta H_r}{H} = \frac{2Lv}{gHT}$. In our example it is $\frac{2 \times 200 \times 6}{32.2 \times 60 \times 3} = 0.414$ or 41.4%.

It can be seen that the maximum pressure rise is $\frac{41.4}{22.9} = 1.8$ or 80% greater than the pressure rise at the beginning of closure. If the pressure rise $\frac{\Delta H_r}{H}$ should not exceed the pressure rise $\frac{\Delta H}{H}$, then the rate of closure at point C (Fig. 10) would have to be:

$$T_r = \frac{2Lv}{gH\frac{\Delta H}{H}} = \frac{2 \times 200 \times 5}{32.2 \times 60 \times 0.229} \text{ or } 5.42 \text{ sec and not only 3 sec.}$$

Therefore, to obtain a fixed relative pressure rise $\frac{\Delta H}{H}$ over the entire period T of closure — i. e., over the entire range of stroke of the mechanism controlling the flow through the turbine — it is necessary to provide means for fixing the rate at which the control mechanism is moved over its entire stroke.

This was accomplished successfully in 1923, and the device called "rate-limiting device" has become a generally adopted standard device protected by valid U. S. patents granted to Allis-Chalmers. It was first used on the three 18,000 hp, double-overhung impulse wheels for Nakatsugawa Plant No. 2 in Japan. The units are of the double-overhung type, each wheel per unit having one jet directly actuated by its own governor.

While tests showed that the maximum relative pressure rise $\frac{\Delta H}{H}$ for full load rejection (full deceleration of water in pipe line) was well within the guarantee (point A of Fig. 10), it was found that on partial load rejections Y the relative pressure rise at B exceeded the maximum value guaranteed. The governor actuating the needle controlling the flow of water was set for a rate T of 3½ sec. Investigation directed from this end by mail disclosed that at intermediate needle positions the rate of closure was not longer but even somewhat shorter than 3½ sec, which at once explained the excessive pressure rise.

Since the rate of movement of the needle by the servomotor (regulating cylinder and piston moving therein and actuating the needle) depends upon the quantity of oil supplied by the governor into and out of the cylinder, all that was required was to provide means whereby this quantity could be delivered at such a rate as to cause the piston (needle-flow of water) to move at the proper rate toward zero position of needle. This device has proved of value also in connection with remote automatic control because it permits starting a turbine promptly to attain normal speed, then practically stopping further increase in discharge; i. e., further increase in speed of the unit until it is synchronized with the network, after which full load can be picked up as rapidly as is practicable.

In a similar measure, as pressure rises take place on deceleration of water due to load rejections, pressure decreases occur on acceleration of water due to load increases. Pressure rise, if excessive, may have a destructive effect on the pipe line, many serious accidents being on record. Likewise pressure decreases may cause a collapse of the pipe line. In both cases such pressure changes affect the speed control of a unit and must, therefore, be limited to practical values. Pressure variations exceeding 30% should be avoided in normal operation.

It can be readily seen that pipe lines of considerable length, or involving high water velocities, would require relatively slow rates T; and, since the momentary speed change is directly dependent on this rate and also on the flywheel effect of rotating parts of the unit, it follows that, to hold the momentary speed changes within commercially satisfactory limits, large flywheel effects (WR^2 — moment of inertia) are involved. This is particularly so with moderate and high head developments.

Example

Head $H = 800$ ft

Length of pipe $L = 5{,}000$ ft

Velocity $v = 15$ ft/sec

$a = 3{,}600$ ft/sec

Thus:

$T_c = \frac{2L}{a} = \frac{2 \times 5000}{3{,}600} = 2.8$ sec critical time. Maximum pressure rise for T_c:

$$\frac{av}{gH} = \frac{36000 \times 15}{32.2 \times 800} = 2.1 \text{ or } 210\% \text{ of normal.}$$

If the flow were stopped in 2.8 sec, a destructive pressure rise would result. Allowing a maximum pressure rise of only 20% above normal, the rate of closure would have to be

$$T_r = \frac{2Lv}{gH\frac{\Delta H}{H}} = \frac{2 \times 5000 \times 15}{32.2 \times 800 \times 0.20} \text{ or } 29.2 \text{ sec.}$$

It is evident that, at such a slow rate of closure, the unit would assume full runaway speed if detached from the network while carrying full load. Likewise, if it remained connected to the network and if the flywheel effect of the network were only moderate, it would cause a serious increase in the speed or frequency of that network.

To obtain a practical picture of the situation, a concrete example may serve, based on the data previously used.

Pressure rise 20%. This increase in head, $\frac{\Delta H}{H} = 0.2$, increases the output $\left(\frac{\Delta H}{H}\right)^{3/2} = 0.3$ or 30%. The governor time is 29.2 sec net, as concerns effect on penstock, or 29.45 sec, taking in consideration 0.25 sec partially dead time ($T_F = 0.1$ and $T_0 = 0.15$, or total 0.25). The simplified formula for speed rise as used before is $\frac{n_2}{n_1} = \sqrt{1 + \frac{3,240,000}{C} \frac{T_g}{2} 1.3}$, where 1.3 is output increase due to increased head. Solving for C we obtain:

$$C = \frac{2,106,000 T_g}{\left(\frac{n_2}{n_1}\right)^2 - 1}$$

Assuming a speed rise of 36%, $\frac{n_2}{n_1} = 1.36, \left(\frac{n_2}{n_1}\right)^2 - 1 = 0.8496.$

Since $T_g = 29.45$ sec,

$$C = \frac{2,106,000 + 29.45}{0.8496} = 73,000,000 \text{ (slide rule figure).}$$

The flywheel constant requires an exceedingly great WR^2 of rotating masses.

For standard designs of generators the value of C varies between 2.5 and 5 million, depending on the size and type of generator. Higher values of C require a special design of rotor or an auxiliary flywheel.

With a moderate flywheel constant, therefore, the change in velocity would have to be considerably reduced from the value, V=15 ft/sec of the example. In other words, only a moderate portion of load could be rejected.

Auxiliary Relief—Pressure Regulators

An auxiliary relief (or pressure regulator) must be provided which by-passes the major portion of the water shut off from the turbine, temporarily if water-saving or permanently if water-wasting. If the governor had to close in 2.5 sec in order to maintain a reasonably low speed rise with a flywheel constant of only 5 million for example, then the allowable change of velocity in the example would have to be for a maximum 20% pressure rise:

$$\frac{\Delta H_r}{H} = \frac{2L\Delta V}{gHT_r} \text{ or } \Delta V = \frac{\Delta H_r gHT_r}{H2L} = \left(\frac{\Delta H_r}{H}\right)\frac{HgT_r}{2L}$$

$$\text{or, } \Delta V = \frac{0.2 + 800 \times 32.2 \times (2.5 - 0.25)}{2 \times 5,000} = 1.16 \text{ ft/sec}$$

or $\frac{100 \times 1.16}{15} = 7.73\%$ of the full flow velocity of 15 ft/sec of the example. Thus 100−7.73 or 92.27% of the full flow (or full gate turbine discharge) would have to be by-passed through the relief.

With impulse wheels this relief device can be a jet deflector, which simply directs the water jet away from the buckets without change of flow in the pipe line. To save water all that is necessary then, in the example, is to close the flow controlling needle at a rate of 29.2 sec to avoid pressure rises exceeding 20%. It is evident that this deflector, since it does not affect the water velocity in the pipe, can be moved as fast as the governor can move it so that, with 1 to 1½ sec governor time, the maximum relative speed rise of our example would be much less than 36%. Hence a lighter WR^2 of rotating parts can be allowed for the limiting value of 36%.

With Francis and propeller type turbines the water cannot be directed away from the runner. It must, therefore, be directly controlled by the governor according to load demand. Here a governor-actuated pressure regulator is required, which discharges the difference between full-load discharge and discharge required for the output called for by the load. This auxiliary acts either as a synchronous by-pass, wasting water when the turbine does not carry full load, or as a water-saving device, which temporarily wastes water and closes automatically at a rate slow enough not to exceed the permissible pressure rise in the pipe line.

If the rate of discharge of the pressure regulator is inverse to the rate of discharge of the turbine at all times, then no change of water flow takes place in the pipe line; and the effect is the same as that obtained with a jet deflector of the impulse wheel. This is illustrated in Fig. 11 for water-wasting and in Fig. 11-A for water-saving action.

If, however, the discharge rates Q_T and Q_r are unequal, the resultant effect will be a change in velocity in the pipe line, causing either a pressure rise or a pressure drop, as shown in Fig. 12. The turbine discharges along curve ABC, and the pressure regulator along DBE so that the total discharge $Q_{Tx} + Q_{rx} > Q_T$ because 2BX or XF is greater than AD or CE. From A to F excess water is drawn through the pipe line (XF>AD) causing a pressure drop. From F to E the flow decreases from XF to EC, causing a pressure drop. From F to E the flow decreases, causing a deceleration of velocity in the pipe and resulting in a pressure rise, although

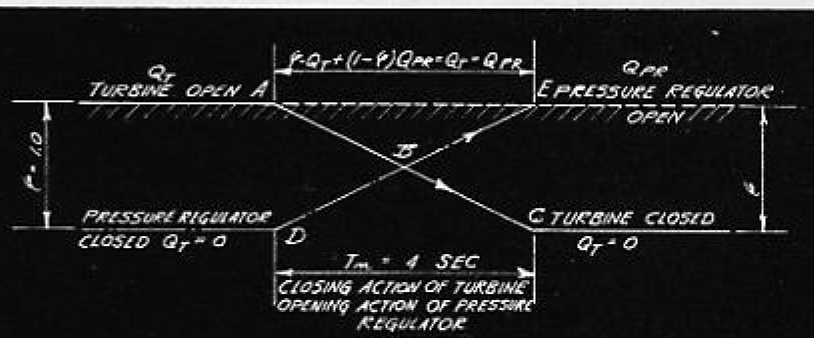


Fig. 11

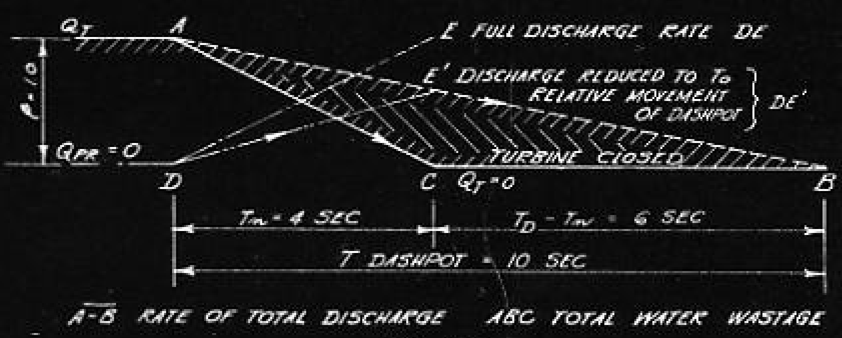


Fig. 11a

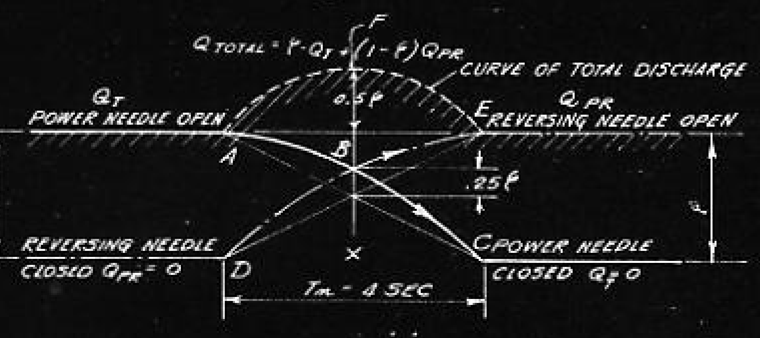


Fig. 12

the full discharge of the pressure regulator is equal to the full turbine discharge. If this discrepancy of synchronous flow is considerable, serious pressure disturbances may be produced in the pipe line.

This has been experienced with reversing needles of impulse wheels of the older type, as illustrated in Fig. 12. This design consisted of a needle directing the jet upon the buckets and a relief needle discharging free. Both needles are so interconnected that one opens when the other closes. An oil dashpot is provided on the relief needle to cause its gradual closure if water cannot be wasted. Any failure of the relief to open while the discharge through the turbine is reduced will produce a serious pressure rise in the pipe.

For instance, if the velocity of 15 ft/sec in the example were reduced to zero in 3 sec with relief valve failing to open, a pressure rise of about 200% of normal would result. In cases of such serious nature it is imperative that the design be such that, in case of failure of the relief to open, it causes a retardation of the closing rate of the governor — in the above example from 3 sec to 29.2 sec so that the relative pressure rise will then be only 20% and not 100%.

Such a design has been placed in eminently satisfactory service with Allis-Chalmers Units A6 and A7 at the Boulder Plant of the U. S. Reclamation Service. These pressure regulators discharge about 80% of the full discharge of the respective turbines, causing a deceleration of 20% of the full load velocity of water through the pipe line. They open as fast as the governor shuts off the water from the turbine and can do so in 3 sec, then discharging about 2,000 cfs each, equivalent to an output of 92,000 hp. They are set to close gradually in 50 sec. Tests were made with the pressure regulator blocked closed to imitate failure of action. The full load of 85,000 kw was then dropped, but the governor was retarded by the power dashpot of the pressure regulator and did not shut the water off from the turbine in 3 sec as is normally the case with the pressure regulator in action, but in 48 seconds. The power dashpot produced a counter action on the servomotor of the turbine equal to about 300,000 ft lb.

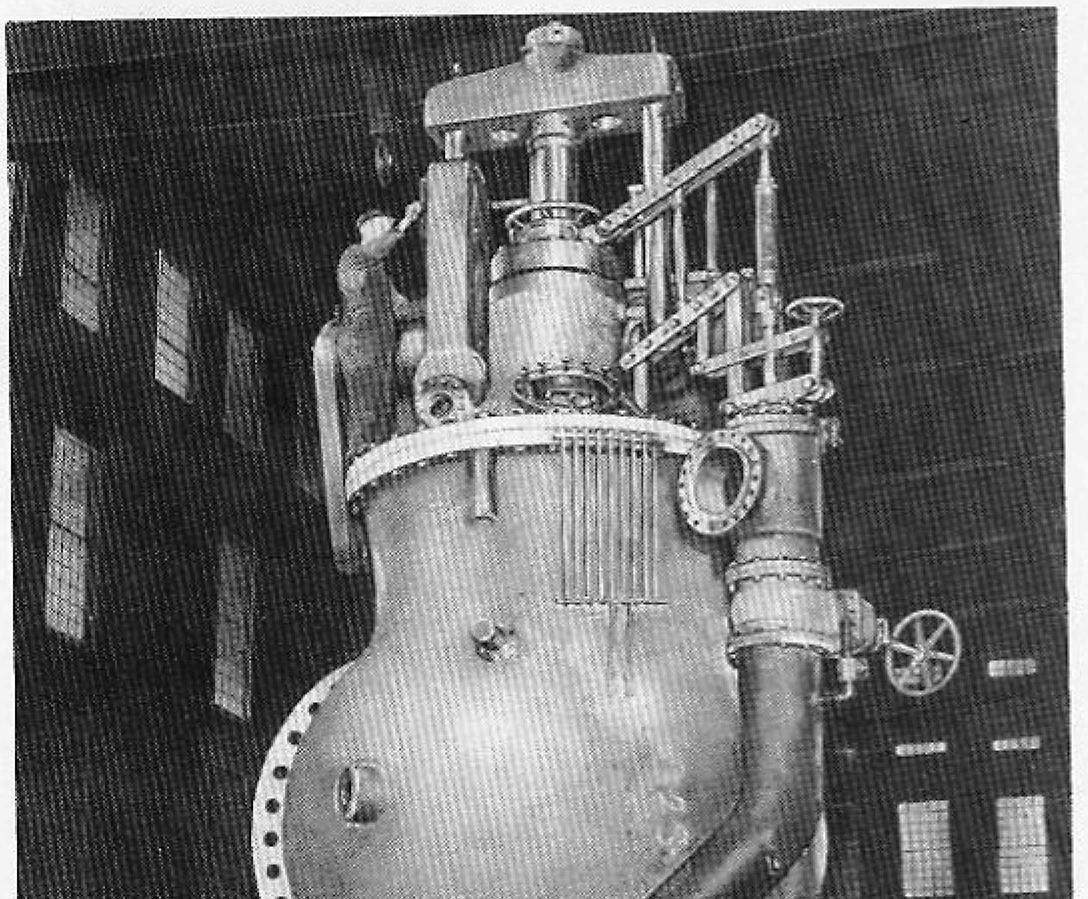


Fig. 13 — Vital to efficient, safe hydraulic turbine operation is this scientifically-engineered pressure regulator.

Figure 13 shows one of these pressure regulators assembled at the Company's works before shipment. Fig. 14 shows the connection between pressure regulator and turbine. The crank is adjustable for the purpose of varying the rate of discharge of the pressure regulator in keeping with the operating head. When the pressure regulator stalls the turbine control mechanism, the connecting rod is subjected to a tension force of about 600,000 lb.

The 92,000 hp passed through the pressure regulator would cause considerable disturbance in the tailrace due to the kinetic energy contained in the 2,000 cfs discharging through the orifice of the pressure regulator at high velocity. To prevent destructive effect, "energy translators" are provided, in which the velocity of water is greatly reduced before discharging into the tailrace. The kinetic energy thus freed is transformed into heat; however, the temperature of the discharging water is thereby raised only about 0.2 F because about 2,000 cfs of water are on hand to absorb this heat. Fig. 15 shows the condition of the tailrace with pressure regulator wide open.

For the same reason that the water must be decelerated very gradually (in the example 29.2 sec) in order to avoid excessive pressure rise, it is necessary to increase only gradually the discharge through the turbine and conduit on sudden load increases. To avoid a pressure drop exceeding 20% in the example, the needle should not be opened in less than about 18 sec. With a flywheel constant C of 5 million, the momentary relative speed drop would be according to previously given formulae:

$$\frac{\Delta n}{n} = 1 - \sqrt{1 - \frac{3{,}240{,}000}{C}\left(1 - \frac{\Delta H}{H}\right)^{3/2} \int_{t=0}^{t=T_g} \phi \, dt}$$

Where $\left(1 - \frac{\Delta H}{H}\right) = 0.8$ for 20% pressure drop, $C = 5{,}000{,}000$. $T_g = 18$ sec for $\phi = 1.0$.

Thus:

$$\int_{t=0}^{t=T_g} \phi \, dt = \frac{1 \times T_g}{2} = \frac{1 \times 18}{2} = 9.$$

$$\frac{\Delta n}{n} = 1 - \sqrt{1 - \frac{324}{500} \times 0.716 \times 9}$$

$$\frac{\Delta n}{n} = 1 - i \times 1.782$$

$$\frac{\Delta n}{n} - 1 = -\sqrt{1 - 4.175712} = -\sqrt{-3.175712} = -i \times 1.782$$

In other words, the unit would come to a dead stop (more so since the factor $\phi_n H$, Fig. 7, also enters into the picture) if full load were suddenly thrown on and if the water could be drawn from the pipe line at a rate of only 18 sec for full flow, from zero to full discharge.

From above it can be seen that only "partial" sudden load increases can be taken care of commercially. For a maximum relative momentary speed drop of 36% with a full load rate of governor opening time of 18 sec and a pressure drop of 20%, we would have:

Governor opening time $T_x = \phi_x T_g = \phi_x \times 18$

Approximately:

$$\frac{\Delta n}{n}=0.36=1-\sqrt{1-\frac{324}{500}\times\left(1-\frac{\Delta H}{H}\right)^{3/2}\phi_x\,\phi_x\,\frac{18}{2}}$$

Assuming linear change of output ϕ_x, so that

$$\int_{t=0}^{t=\phi_x T}\phi_x\,dt=\phi_x^2\times\frac{18}{2}=9\,\phi_x^2$$

$$1-\frac{\Delta H}{H}=0.8$$

$$\left(1-\frac{\Delta H}{H}\right)^{3/2}=0.716$$

Thus:

$$0.36=1-\sqrt{1-0.648\times0.716\times9\,\phi_x^2}$$

$$0.64=-\sqrt{1-4.175712\,\phi_x^2}$$

$$(0.64)^2=0.4096=1-4.175712\,\phi_x^2$$

$$-0.5904=-4.175712\,\phi_x^2$$

$\phi_x=\sqrt{\frac{0.5904}{4.175712}}=0.376$ or 37.6% of full load can be picked up with a relative speed drop of 36%.

If 37.6% of full load is picked up by the governor in 18 sec for full governor stroke and 20% pressure drop, assuming linear reduction of output by the governor, the speed drop will be 36% for a flywheel constant of 5,000,000, also assuming that during this speed change the output for constant head does not change.

Under commercial operation, when the unit is tied into the network, a large total flywheel effect WR^2 of rotating masses operating in synchronism is available so that the speed drop will not be as serious. Besides, the load will not come on suddenly — i. e., in 0 sec — as is assumed in the formulae used and as would be nearer the case if the load were carried by a water rheostat test. It is, therefore, not necessary to provide flywheel effect sufficient to permit a sudden full-load increase unless the unit must carry all load alone.

By referring again to Fig. 4 it can be readily seen that for a full load increase within 2.25 sec instead of suddenly, the required total kinetic energy demand would be represented by the areas ABHC — ABH' so that the shortage of energy would be ADEHB — AH'B or about one-half of the sudden full-load increase; and this will, naturally, greatly reduce the resultant speed drop over that of a sudden full-load increase.

Applied to the previous numerical example we would obtain by approximate end result formula:

$$\frac{\Delta n}{n}=0.36=1-\sqrt{1-\frac{324}{500}\left[\left(1-\frac{\Delta H}{H}\right)^{3/2}\phi_x\,\phi_x\,\frac{T_g}{2}-\phi_x\,\phi_x\,\frac{9.0}{2}\right]}$$

For $\left[1-\frac{\Delta H}{H}\right]^{3/2}=0.716$, and $T_g=18$ sec.

$$0.36-1=-0.64=-\sqrt{1-\left(4.175712\,\phi_x^2-2.916\,\phi_x^2\right)}$$

$$=-\sqrt{1-1.259712\,\phi_x^2}$$

$$(-0.64)^2=0.4096=1-1.259712\,\phi_x^2$$

$$-0.5904=-1.259712\,\phi_x^2$$

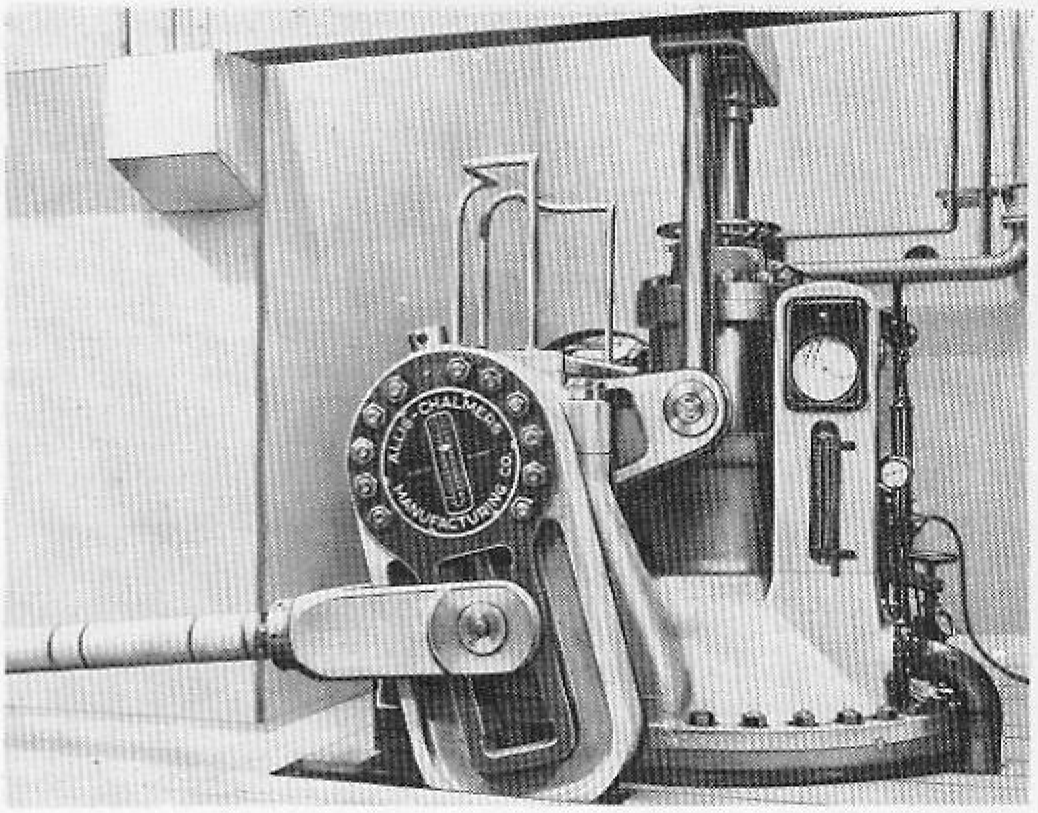


Fig. 14 — Pressure regulator's rate of discharge can be controlled with the mechanism shown. Rate can be adjusted.

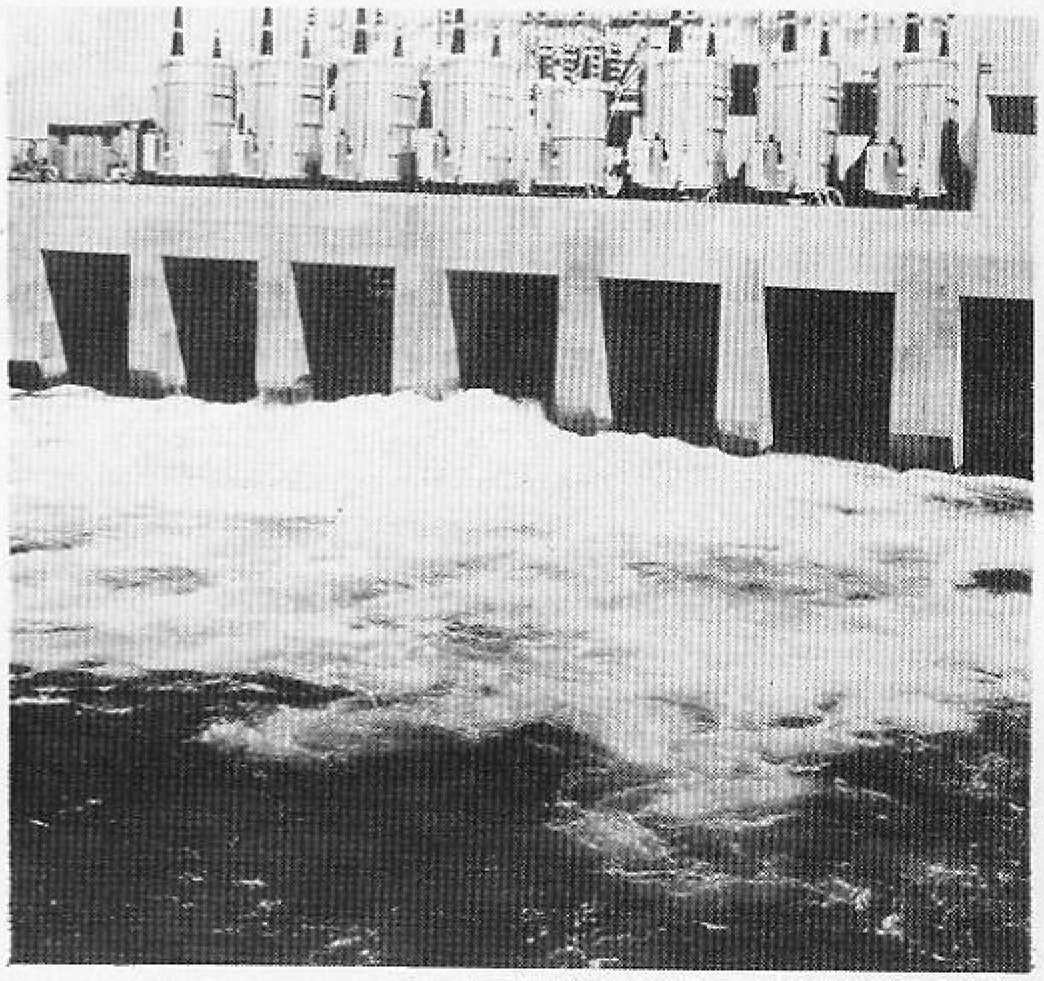


Fig. 15 — With pressure regulator wide open, discharge is about 2000 cfs under 525 ft head at a large western dam.

$\phi_x=\sqrt{\frac{0.5904}{1.259712}}=\sqrt{0.4687}=0.6846$ or 68.46% of full load.

With the load coming on not suddenly but in 9 sec (for full load), the unit can pick up 68.5% of full load as against only 37.6%, or about 82% more than the former value figured for a sudden full-load increase. From all of above it can be realized that, whenever speed control is essential, the governing equipment is the most important accessory of a hydro-electric unit.

Part IV of this article will deal with the complete problem of speed control, taking into consideration the modifications of the kinetic energies caused by pressure changes in the conduit admitting water to the turbine and by the changes of output due to changes in speed. It will conclude this series with comments on other accessories such as gate valves, butterfly valves, etc.

Finish machining gives a glass-smooth surface to the runner for a 35,000 hp Francis type hydraulic turbine destined for TVA.

Part IV

SPEED AND PRESSURE CONTROL [Cont.]
SPECIFIC EXAMPLES

In Part III there was emphasized the necessity of computing speed and pressure variations by the "step-by-step" method, rather than using end formulae, because the latters' accuracy may vary materially in individual cases.

The step-by-step method can best be illustrated with a concrete example. Assume:

Two units operate on one common pressure pipe line.

Length of pipe $(L) = 573$ ft.

Static Head $(H_o) = 180$ ft.

Net Head $(H_n) = 167.5$ ft,

Maximum flow $(Q) = 2 \times 980 = 1960$ cfs.

Maximum water velocity in pipe line $V = 13.217$ ft/sec (based on average pipe area).

Full gate capacity = 16,000 hp each at 86% efficiency ($\varphi = 1.0$).

Speed $(n) = 225$ rpm.

Flywheel effect $(WR^2) = 4{,}377{,}500$ ft^2 lb.

Flywheel constant $(C) = 13{,}850{,}000$.

Velocity of wave $(a) =$ about 3600 ft/sec.

Thus, critical time $T_c = \frac{2L}{a} = \frac{2 \times 573}{3600} = 0.3185$ sec.

For this critical speed interval the momentary head change

$$\Delta H = \frac{a\Delta v}{g} = \frac{3600\Delta v}{32.2} = 111.8\Delta v$$

For convenience of computation, write $\Delta H = 112\Delta v$. The error is:

$$112 - 111.8 = 0.2 \text{ or } \frac{0.2 \times 100}{111.8} = 0.179\% \text{ only.}$$

Assume that the full change of velocity from 13.217 ft/sec to 0 requires 16 intervals of T_c. The time to accomplish this is, therefore, $T_p = 16 \times 0.3185 = 5.096$ sec. The computations are now made step-by-step for 16 intervals. $\Delta H = 112\Delta v$ thus giving us the increase (for load rejections — deceleration of water in pipe) or the decrease (for load increases — acceleration of water in pipe).

During these changes the friction losses in the pipe change accordingly. The total friction loss for 1960 cfs flow is $H_o - H_n = F = 180 - 167.5 = 12.5$ ft.

Figure 2 shows the values of friction loss F in feet as ordinates and the flow Q in cfs as abscissae. In Fig. 2 the curve H_n shows the respective net heads for each flow Q. These values must be used as a correction of the head for each step computed.

Table I shows the computations, assuming a momentary full load rejection of 16,000 hp on each unit simultaneously so that the full flow of 1960 cfs must be stopped by the governors, decelerating the water from 13.217 to zero ft/sec. The table shows the following columns:

$Q = 1960$ cfs

$V = 13.217$ ft/sec

$F = 12.5$ ft

$L = 573$ ft

$\frac{2L}{a} = \frac{1146}{3606} = 0.3185$

$\Delta H = \frac{a \Delta V}{g} = \frac{3606}{32.2} \Delta V = 112 \Delta V$

$B_o = \frac{V_o}{\sqrt{H_o}} = \frac{13.217}{\sqrt{167.5}} = 1.0217$

$Q_x = \frac{1960}{13.217} V_x = 148.3 \; V_x$

a	b	c	d	e	f	g	h	i	k	l	m	n	o	p	q	r s
INTERVAL	T_p in sec	B	V ΔV	$\Delta H = 112 \; \Delta V$	$H + \overline{\sum h}$	F for cfs	$H_n = H_o + F$	$H + \overline{\sum h} + \overline{\sum \Delta F}$	$\overline{\sum h}$	% press. rise	Q cfs	$\frac{Q}{2\sqrt{H + \overline{\sum h + \Delta F}}}$	% eff.	hp	$\varphi = \frac{hp}{32{,}000}$	φ_m, T_n, $100 \frac{\Delta n}{n}$
0	0	1.0217	13.217 .052	0	167.5	0	167.5	167.5	0	0	1960	75.6	86	32,000	1.0	$\frac{\Delta n}{n} = 2.88\%$
1	.3185	1.00	13.165 .220	5.824	173.324 .15	0.15	167.65	173.47	5.824	3.47	1952	74.0	86.4	33,200	1.0375	$\varphi_m = 1.0658$ $T_n = .637$ $100 \frac{\Delta n}{n} = 10.34\%$
2	.6370	.933	12.945 .541	24.64	192.29 .45	0.60	168.1	192.74	24.64	14.66	1915	69.0	88.5	37,280	1.160	
3	.9555	.866	12.404 .710	60.59	204.05 .9	1.50	169.00	204.95	35.95	21.27	1840	64.2	89.9	38,500	1.200	$\varphi_m = 1.813$ $T_n = .637$ $100 \frac{\Delta n}{n} = 18.28\%$
4	1.280	.800	11.694 .821	79.50	212.55 1.20	2.7	170.2	213.75	43.55	25.59	1730	59.3	90.0	37,850	1.184	
5	1.60	.733	10.873 .893	91.95	218.60 1.40	4.1	171.6	220.00	48.40	28.20	1610	54.4	89.0	35,850	1.120	$\varphi_m = 1.109$ $T_n = .637$ $100 \frac{\Delta n}{n} = 24.83\%$
6	1.92	.666	9.980 .936	100.00	223.21 1.3	5.4	172.9	224.51	51.61	29.90	1476	49.3	87.0	32,750	1.023	
7	2.24	.600	9.044 .961	104.83	226.02 1.3	6.7	174.2	227.32	53.12	30.49	1340	44.4	84.5	29,250	0.914	$\varphi_m = 0.909$ $T_n = .637$ $100 \frac{\Delta n}{n} = 30.16\%$
8	2.56	.533	8.083 .983	107.63	228.71 1.2	7.9	175.4	229.91	54.51	31.07	1199	39.55	81.0	25,300	0.790	
9	2.88	.466	7.100 .996	110.10	230.99 1.1	9.0	176.5	232.09	55.59	31.49	1054	34.6	79.5	22,100	0.691	$\varphi_m = 0.6813$ $T_n = .637$ $100 \frac{\Delta n}{n} = 33.98\%$
10	3.20	.400	6.104 1.005	111.55	232.46 .90	9.9	177.4	233.36	55.96	31.54	906	29.7	74.8	18,000	0.563	
11	3.52	.333	5.099 1.015	112.51	234.00 .80	10.7	178.20	234.8	56.61	31.75	756	24.65	71.0	14,500	0.453	$\varphi_m = 0.450$ $T_n = .637$ $100 \frac{\Delta n}{n} = 36.48\%$
12	3.84	.266	4.084 1.013	113.68	235.27 .65	11.35	178.85	235.9	57.07	31.96	606	19.7	66.0	10,710	0.334	
13	4.16	.200	3.011 1.02	113.456	235.236 .45	11.8	179.3	235.686	56.38	31.44	454	14.80	59.0	7,170	0.214	$\varphi_m = 0.22596$ $T_n = .637$ $100 \frac{\Delta n}{n} = 37.71\%$
14	4.48	.133	2.051 1.026	114.24	237.166 .45	12.25	179.75	237.616	57.86	32.19	308	10.0	50.0	4,155	0.1299	
15	4.80	.0666	1.025 1.025	114.91	236.80 .25	12.40	179.90	237.05	57.15	32.15	152	4.93	36.0	1,415	0.0442 Friction Load $\varphi = 0.030$	$\varphi_m = 0.05803$ $T_n = .414$ $100 \frac{\Delta n}{n} = 38.0\%$
16	5.12	0	0	114.80	237.55 .1	12.5	180	237.65	57.65	32.03	0	0	0	0	0	

(Slide rule computations)

Table I — 2 x 16,000 H_p Units, Full Load Off

Column a — Intervals T_c

Column b — Time (T_p) of intervals

Column c — Values of B

The value of B is: $B_o = \frac{v_o}{\sqrt{H_o}} = \frac{13.217}{\sqrt{167.5}} = 1.0217$ and B_x for each step $= \frac{v_x}{\sqrt{H_x}}$, where v_x is the velocity prevailing at the respective interval and H_x the respective head.

The values of B_x are taken from Fig. 3, which represents the character of the movement of the control-mechanism, which adjusts the opening through which the water discharges upon the runner of a Francis turbine. In the case of an impulse wheel it is the movable needle in the orifice through which the water issues, and in a propeller turbine it is the movable guide vanes admitting water to the propeller, influenced also in the case of a propeller with movable runner vanes (tilt) by the changed discharge openings between the propeller vanes.

For a fixed design the character of curve B_x is fixed for each time interval $T_c = \frac{2L}{a}$, of which, as stated before, there are 16 intervals in the example, Table I, for a complete gate closure; i. e., for each 0.3185 sec of gate movement.

A certain change Δv of velocity is now assumed. See Fig. 3, line interval 0; Table I, column d, $\Delta v = 0.052$. Thus the new velocity (second line) is $13.217 - 0.052$ or 13.165 ft/sec.

The first momentary rise in head is: $H = 112 \times 0.052 = 5.824$ ft (column e). This is the amount of positive pressure wave which travels from the guide vanes up through the pipe line. It increases the head to $167.5 + 5.824$ or 173.324 ft, as shown in column f. For the new water velocity of 13.165 ft/sec the new discharge Q (column m) is $\frac{1960 \times 13.165}{13.217}$ or 1952 cfs, or an initial reduction of flow of only 8 cfs, as caused by the character of curve B_x, Fig. 3. For this new discharge a friction loss decrease of 0.15 ft can be taken from curve F, Fig. 2; namely $12.5 - 12.35$ or 0.15 ft. This value must be added to the value in column f of the table and is inserted in column i as 173.47 ft on line 1 at beginning of the second interval.

Likewise, the new net head of $167.5 + 0.15$ or 167.65 ft is inserted in column h. Column k shows the total head increase (5.824 ft) at the end of the first interval. Column l shows the percent increase in head, namely $\frac{173.47 - 167.65}{167.65} \times 100$ or 3.47%. Note that this pressure rise is not based on the original net head but on the net head prevailing at the respective time.

If Δv is correct, then B (interval 1) $\times \sqrt{173.47} = 13.165$.

It is a somewhat tedious process, by trial, to arrive at the

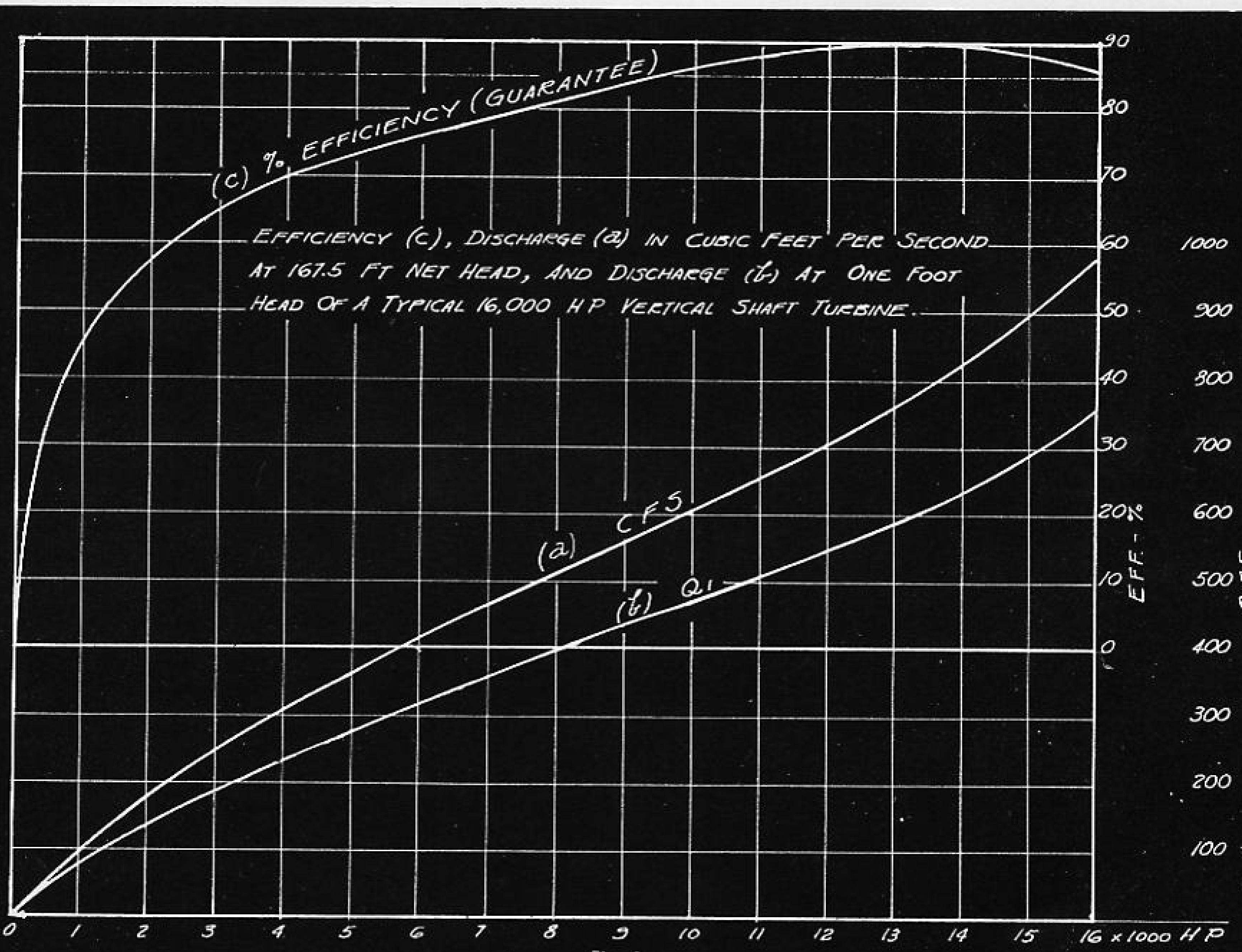

(c) % EFFICIENCY (GUARANTEE)
EFFICIENCY (c), DISCHARGE (a) IN CUBIC FEET PER SECOND
AT 167.5 FT NET HEAD, AND DISCHARGE (b) AT ONE FOOT
HEAD OF A TYPICAL 16,000 HP VERTICAL SHAFT TURBINE.
(a) CFS
(b) Q1
90
80
70
60
50
40
30
20
10
0
EFF. %
1000
900
800
700
600
500
400
300
200
100
CFS
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16 x 1000 HP

Fig. 1

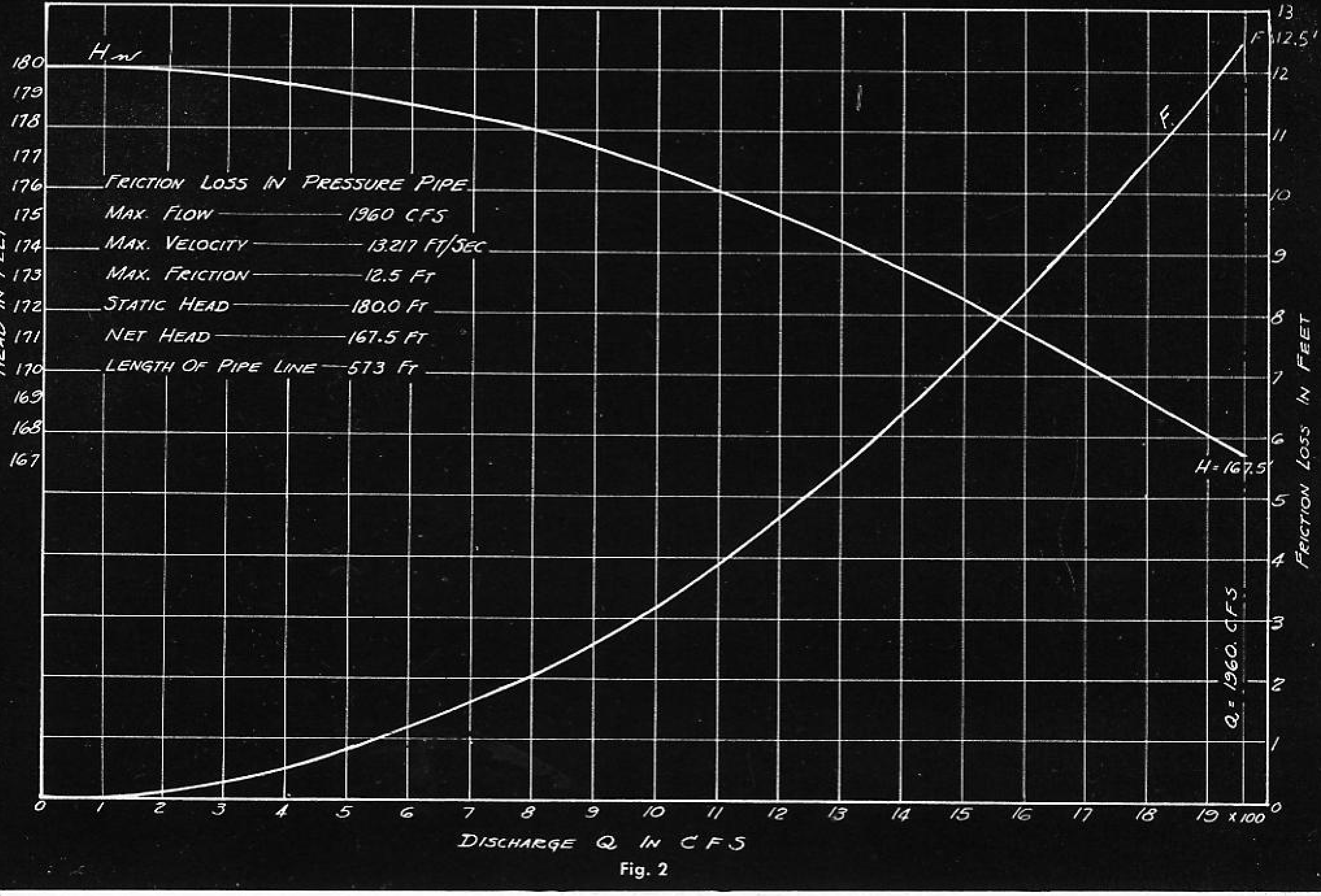

H N
F
F 12.5'
H = 167.5
Q = 1960. CFS
FRICTION LOSS IN PRESSURE PIPE
MAX. FLOW 1960 CFS
MAX. VELOCITY 13.217 FT/SEC
MAX. FRICTION 12.5 FT
STATIC HEAD 180.0 FT
NET HEAD 167.5 FT
LENGTH OF PIPE LINE 573 FT
HEAD IN FEET
180
179
178
177
176
175
174
173
172
171
170
169
168
167
13
12
11
10
9
8
7
6
5
4
3
2
1
0
FRICTION LOSS IN FEET
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19 x 100
DISCHARGE Q IN CFS

Fig. 2

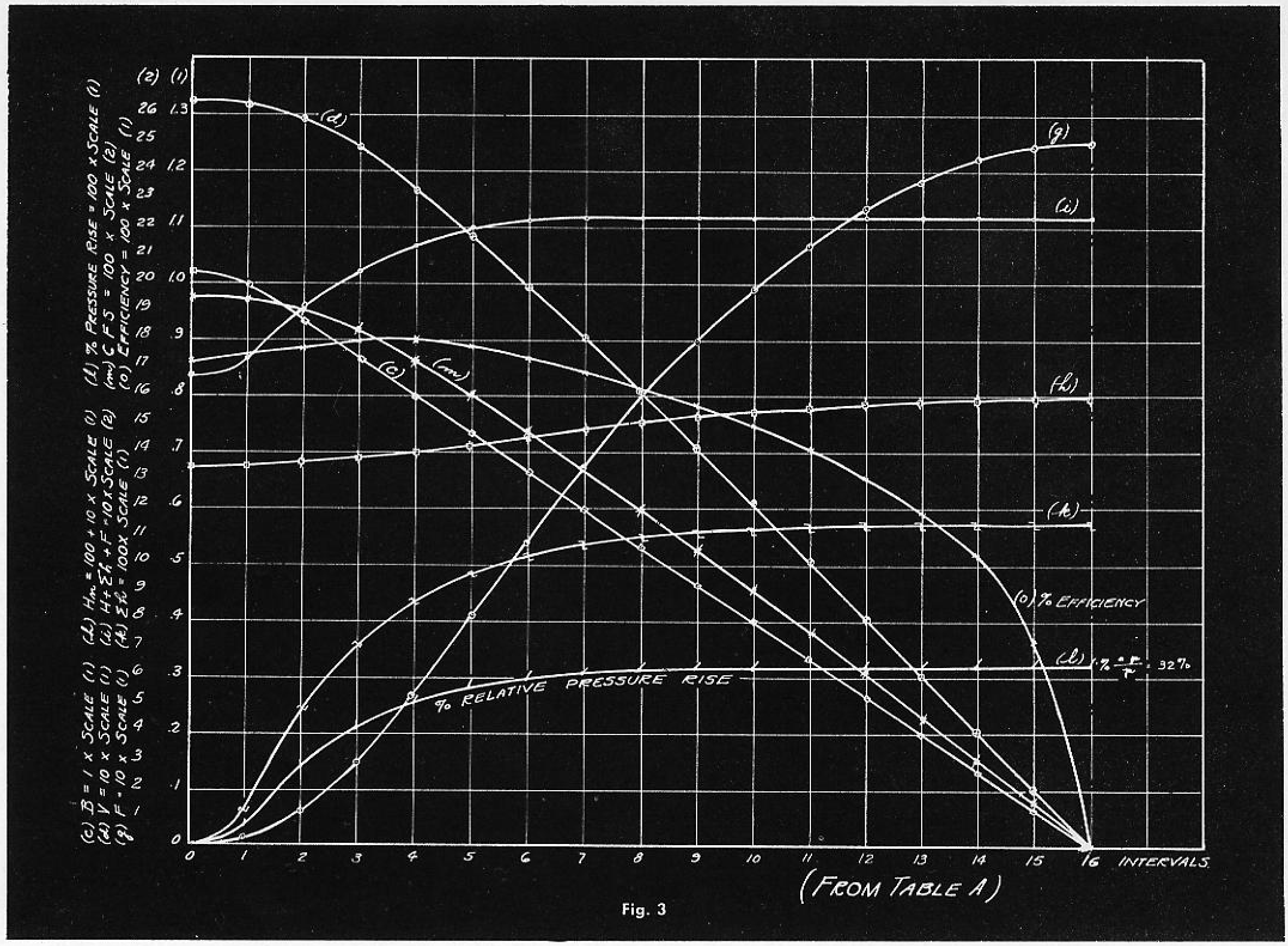


Fig. 3

correct value of Δv. Practice, however, soon allows of a fairly quick correction.

Likewise the values of the next interval are figured; starting with $\Delta v=0.220$, obtaining the new pressure wave $\Delta H=112\times\Delta v=24.64$ ft inserted in column e. If the first pressure wave of 5.824 ft did not return, the new head would be 173.47+24.64 or 198.11 ft. The first wave returns as a "negative" wave so that the final head increase is 173.47+24.64−5.824 or 192.286 (say 192.29) ft as inserted in column f, the new velocity v being 13.165−0.220=12.945 ft/sec (column d) and the new flow Q=1915 cfs (column m). The new reduction in friction loss is 0.45 ft (column f), or a total of 0.6 ft (column g), so that the final percent pressure rise is 14.66 inserted in column l, and so forth.

All other intervals are computed similarly, ending up with interval 16; $B_x=0$; $v=0$; $F=12.5$ ft; $H_n=180$ ft; $Q=0$; percent final pressure rise 32.03. These values are plotted in Fig. 3. It will be noted that the percent pressure rise (curve l) reaches its maximum at about interval 10 and remains almost constant after that for the specific example.

Determination of Efficiencies

In order to determine the respective efficiencies, the values of $Q_1=\dfrac{Q}{2\sqrt{H+\sum h+F}}$ (for one unit) are computed in column n, and the efficiencies applied from curves c and a of Fig. 1. With these efficiencies and the discharges Q of column m and the actual heads of column i the actual total horsepower outputs of the two units for the various intervals are computed as per column p.

By dividing these values by 32,000 (the total full capacity output of two units), the output factors φ for each interval is obtained, as shown in column q, necessary for computations of the relative momentary speed rises of each interval.

Before proceeding with the above computations an analysis of Table I and the various curves plotted on Fig. 3, columns c, d, l, m, o, and q may prove of interest.

Curve column c shows a gradual reduction during the first interval and a practically linear reduction thereafter down to 0, as a characteristic curve of gate movement. Column m shows the total discharges Q, and it is noted that, because of the increased heads (curve column i), the discharge is not reduced so rapidly; whereas curve of Q_1 (column n) approaches closer to curve B (column c).

On account of the increased heads and the more gradual discharges Q, the total outputs (hp) (column p) deviate materially from the linear reduction previously assumed in the end result calculations outlined in previous parts of this article.

IT IS THIS INCREASE IN HORSEPOWER, ABOVE THE LINEAR DECREASE, WHICH CAUSES A RELATIVE SPEED RISE IN EXCESS OF THAT RESULTING FROM A LINEAR DECREASE OF OUTPUT, SUCH AS IS THE CASE IF THE HEAD REMAINS PRACTICALLY CONSTANT, AND WHICH IS NEVER

ATTAINABLE WITH A TURBINE UNLESS IT BE PLACED DIRECTLY IN A LARGE LAKE, SO THAT A REDUCTION OF DISCHARGE THROUGH THE UNIT PRODUCES NO HEAD INCREASE.

Effect on Speed Rise of Percent Load Carried

It can be seen that the output curve φ (column q) (see Fig. 4) materially exceeds the value 1.0 and, in the example, does not return to normal (full load output) until after six intervals have passed. This is caused by the increased heads (column i). In the example a peak output of 18% "above full" is reached. (This may explain a case where the overload caused the circuit breaker to trip the entire load!)

Referring to Fig. 4 for a load change of 25% from full output ($\varphi = 1.0$) to ¾ output ($\varphi = 0.75$), it requires 8.5 intervals for the governor, whereas a linear decrease of φ would require only four intervals. The actual relative speed rise, as will be seen later, will be much higher for a 25% load change from full to ¾ than for the same percent load change from ¼ to 0.

Likewise for a load change of 50% from full to half it requires about 10.5 intervals actual as against eight along linear reduction. Load rejections usually take place with the unit operating nearer to full load than to small fractions thereof. A 25% load change will take place oftener from full to ¾ than from ¼ to 0. Therefore, attention should be concentrated on such load changes that are more frequent than others.

It can be seen at once that, to avoid this condition ($\varphi > 1$), the pressure rises must be avoided; and this is accomplished with pressure regulators, as previously explained, whereby the φ curve is lowered toward the straight line, a linear decrease.

It is noticed that, for the last quarter output from interval 12.7 to 16, the φ curve even undercuts the straight line so that for this last quarter the additional speed rise will be much smaller than for the first quarter.

The same conditions prevail in connection with load increases. Here the decreased heads cause a shortage of output, pressing the φ curve materially below the straight line of linear change of φ. From this it can be concluded at once that, unless a pressure regulator acting as a "water-wasting device" is provided, thereby practically eliminating acceleration of the water in the pipe, it is of no assistance in improving speed regulation on load increases.

Where water-wasting pressure regulators cannot be employed and where the velocity of water in the pipe is high, the speed decreases resulting from sudden load increases are bound to be very serious. However, such load increases occur when the unit operates in parallel with an electric power system. In that case — unless exclusively for chemical process load, etc. — the WR^2 of rotating masses operating in parallel is much greater than those of the unit above, and also the load will not increase instantaneously. The effect of this has been illustrated in previous parts of this article.

In order to maintain normal (or synchronous) speed the turbine at no-load condition discharges a certain cfs. If the governor reduces the discharge further, the turbine will act as a brake. This would be an advantage because it would

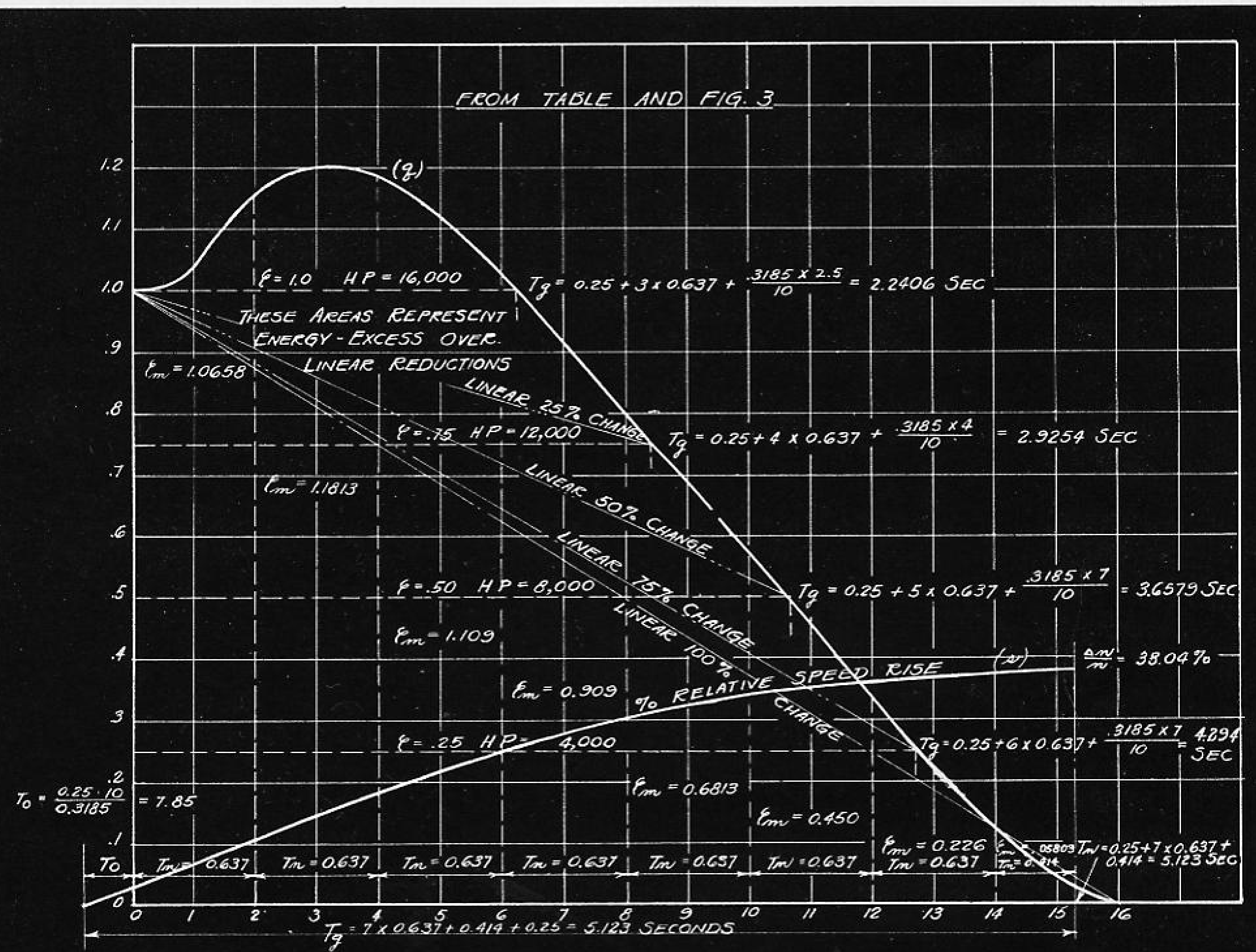


Fig. 4

assist in bringing the speed back to normal after a load rejection to zero-load has caused a speed rise; however, practically, it may introduce danger since the kinetic energy in the draft tube would cause a vacuum under the runner which may have a disturbing, if not destructive, effect. It is for this reason that turbo-vents are employed, which admit air to the discharge side of the runner, although they defeat the advantages otherwise gained for speed full load-off control. The actual governor time is, therefore, somewhat less than 16 intervals in the example, zero output being attained at about the fifteenth interval, namely when $\varphi=0.03$ or about 3% of the normal full output of 16,000 hp.

The relative speed rises are computed as follows: Referring to Fig. 4, for a total gate closure with 16 intervals, the time is 16×0.3185 or 5.096 sec. For computing step-by-step, two intervals each were used here, of a total of 0.637 sec, and the average value φ_m for each portion was thus determined, as shown in column r of Table I. If there were no insensitiveness ($T_r=0$) and if the governor would move the control of water instantly to the full rate of control movement, then the speed at the beginning of the first interval would still be normal, 225 rpm in the example, so that the percent speed rise curve s (Table I and Fig. 4) would begin at zero. However, T_r is not zero; therefore $T_r\times(\varphi=1.0)>0$. Likewise $T_n\times\varphi>0$, the kinetic energy surplus due to acceleration of control movement. For simplicity's sake, and as a safe figure, assume that $T_r\varphi+T_n\varphi_o$, as outlined in previous part of this article, be 0.25 sec×1.0 (φ), or 0.25. Thus is obtained an initial momentary speed rise after 0.25 sec, as will be shown below.

The maximum speed rise is attained when the output of the turbine has been reduced to the value of the friction load output, which, as explained before, may in the example be taken as $\varphi_r=0.03\times(\varphi=1.0)$ as can be seen from Fig. 4; this takes place between interval 15 and 16. The last interval that concerns governor time T_n and φ_m (column r) is, therefore, not 0.637 sec but 0.3185 plus 0.0955 or 0.414 sec (see item 8 below), and φ_m is 0.05803.

The total governor time also is 5.123 sec, as shown, resulting in a maximum relative speed rise of 38.0%, as shown in the following computations.

Computation of Percent Relative Speed Rise for 2x16,000 Hp Load Rejection

($\varphi=1.0$ to 0)

Values of φ_m taken from Table I.

(Slide rule computations.)

General formula:

$$\frac{\Delta n}{n}-1=\sqrt{\left(\frac{n_2}{n_1}\right)^2+\frac{3{,}233{,}000}{C}\times\varphi_m T_n}-1$$

$C=13{,}816{,}000 \quad n=225$ rpm

Speed rise due to insensitiveness and delay in gate movement.

$\varphi=1.0 \quad (T_r=0.10)+(T_n=0.15)=0.25$

$$\frac{\Delta n}{n}-1=\sqrt{1+0.234\times1.0\times0.25}-1=1.0288-1,\text{ or } 2.88\%$$

after 0.25 sec.

Values of φ_m and T_n

1)	$\varphi_m=1.0658$	$T_n=0.637$	5)	$\varphi_m=0.6813$	$T_n=0.637$
2)	1.1813	0.637	6)	0.450	0.637
3)	1.109	0.637	7)	0.22596	0.637
4)	0.909	0.637	8)	0.05803	0.414

1) $$\frac{\Delta n}{n}-1=\sqrt{\underset{\frac{0.159}{1.2175}}{1.0585}+\underset{0.159}{0.234\times1.0658\times0.637}}-1=1.104-1,\text{ or}$$

10.34% after 0.887 sec.

2) $$\frac{\Delta n}{n}-1=\sqrt{\underset{\frac{0.1764}{1.3939}}{1.2175}+\underset{0.1764}{0.234\times1.1813\times0.637}}-1=1.1828-1,\text{ or}$$

18.28% after 0.887+0.637=1.524 sec.

3) $$\frac{\Delta n}{n}-1=\sqrt{\underset{\frac{0.1655}{1.5585}}{1.393}+\underset{0.1655}{0.234\times1.109\times0.637}}-1=1.2483-1,\text{ or}$$

24.83% after 1.524+0.637=2.161 sec.

4) to 7) etc.

8) $$\frac{\Delta n}{n}-1=\sqrt{\underset{\frac{0.00565}{1.90205}}{1.89640}+\underset{0.00565}{0.234\times0.05803\times0.414}}-1=1.380-1,\text{ or}$$

38.0% after 4.709+.414=5.123 sec.

Since this speed is materially above the normal no-load speed, the governor will close the turbine gates completely, thereby causing the turbine runner to act partly as a brake, which will gradually reduce the overspeed. When the flow of water is stopped, there will also be no further deceleration of water in the pipe line so that the head will also return to the normal static head ($H_o=180$ ft), again contributing to the return of the turbine speed to 225 rpm plus the small percent (about 1½%) caused by the characteristic of the governor relay, as previously explained.

Likewise on partial load rejections from full load, the governor will close the turbine gates somewhat farther than is necessary for holding the new load at normal speed. This small over-travel will cause a slight temporary oscillation of the speed.

The various step values are thus, as tabulated below:

Interval	Percent Relative Speed Rise	Time
0	2.88%	after 0.25 sec (due to insensitiveness)
0 to 2	10.34%	after 0.887 sec
2 to 4	18.28%	after 1.524 sec
4 to 6	24.83%	after 2.161 sec
6 to 8	30.16%	after 2.798 sec
8 to 10	33.98%	after 3.435 sec
10 to 12	36.48%	after 4.072 sec
12 to 14	37.71%	after 4.709 sec
14 to 15 plus	38.0 %	after 5.123 sec

Relative Speed Rise

Compare this 38.0% with only 28.75% computed for no increase in head and obtained from equation:

$$\frac{\Delta n}{n}-1=\sqrt{1.0585+0.234\times\underset{(\varphi_m)}{\frac{1.0}{2}}\times\underset{(T_n)}{5.123}}-1=0.2875.$$

From Fig. 4 it can be seen that the speed has already risen 25% above normal before the turbine output returns to full capacity! ($\varphi=1.0$, interval 6 plus.)

For a sudden load rejection of 25% from full load to ¾ load ($2\times16{,}000$ to $2\times12{,}000$ hp, $\varphi=1.0$ to $\varphi=0.75$), the relative speed rise is computed by deducting from the above tabulated values of φ_m the constant value $\varphi=0.75$.

The various values so computed are shown below.

Computation of Percent Relative Speed Rise for 2x16,000 to 12,000 Hp Load Rejection

$(\varphi=1.0-0.75=0.25)$

Values of φ_m taken from Table I.

(Slide rule computations.)

General formula:

$$\frac{\Delta n}{n}-1=\sqrt{\left(\frac{n_2}{n_1}\right)^2+\frac{3{,}233{,}000}{C}\times\varphi_m\,T_n}-1$$

$C=13{,}816{,}000 \quad n=225$ rpm

Speed rise due to insensitiveness and delay in gate movement.

$\varphi=0.25 \quad (T_P=0.10)+(T_o=0.15)=0.25$

$\frac{\Delta n}{n}-1=\sqrt{\underset{1.0146}{1+0.234}\times\underset{0.0146}{0.25\times0.25}}-1=1.0072-1$, or 0.72% after 0.25 sec.

Values of φ_m and T_n

1)	$\varphi_m=1.0658-.75=0.3158$	$T_n=0.637$
2)	$1.1813-.75=0.4313$	0.637
3)	$1.109-.75=0.359$	0.637
4)	$0.909-.75=0.259$	0.637
5)	$0.790-.75\div2=0.02$	0.1274

1) $\frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.0471}\\1.0617}}{1.01461}+0.234\times\underset{0.0471}{0.3158\times0.637}}-1=1.0304-1$, or 3.04% after $0.25+0.637=0.887$ sec.

2) $\frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.0643}\\1.1260}}{1.0617}+0.234\times\underset{0.0643}{0.4313\times0.637}}-1=1.0611-1$, or 6.11% after $0.887+0.637=1.524$ sec.

3) $\frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.05355}\\1.17955}}{1.1260}+0.234\times\underset{0.05355}{0.359\times0.637}}-1=1.0860-1$, or 8.60% after $1.524+0.637=2.161$ sec.

4) $\frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.0237}\\1.20325}}{1.17955}+0.234\times\underset{0.0237}{0.159\times0.637}}-1=1.0969-1$, or 9.69% after $2.161+0.637=2.798$ sec.

5) $\frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.000596}\\1.203846}}{1.20325}+0.234\times\underset{0.000596}{0.02\times0.1274}}-1=1.0971-1$, or 9.71% after $2.798+0.1274=2.9254$ sec.

Again it takes about 8.5 intervals as against only about four for a 25% load change from 4000 to 0 hp, as shown later, because for such computation it is necessary to compute the pressure increases again step by step.

For a 50% sudden load rejection from $2\times16{,}000$ to 2×8000 hp ($\varphi=1.0$ to $\varphi=0.5$), the relative speed rise is computed as follows.

Computation of Percent Relative Speed Rise for 2x16,000 to 8,000 Hp Load Rejection

$(\varphi=1.0-0.5=0.5)$

Values of φ_m taken from Table I.

(Slide rule computations.)

General formula:

$$\frac{\Delta n}{n}-1=\sqrt{\left(\frac{n_2}{n_1}\right)^2+\frac{3.233{,}000}{C}\times\varphi_m\,T_n}-1$$

$C=13{,}816{,}000 \quad n=225$ rpm.

Speed rise due to insensitiveness and delay in gate movement.

$\varphi=0.50 \qquad (T_P=0.10)+(T_o=0.15)=0.25$

$\frac{\Delta n}{n}-1=\sqrt{\underset{1.02925}{1+0.234}\times\underset{0.02925}{0.50\times0.25}}-1=1.0145-1$, or 1.45% after 0.25 sec.

Values of φ_m and T_n

1)	$\varphi_m=1.0658-.5=0.5658$	$T_n=0.637$
2)	$1.1813-.5=0.6813$	0.637
3)	$1.109-.5=0.609$	0.637
4)	$0.909-.5=0.409$	0.637
5)	$0.6813-.5=0.1813$	0.637
6)	$0.563-.5\div2=0.0315$	0.22295

1) $\frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.0844}\\1.11365}}{1.02925}+0.234\times\underset{0.0844}{0.5658\times0.637}}-1=1.0553-1$, or 5.53% after $0.25+0.637=0.887$ sec.

2) $\frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.1016}\\1.21525}}{1.11365}+0.234\times\underset{0.1016}{0.6813\times0.637}}-1=1.1023-1$, or 10.23% after $0.887+0.637=1.524$ sec.

3) $\frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.0908}\\1.30605}}{1.21525}+0.234\times\underset{0.0908}{0.609\times0.637}}-1=1.1428-1$, or 14.28% after $1.524+0.637=2.161$ sec.

4) and 5) etc.

6) $\frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.001643}\\1.395793}}{1.39415}+0.234\times\underset{0.001643}{0.0315\times0.223}}-1=1.1814-1$, or 18.14% after $3.435+0.22295=3.65795$ sec.

Again it takes between 10 and 11 intervals, as against only eight for linear discharge and no head increases.

$V=\frac{13.217\times1415}{1960}=9.61$ ft/sec

$B=1.0217\times\frac{1415}{1960}=.738$

$Q=\frac{24,000\times8.8}{167.5\times.89}=1415$ cfs

$F=12.5-6.5=6.0$ ft

$\Delta H=112\,\Delta V$ as before

T_p for 4a to 16 intervals$=0.3185+11\times0.3185=3.53535$ sec

a	b	c	d	e	f	g	h	i	k	l	m	n	o	p	q	r	s
INTERVAL	T_p in sec	B	V ΔV	$\Delta H= 112\ \Delta V$	$H+\overline{\sum h}$	F for cfs	$H_n= H_o+F$	$H+\overline{\sum h}$ $+\overline{\sum \Delta F}$	$\overline{\sum h}$	% press. rise	Q cfs	$\frac{Q}{2\sqrt{\text{col. i}}}$	% eff.	hp	$\varphi=\frac{hp}{32,000}$	φ_m T_n	$100\ \frac{\Delta n}{n}$
4a	0	.738	9.61 .03	0	167.5	0	167.5	167.5	0	0	1415	54.7	89.0	2 x 12,000 =24,000	.75	$\varphi_m=.755$ $T_n=0.03185$	2.44%
5	0.03185	.733	9.58 .274	3.36	170.86	~0	167.5 .45	170.86	3.36	.20	1410	53.9	88.8	24,300	.76		
6	0.35035	.666	9.306 .615	30.69	194.83 .45	.45	167.95	195.28	27.33	10.31	1370	49.0	87.0	26,450	.826	$\varphi_m=0.797$ $T_n=0.637$	8.07%
7	0.66885	.600	8.691 .81	68.88	209.50 .75	1.3	168.8	210.25	41.45	24.55	1280	44.1	84.5	25,700	.805		
8	0.98735	.533	7.881 .897	90.70	218.15 .90	2.2	169.7	219.05	49.35	29.08	1162	39.3	81.0	23,460	.734	$\varphi_m=0.7257$ $T_n=0.637$	12.99%
9	1.30585	.466	6.994 .98	100.16	224.71 .90	3.1	170.6	225.61	55.01	32.24	1028	34.2	77.5	20,400	.638		
10	1.62435	.400	6.014 .986	109.76	225.35 1.0	4.1	171.6	226.31	54.75	31.91	875	29.05	74.0	16,640	.52	$\varphi_m=0.5273$ $T_n=0.637$	16.41%
11	1.94285	.333	5.028 .998	110.43	227.28 .6	4.7	172.2	227.88	55.68	32.33	741	24.55	70.5	13,510	.424		
12	2.26135	.266	4.030 1.0	111.77	228.29 .7	5.4	172.9	228.99	56.09	32.44	593	19.60	66.0	10,190	.318	$\varphi_m=0.3187$ $T_n=0.637$	18.44%
13	2.57985	.200	3.030 1.01	112.0	228.81 .5	5.9	173.4	229.31	55.91	32.24	445	14.85	59.0	6,845	.214		
14	2.89835	.133	2.020 1.015	113.12	230.61 .35	6.25	173.75	230.96	57.21	32.92	298	9.81	50.0	3,910	.1225	$\varphi_m=0.1262$ $T_n=0.637$	19.24%
15	3.21685	.665	1.005 1.005	113.68	230.22 .2	6.45	173.95	230.42	56.47	32.46	147	4.74	35.0	1,345	.042 .30		
16	3.53535	0	0	112.56	230.04 .05	6.5	174.0	230.09	56.09	32.23	0	0	0	0	0	$\varphi_m=0.036$ $T_n=0.09555$	19.26%

(Slide rule computations)

Table II — Two Units—2 x 12,000 hp off

Likewise a 75% sudden load rejection from 2×16,000 to 2×4000 hp ($\varphi=1.0$ to $\varphi=0.25$) is computed below.

Computation of Percent Relative Speed Rise for 2x16,000 to 4,000 Hp Load Rejection

($\varphi=1.0-0.75=0.25$)

Values of φ_m taken from Table I.

(Slide rule computations.)

General formula:

$$\frac{\Delta n}{n}-1=\sqrt{\left(\frac{n_2}{n_1}\right)^2+\frac{3,233,000}{C}\times\varphi_m\,T_n}-1$$

$C=13,816,000 \qquad n=225$ rpm.

Speed rise due to insensitiveness and delay in gate movement.

$\varphi=0.75 \qquad (T_F=0.10)+(T_o=0.15)=0.25$

$$\frac{\Delta n}{n}-1=\sqrt{\underset{1.0439}{1+0.234}\times\underset{0.0439}{0.75\times0.25}}-1=1.0217-1,\text{ or }2.17\%$$

after 0.25 sec.

Values of φ_m and T_n

1) $\varphi_m=1.0658-.25=0.8158 \qquad T_n=0.637$
2) $1.1813-.25=0.9313 \qquad 0.637$
3) $1.109\ -.25=0.859 \qquad 0.637$
4) $0.909\ -.25=0.659 \qquad 0.637$
5) $0.6813-.25=0.4313 \qquad 0.637$
6) $0.450\ -.25=0.200 \qquad 0.637$
7) $0.334-.25\div2=0.042 \qquad 0.22295$

1) $$\frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.1219}\\ 1.1658}}{1.0439}+\underset{0.1219}{0.234\times0.8158\times0.637}}-1=1.0797-1,\text{ or}$$

7.97% after 0.25+0.637=0.887 sec.

2) $$\frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.1388}\\ 1.3046}}{1.1658}+\underset{0.1388}{0.234\times0.9313\times0.637}}-1=1.1421-1,\text{ or}$$

14.21% after 0.887+0.637=1.524 sec.

3) $$\frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.1281}\\ 1.4327}}{1.3046}+\underset{0.1281}{0.234\times0.859\times0.637}}-1=1.1969-1,\text{ or}$$

19.69% after 1.524+0.637=2.161 sec.

4) to 6) etc.

7) $$\frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.00219}\\ 1.62720}}{1.62501}+\underset{0.00219}{0.234\times0.042\times0.223}}-1=1.2756-1,\text{ or}$$

27.56% after 4.072+0.22295=4.29495 sec.

Table II shows the step-by-step computation for a 75% sudden load rejection of 2×12,000 hp to zero ($\varphi=0.75$ to $\varphi=0$). The initial discharge for 2×12,000 hp begins between the fourth and fifth interval of Table I; and to reduce gate movement to friction load it takes about 10.5 intervals, shown also graphically in Fig. 5, as against 12.75 intervals, as per Fig. 4 for the same 75% load rejection. However, from full

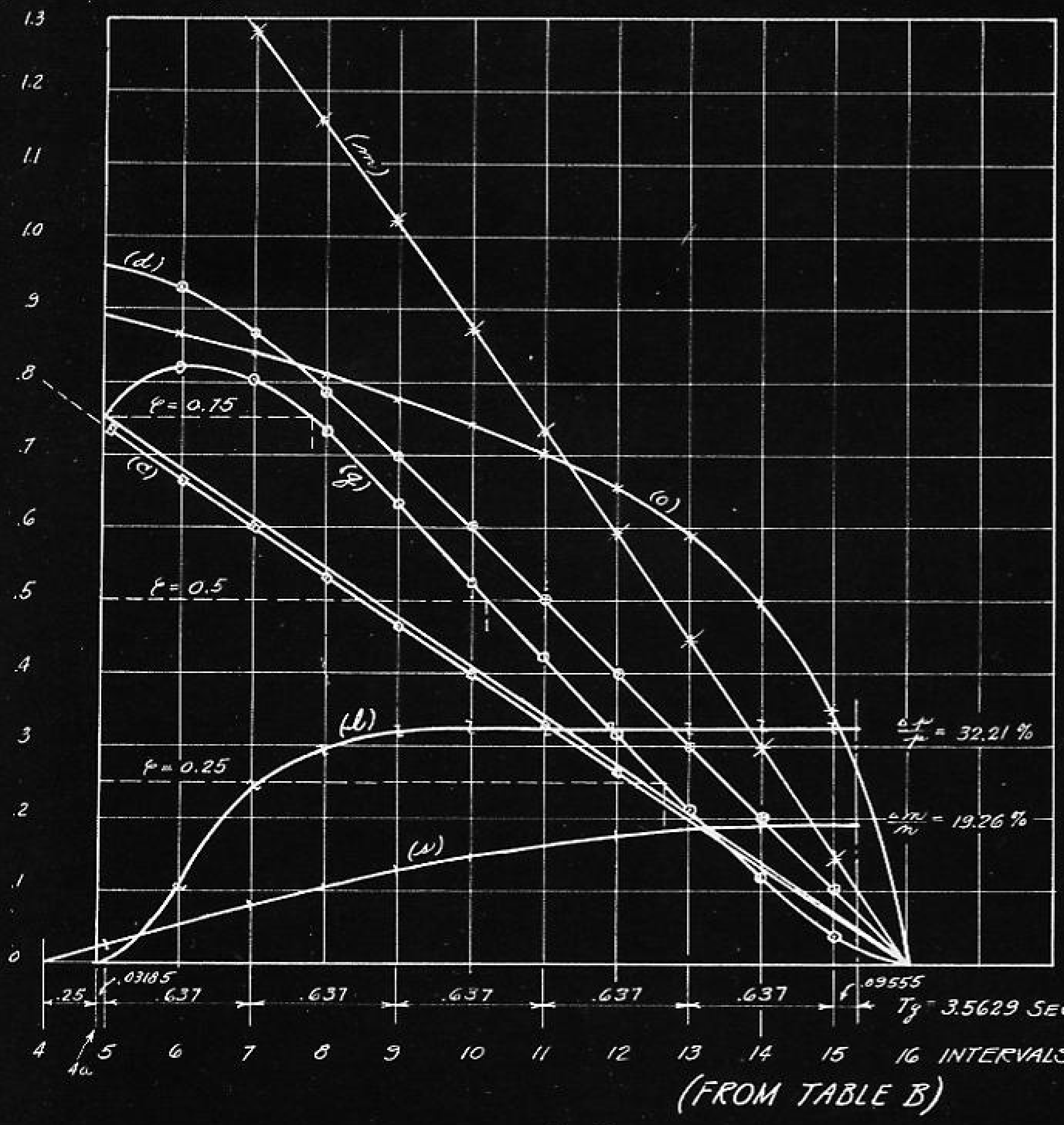


Fig. 5

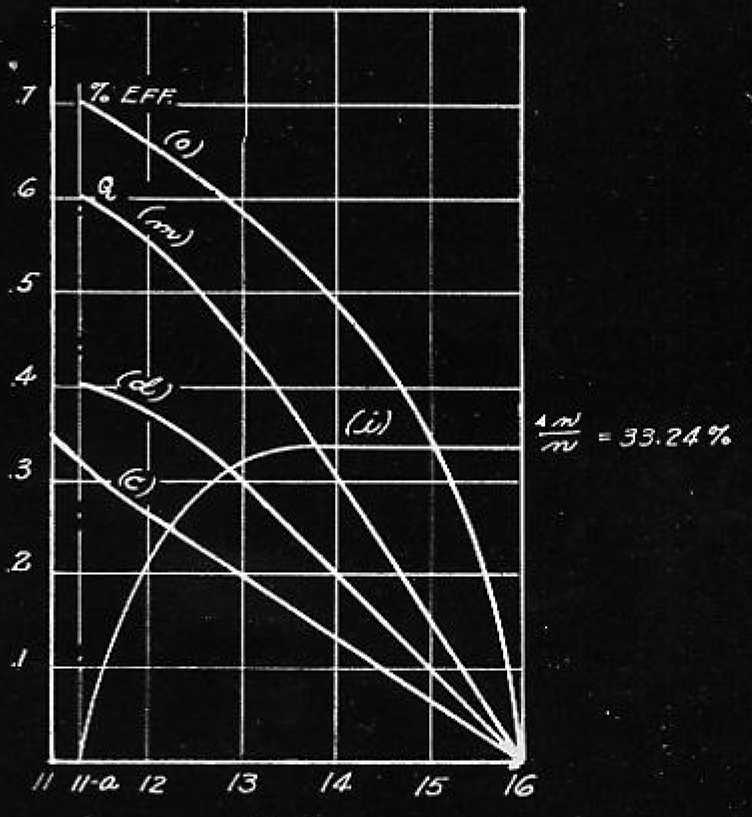


Fig. 7

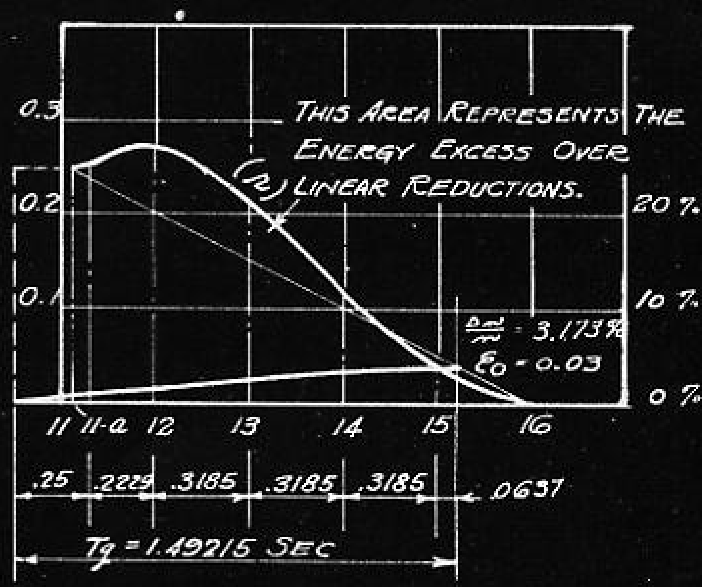


Fig. 7a

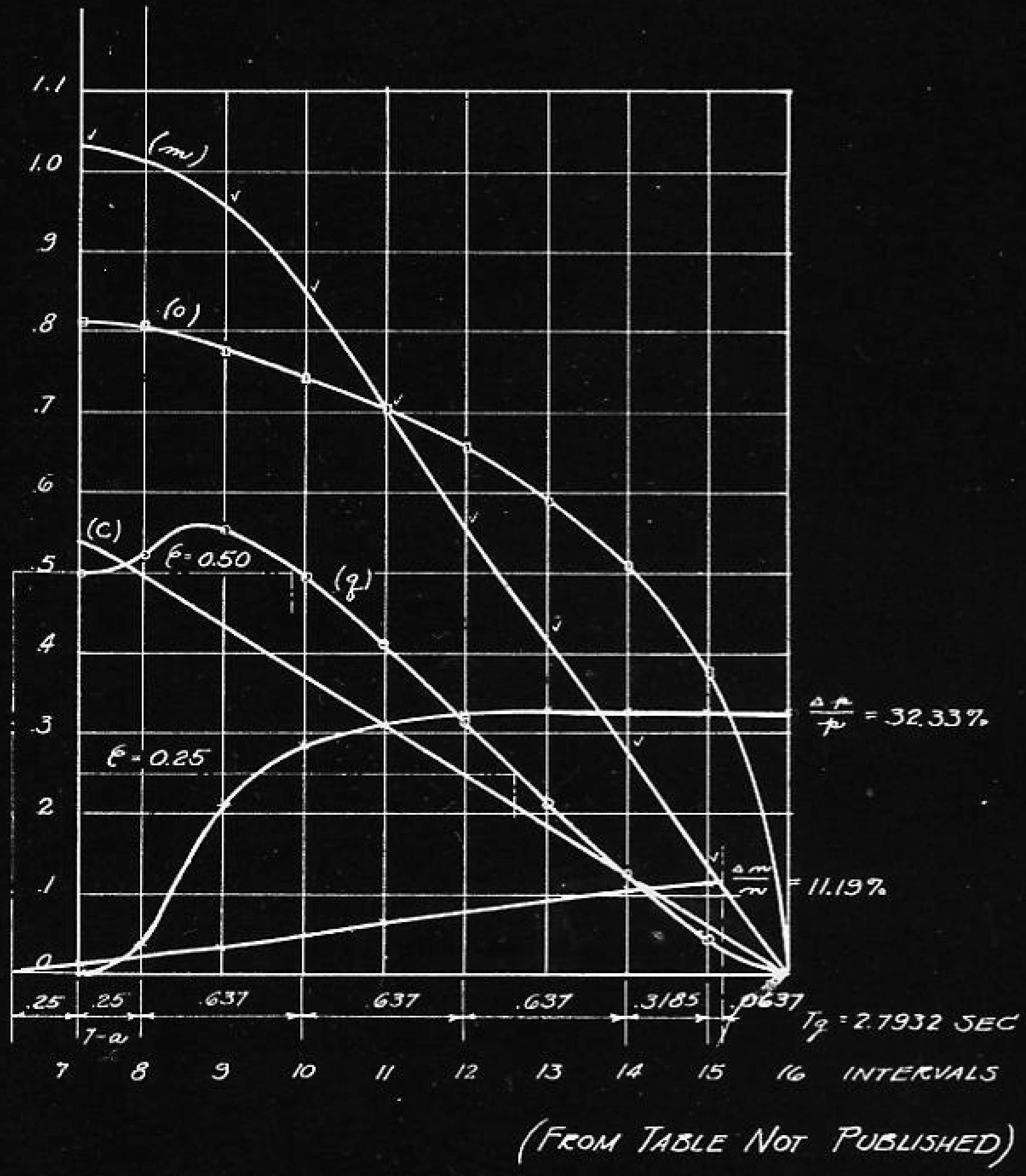


Fig. 6

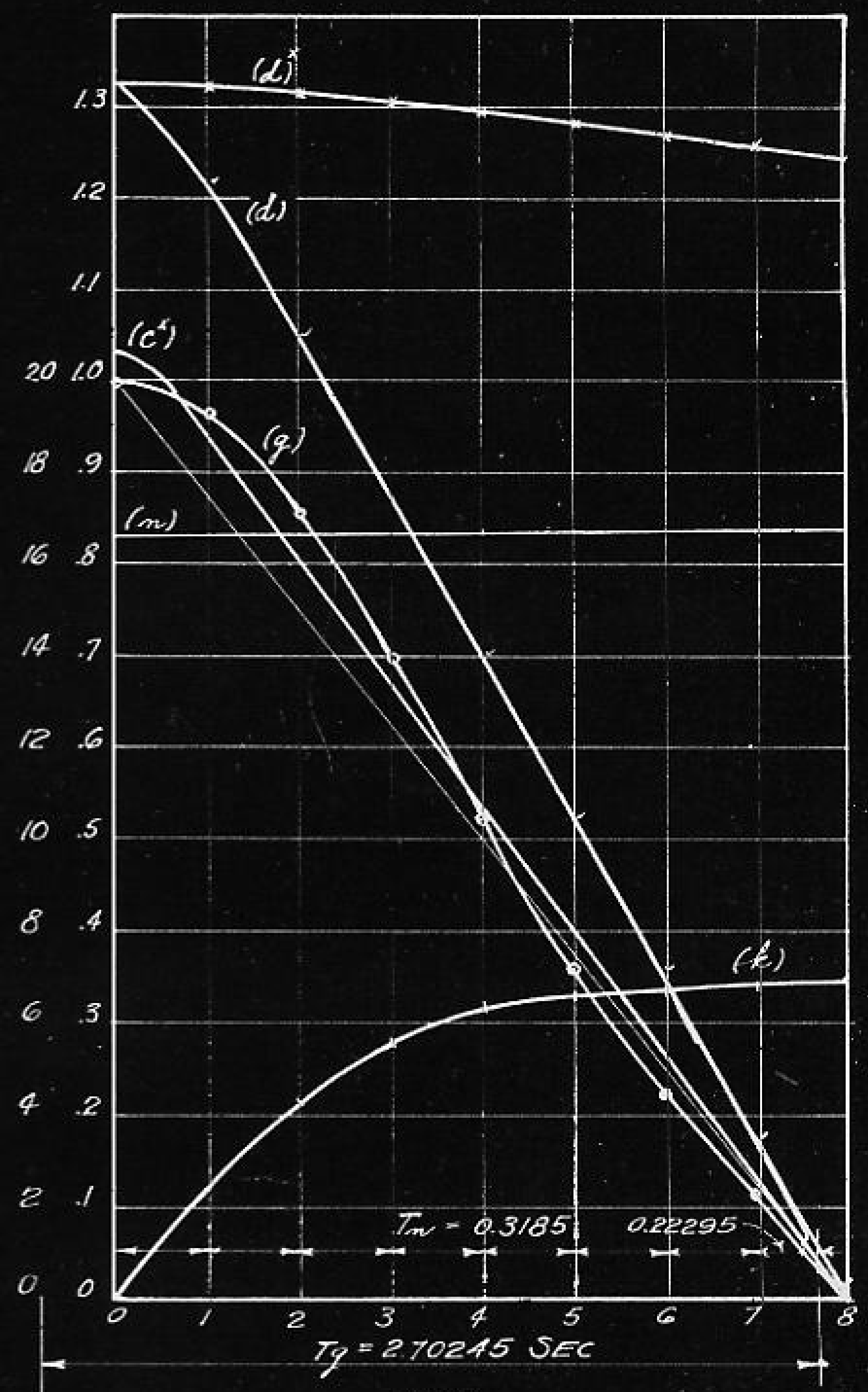


Fig. 8

to ¼ ($\varphi=1.0$ to $\varphi=0.25$) the computations for the relative speed rises are shown below.

Computation of Percent Relative Speed Rise for 2x12,000 to 0 Hp Load Rejection

($\varphi=0.75$ to 0)

Values of φ_m taken from Table II.

(Slide rule computations.)

General formula:

$$\frac{\Delta n}{n}-1=\sqrt{\left(\frac{n_2}{n_1}\right)^2+\frac{3{,}233{,}000}{C}\times\varphi_m T_n}-1$$

$C=13{,}816{,}000 \qquad n=225$ rpm.

Speed rise due to insensitiveness and delay in gate movement.

$\varphi=0.75 \qquad (T_F=0.10)+(T_o=0.15)=0.25$

$$\frac{\Delta n}{n}-1=\sqrt{\underset{1.0439}{1}+\underset{0.0439}{0.234\times0.75\times0.25}}-1=1.0217-1,\text{ or } 2.17\%$$

after 0.25 sec.

Values of φ_m and T_n

1) $\varphi_m=0.755 \qquad T_n=0.03185$

2) $0.797 \qquad 0.637$

3) $0.7257 \qquad 0.637$

4) $0.5273 \qquad 0.637$

5) $\varphi_m=0.3187 \qquad T_n=0.637$

6) $0.1262 \qquad 0.637$

7) $0.036 \qquad 0.09555$

$$1)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.00563}\\1.04953}}{1.0439}+\underset{0.00563}{0.234\times0.755\times0.03185}}-1=1.0244-1,\text{ or}$$

2.44% after $0.25+0.03185=0.28185$ sec.

$$2)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.1188}\\1.16833}}{1.04953}+\underset{0.1188}{0.234\times0.797\times0.637}}-1=1.0809-1,\text{ or}$$

8.09% after $0.28185+0.637=0.91885$ sec.

$$3)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.1084}\\1.27673}}{1.16833}+\underset{0.1084}{0.234\times0.7257\times0.637}}-1=1.1299-1,\text{ or}$$

12.99% after $0.91885+0.637=1.55585$ sec.

4) to 6) etc.

$$7)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.000805}\\1.422445}}{1.42164}+\underset{0.000805}{0.234\times0.036\times0.09555}}-1=1.1926-1,\text{ or}$$

19.26% after $3.46685+0.09555=3.56240$ sec.

Comparing these two cases of 75% load rejection:

Load Rejection in Hp	Governor Time in Sec	Max. Relative Speed Rise	Max. Relative Pressure Rise
2×16,000 to 12,000	4.294	27.56%	About 32%
2×12,000 to 0	3.5624	19.26%	About 33%

Here the shorter governor time required for 2×12,000 to 0 hp produces a somewhat higher maximum relative pressure rise but cuts down materially the maximum relative speed rise.

Similar computations are made for 50% sudden load rejection, 2×8000 to 0 hp ($\varphi=0.5$ to $\varphi=0.0$). The step-by-step tabulation is not published here, but only the graphic results, as per Fig. 6, and the computation of relative speed rises.

Computation of Percent Relative Speed Rise for 2x8,000 to 0 Hp Load Rejection

($\varphi=1.0-0.5=0.5$)

Values of φ_m taken from table not published.

(Slide rule computations.)

General formula:

$$\frac{\Delta n}{n}-1=\sqrt{\left(\frac{n_2}{n_1}\right)^2+\frac{3{,}233{,}000}{C}\times\varphi_m T_n}-1$$

$C=13{,}816{,}000 \qquad n=225$ rpm.

Speed rise due to insensitiveness and delay in gate movement.

$\varphi=0.50 \qquad (T_F=0.10)+(T_o=0.15)=0.25$

$$\frac{\Delta n}{n}-1=\sqrt{\underset{1.02925}{1}+\underset{0.02925}{0.234\times0.50\times0.25}}-1=1.0145-1,\text{ or } 1.45\%$$

after 0.25 sec.

Values of φ_m and T_n

1) $\varphi_m=0.511 \qquad T_n=0.25$

2) $0.524 \qquad 0.637$

3) $0.408 \qquad 0.637$

4) $\varphi_m=0.215 \qquad T_n=0.637$

5) $0.0812 \qquad 0.3185$

6) $0.0057 \qquad 0.0637$

$$1)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.0299}\\1.05915}}{1.02925}+\underset{0.0299}{0.234\times0.511\times0.25}}-1=1.0291-1,\text{ or}$$

2.91% after $0.25+0.25=0.50$ sec.

$$2)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.0782}\\1.13735}}{1.05915}+\underset{0.0782}{0.234\times0.524\times0.637}}-1=1.0665-1,\text{ or}$$

6.65% after $0.50+0.637=1.137$ sec.

$$3)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.0608}\\1.19815}}{1.13735}+\underset{0.0608}{0.234\times0.408\times0.637}}-1=1.0947-1,\text{ or}$$

9.47% after $1.137+0.637=1.774$ sec.

4) to 5) etc.

$$6)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.0000848}\\1.2362848}}{1.23620}+\underset{0.0000848}{0.234\times0.0057\times0.0637}}-1=1.1119-1,\text{ or}$$

11.19% after $2.7295+0.0637=2.7932$ sec.

Comparing the two 50% load rejections:

Load Rejection in Hp	Governor Time in Sec	Max. Relative Speed Rise	Max. Relative Pressure Rise
2×16,000 to 8000	3.658	18.14%	About 32%
2×8000 to 0	2.7932	11.19%	About 33.5%

Again the shorter governor time causes a higher pressure rise but cuts materially the maximum relative speed rise.

At last a step-by-step computation is made for a 25% sudden load rejection, from 2×4000 to 0 hp ($\varphi=0.25$ to $\varphi=0.0$).

Table III shows the detail computation, the discharge beginning with interval 11a, thus requiring only about 4.5 intervals, as against about 8.5 intervals (Fig. 4) in the case of a load rejection from 2×16,000 to 2×12,000 hp.

The corresponding graphics are shown in Figs. 7 and 7a, and the detail computations of relative speed rise are given below.

Computation of Percent Relative Speed Rise for 2x4,000 to 0 Hp Load Rejection

(φ=0.25 to 0)

Values of φ_m taken from Table III.

(Slide rule computations.)

General formula:

$$\frac{\Delta n}{n}-1=\sqrt{\left(\frac{n_2}{n_1}\right)^2+\frac{3{,}233{,}000}{C}\times\varphi_m T_n}-1$$

C=13,816,000 n=225 rpm.

Speed rise due to insensitiveness and delay in gate movement.

$\varphi=0.25$ $(T_F=0.10)+(T_o=0.15)=0.25$

$$\frac{\Delta n}{n}-1=\sqrt{\underset{1.01461}{1+0.234\times0.25\times0.25}}-1=1.0072-1,\text{ or }0.72\%$$

after 0.25 sec.

Values of, φ_m and T_n

1) $\varphi_m=0.26042$ $T_n=0.22295$ 4) $\varphi_m=0.0787$ $T_n=0.3185$

2) 0.23708 0.3185 5) 0.0055 0.0637

3) 0.165 0.3185

$$1)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.01396}\\1.02857}}{1.01461}+\underset{0.01396}{0.234\times0.26042\times0.22925}}-1=1.0141-1,$$

or 1.41% after 0.25+0.22295=0.47295 sec.

$$2)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.01766}\\1.04623}}{1.02857}+\underset{0.01766}{0.234\times0.23708\times0.3185}}-1=1.02285-1,$$

or 2.28% after 0.47295+0.3185=0.79145 sec.

$$3)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.0123}\\1.05853}}{1.04623}+\underset{0.0123}{0.234\times0.165\times0.3185}}-1=1.0288-1,\text{ or}$$

2.88% after 0.79145+0.3185=1.10995 sec.

$$4)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.00587}\\1.06440}}{1.05853}+\underset{0.00587}{0.234\times0.0787\times0.3185}}-1=1.0317-1,\text{ or}$$

3.17% after 1.10995+0.3185=1.42845 sec.

$$5)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.00008}\\1.06448}}{1.06440}+\underset{0.00008}{0.234\times0.0055\times0.0637}}-1=1.03173-1,$$

or 3.173% after 1.42845+0.0637=1.49215 sec.

Comparing the two 25% load rejections:

Load Rejection in Hp	Governor Time in Sec	Max. Relative Speed Rise	Max. Relative Pressure Rise
2×16,000 to 12,000	3.173	9.71%	About 32%
2×4000 to 0	1.49215	3.173%	About 33.5%

Here the maximum relative speed rise of the first case is fully three times as high as that for the second case.

Attention is called to column o of the various tables. These efficiencies are taken from Fig. 1, curve c, and are the values based on normal net head of 167.5 ft and normal speed of 225 rpm. Since the computed speeds (column s) of the tables are materially higher, it will be necessary to introduce an additional correction. Fig. 8 of Part III (POWER REVIEW, Aug., 1942) shows the percent loss of efficiency for speeds above and below normal. However, these values apply only to the full load characteristic and not to any part loads ($\varphi<1$) as prevailing. It would lead too far afield to illustrate also these corrections here for "load-rejection conditions."

It is evident that any lower efficiency than shown in column o of the tables reduces the respective value φ_m of the computation and, therefore, depresses the φ curves, all of which will cause a somewhat smaller relative speed rise value than computed. Referring again to Table I, the maximum relative speed rise is given in column s as 38.0%; the actual speed then is 1.3804×225 or 310.59 rpm. It prevails under a

$Q=\frac{4{,}000\times8.8}{167.5\times0.7}=300.2$ cfs

$B=\frac{1.0217\times4.075}{13.217}=0.313$

Q = 600.4 cfs (for two units)

$V_m=\frac{13.217\times600.4}{1960}=4.075$ ft/sec

$T_c=0.3185$

H=167.5 ft—eff. 70% as per Fig. 2.

a	b	c	d	e	f	g	h	i	k	l	m	n	o	p	q	r	s
INTERVAL	T_p in sec	B	$\frac{V}{\Delta V}$	$\Delta H=$ 112 ΔV	$H+\overline{\sum h}$	F	$H_n=H_o+F$	$H+\overline{\sum h}+\overline{\sum\Delta F}$	$\overline{\sum h}$	% press. rise	Q in cfs	$Q_1=Q/2\sqrt{\text{col. i}}$	% eff.	hp	$\varphi=\frac{hp}{32{,}000}$	φ_m T_n	$100\frac{\Delta n}{n}$
11a	0	.313	4.075 .30	0	167.5	0	167.5	167.5	0	0	600.4	23.18	70.0	8,000	.25	φ_m=0.26042	0.72%
12	0.22295	.266	3.775 .787	33.6	201.1 .15	0.15	167.65	201.25	33.60	20.04	556	19.6	66.0	8,370	.26125	T_n=0.22295 φ_m=0.023708	1.41%
13	0.54143	.200	2.988 .99	88.14	222.19 .35	0.50	168.0	222.54	54.54	32.46	440	14.7	59.0	6,555	.210	T_n=0.3185 φ_m=0.165	2.28%
14	0.85993	.133	1.998 1.0	110.88	224.34 .35	0.85	168.35	224.69	56.34	33.46	294	9.81	49.0	3,680	.115	T_n=0.3185 φ_m=0.0787	2.88%
15	1.17843	.066	.998 .998	112.0	224.01 .2	1.05	168.55	224.21	55.66	33.22	147	4.9	35.0	1,313 Friction Load —	.041 .030	T_n=0.3185 φ_m=0.0055	3.17%
16	1.49693	0	0	111.776	224.66	1.15	168.65	224.66	56.01	33.24	0	0	0	0	.011 ÷ 2	T_n=0.0637	3.173%

(Slide rule computations)

Table III — Two Units—2 x 4,000 Hp to 0

$V = 13.217$ ft/sec

$B_o = \frac{V_o}{\sqrt{H_o}} = \frac{13.217}{\sqrt{167.5}} = 1.0217$

$(c^x)\ B_s = 1.0217 \times 0.92 = 0.9411$

$Q = 2 \times 980 = 1960$ cfs

$V - V_{P.R.} = (1 - .08)\ V = 12.15964$ ft

$Q_{P.R.} = 1960 \times 0.92 = 1803.2$ cfs at 167.5 ft

a	b	c^x	d^x	e	f	g	h	i	k	l	m^x	m^x_1	c	d	m	n	o	p	q	r	s
INTERVAL	T_p in sec	Gate B (T—P.R.)	ΔV (T—P.R.)	$\Delta H = 112\ \Delta V$	$H + \overline{\sum h}$	F	$H_n = H_o + F$	$H + \overline{\sum h} + \overline{\sum \Delta F}$	$\overline{\sum h}$	% press. rise	Q (T—P.R.)	$\sqrt{\frac{Q}{H + \overline{\sum h + \Delta F}}}$	B_T	$V_T = B_T \sqrt{\text{col. i}}$	Q_T	$\frac{Q_T}{2\sqrt{\text{col. i}}}$	% eff.	hp_T	$\varphi = \frac{hp_T}{32{,}000}$	$\frac{\varphi_m}{T_n}$	% speed rise $100\ \frac{\Delta n}{n}$
0	0	1.0217	13.217 0.0225	0	167.5	0	167.50 +0.1	167.50	0	0	1960	15.146	1.0217	13.217	1960	75.73	86.0	32,000	1.0		
1	0.3185	1.0118	13.1945 .061	2.5604	170.06 +0.1	0.1	167.60	170.16	2.5604	1.52	1955	14.95	0.931	12.15	1800	68.95	89.0	31,000	0.97	$\varphi_m = 0.9413$ $T_n = 0.637$	9.44%
2	0.637	1.0117	13.1335 0.0865	6.83	171.87 +0.10	0.20	167.70	171.97	4.27	2.54	1945	14.85	0.798	10.45	1550	59.1	90.0	27,300	0.854		
3	0.955	0.9916	13.0470 0.105	9.688	173.31 +0.20	0.4	167.90	173.51	5.61	3.34	1933	14.70	0.665	8.77	1298	49.25	87.0	22,300	0.696	$\varphi_m = 0.691$ $T_n = 0.637$	14.10%
4	1.2740	0.9815	12.9420 .108	12.096	174.386 +0.20	0.6	168.10	174.58	6.48	3.85	1919	14.60	0.532	7.03	1041	39.45	81.0	16,750	0.524		
5	1.5925	0.9714	12.8340 0.117	13.104	174.65 +0.30	0.90	168.40	174.95	6.55	3.89	1902	14.39	0.399	5.28	782	29.57	74.5	11,580	0.364	$\varphi_m = 0.372$ $T_n = 0.637$	16.49%
6	1.9110	0.9613	12.7170 0.118	13.216	175.06 +0.30	1.20	168.70	175.36	6.66	3.94	1885	14.25	0.266	3.52	521	19.65	66.0	7,270	0.227	$\varphi_m = 0.1737$ $T_n = 0.3185$	17.04%
7	2.2295	0.9512	12.5990 0.121	13.55	175.59 +0.25	1.45	168.95	175.84	6.89	4.07	1866	14.09	0.133	1.764	262	9.87	47.0	3,860	0.1203	$\varphi_m = 0.0451$ $T_n = \frac{.3185 \times 7}{10} =$	
8	2.548	0.9411	12.4780 0.123	13.776	175.83 +0.20	1.65	169.15	176.03	6.93	4.09	1848	13.929	0	0	0	0	0	0	0	0.22295	17.15%

(Slide rule computations)

Table IV — Two Units Each 16,000 hp at 167.5 ft Head with Pressure Regulator of 92% Net Turbine Discharge Capacity Fuii Load Rejection

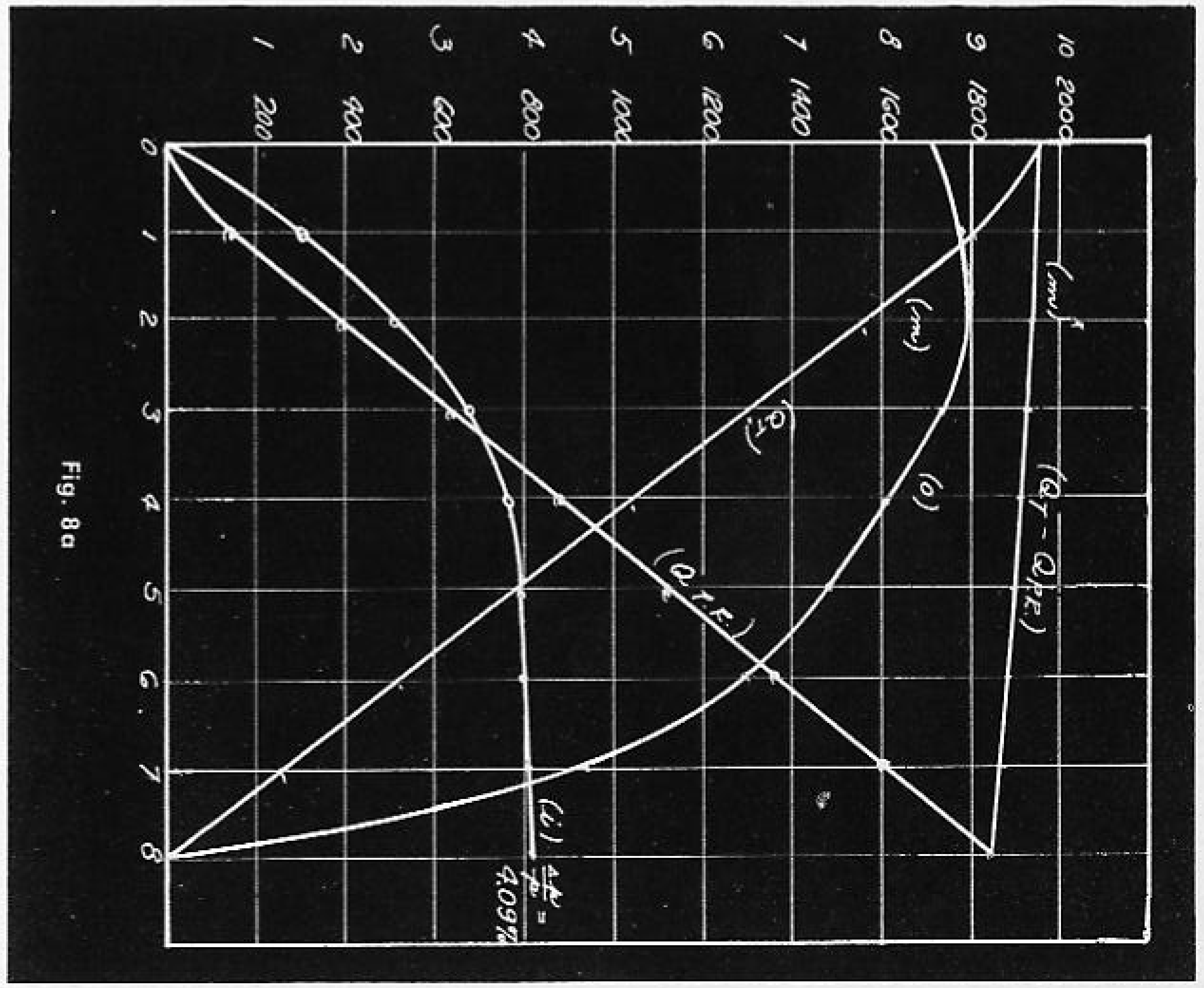


Fig. 8a

head of about 237.6 ft and not under 167.5 ft. The normal speed of 225 rpm under the net head of 167.5 ft corresponds to a speed of $\frac{225}{\sqrt{167.5}}$ or 17.3 rpm at 1 ft head (called unity-head-speed). Thus the maximum speed of 310.59 under 237.6 ft is $\frac{310.59}{\sqrt{237.6}}$ or 20.58 rpm unity speed, and the corrected relative speed rise is $100 \times \left(\frac{20.58}{17.3} - 17.3\right)$ or 18.96%.

It is this value and not 38.0% which must be used as basis for determining the additional loss in efficiency of the respective turbine runner characteristic. Also the last value of φ_m (Table I) is only 0.05803 so that even a relatively large additional loss of efficiency, say 10%, would affect this last item relatively little. Where φ_m is still larger, for instance the first item of Table A ($\varphi_m = 1.0658$), the absolute relative speed rise of 10.34% would require only a small percent additional correction.

From the above it can be concluded that this additional correction does not alter the results materially and, if disregarded, leaves a small margin in the safe direction.

Table IV is a computation to illustrate the beneficial effect of a pressure regulator discharging 92% of the full turbine discharge when the latter is reduced to zero discharge. This may be accomplished either by a synchronous or water-wasting by-pass of 92% turbine discharge capacity, or by a water-saving pressure regulator of 100% turbine discharge capacity. The latter, however, begins to close (on the water-saving principle at a slow rate) so that it has lost relatively 8% of its full capacity when the flow through the turbine has been stopped.

A full sudden load rejection of 2×16,000 to 0 hp ($\varphi = 1.0$ to $\varphi = 0$) has been used as basis, but with only eight intervals of time $T_c = \frac{2L}{a} = 0.3185$ sec, and not 16 as before, because the resultant relative pressure rise will be so moderate that the governor time can be materially reduced over that necessary in the case of Table I, namely from $T_g=5.123$ sec to $T_g=2.70245$ sec.

The 8% shortage of discharge, when the flow through the turbine has stopped and the pressure regulator opened, causes a corresponding deceleration of velocity in the pipeline from 13.217 to 12.15964 ft/sec as indicated in Table IV. Thus the discharge through the pipe is reduced not from 1960 cfs to zero as in Table I, but in the pipeline only to 1960×0.92 or 1803.2 cfs, namely along curve of column m^x, Fig. 8a, the resultant velocities in the pipe being represented in column d^x and plotted in Fig. 8.

The turbine discharge is reduced along the characteristic gate movement rate represented by the values B of column c, the value of B_8 at the end of eighth interval being $B_8=B_o\frac{1803.2}{1960}=1.0217\times0.9411$. The pressure regulator opens along a characteristic curve which is inverse to that of the turbine; therefore, the characteristic of the resultant deceleration of water in the pipe is the same as for the turbine, not down to zero but only to 0.9411.

The results of the various columns are computed in the same manner as for Table I. The percent relative pressure rise for the eighth interval (column l) is only 4.09 compared to about 32% without pressure regulator, as plotted in Fig. 9. The resultant discharge at end of interval 8 (column m^x) is not 1803.2 cfs but 1848. This is due to the fact that the prevailing head (column i) is 176.03 ft and not 167.5 ft. In fact, $1848\times\sqrt{\frac{167.5}{176.03}}$ is 1803.2.

To compute the speed rises it is necessary to compute the surplus energies developed in the turbine. Here B (column c) is reduced from 1.0217 to 0 in eight intervals along the same gate movement characteristic curve as for Table I but applying here to only eight intervals instead of 16. The turbine discharge decreases from 1960 to 0 (column m) and the output from 32,000 to 0 hp or from $\varphi=1.0$ to $\varphi=0$ (column q). The ultimate percent maximum relative speed rise is 17.15% compared to 38.04% without a pressure regulator.

On Fig. 9 are plotted for comparison the essential data of both cases, Tables I and IV. Note that the φ curve with pressure regulator does not exceed the value 1.0. Also, for the partial load rejections from full, a materially shorter governor time is required with the pressure regulator.

The detail computations for the full load rejection, $\varphi=1.0$ to $\varphi=0$, are shown below.

Computation of Percent Relative Speed Rise for 2x16,000 to 0 Hp Load Rejection

(With pressure regulator of 92% net turbine discharge capacity)

Values of φ_m taken from Table IV.

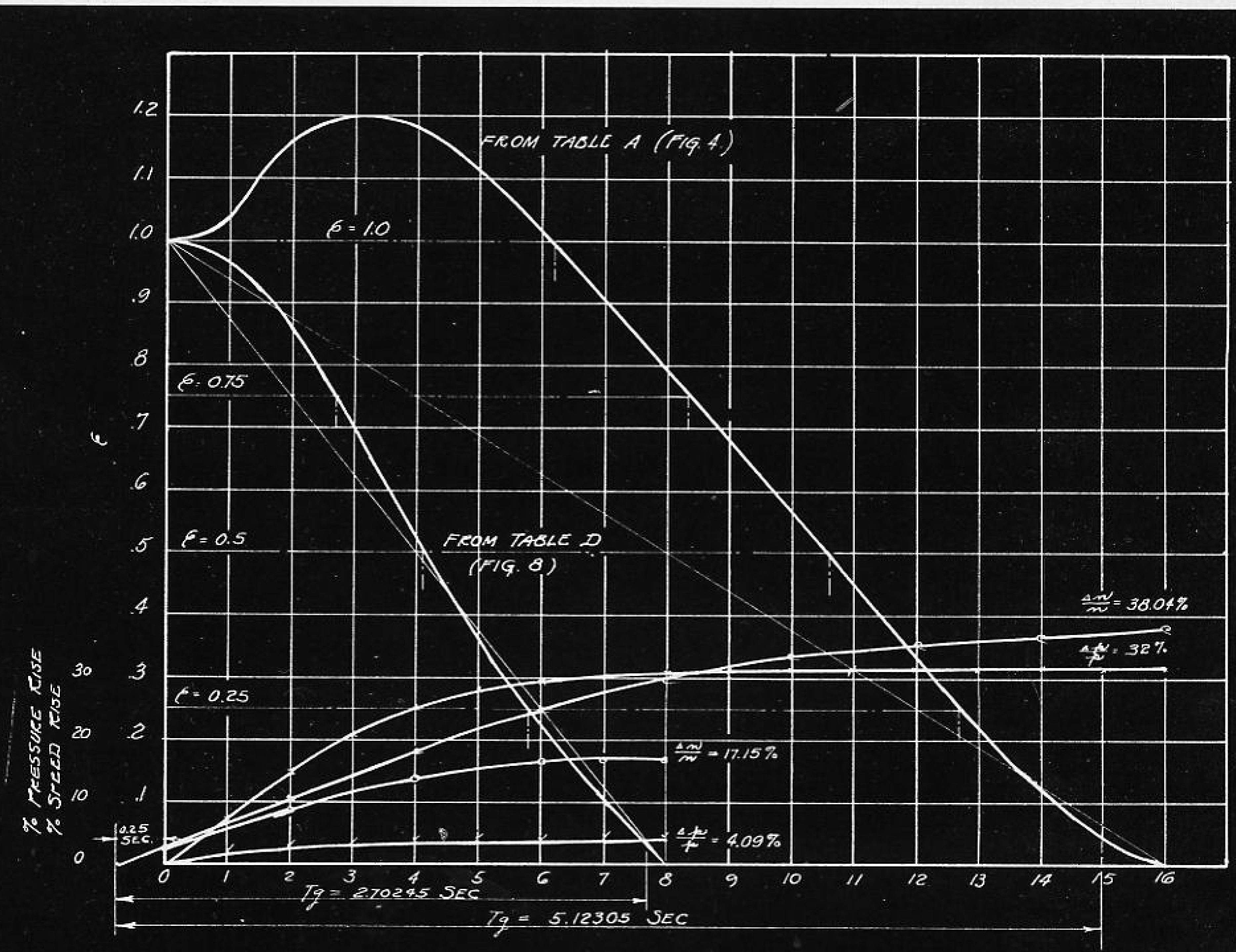


Fig. 9

$B = \frac{1.0217 \times 235}{1960} = 0.122604$

$V = \frac{13.217 \times 235}{1960} = 1.58604$ ft/sec

$Q = \frac{960 \times 8.8}{179.9 \times 0.2} = 235$ cfs (20% no-load eff.)

Friction load 3% = 960 hp (two units)

$Q = 1960$ cfs $H_n = 167.5$ ft

$V = 13.217$ ft/sec

$H_o = 180$ ft

For $Q = 235$ cfs

Net Head = 179.8 ft

$V = 1.5860$ ft/sec

F (Fig. 2) = 0.2 ft

a	b	c	d	e	f	g	h	i	k	l	m	n	o	p	q	q'	r	s	t	u	v	w
INTERVAL	B gate	T_p in sec	V ΔV	$\Delta H = 112\Delta V$	$H-\overline{\sum h}$	F	$H_n = H_o - F$	$H-\overline{\sum h}$ $-\overline{\sum \Delta F}$	$\overline{\sum h}$	% press. drop	Q in cfs	$\frac{Q}{2\sqrt{\text{col. i}}}$	% eff.	hp	$Q=\frac{hp}{32{,}000}$	$1-\varphi$	$1-\varphi_m$	% speed drop $100\,\frac{\Delta n}{n}$	% unit speed drop = corrected	% hp output loss	$1-\varphi_m'$	% net speed drop
1a	0.122604	0	1.5860 0.13	0	0	0	179.8	179.8	0	0	235	8.77	20	960	0.03	0.97	0.97 $1-\varphi_m=0.9537$ $T_n=0.0637$	2.97	2.97	0	0.97	2.97
2	0.133	0.0637 +.3185	1.7160 0.565	13.44	166.36	~ 0	179.8	166.36	13.44	7.47	256	9.925	42.0	2,005	0.0626	0.9374		3.71	3.42	0.25	$1-0.0463\,(1-0.0025)$ $=0.9538$	3.71
3	0.200	0.3822 +.7007	2.281 0.826	63.286	129.96 .15	.15	179.65	129.81	49.84	27.79	338	14.85	59.0	3,170	0.099	0.901	$1-\varphi_m=0.8965$ $T_n=0.637$	10.92	1.477	~ 0	$1-0.1035\,(1-\sim 0)$ $=0.8965$	10.92
4	0.266	1.0192	3.107 0.770	92.512	136.98 .35	.50	179.30	136.63	42.67	24.32	465	19.90	66.0	4,770	0.149	0.851						
5	0.333	1.3377	3.877 0.775	86.24	135.73 .40	.90	178.90	135.33	43.57	24.34	574	24.65	71.0	6,270	0.1956	0.8044	$1-\varphi_m=0.8012$ $T_n=0.637$	17.9	8.05	1	$1-1.988\,(1-0.01)$ $=0.8032$	17.92
6	0.40	1.6562	4.652 0.770	86.80	135.67 .40	1.30	178.50	135.27	43.23	24.22	690	29.65	76.0	8,060	0.252	0.748						
7	0.466	1.9747	5.422 0,765	86.24	135.49 .45	1.75	178.05	135.04	43.01	24.15	803	34.55	77.5	9,530	0.298	0.702	$1-\varphi_m=0.70$ $T_n=0.637$	24.84	16.2	3	$1-0.3\,(1-0.03)$ $=0.709$	24.62
8	0.533	2.2932	6.187 0.762	85.68	135.38 .70	2.45	177.35	134.68	42.67	24.06	916	39.45	81.0	11,350	0.359	0.641						
9	0.60	2.6112	6.949 0.7585	85.34	134.68 .65	3.1	176.70	134.03	42.67	24.14	1009	43.55	84.0	12,900	0.407	0.593	$1-\varphi_m=0.5896$ $T_n=0.637$	30.59	22.26	5	$1-0.4104\,(1-0.05)$ $=0.61012$	30.94
10	0.666	2.9307	7.7075 0.7585	84.95	134.42 .70	3.8	176.00	133.72	42.28	24.01	1118	48.9	87.0	14,900	0.465	0.535						
11	0.733	3.2487	8.4660 0.750	84.95	133.33 .80	4.4	175.40	132.53	42.67	24.23	1228	53.3	88.5	16,400	0.512	0.488	$1-\varphi_m=0.4523$ $T_n=0.637$	35.62	27.64	7	$1-0.5477\,(1-0.07)$ $=0.4906$	36.47
12	0.80	3.5672	9.2160 0.7342	84.00	133.87 1.15	5.55	174.25	132.72	41.53	23.83	1336	58.0	89.9	18,100	0.566	0.434						
13	0.866	3.8857	9.9502 0.7280	82.23	133.55 1.25	6.80	173.00	132.30	40.70	23.52	1471	64.0	89.9	19,860	0.622	0.378	$1-\varphi_m=0.387$ $T_n=0.637$	40.27	32.5	10	$1-0.613\,(1-0.10)$ $=0.4483$	42.01
14	0.933	4.2042	10.6782 0.728	81.536	132.17 1.15	7.95	171.85	131.02	40.83	23.75	1582	69.05	88.5	20,810	0.651	0.349						
15	1.0	4.5227	11.4062 0.638	81.536	131.15 1.15	9.1	170.70	130.00	40.70	23.84	1690	74.2	86.4	21,160	0.675	0.320	$1-\varphi_m=0.3045$ $T_n=0.637$	42.3	36.53	12	$1-0.6955\,(1-0.12)$ $=0.388$	47.31
16	1.0217	4.8412	12.0442 0.448	71.45	139.95 1.05	10.15	169.65	138.90	30.75	18.12	1782	75.6	86.0	24,400	0.7555	0.2445						
17	1.0217	5.1597	12.896 0.205	50.176	150.21 .75	10.9	168.90	149.46	19.44	11.51	1850	75.6	86.0	27,000	0.842	0.158	$1-\varphi_m=0.1422$ $T_n=0.637$	46.15	45.61	19	$1-0.8578\,(1-0.19)$ $=0.3052$	51.82
18	1.0217	5.4782	13.101 0.05	22.96	165.38 1.10	12.0	167.80	164.28	3.52	0.21	1942	75.6	86.0	31,200	0.976	0.024						
19	1.0217	5.7967	13.151 .03	5.6	165.72 0.1	12.1	167.70	165.62	2.08	0.124	1953	75.6	86.0	31,600	0.988	0.012	$1-\varphi_m=0.01473$ $T_n=0.637$	48.22	48.05	21	$1-0.9853\,(1-0.21)$ $=0.2227$	55.41
20	1.0217	6.1152	13.181	3.36	166.42 .05	12.15	167.65	166.37	0.28	0.16	1954.7	75.6	86.0	31,710	0.9918	0.0082						

(Slide rule computations)

Table V—Two Units Simultaneously Loaded to Full Gate

(Slide rule computations.)

General formula:

$$\frac{\Delta n}{n}-1=\sqrt{\left(\frac{n_2}{n_1}\right)^2+\frac{3,233,000}{C}\times\varphi_m T_n}-1$$

$C=13,816,000 \qquad n=225$ rpm.

Speed rise due to insensitiveness and delay in gate movement.

$\varphi=1.0 \qquad (T_F=0.10)+(T_o=0.15)=0.25$

$$\frac{\Delta n}{n}-1=\sqrt{\underset{1.0585}{1+0.234\times1.0\times0.25}}-1=1.0288-1,$$ or 2.88% after 0.25 sec.

Values of φ_m and T_n

1)	$\varphi_m=0{,}9413$	$T_n=0.637$	4)	$\varphi_m=0.17375$	$T_n=0.3185$
2)	0.691	0.637	5)	0.0451	0.22295
3)	0.372	0.637			

$$1)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.1403}\\1.1988}}{1.0585}+\underset{0.1403}{0.234\times0.9413\times0.637}}-1=1.0944-1,$$ or

9.44% after 0.25+0.637=0.887 sec.

$$2)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.103}\\1.3018}}{1.1988}+\underset{0.103}{0.234\times0.691\times0.637}}-1=1.141-1,$$ or

14.10% after 0.887+0.637=1.524 sec.

$$3)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.0554}\\1.3572}}{1.3018}+\underset{0.0554}{0.234\times0.372\times0.637}}-1=1.1649-1,$$ or

16.49% after 1.524+0.637=2.161 sec.

$$4)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.01292}\\1.37012}}{1.3572}+\underset{0.01292}{0.234\times0.17375\times0.3185}}-1=1.1704-1,$$ or

17.04% after 2.161+0.3185=2.4795 sec.

$$5)\ \frac{\Delta n}{n}-1=\sqrt{\underset{\substack{\underline{0.00235}\\1.37247}}{1.37012}+\underset{0.00235}{0.234\times0.0451\times0.22295}}-1=1.1715-1,$$

or 17.15% after 2.4795+0.22295=2.70245 sec.

Table V is a computation of relative pressure and speed drop for a full sudden increase of load from 0 to 2×16,000 hp ($\varphi=0$ to $\varphi=1.0$), requiring a full acceleration of flow to 1960 cfs and starting from the static head of 180 feet, again in 16 intervals each of 0.3185 seconds as used in Table I.

Instead of starting with interval 0 at zero flow ($\varphi=0$, column m), it begins with interval 1a at a total flow of 235 cfs at the net head of 179.8 ft (0.2 ft friction loss as per Fig. 2), assuming 20% turbine efficiency at which the output is 3% of the full output of both units, namely hp=960 (column p) $\varphi=0.03$ (column q). The B curve (column b) is inversed the same as in Table I. It remains at its maximum after interval 16 when the turbine is wide open.

The friction loss values F (column g) are here deducted from previous values (columns f and i). At the end of the 16th interval the net head (column n) has dropped to 169 65 ft, but the momentary head (column i) has dropped to 138.90 ft so that the full gate discharge (column m) is only 1782 cfs not 1960; in fact, $1782\sqrt{\frac{167.5}{138.9}}=1960$ cfs. The full discharge per turbine at 1 ft head, as per column n, is

$$\varphi_1=\frac{1960}{\sqrt{167.5}}=\frac{1782}{\sqrt{138.9}}=75.6.$$

The percent maximum relative pressure drops (column l), as well as points of columns b, d, m, and o of Table V, are plotted in Fig. 10. The maximum pressure drop occurs at the third interval, and the points following remain fairly constant up to interval 15. The curve drops to interval 16 because the gate opening characteristic B (column b) also rises less rapidly.

Since the available head at end of interval 16 is less than the normal net head, the units are still short of hp output, namely 24,400, or only 75.5% ($\varphi=0.7555$) of full, so that a shortage of 24.45% ($1-\varphi=0.2445$) still prevails, as graphically shown in Fig. 10a.

From subsequent intervals 17 to 20 of Table V the net head gradually recovers from 138.9 ft toward the full net head of 167.5 ft so that the flow (column m) comes up in spite of the fact that the turbine gates (already wide open, B=1.0217) cannot open farther. Theoretically, therefore, the output (column p) gradually approaches the full value of 32,000 hp ($\varphi=1.0$), thus also reducing the shortage ($1-\varphi$) asymptotically to zero, whereby the relative speed rise (column s) also approaches asymptotically a maximum value as indicated in Fig. 10a.

Since there is no additional surplus of output ($\varphi=1.0$, and not >1.0), the maximum speed drop will remain and cannot recover to the normal speed.

Since the efficiencies (column o) are based on a constant speed of 225 rpm under the normal net head of 167.5 ft, i. e., as previously set forth, on a speed of $\frac{225}{\sqrt{167.5}}=17.385$ rpm at one foot head, and since the values of unit speeds computed for each respective speed of column s and corresponding heads (column i) deviate materially, an additional correction must be applied. Column t shows the corrected value (s) obtained as follows:

Taking the value (s) of 42.3%, the net head being 138.9 ft (column i, interval 16):

Actual speed $n'=225\ (1-0.423)=129.8$ rpm.

Thus unity speed $n'_1=\frac{129.8}{\sqrt{138.9}}=11.014$ rpm.

The normal unity speed n_1 is $\frac{225}{\sqrt{167.5}}=17.385$ rpm.

Thus $\frac{n'_1}{n_1}=1-\frac{11.014}{17.385}=0.3653$ or 36.53%.

$$\text{Values of column t}=\left[1-\left(1-\frac{\text{values of column s}}{100}\right)\times\sqrt{\frac{167.5}{\text{values of column i}}}\right]\times100.$$

For these corrected percent relative speed drops (t), the corresponding losses in efficiency for the respective values φ_m (taken from the runner characteristic, not listed here in detail) are shown in column u.

Accordingly, the respective values of $1-\varphi_m$ in column r must be corrected as follows:

For example: $1-\varphi_m=0.7$ in column r; efficiency loss 3% (column u). $\varphi_m=1-0.7$ or 0.3 and $\varphi'_m=0.3\ (1-0.03)=0.291$; therefore $1-\varphi'_m$ (column v)$=0.709$ instead of 0.70 as before.

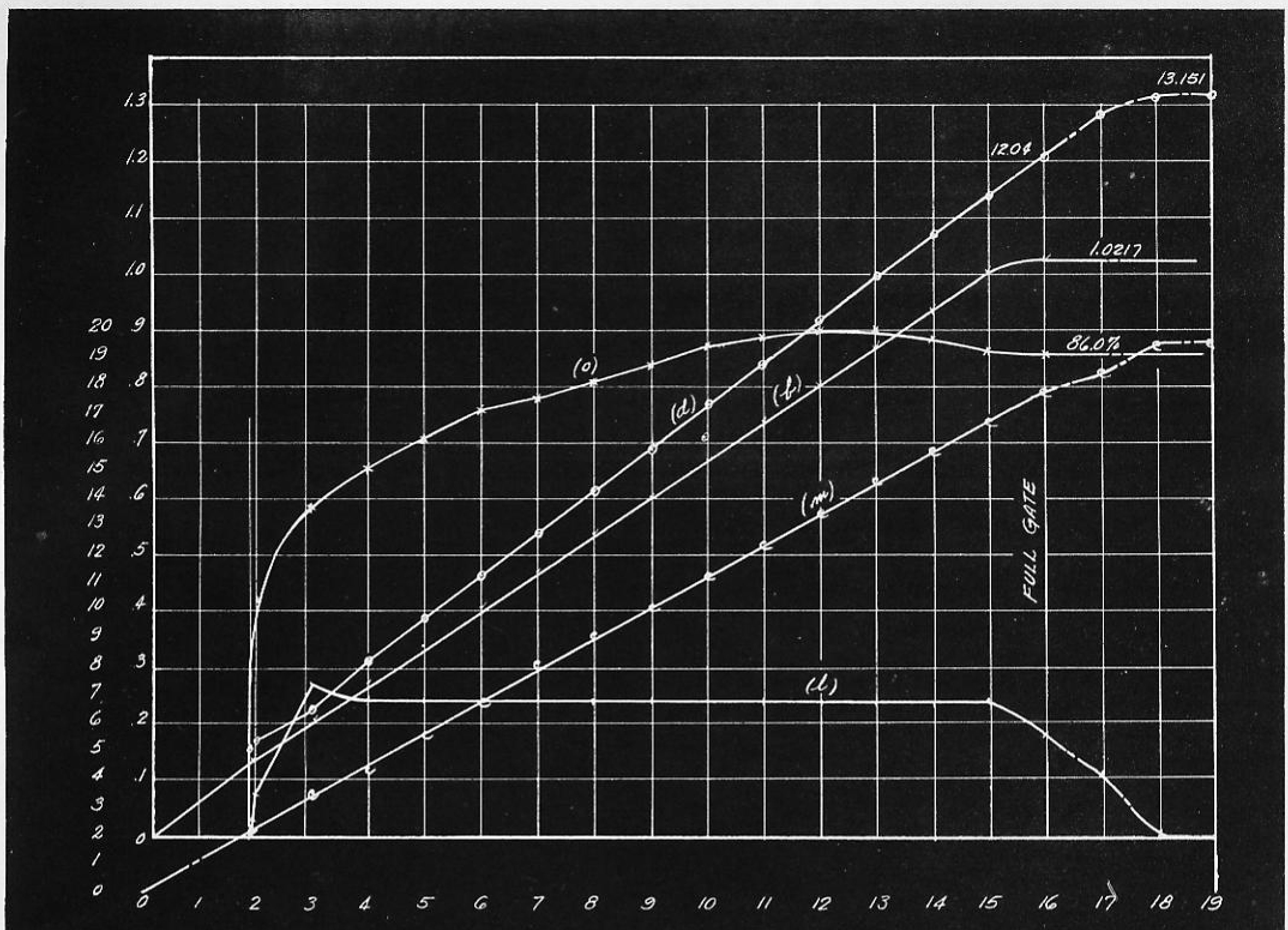


Fig. 10

Computations of percent relative speed drop (column s) are shown below.

Computation of Percent Relative Speed Drop for 2x0 to Full Hp

($\varphi=0.1$ to full)

Values of φ_m taken from Table V.

(Slide rule computations.)

General formula:

$$\frac{\Delta n}{n}=1-\sqrt{\left(\frac{n_2}{n_1}\right)^2-\frac{3{,}233{,}000}{C}\times\varphi_m T_n}$$

$C=13{,}816{,}000$ $n=225$ rpm.

Speed drop due to insensitiveness and delay in gate movement.

$\varphi=0.10$ $(T_p=0.10)+(T_o=0.15)=0.25$

$$\frac{\Delta n}{n}=1-\sqrt{\underset{0.9415}{1-0.234}\underset{0.0585}{\times1.0\times}0.25}=1-0.9703=0.0297,\text{ or }2.97\%$$

after 0.25 sec.

Values of φ_m and T_n

1)	$1-\varphi_m=0.9537$	$T_n=0.0637$	6)	$1-\varphi_m=0.4523$	$T_n=0.637$
2)	0.8965	0.637	7)	0.387	0.637
3)	0.8012	0.637	8)	0.3045	0.637
4)	0.70	0.637	9)	0.1422	0.637
5)	0.5896	0.637	10)	0.01473	0.637

$$1)\ \frac{\Delta n}{n}=1-\sqrt{\underset{\begin{array}{c}\underline{0.01421}\\0.92729\end{array}}{0.9415}-0.234\times\underset{0.01421}{0.9537}\times0.0637}=1-0.9629=0.0371,$$

or 3.71% after $0.25+0.0637=0.3137$ sec.

$$2)\ \frac{\Delta n}{n}=1-\sqrt{\underset{\begin{array}{c}\underline{0.1337}\\0.79359\end{array}}{0.92729}-0.234\times\underset{0.1337}{0.8965}\times0.637}=1-0.8908=0.1092,$$

or 10.92% after $0.3137+0.637=0.9507$ sec.

3) to 9) etc.

$$10)\ \frac{\Delta n}{n}=1-\sqrt{\underset{\begin{array}{c}\underline{0.02194}\\0.26811\end{array}}{0.29005}-0.234\times\underset{0.02194}{0.01473}\times0.637}=1-0.5178=0.4822,$$

or 48.22% after $5.4097+0.637=6.0467$ sec.

The corrected relative speed changes tabulated in column w are computed below.

Computation of Percent Relative Speed Drop for 2x0 to Full Hp

(Corrected for loss due to loss in efficiency caused by speed drop.)

($\varphi = 1.0$ to full)

Values of φ_m taken from Table V.

(Slide rule computations.)

General formula:

$$\frac{\Delta n}{n} = 1 - \sqrt{\left(\frac{n_2}{n_1}\right)^2 - \frac{3.233,000}{C} \times \varphi_m T_n}$$

C = 13,816,000   n = 225 rpm.

Speed drop due to insensitiveness and delay in gate movement.

$\varphi = 1.0$ $(T_F = 0.10) + (T_o = 0.15) = 0.25$

$$\frac{\Delta n}{n} = 1 - \sqrt{1 - 0.234 \times 0.5 \times 0.25} = 1 - 0.98526 = 0.01474, \text{ or } 1.4174\%$$

after 0.25 sec.

Values of φ'_m and T_n

1)	$1-\varphi'_m = 0.9538$	$T_n = 0.0637$	6)	$1-\varphi'_m = 0.4906$	$T_n = 0.637$
2)	0.8965	0.637	7)	0.4483	0.637
3)	0.8032	0.637	8)	0.388	0.637
4)	0.709	0.637	9)	0.3052	0.637
5)	0.61012	0.637	10)	0.2227	0.637

$$1)\ \frac{\Delta n}{n} = 1 - \sqrt{\underset{\frac{0.01421}{0.92729}}{0.9415} - 0.234 \times \underset{0.01421}{0.9538 \times 0.0637}} = 1 - 0.9629 = 0.0371,$$

or 3.71%.

$$2)\ \frac{\Delta n}{n} = 1 - \sqrt{\underset{\frac{0.1337}{0.79359}}{0.92729} - 0.234 \times \underset{0.1337}{0.8965 \times 0.637}} = 1 - 0.8908 = 0.1092,$$

or 10.92%.

3) to 9) etc.

$$10)\ \frac{\Delta n}{n} = 1 - \sqrt{\underset{\frac{0.0333}{0.19884}}{0.23214} - 0.234 \times \underset{0.0333}{0.2227 \times 0.637}} = 1 - 0.4469$$

= 0.5541, or 55.41%.

Note that the values of $(1-\varphi'_m)$ column v as plotted in Fig. 10a drop materially below the values $(1-\varphi_m)$, column r, as the relative speed drop (s) increases, especially after the 16th interval where the turbine gates are wide open. As a result of this the corrected relative speed drop, curve w, also rises above curve s as the loss in efficiency (column u) becomes greater and greater (beyond interval 20 at which point the net head has become very close to the final value, 167.5 ft) so that the output shortage increases again — v drops — and causes an upturn of the relative speed drop, curve w, as indicated by the arrow in Fig. 10a.

As previously stated, this case in its full severity — full load thrown on "suddenly" — cannot be produced in actual practice, not even by means of a water rheostat load.

No computations or guarantees should, therefore, be named for sudden load increases equalling the full output of the unit, but only for fractional load increases, enabling the governor to open up temporarily beyond that discharge which is ultimately required to hold the new, increased load at normal rpm.

Even such part-load calculations are of interest only in case the unit operates as an independent energy source, not interconnected with an electrical network.

In the next issue of POWER REVIEW this article will be concluded, with a brief reference to other hydraulic auxiliaries, such as racks, headgates, sandtraps, inlets to pressure pipes, butterfly valves, gate valves, and tailrace gates.

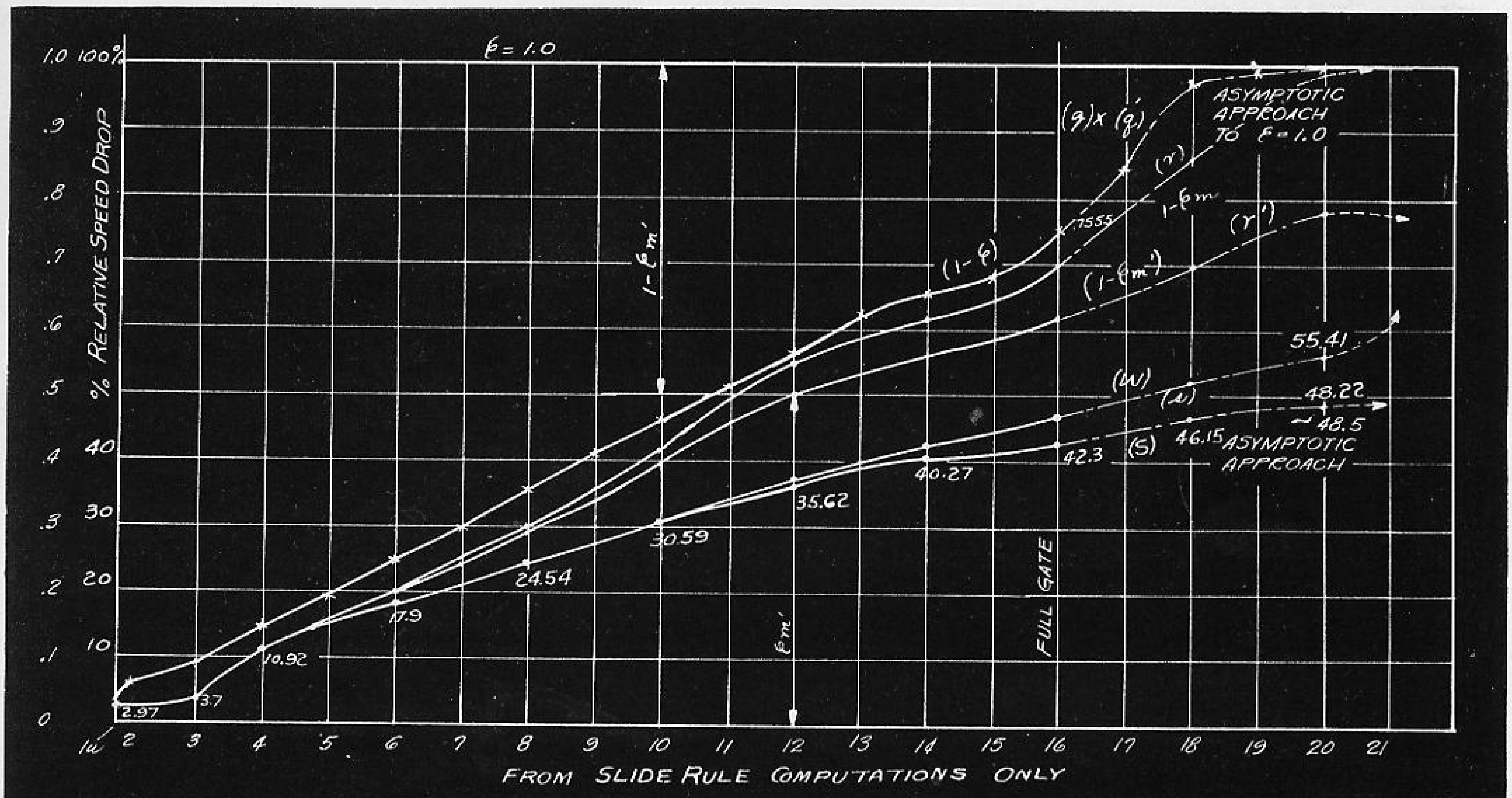
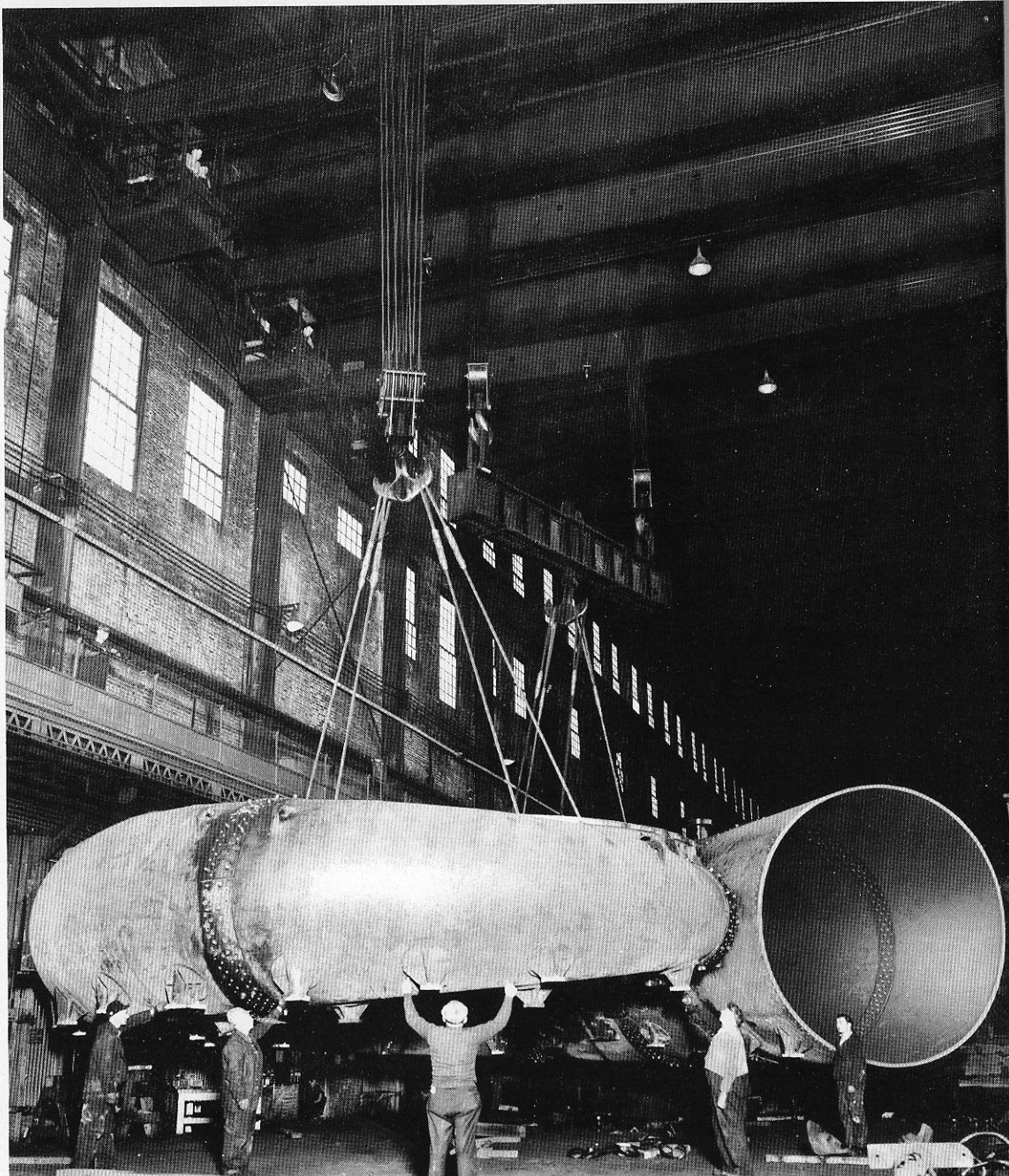


Fig. 10a



Weighing approximately 350,000 lb, a 152 in. inlet diameter hydraulic turbine casing is supported by hooks of three cranes.

Part V

ACCESSORIES

PART V — ACCESSORIES

Although belonging not strictly to the prime-mover equipment within the scope of manufacture of the Allis-Chalmers Mfg. Company, the accessories pertaining to the hydraulic part of an hydro-electric development are also of some importance since they affect the successful operation of a plant — their failure impairs the revenue derived from the prime movers.

These accessories embrace all the items from the intake of the water to the end of the tailrace downstream of the hydraulic prime mover and involve not only mechanical engineering but also civil engineering.

Intakes

Intakes are located at the head of the hydro development, being a pond or storage reservoir, or a river. In a pond of fair capacity the water flows at low velocity in the direction of the intake. Therefore, sand or silt may be deposited there so that it is not carried through the intake. No matter how low this velocity may be, driftwood or other floating or suspended matter will, however, finally reach the intake and may eventually clog up the water passages through the prime mover (orifice of an impulse wheel, or guide vane and runner passages of a Francis or propeller turbine). Even grass or leaves, if arriving simultaneously in large quantities, can lead to material reduction in discharge and, consequently, in reduction of output of the prime mover. Fish and eels may also become lodged in these passages and may be caught in the control mechanism of the turbine. In one case over 40 years ago, a governor servomotor stuck, and even high oil pressure would not move the mechanism of the cylinder gate controlling the water to the guide case. Examination disclosed that a large salmon was "caught," and its body was sufficiently tough to stall the servomotor.

Where an intake is located on a river carrying sand and silt, provision should be made by sand traps and sluice gates which, when opened periodically, permit flushing the material out again.

Figure 1 shows a typical arrangement. Naturally the bottom of such a sand trap must be inclined toward the sluice opening. The area of the sand trap basin should be made as liberal as possible so that the water has an opportunity to flow at low velocity, allowing the sand or silt to drop to the bottom.

Suspended and floating material is withheld from the canal or conduit by means of trash racks, placed across the entire cross-section area of the intake. They consist of structural steel bars, usually of rectangular cross-section, extending from the bottom of the intake sill to an elevation either below normal head water level (submerged racks) or above the normal level (open racks).

They must be accessible from a floor or deck which should be placed safely above any flood water level; otherwise it would be impossible to clean the racks at times of flood, the most essential time because much driftwood is held at the racks. Any clogging will reduce the passage area through the racks and will cause a drop in available head. It will increase also the water load on the rack structure, causing collapse under such emergency conditions if the rack is not of adequate design. Such collapse will allow all the drift matter to

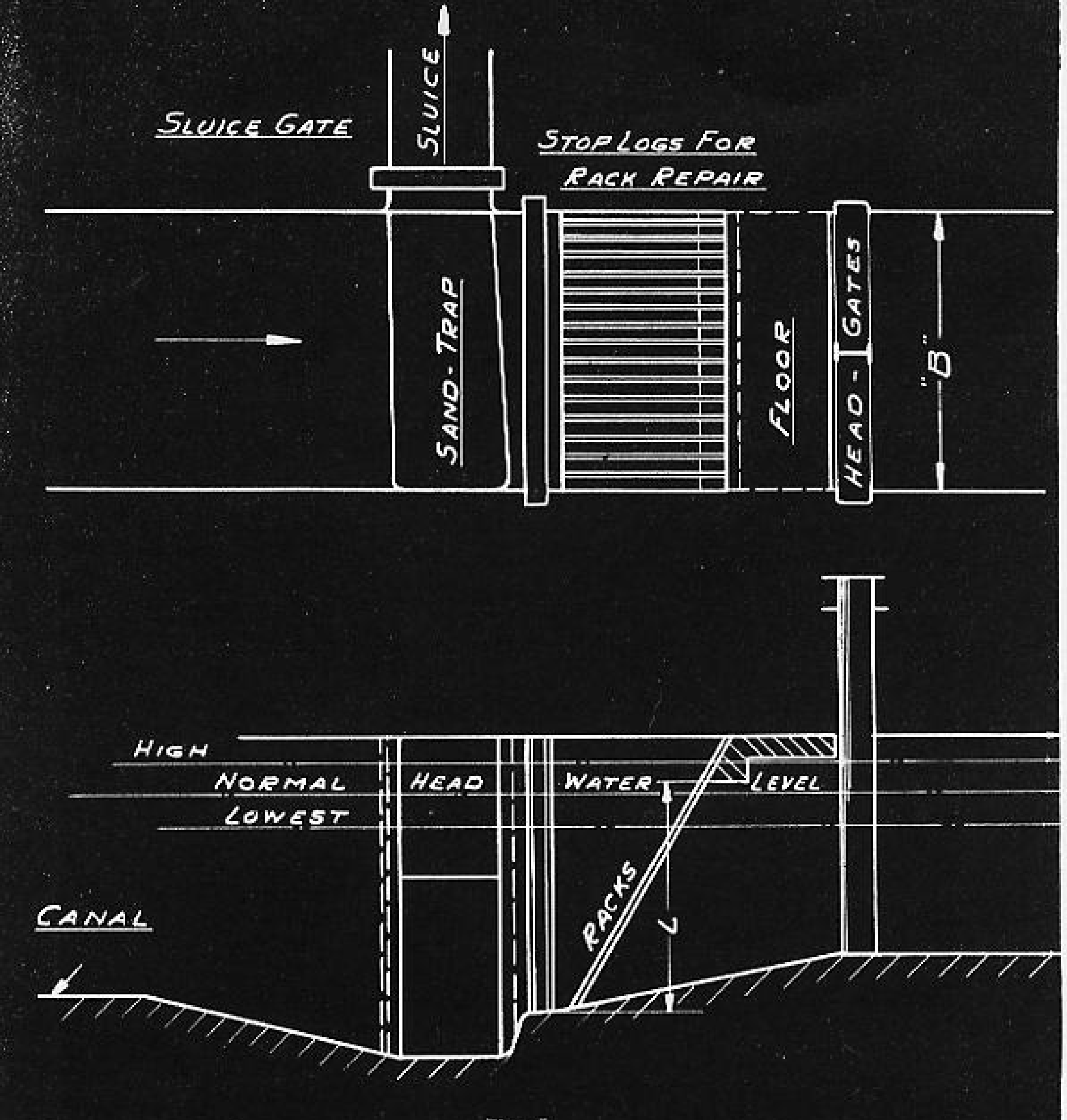


Fig. 1

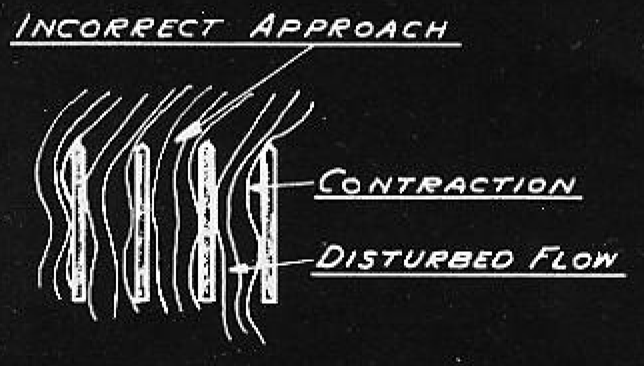


Fig. 2

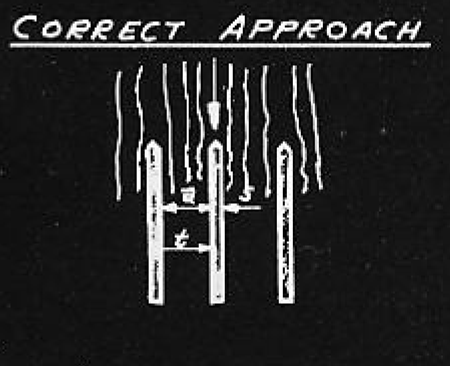


Fig. 2a

clog up the turbine, causing a complete shutdown of all the units supplied with this water.

Naturally the rack bars reduce the effective area. In Fig. 2 let a be the actual distance between two rack bars, s the thickness of each bar and n the number of bars required for the entire surface. The net area for a depth L of rack bar is:

$A_n = n\,a\,L + aL$

The gross areas $A_g = n\,(a+s)\,L + aL$, over the width B of the intake (Fig. 1).

$B = n\,(a+s) + a$

The relative reduction in area is : $\frac{A_n}{A_g} = \frac{n_a + a}{n(a+s)+a}$. For small turbines the width a must be made small because of small passages in the turbine, and s is fixed by the structure so that it may be a material portion of a. The net rack area A_n is then considerably less than the gross area, $n\,a\,L \ll n(a+s)L$.

Since the water velocity is $V = \frac{Q}{A_n}$, where Q represents the quantity of water passing through the racks, it follows that the depth L of the rack must be increased accordingly. The depth L should be taken as the vertical distance between rack-sill and the lowest operating water level above the sill for an open rack arrangement. It can be seen at once that A_n is utilized fully only if the water passes parallel to the rack bars, as indicated by the arrows in Fig. 2. If the water reaches the rack bars at an angle, as in Fig. 2a, then a contraction takes place, thereby reducing the effective area and involving a higher water velocity, which in turn results in a drop h_s of water level past the bars. This constitutes a loss of available head H at the turbines so that the output loss will be $hp\left[1-\left(1-\frac{h_s}{H}\right)^{3/2}\right]$, H being the normal net head on the unit, and hp the corresponding horsepower output. The loss h_s affects the output more severely under low heads H.

For instance: $h_s = 1$ ft, and $H = 20$ ft. The percent output loss is $1-\left(1-\frac{1}{20}\right)^{3/2}\times 100$ or in excess of 7½%. A similar loss is caused by a partial clogging of the racks by driftwood and so forth. An arrangement as shown in Fig. 3 is defective because the water arrives at the racks at an angle which is most pronounced for the upstream units.

An actual experience with such a plant may be cited here. This plant was built in 1897 on the Rhine River. Twenty hydro electric units are placed as shown in Fig. 3. The head race, or intake canal, runs parallel to the river into which the 20 turbines discharge directly at right angles to the flow in the river.

The deck or rack floor was placed only a foot or two above normal high water level, and the structure supporting the racks was not divided into 20 flumes, but allowed free circulation of the water behind the racks along the entire length of the powerhouse.

The first five upstream units furnish energy for the production of aluminum, operating permanently at full capacity, all of which causes a very harmful loss of head at the outset through the racks, even to such extent that large salmon in the canal were washed against the racks with such force that they were unable to get away from the racks when once pressed against them. Naturally, if even large salmon are held to the racks, trash and floating matter will also be held so firmly that it takes much more effort to raise it when cleaning the racks. The effective velocity of water should not exceed 2.5 to 3 ft/sec.

The dam across the Rhine River was not completed at the time the plant had to start operation; the inflow to the canal was temporarily obtained from a pond formed by a coffer dam upstream in the river, the crest of which was naturally above any flood water level expected in the river during construction of the permanent dam formed by control gates. A heavy late snowfall of several feet in the water shed was quickly followed by a Syrocco (a very warm wind blowing from Africa and Italy across the Alps, which causes the snow to melt in a few days). The river rose rapidly far above the maximum level anticipated, and the temporary coffer dam allowed the water in the forebay of the power plant to rise so far above the rack deck that cleaning of the racks became an impossibility.

The flood also brought an enormous amount of debris so that the racks became so clogged that the output of the 20 turbines was greatly reduced. Realizing the danger of a collapse of the rack structure, it was decided to shut the plant down entirely, except that the aluminum concern refused to stop its five units because it would have caused freezing of the furnaces. Gradually the racks of these units became so water-loaded that they caved in. This allowed all debris to enter the turbine flumes, not only of the five units but of some adjacent units. The job of repairing these racks later involved work by divers for several months.

The shutting down of the 15 remaining units caused such an additional static head that the civil engineers in charge became fearful that the powerhouse might be pushed over into the river, and it was decided to blast the upstream coffer dam to relieve this exceedingly precarious situation. Such exciting experiences leave a deep impression and a desire to warn against a repetition of such circumstances. Although, as indicated in Fig. 3, there was a sluice flume at the downstream end of the forebay, it was impossible to raise the sluice gate because of the excessive waterload upon it caused by a flood level far above any expected emergency condition. The absence of individual piers between each of the 20 flumes made the usual provision of stop-logs in front of the racks impossible, so that the divers had to do the repair work adjacent to these units which had to be put back in operation as soon as was practical. Fig. 3a illustrates a hydraulically-correct arrangement.

Intake Gates

It will be noted from Fig. 1 that the sill of the intake gate is higher than the sill of the racks since the full cross area is again available and since a higher velocity can be allowed downstream of the racks. This can also be attained by reducing the width B of the flume. The intake gates must be so designed that they can be raised and lowered against any maximum water level that may prevail, and they should be reasonably water-tight. Where the powerhouse is directly behind the rack floor, the intake gates can be located inside the powerhouse, thereby dispensing with individual gate hoisting mechanisms. This may also be done by enlarging the rack floor and employing a gantry crane along the entire length of the powerhouse. Gate tables can be used which are lowered into place or lifted out by the powerhouse crane or by the gantry crane. These points are discussed here at some length because it is essential that all intakes be laid out so that the water can be safely and completely stopped under any abnormal conditions. Failure of such performance may lead to serious damage to the powerhouse equipment.

Intake to Closed Conduits

In closed conduits, such as pressure tunnels, pipelines, etc., higher water velocities can be allowed than in the case of open canals. The transition from open canal to closed conduit should be so arranged that air is not drawn into the conduit. Therefore, the top of the intake to the conduit must be safely submerged. If V_1 is the velocity of water in ft/sec in the conduit, then $\frac{V_1^2}{2g} = H_a'$ represents the head due to velocity of approach; and, if the lowest water level is not at least H_a' feet above the top of the conduit, it is bound to pull in air. To reduce H_a, therefore, it is advisable to enlarge the intake area of the conduit. Fig. 4 illustrates a typical inlet. Care should be taken that no drift matter enters the closed conduit. Therefore, the racks should be located close to the intake, and the space between racks and intake should be covered.

If the intake of the closed conduit is quite a distance from the intake into the open canal, or open conduit, then provision should again be made for shutting off the water from the pressure conduit. This can be accomplished by a head gate as previously outlined or by a shut-off valve at the top of the conduit, as indicated in Fig. 4. Unless an overflow is provided, the top of this forebay wall should be sufficiently high to prevent flowing of water when the flow in the pressure conduit is stopped.

If a shut-off valve is provided, provision must be made to admit air at the downstream side of the valve; otherwise a vacuum is caused which may lead to a collapse of the pipeline directly downstream of the shut-off valve. Admission of air is accomplished by an open standpipe, or by an air-vent valve directly bolted to the pipe. In case of an open stand-(or vent-) pipe, its top should be safely above the highest water level in the forebay. It is not advisable to use such a pipe if the water velocity V_4 at the junction with the main pipe is high and if there is a considerable distance from water level in forebay to the point of junction. The water level in the standpipe will naturally be lower than the operating level in the forebay. Or $H_a = H_t - F - \frac{V_4^2}{2g}$, where H_t is the vertical distance between point x of junction and water level in forebay, F the friction loss in intake from water level to point x, and $\frac{V_4^2}{2g}$ the velocity head at x.

In case of a break in the pipeline below point x the velocity V_4 will naturally greatly increase so that H_a may drop below point x. In this case the standpipe will draw in air. The same action takes place when the air valve (used instead) functions. The greater flow will also increase the velocity head at the entrance, and air may be drawn in by the forming of vortexes in the forebay.

For pipelines of small diameter a reliable standard design of gate valve may be used at economical cost. Where a large diameter is involved, the less costly butterfly valve type should be employed. The details of such valves will be discussed further below.

Pressure Pipeline

In regard to calculation of thickness of plates of pressure pipes (or of wood staves and bands), joints, anchorage, etc., there are sufficient data available in textbooks to make reference here superfluous. Therefore, only features not always

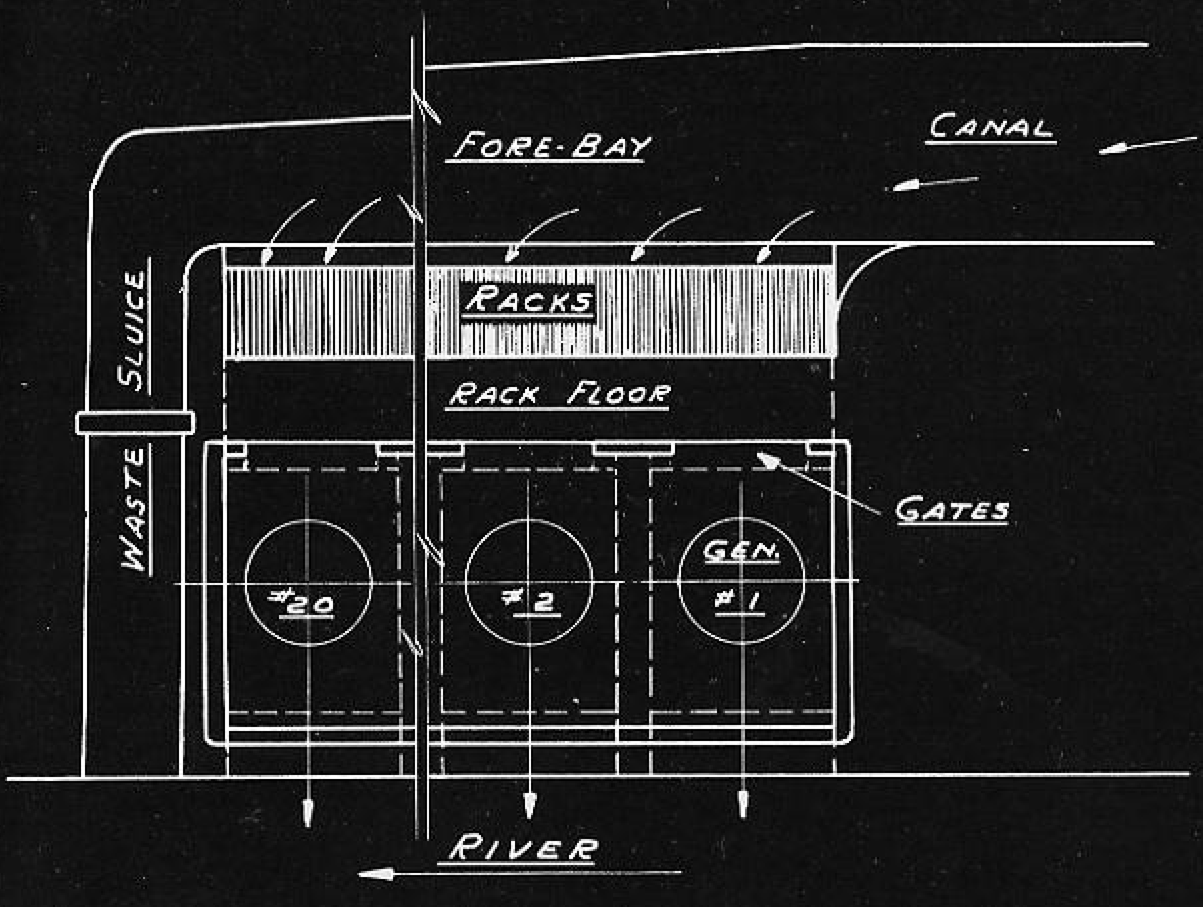


Fig. 3

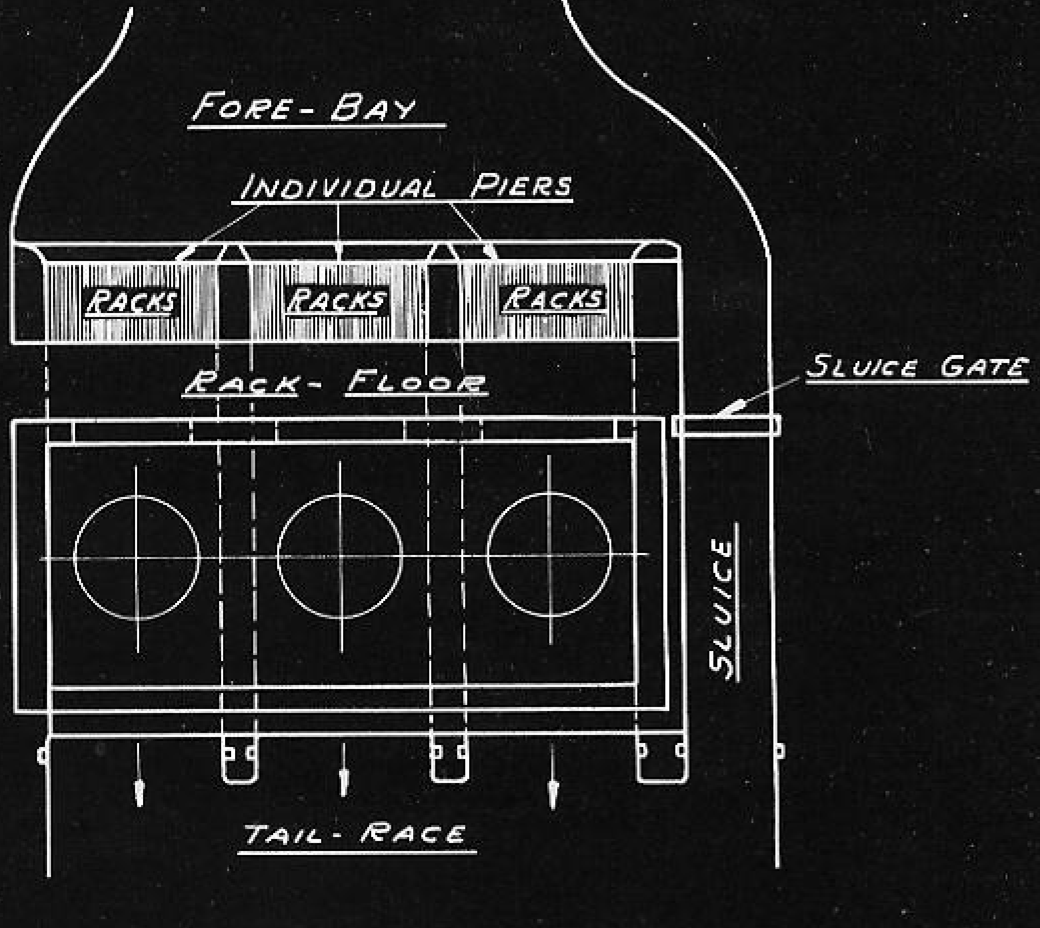


Fig. 3a

referred to and yet of practical value will be discussed.

The shortest distance from forebay to powerhouse level is the straight line between these two points. This is not always possible on account of the geographical conditions so that bends may have to be introduced in both the vertical and horizontal plane.

Figure 5 illustrates a straight line pipe. The static head (no water flowing) is H; the total friction loss in the pipe at full flow is F so that the net effective head at the pipe end E is $H_n=H-F$. The line ab is called the hydraulic gradient. It is also a straight line if the pipe line is of uniform diameter over the entire length L. At any intermediate point x the friction loss is $F_x=\frac{F\,L_x}{L}$, and the net head at point x is $H_{nx}=H_x-F_x$.

If the pipe line has two different diameters (Fig. 5a), then the total friction loss is made up of F_1 and F_2. The hydraulic gradient bc for length L_2 is steeper if the diameter d_2 is smaller than d_1 ($F_2>F_1$).

Figure 6 illustrates a case where the pipeline is not a straight line in the vertical plane but begins with a section L_1 of smaller slope with a diameter d_1, resulting in a friction loss F_1 (gradient line ab) for a flow Q. With a distance (xy) between static water level (Q=0) and end of pipe L_1 the net pressure head would be yb. Let L_2 be the steeper section with a diameter d_2. The friction loss in that section is F_2 (gradient line bc). The total static head is H_t, and the net head for flow Q is $H_n=H_t-(F_1+F_2)$.

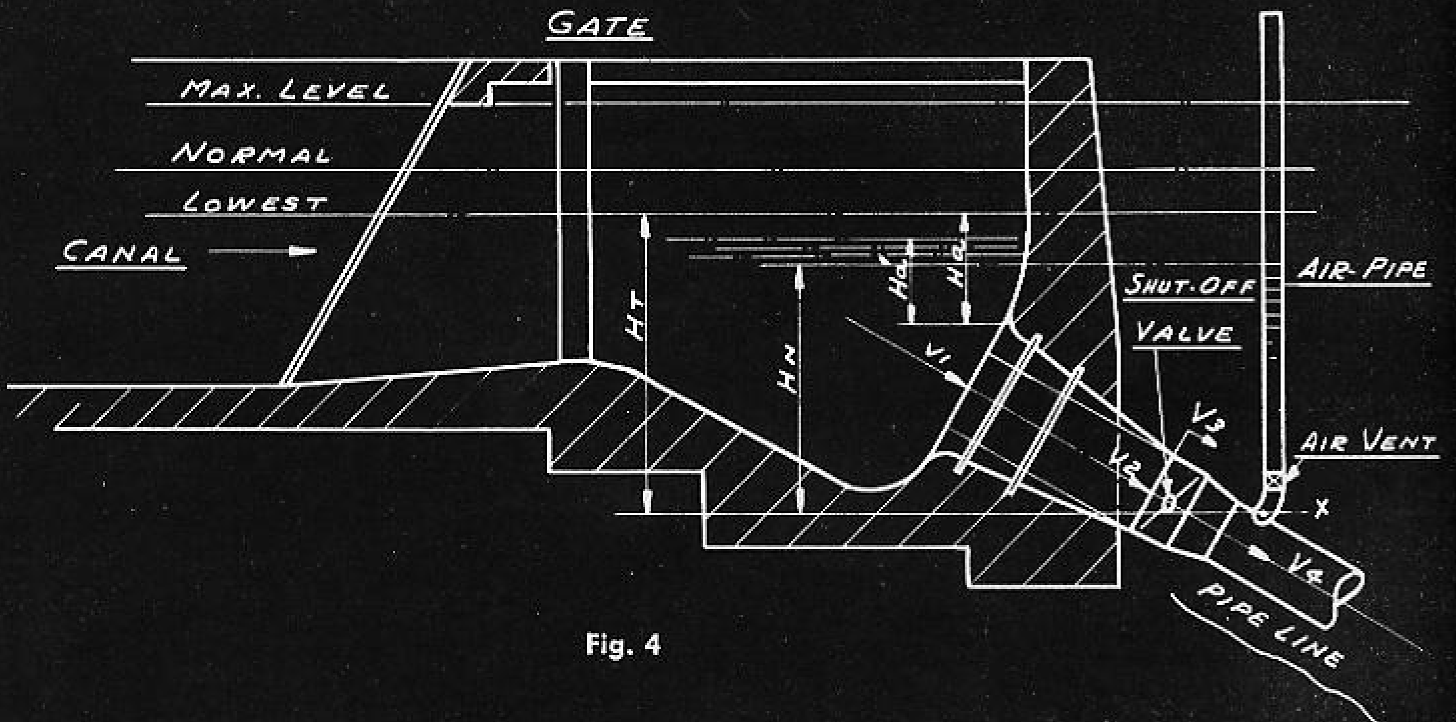


Fig. 4

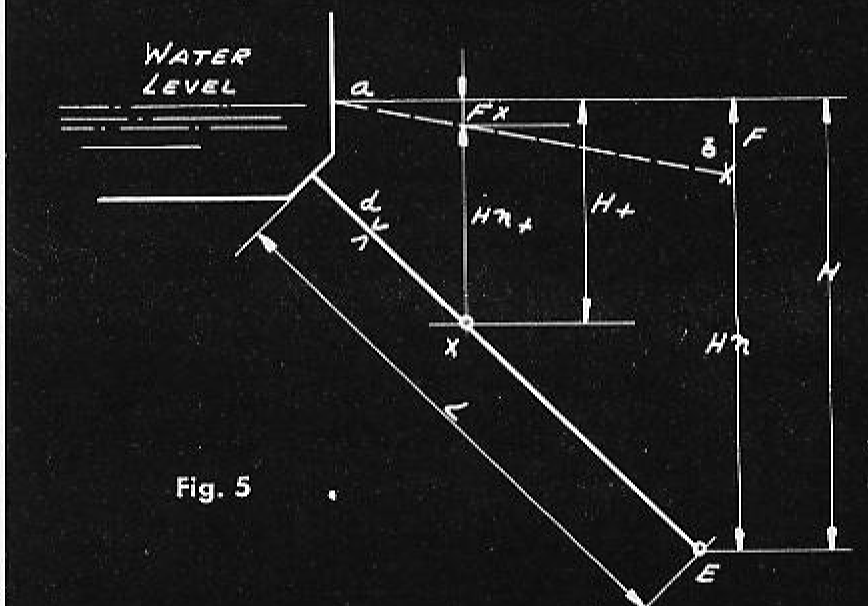


Fig. 5

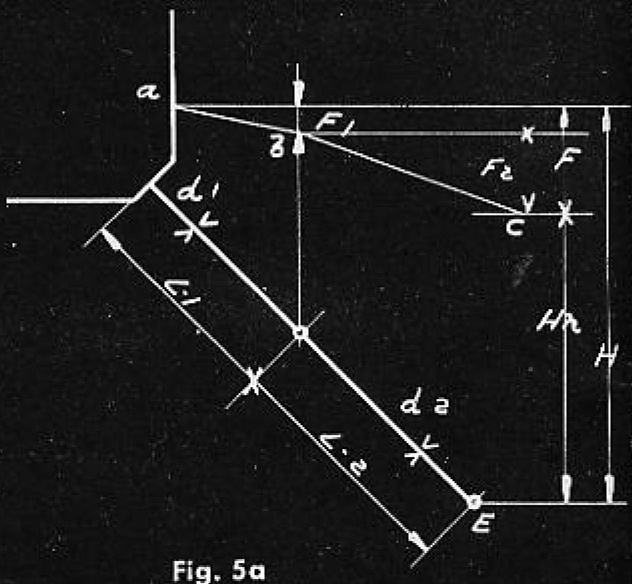


Fig. 5a

If for some reason the flow should greatly exceed the normal value Q_1, which may be caused by either a break in the pipeline below or by a surge due to sudden acceleration of water in the pipe, the hydraulic gradient would be steeper (ab′) and could exceed the distance xy at the knee of the pipeline so that ab′>xy.

This would cause a negative pressure in the pipeline at y and might result in a collapse at y. In fact, the whole shaded portion in Fig. 6 would be subjected to a negative pressure. To protect the pipe against collapse an air valve would have to be provided at point y, and air would be drawn into the pipe until the pipe at point y were again subjected to an internal overpressure.

If a standpipe were provided at y instead of an air valve, the water in that standpipe would flow into the pipeline until the level in the standpipe drops to point y. The above conditions will prevail whenever the section L_1 of small slope is of considerable length as compared with the lower section L_2. The upper section may be a tunnel or woodstave pipe or a common pipe supplying water to more than one pipeline.

Surge Reservoir

The introduction of a standpipe, surge tank, or surge reservoir at point y (Fig. 7) changes the nature of the flow of water in the upper section from a pressure flow to a gravity flow which follows different laws than those of a pressure flow.

With no water flowing in the gravity section L_1 and the pressure section L_2, the friction losses F_1 and F_2 would be 0, the head at end of pipe L_2 being H_t, called the static head. With water flowing at a maximum Q, the friction loss in upper section L_1 is F_1, and in lower section L_2 it is F_2. The water level in the surge tank will then be F_1 feet lower than at the intake, and the net head at end of pressure pipe L_2 will be $H_n=H_t-(F_1+F_2)$.

If the flow is suddenly stopped at the end of pipe L_2, the water will keep on flowing in gravity section L_1 so that the water level in the surge tank rises and may rise above even the water level at the intake. Up-surge $h_s>F_1$, so that water will then flow from surge tank to intake until equal water levels are restored. In other words, the water level in the surge tank will then oscillate to some extent, and this will be transmitted to the pressure pipe L_2, causing a fluctuation of the head H_n on the turbine.

Likewise on a sudden increase in flow at end of pipe L_2, the water will accelerate much faster in the pressure section L_2 than it does in the gravity section L_1, thus resulting in a down-surge of water level in the surge tank. Care must, therefore, be taken that the water level never drops as low as the junction point y (Fig. 7) of pipe because air would then be drawn into the pressure pipe and turbine. For a simple surge tank of constant area A (Fig. 8), the maximum surge h_s can be figured sufficiently accurately from the equation given below:

$h_s=\sqrt{\frac{PLV^2}{Ag}+F^2}$ where h_s=surge in ft; P=area in sq ft of (gravity flow) section of tunnel or conduit; L=length of conduit or tunnel in ft; V=velocity of water in ft/sec for full

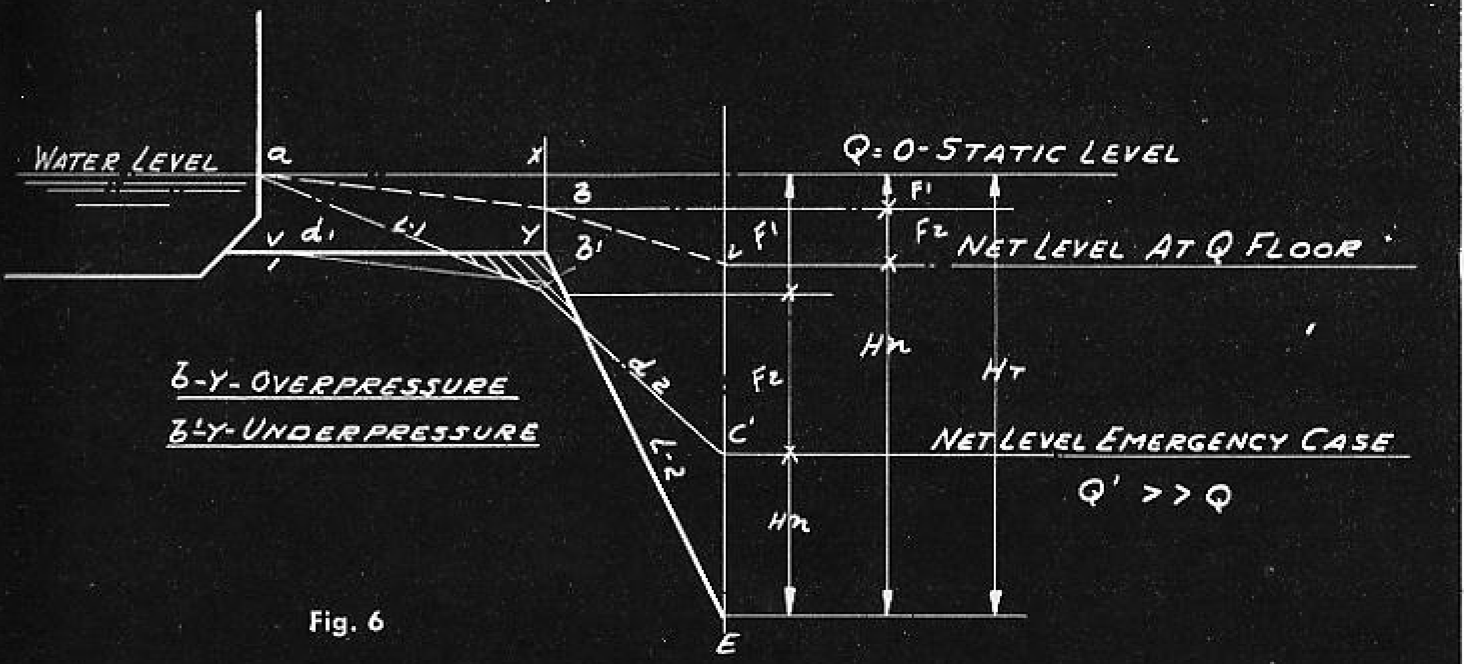


Fig. 6

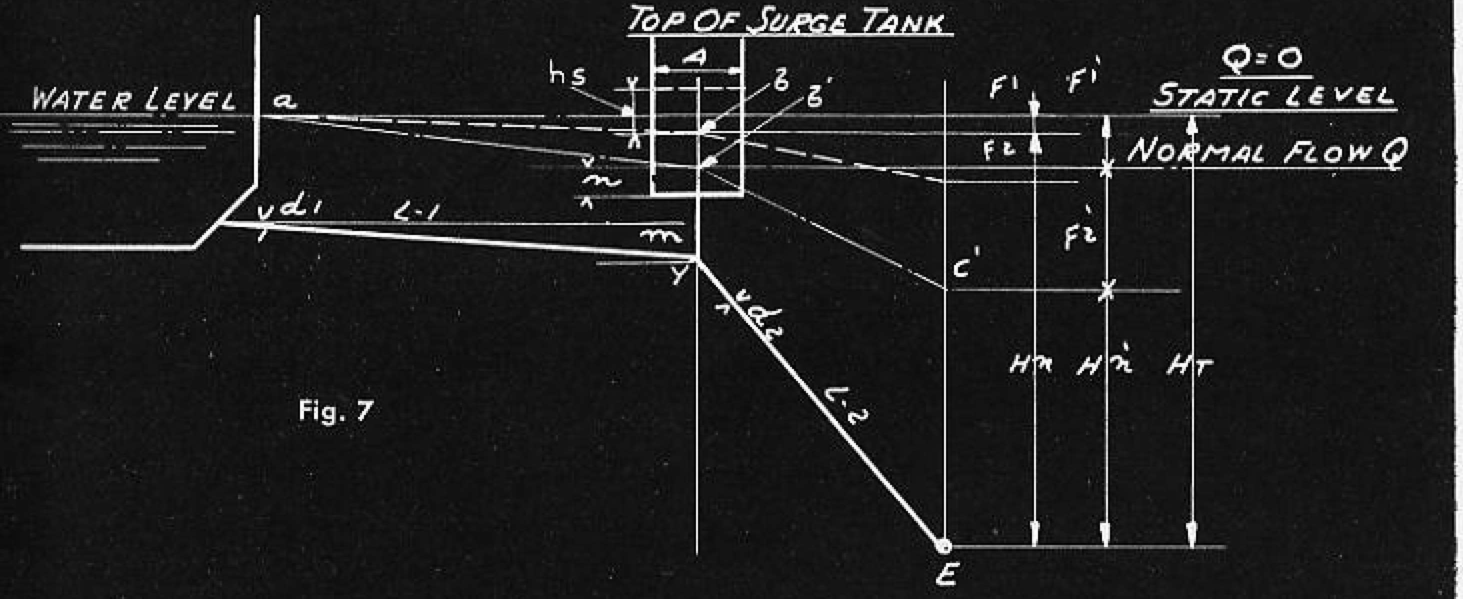


Fig. 7

flow Q in cfs; A=area in sq ft of surge tank; g=32.2 ft/sec/sec; F=friction loss in ft for flow Q, i. e., velocity V.

For example:

P=100 sq ft
L=2500 ft
V=6 ft/sec
A=2000 sq ft
F=3 ft

$$Q=100\times 6=600 \text{ cfs}$$

$$h_s=\sqrt{\frac{100\times 2500\times 36}{2000\times 32.2}+9}=12.1 \text{ ft.}$$

The time from beginning of flow Q to the maximum surge h_s is $\frac{t_1}{2}=\frac{\pi}{2}\sqrt{\frac{LA}{gP}+\left(\frac{CVA}{P}\right)^2}$, t_1 being the time for a total surge wave and $C=\frac{F}{V^2}$; in our example $C=\frac{3}{36}=\frac{1}{12}$.

$$\frac{t_1}{2}=\frac{\pi}{2}\sqrt{\frac{2500\times 2000}{32.2\times 100}+\left(\frac{1\times 6\times 2000}{12\times 100}\right)^2}=64 \text{ sec.}$$

After 64 sec the water level beings to rise again.

It can be seen that, in the example, this maximum surge is reached long after the governor has opened the turbine in T_g sec (5.096 sec in the example Table V, Part IV of this article). Therefore, only a part of the surge h_s affects the governor.

Assuming a linear increase of Q cfs in 5.0 sec, the flow would be

0-1 sec	120 cfs	3-4 sec	480 cfs
1-2	240	4-5	600
2-3	360		

or a total of 1800 cu ft.

With a surge tank of area A=2000 sq ft, the water level would drop $\frac{1800}{2000}=0.9$ ft, or only about 7½% of the total surge h_s. Assuming the governor had opened the turbine wide after five sec and that it would remain open, the available head would drop another 12.1−0.9 or 11.2 ft; and the output of the unit would decrease gradually during the 64−5 or 59 sec. Since the total friction loss is only 3 ft, the water level is 12.1−3 or 9.1 ft below the final level.

Under those conditions the velocity of water in the tunnel has reached the normal full load value of 6 ft/sec but increases further, so that a steady rise of level in the forebay will take place and, depending upon the characteristics of tunnel and surge tank, may rise above even the static or no flow level. In other words, there may be a series of fluctuations up and down before the water level comes to rest. Conditions may even be such that a constant oscillation will result or even one increasing in amplitude. It is, therefore, necessary to study each case and determine the surge curve in detail over a period of time long enough to show its character.

To determine this curve a step-by-step method must be adopted since a correct mathematical calculation involves complicated differential equations.

Let Δt be the time interval of the step, Δy the respective change in water level in surge tank and Δv the respective change in velocity in the tunnel (or gravity flow section): $\Delta y=\left(\frac{Q}{A}-\frac{P}{A}v\right)\Delta t$ for each initial drop.

If the governor opened up full and instantly, Q would be the maximum flow, or $\underset{(v)}{6}\times\underset{(p)}{100}$ or 600 cfs in the example.

For a linear increase of flow, in 5 sec the values of $\frac{600}{5}=120$, 240, 360, 480, and 600 cfs, respectively, must be used in above formula.

As soon as the water level drops to Δy, the flow through the tunnel starts. The velocity change is $\Delta v=\frac{g}{L}(y-cv^2)\times\Delta t$.

For the first interval Δv is still 0. In both equations y and v are the end values, i. e., the respective sum of all Δy and Δv up to that time.

Tables I, II, and III (see pages 6 and 7) show the various values of Δy, Δv, y and v for instantaneous full draw of Q.

The maximum surge occurs between 60 and 65 sec. Here the velocity v becomes 6 ft/sec. It increases to slightly over 9 ft/sec between 115 and 120 sec, at which point the up-surge intensity has reached a maximum. The velocity v now decreases, causing a gradual deflection of the surge curve, until v reaches normal (full flow) value of 6 ft/sec, at which the level in the surge tank has reached a maximum — even 0.77 ft higher than the static or no flow level, etc., all as illustrated in Fig. 8. If the surge tank is of a different design, for instance, if A is not constant throughout (not a cylindrical tank), the general end formula is not applicable; and the step-by-step method must be applied, using the respective values of A for each water level y.

Differential Surge Tank

In place of the simple surge tank a differential surge tank may be used. This consists of a large tank into which is placed another tank of smaller diameter, communicating with the large tank by a series of openings at the bottom. This smaller tank communicates throughout its area with the pressure pipe. On an increased draw of water through the pressure pipe, the water level y' in the smaller tank drops faster than it does in the large tank, y'>y. This causes a greater pressure drop in relation to the turbine but results in a quicker acceleration of the water in the gravity section compared to the simple tank design. The effect is that momentarily y is greater, but the velocity v in the gravity section recovers more quickly so that the wave time T' will be shorter. Important problems should be submitted for calculation by Dr. Norman Gibson at Niagara Falls, N. Y., who specializes on differential surge tanks.

Shut-off Valves

Any means for shutting off the water should be designed and built to withstand safely operation under any emergency condition as may arise. Failure at such critical moment may cause much greater damage and momentary loss than the costs of the best design and make.

There are two principal types, the gate valve and the butterfly valve.

Gate Valves

There are numerous standard makes, but they are not all well fitted for use as shut-off valves in connection with hydraulic plants. The usual standard designs (Fig. 9) consist of a valve housing or body (1) with bonnet (1a). The bonnet and the portion of valve body below are of elliptical or even rectangular cross-section, introducing uneven stresses in the material when subject to the internal water pressure. A valve plug (2) is loosely guided sideways for vertical movement by means of guides (3) and operated by stem (2a) either directly mechanically outside of valve bonnet or by electric motor drive, or hydraulically by placing a piston on stem (2a) operating in a cylinder mounted on top of bonnet (1a). With standard designs the valve plug (2) usually has tapered surfaces F lined with renewable rings seating in closed position against seat rings in the valve housing (1). Thus the valve wedge can be pressed against the two tapered seats closing off tightly against flow and over-pressure in both directions.

Since the water flows only in one direction from pipeline to turbine, a double seating is unnecessary. The taper feature of the seat also introduces poor hydraulic conditions at partial opening of the valve and is likely to involve vibrations, especially under emergency conditions arising when the valve must be closed against full flow and pressure. A correct design of valve should have one single seat surface on the downstream side of valve, and the seat should be parallel to the axis of the stem so that the plug can slide along the seat surface, as illustrated in Fig. 10.

For high heads, or in case of high water velocities through the valve and especially in case of sand or silt carried in suspension by the operating water, an improved design, as shown in Fig. 11, has been used by Allis-Chalmers since 1910. To

Time Interval		Δy		Δv	y	v	Remarks
0-5		1.5		0.097	1.5	0.10	$y_0=0$ $v_0=0$ Starting
5-10		1.48		0.19	2.98	0.29	
10-15	Results figured from equation $\Delta y=1.5—0.25v$	1.43	Results figured from equation $\Delta v=0.064\left(\frac{y-v^2}{12}\right)$	0.28	4.40	0.57	
15-20		1.36		0.37	5.76	0.94	
20-25		1.28		0.45	7.04	1.39	
25-30		1.06		0.51	8.09	1.90	
30-35		1.02		0.57	9.12	2.47	
35-40		0.88		0.61	10.00	3.08	
40-45		0.73		0.64	10.73	3.72	
45-50		0.58		0.66	11.30	4.38	
50-55		0.41		0.66	11.71	5.03	
55-60		0.25		0.64	11.95	5.67	Max. surge h_s v=6.0 ft
60-65		+0.08		0.61	12.09	6.27	

Table I

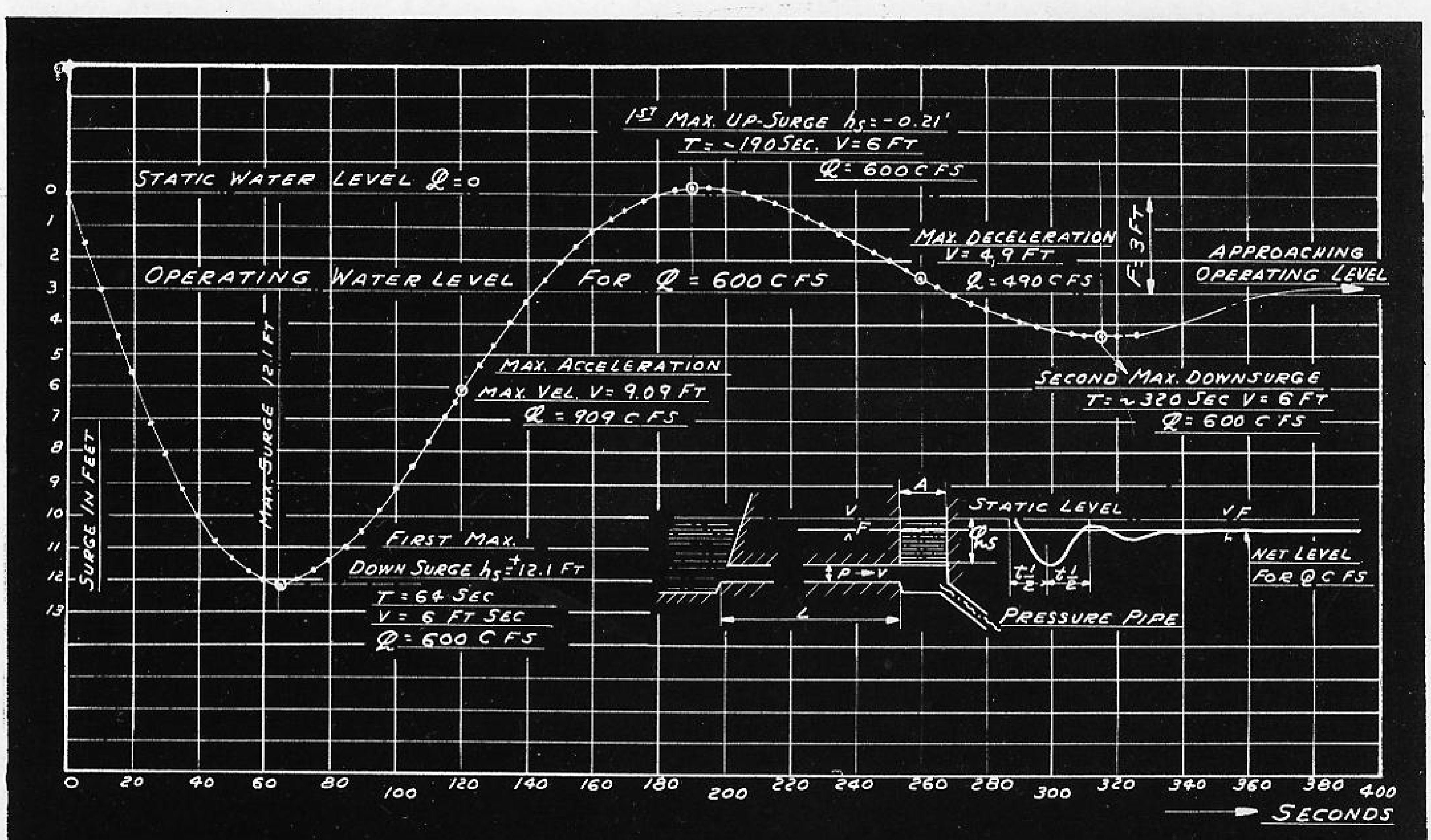


Fig. 8

	Results figured from equation $\Delta y = 1.5—0.25v$	Results figured from equation $\Delta v = 0.064 \left(\frac{y - v^2}{12} \right)$			
65-70	—0.07	0.56	11.97	6.84	Surge curve rises because $v > 6$ ft Δy negative
70-75	—0.21	0.51	11.77	7.34	
75-80	—0.33	0.45	11.44	7.79	
80-85	—0.45	0.39	11.00	8.18	
85-90	—0.54	0.32	10.45	8.49	
90-95	—0.62	0.25	9.83	8.74	
95-100	—0.68	0.18	9.15	8.92	
100-105	—0.73	0.12	8.42	9.03	
105-110	—0.76	0.06	7.66	0.09	
110-115	—0.77	0.0	6.90	9.09	
115-120	—0.77	—0.05	6.13	9.03	
120-125	—0.75	—0.09	5.37	8.93	
125-130	—0.74	—0.13	4.63	8.80	
130-135	—0.70	—0.14	3.93	8.66	
135-140	—0.66	—0.19	3.27	8.50	
140-145	—0.62	—0.22	2.68	8.28	
145-150	—0.57	—0.23	2.11	8.04	
150-155	—0.51	—0.25	1.60	7.80	
155-160	—0.45	—0.26	1.15	7.54	
160-165	—0.38	—0.26	0.77	7.28	
165-170	—0.32	—0.26	0.45	7.02	
170-175	—0.25	—0.26	0.20	6.76	
175-180	—0.19	—0.24	0.01	6.52	
180-185	—0.13	—0.24	—0.13	6.28	
185-190	—0.07	—0.23	—0.20	6.06	Positive max. surge Δy changes
190-195	—0.02	—0.21	—0.21	5.84	

Table II

	Results figured from equation $\Delta y = 1.5—0.25v$	Results figured from equation $\Delta v = 0.064 \left(\frac{y - v^2}{12} \right)$			
195-200	+0.04	—0.20	—0.19	5.65	
200-205	0.09	—0.18	—0.11	5.47	
205-210	0.14	—0.16	0.03	5.31	
210-215	0.17	—0.14	0.20	5.17	
215-220	0.21	—0.12	0.41	5.06	
220-225	0.24	—0.10	0.64	4.96	
225-230	0.26	—0.08	0.90	4.89	
230-235	0.28	—0.05	1.18	4.88	
235-240	0.28	—0.03	1.47	4.88	
240-245	0.29	—0.01	1.75	4.85	
245-250	0.29	—0.005	2.04	4.90	
250-255	0.28	+0.02	2.31	4.92	
255-260	0.28	0.04	2.59	4.96	
260-265	0.26	0.05	2.84	5.01	
265-270	0.25	0.06	3.08	5.07	
270-275	0.23	0.08	3.31	5.15	
275-280	0.22	0.09	3.52	5.24	
280-285	0.19	0.09	3.71	5.33	
285-290	0.17	0.10	3.88	5.43	
290-295	0.15	0.10	4.03	5.53	
295-300	0.12	0.10	4.15	5.64	
300-305	0.10	0.10	4.24	5.74	
305-310	0.07	0.10	4.31	5.86	
310-315	0.05	0.10	4.36	6.00	3rd surge $h_{s_2} = 4.37\ \Delta y$ changes
315-320	+0.02	0.09	4.37	6.04	
	—0.01		4.36		

Table III

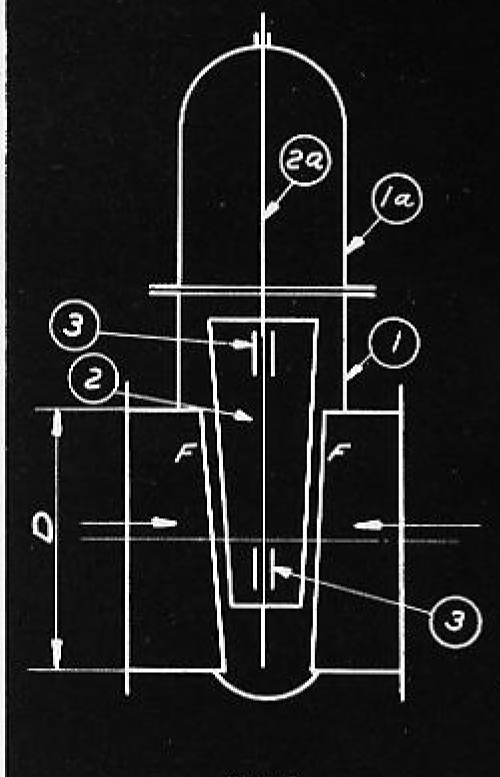


Fig. 9

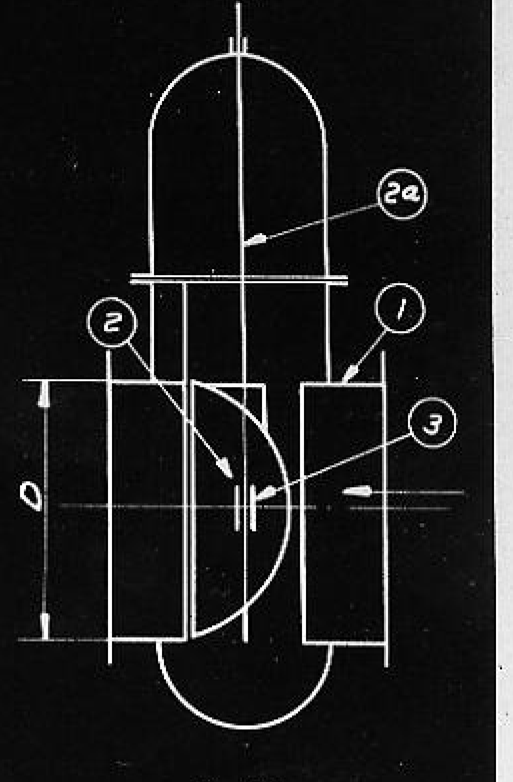


Fig. 10

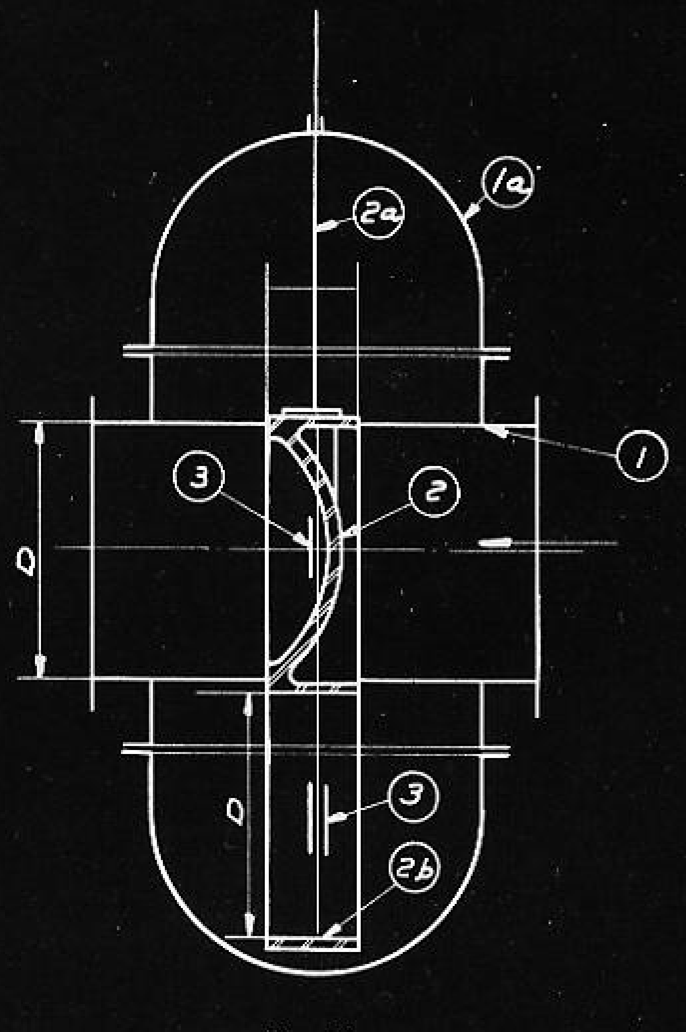


Fig. 11

eliminate unequal stresses in the valve housing and bonnet, a circular section and not an elliptical or rectangular is used although this somewhat increases the overall face-to-face distance between connecting flanges.

To protect the seat surfaces against damage by floating sand, silt, or grit under high velocity, the valve plug has a cylindrical ring-shaped extension (2b) which in wide open position fills the gap otherwise left by the valve plug. Thus it not only covers the seat surface but restores a continuous pipe, eliminating eddies and at the same time preventing chattering of the valve plug when going through the intermediate position.

Figure 12 shows one of the eight valves installed in 1913 under heads as high as 2160 ft. In emergency cases these valves have been closed against full over-pressure on the valve plug, such as becomes necessary when either the orifice or nozzle has become partly clogged up by debris or when for similar reasons the pressure regulator attached to the same nozzle cannot be completely closed. These valves have been in continuous service for the past 29 years and have required no repair, thus compensating richly for any excess costs over a standard type valve.

In hydraulic plants inlet valves should be either closed or wide open and should not remain at an intermediate position since such operation would throttle the available head of the turbine. Hydraulically operated valves, therefore, need not have a relay-controlled operating mechanism; a simple four-way cock, or distributor, is sufficient for admitting water pressure to one side of the cylinder while draining the opposite side, or reversed for reversed movement of valve plug.

The valve is opened or closed by means of a control valve instead of a four-way cock, arranged either for hand-operation or, by means of a small electric motor, for switchboard or other remote control.

Figure 13 shows two hydraulically operated gate valves of 36 in. clear diameter for the 30,000 hp, double overhung impulse wheel units at the Glenville plant, North Carolina, operating under 1200 ft head.

The control valve attached to the side of the operating cylinder is of an improved design. It operates with penstock pressure for closing the valve only, and the motor of the control valve is wired to the switchboard only for closing the gate valve. For opening the valve, instead of utilizing penstock pressure, pressure built up in the nozzle pipe (downstream of the gate valve) is used. Thus it is impossible to open the gate valve accidentally when any part downstream of the gate would permit of such an amount of discharge of water that the hand-operated bypass valve could not maintain an appreciable pressure downstream of the (still) closed gate valve.

Butterfly Valves

For large diameter pipes and medium heads the butterfly valve is less costly than an equally dependable design of gate valve. The gate valve has a sliding element for shutting-off the water, whereas the butterfly valve has a rotating element. Fig. 14 illustrates this type of valve. It consists of a valve housing (1), a wicket (2) fastened to a shaft (3) supported in two trunnion bearings (1a and 1b) integral with the valve housing. In closed position, the wicket (2) seats against the cylindrical bore of the valve housing (1), the contact surface (2a) of the wicket being also cylindrical and of a diameter equal to that of the bore in the housing. To assure that the wicket contact surface can be pressed against the adjacent surface of the housing, the wicket does not stand at right angle to the centerline of the housing ($\alpha=90°$), but at an angle $\alpha < 90°$.

The full pressure force upon the wicket is $P=H\times 0.433\times D^2\frac{\pi}{4}$ lb, H being the head in ft, D the diameter of the contact cylinder in in., and p the pressure in lb/sq in.$=0.433H$. The load P causes a pressure upon the trunnions 1a and 1b, requiring a force Q on the lever R operating the wicket shaft (3). This force R is: $\frac{P\mu d/2}{R}$ lb, where μ is the friction coefficient in the bearings (μ about 0.25), d the diameter in in. of the trunnions 1a and 1b, and R the length in in. of the operating lever.

With a gate valve (Figs. 9 and 10), the force Q on stem 2a is $Q=0.433H\mu\frac{\pi(D+2a)^2}{4}$, where a is the width of the seat ring surface 1b. This force Q decreases in proportion to the gradually reduced area covered by the valve plug during the movement from closed to open position of valve, even in the emergency when the valve discharges free into atmosphere (no back pressure on downstream side of valve plug).

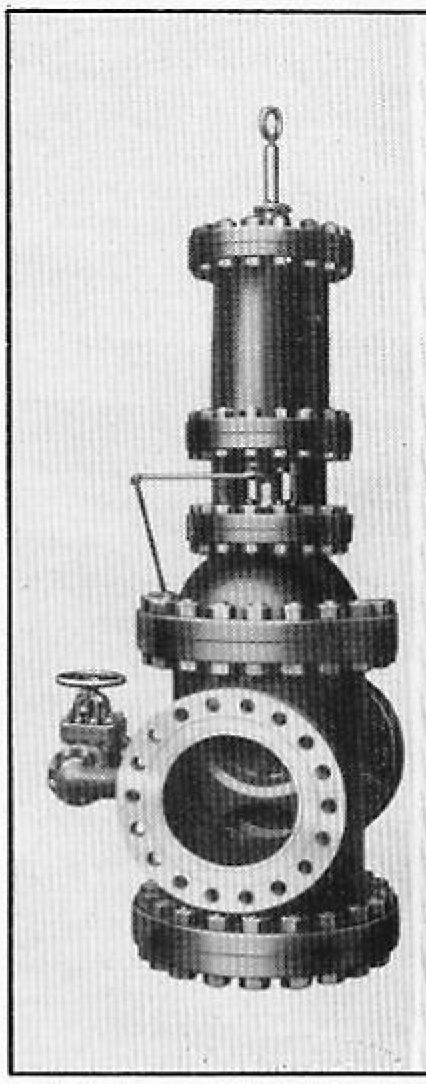


Fig. 12 — Twenty-four in. hydraulically operated gate valve.

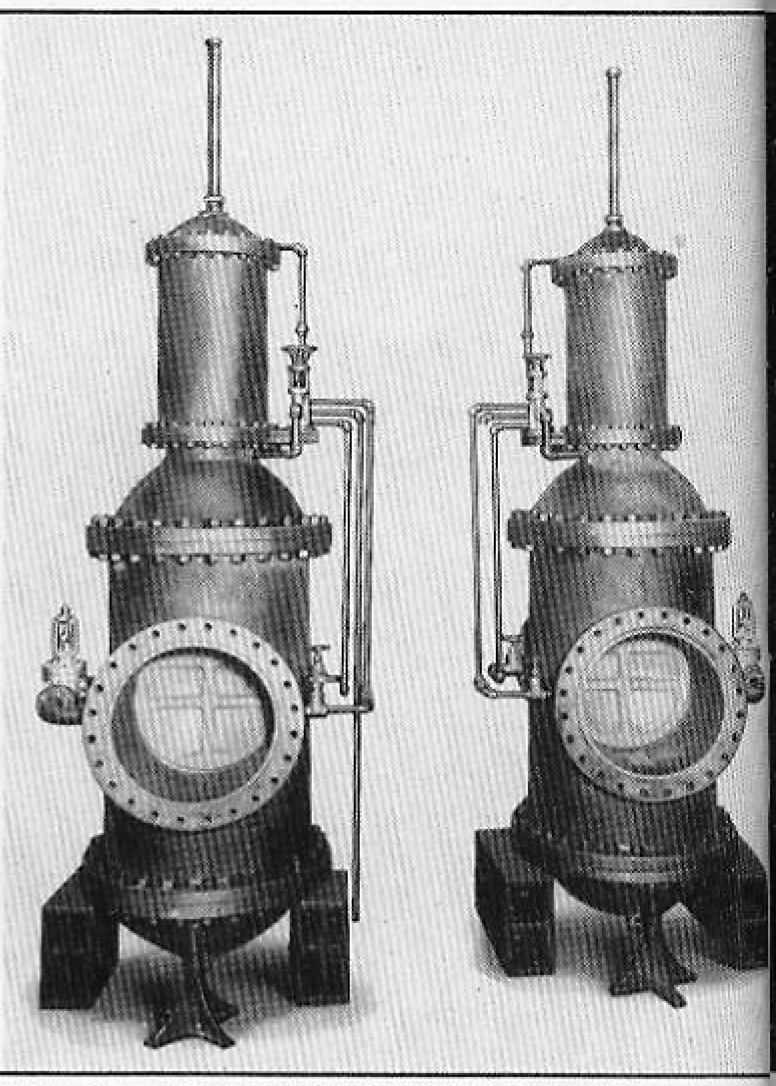


Fig. 13 — Two 36 inch follower type gate valves for 30,000 hp impulse hydraulic turbine with 1150 ft head.

With a butterfly valve the force Q changes according to entirely different causes. The velocity V_1 at the orifice (Fig. 15) changes with the position or angle α of the wicket; and, of course, with the ½ power of H because $V_1=M\sqrt{2gH}$, M being a function of α. The velocity head at the orifice is $\frac{V_1^2}{2g}$, leaving a pressure head $H_{n_1}=H-\frac{V_1^2}{2g}$ pressing upon the wicket, tending to open it. As the passage widens toward the center of the wicket, the velocity V_2 becomes lower so that $H_{n_2}=H-\frac{V_2^2}{2g}$ is greater, as shown by curve X_1 in Fig. 15. The upstream part of orifice formed by the lower half of the wicket and the valve housing forms a less perfect orifice because some of the water must turn around as indicated by the stream lines shown in Fig. 15. The velocity V_3 is lower than V_1, and V_4 is much lower than V_2 so that the total pressure upon that portion of wicket is greater, as indicated by pressure curve X_2, and this causes the tendency of the wicket to be slammed shut.

The character of the curves X_1 and X_2 changes with the position of the wicket; it reaches about balanced condition when the wicket is wide open ($\alpha=0$) and is zero in closed position of the wicket. ($\alpha=$about 78° with Allis-Chalmers designs.) Thus a hydraulic, unbalanced moment M_{hu} enters into the problem of proper design of the wicket, wicket shaft, bearings, and operating mechanism! It can be computed:

$M_{hu}=KD^3V_n^2$, where M_{hu} in ft lb

D diameter of valve in ft

V_n velocity in ft/sec

K a coefficient which is a function of α.

When the wicket 2 is in closed position (Fig. 14), the total load upon the wicket is: $P=\frac{\pi D^2}{4}0.433H$, just as in the case of the gate valve. This load is transmitted to the two trunnions (1a) and (1b), producing a surface pressure $p=\frac{P}{2Ld}$, where L

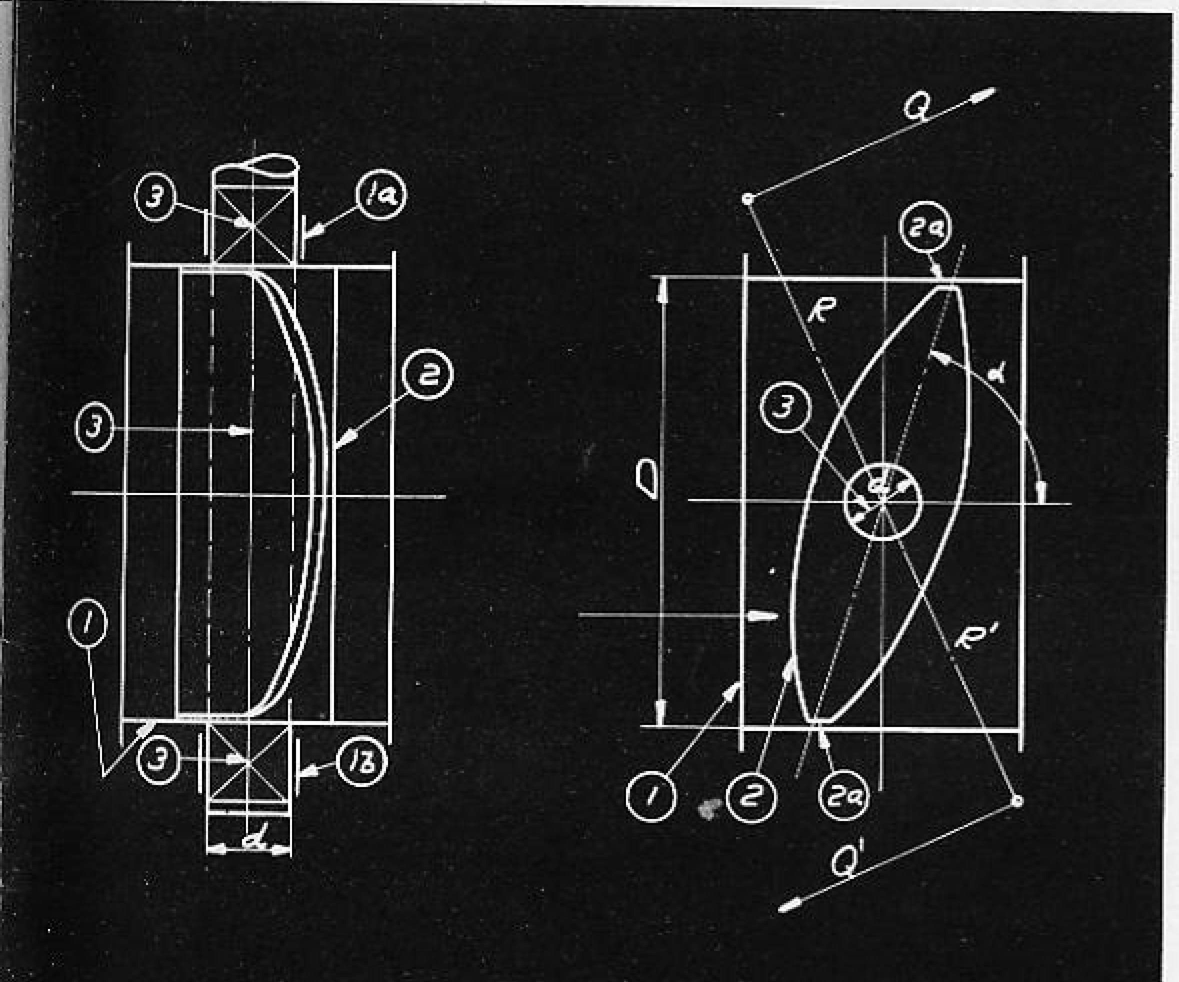


Fig. 14

is the length of the trunnion bearing and d the diameter, each in in. The friction moment on the shaft to open the wicket is:

$M_f = P\mu d/2$ in in. lb if d is in in.

The force on lever end R is $Q = \frac{M_f}{R}$, or

$$Q = \frac{\pi D^2}{4} \times \frac{0.433 H \mu d}{2R} = \frac{0.17 D^2 H d}{R}$$

This is the static force required to move the wicket away from closed position.

When the wicket is in wide open position ($\alpha = 0°$), this force disappears except for that part P caused by the weight of the wicket and shaft if the shaft is not in vertical position. As soon as the wicket opens and a discharge begins, the hydraulic conditions illustrated in Fig. 15 arise.

The upper half of the wicket forms a nozzle, so to speak, with gradually increasing water velocities reaching a maximum V_1 at the orifice formed by the wicket rim and the housing.

At the upstream side of the valve housing, there is a head H or a pressure $p = 0.433H$. The velocity V_n is obtained by dividing the flow of water through the valve Q by the projected net area A_n of passage of the flow (shaded area in Fig. 15a).

$$V_n = \frac{Q}{A_n} = \frac{Q}{\frac{D^2}{4}(1 - \sin\alpha)}, \text{ where } \alpha = 0, \text{ wide open wicket } (\alpha = 78°, \text{ closed wicket}).$$

The hydraulic unbalanced moment M_{hw} is a maximum when the wicket has closed about 18 to 20° from wide open position. Depending upon operating conditions, it may amount to over ten times the static friction moment M_f with the wicket in closed position.

It is, therefore, absolutely necessary to investigate the conditions under which such a butterfly valve is to operate, whether for entirely free discharge under the available head H, or only for a maximum flow (cfs), etc., with penstock valves which must be capable of closing without defect under emergency conditions, such as breakage of pipe at any point downstream of the valve. Very serious accidents with loss of lives are on record from earlier days of the art when the forces were not yet known and designs were inadequate.

Structural Features

It should be needless to state that the valve housing should be so built that it will not deform "out of the round" at the

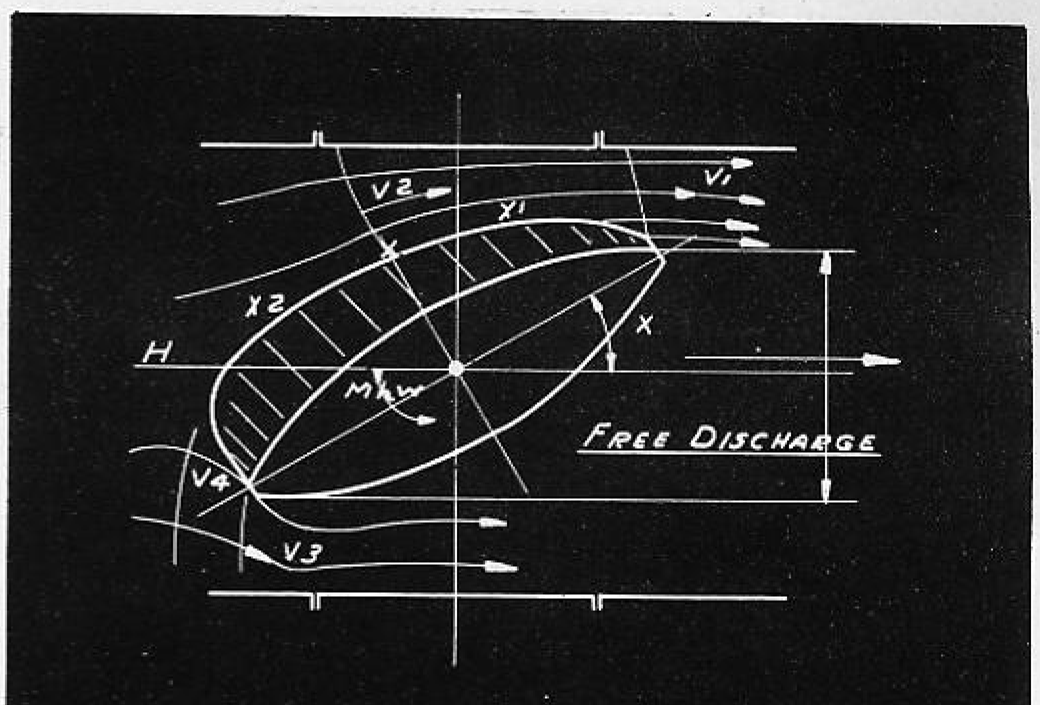


Fig. 15

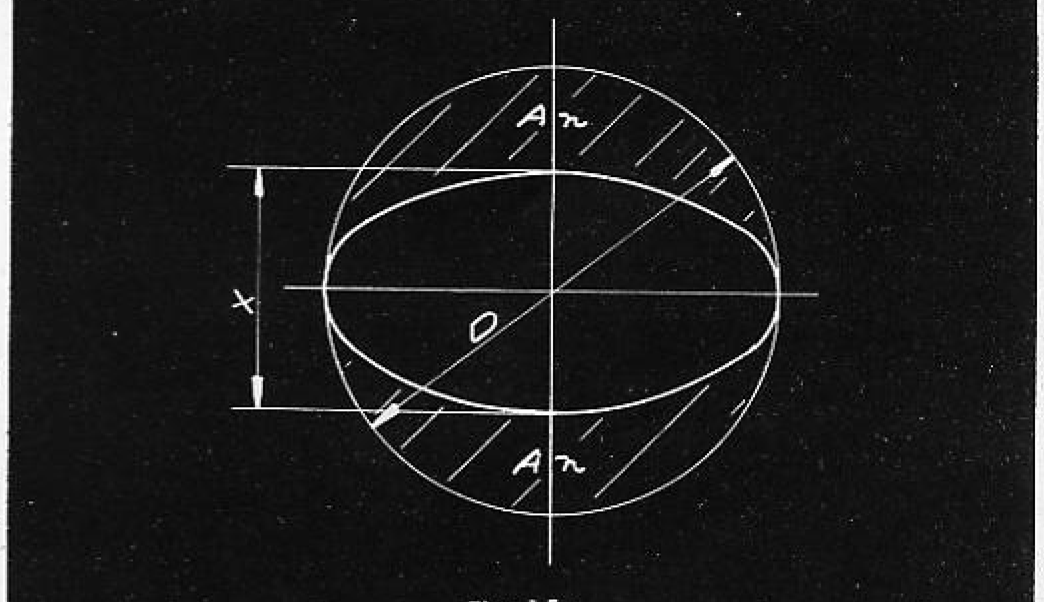


Fig. 15a

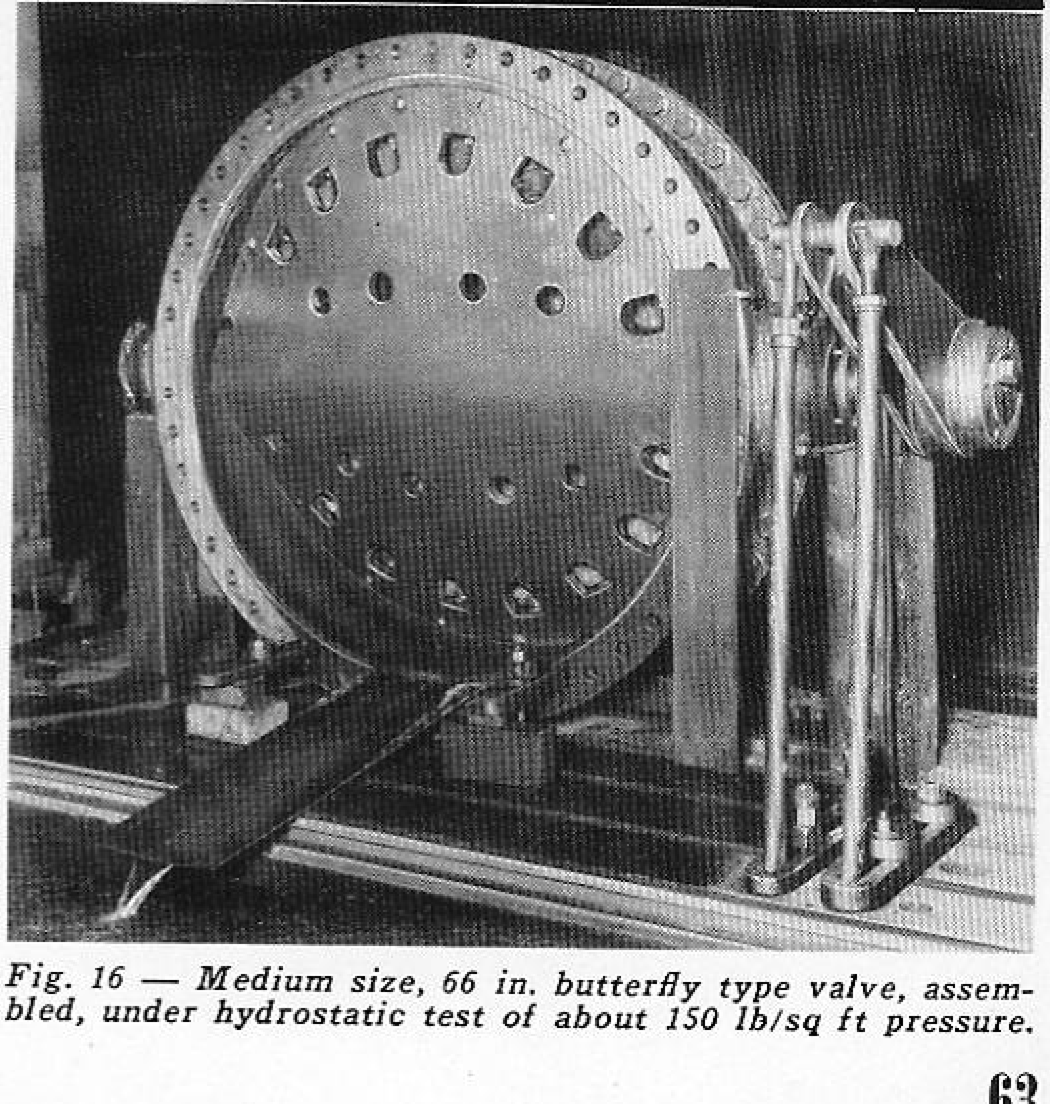


Fig. 16 — Medium size, 66 in. butterfly type valve, assembled, under hydrostatic test of about 150 lb/sq ft pressure.

Fig. 17 — One 11 and two 9 foot butterfly valves for two 39,000 hp vertical steel plate spiral cased turbines.

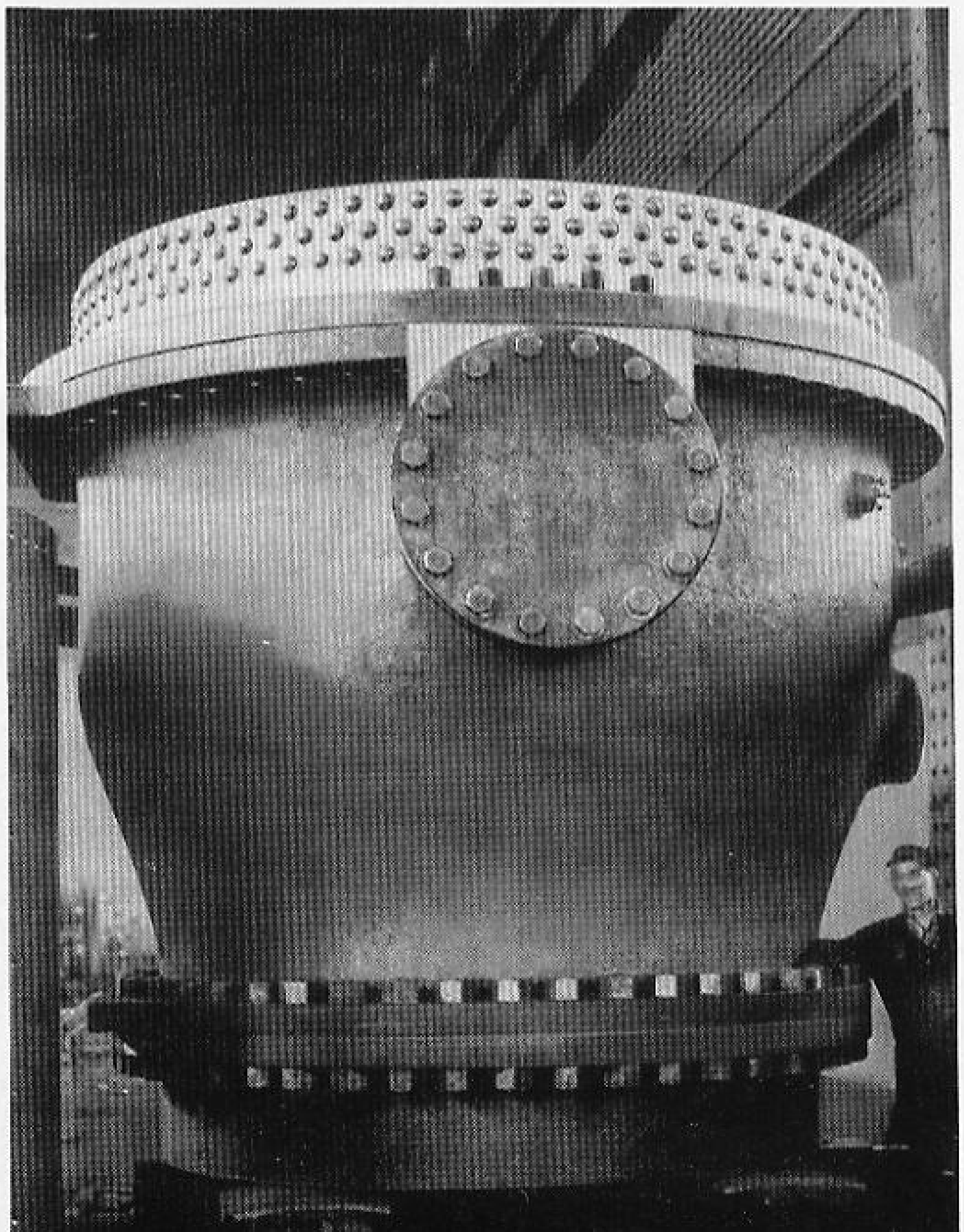


Fig. 18 — Cast steel housing for 108 inch butterfly type valve. Valve operates under a static head exceeding 900 ft.

surface of contact with the wicket in closed position. These surfaces should be made rënewable and of non-corrosive material. One should be made adjustable to permit individual adjustment in the field. Fig. 16 shows a medium size valve of 66 in. diameter for about 150 lb test pressure. Housing and wicket are of annealed cast steel, the latter being of a fish-shaped profile. The outer circumference of the wicket contains renewable metal rings which can be individually pressed against the contact surface in the housing and locked, all from the downstream side of valve with full penstock pressure on inlet side.

Larger valves are made with housings of welded steel plate as shown in Fig. 17 (three shop-assembled valves, one of 11 ft and two of 9 ft diameter, for an operating head of 390 ft, supplying water to two 39,000 hp turbines). Here the operating lever is a gear segment engaging a pinion and worm gear for electric motor operation.

For high heads, cast steel housings are used; and, since the wicket displaces considerable area, the downstream side is contracted to maintain approximately constant water velocity past the downstream portion of wicket and housing. Fig. 18 shows one of two valves for a static head exceeding 900 ft. Diameter of the valve at the seat of the wicket is 108 in. If this valve housing were made cylindrical, the increase in passage area downstream of the wicket would be so abrupt that the water could not decelerate properly. Eddies and pitting as well as possible vibrations would result.

Figure 19 shows two valves under hydrostatic shop test of 150 lb. The housings are of welded steel, 15 ft clear inside diameter. The two valves were assembled with upstream sides temporarily riveted together so that test pressure could be applied to both housings and wickets simultaneously; at the same time leakage of water past the wickets was measured. The valves are 15 ft in diameter, resulting in a load P of 3,800,000 lb on each wicket; and the leakage did not exceed 65 gpm, thanks to the excellent design of seal rings in the wickets and a minimum of deformation of housing and wicket.

Figure 21 shows the shop-assembled, automatic, electric

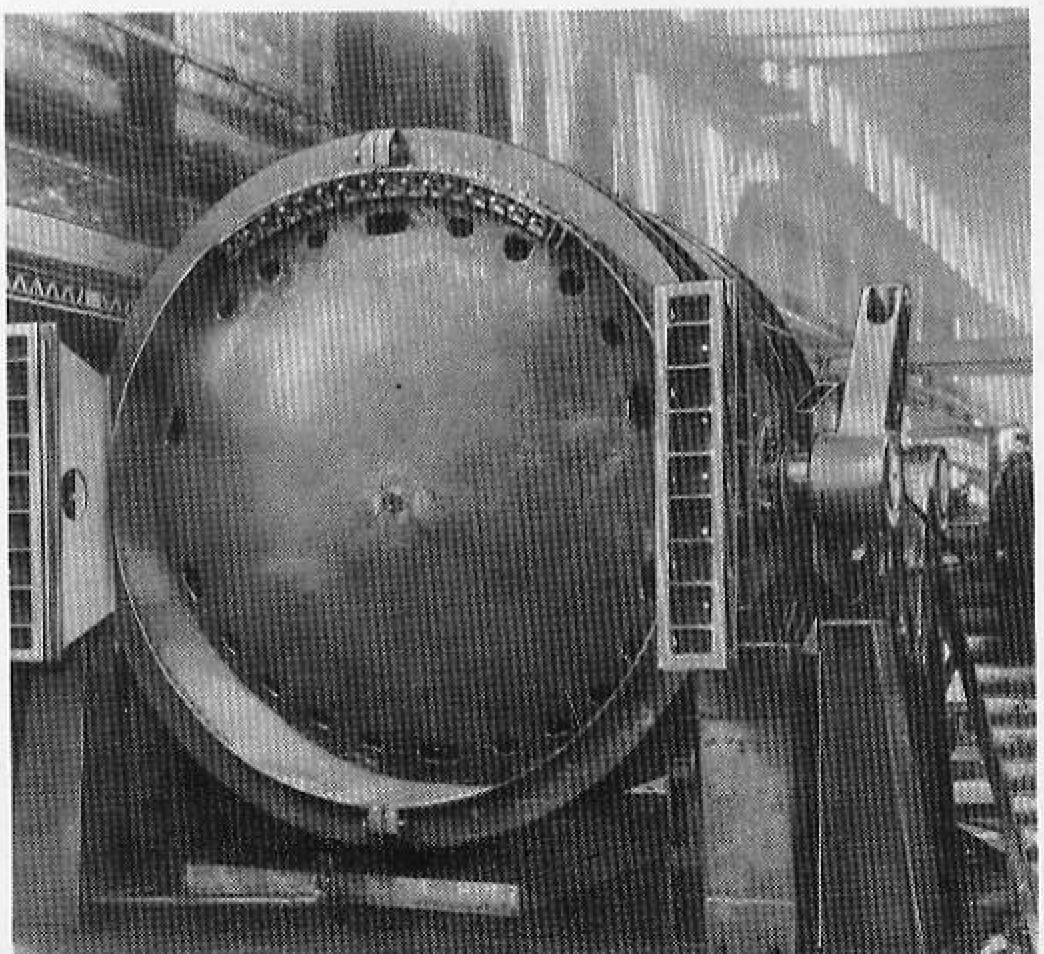


Fig. 19 — Two 15 ft butterfly valves under 150 lb/sq in. hydraulic test pressure. Leakage did not exceed 65 gpm.

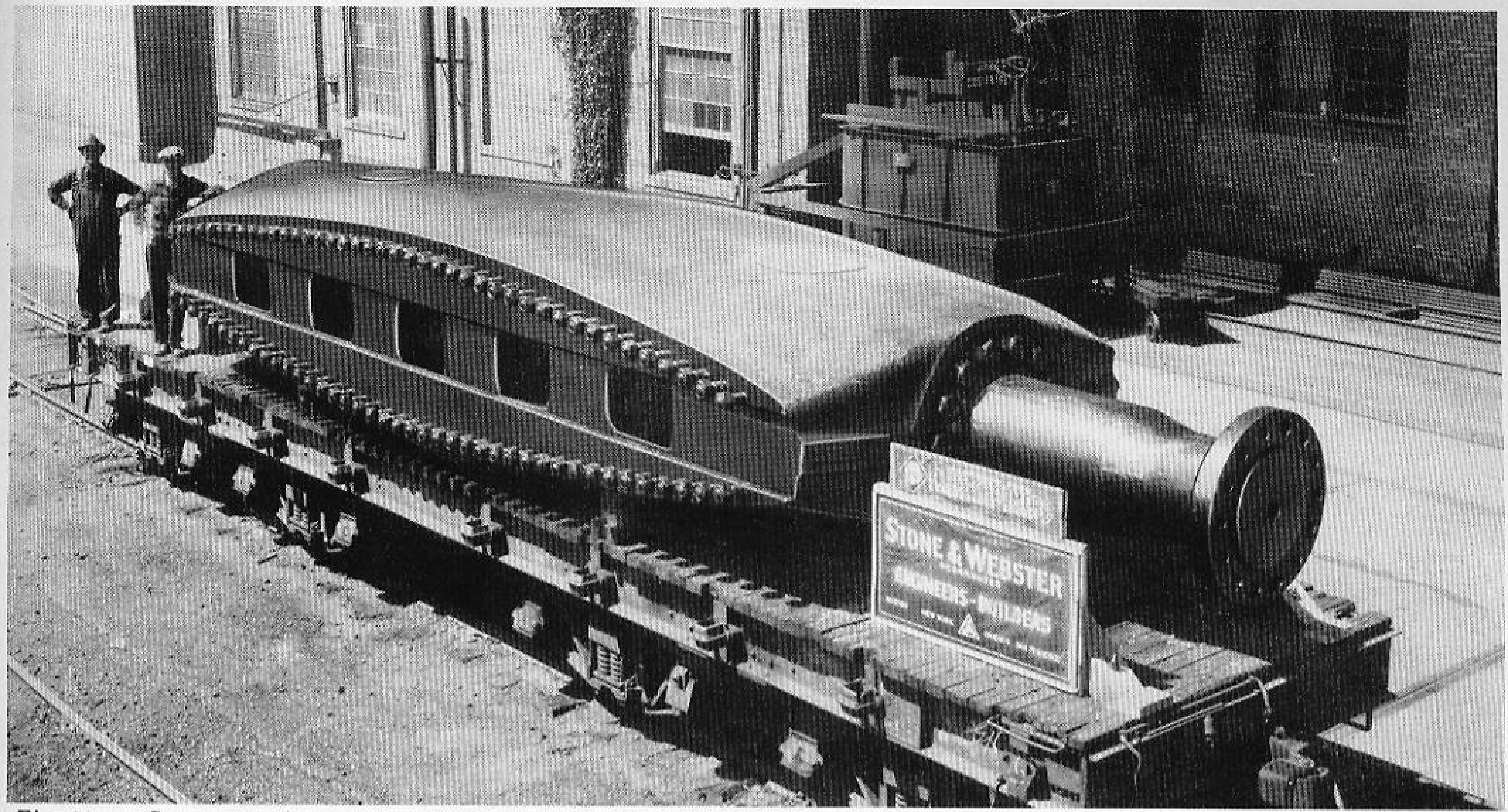


Fig. 20 — Center section of the cast steel wicket for a 27 ft butterfly type valve. This section alone weighs 151,000 lb.

operating mechanism of one of these valves. The threaded rod to the left of the picture is pivoted to the lever operating the wicket. This mechanism is built to operate safely under emergency conditions when, because of a break in the pipeline, the water passes over the wicket at a velocity of about 28 ft/sec compared to a velocity of 13.55 ft/sec for full turbine discharge at 115 ft net head. The normal torque capacity of the electric motor is 6,000,000 in. lb, and the stalling capacity is 12,000,000 in. lb. Four such valves were placed in successful operation in 1939.

Figure 20 shows the center portion of the wicket of one of the four butterfly valves of 27 ft clear diameter furnished in 1926, which still hold the world record for size. Of the butterfly type, this center piece weighs 151,000 lb. Two leaves, one on each side, are bolted to it.

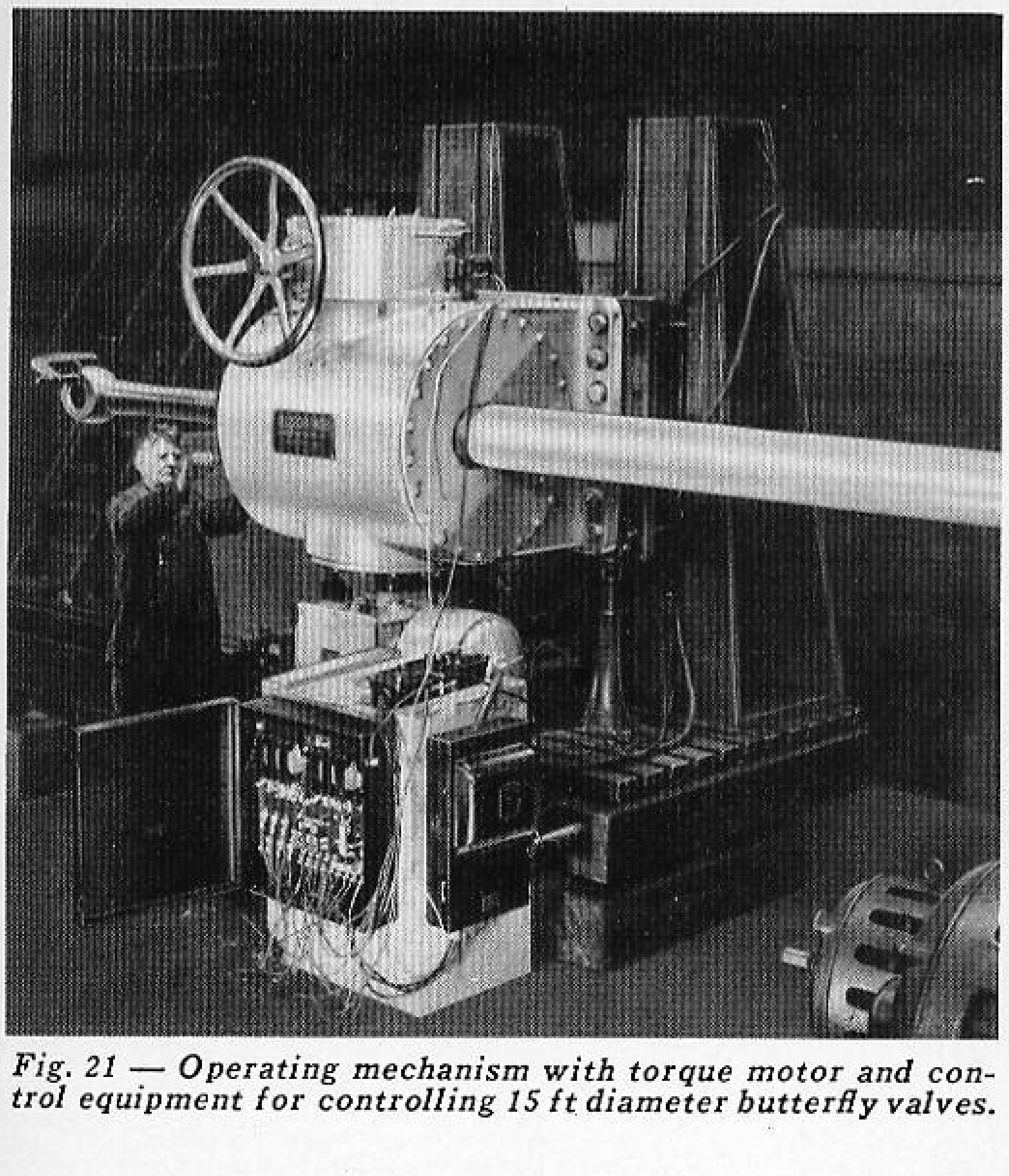


Fig. 21 — Operating mechanism with torque motor and control equipment for controlling 15 ft diameter butterfly valves.

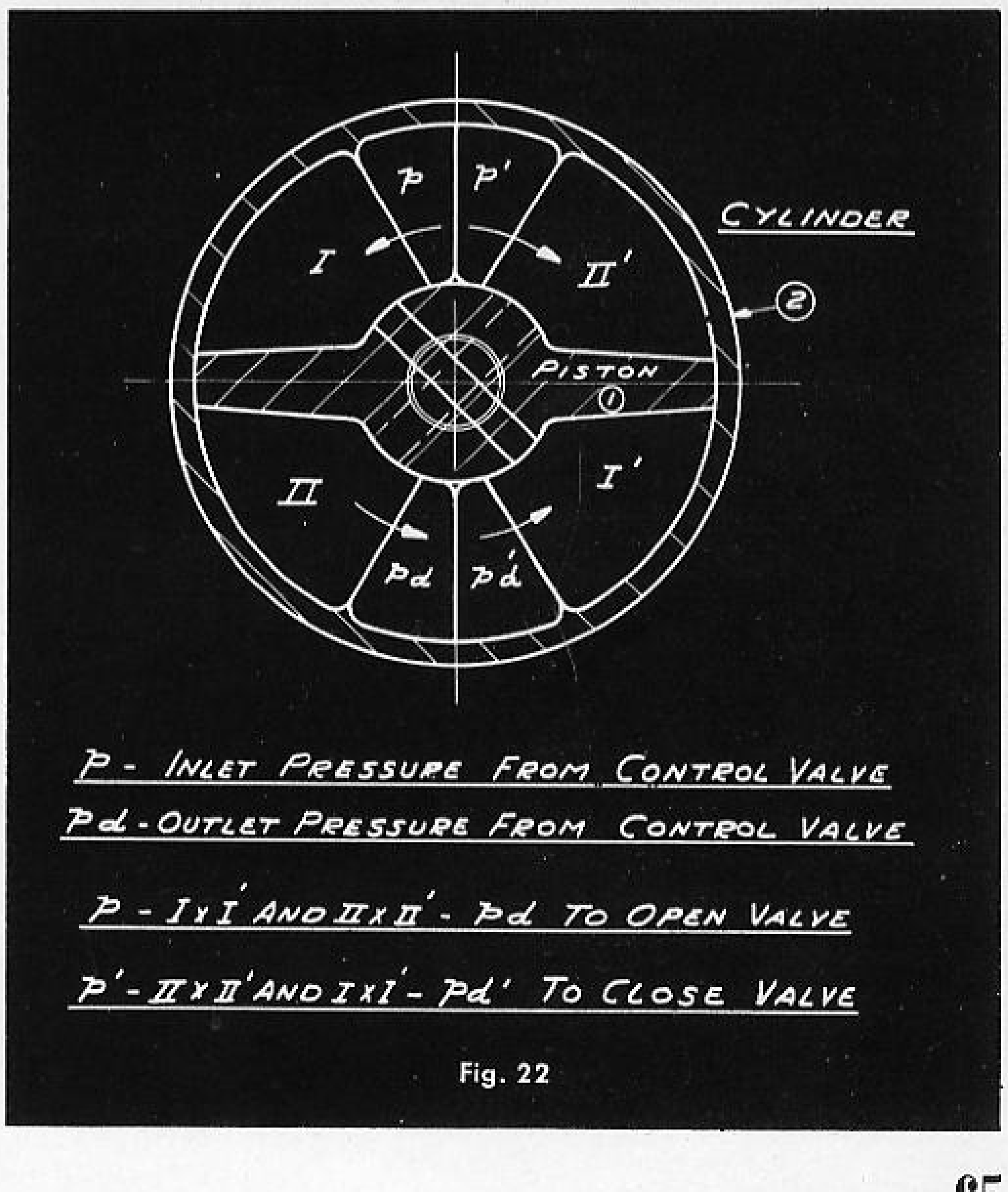


Fig. 22

When the force Q for operation of the wicket is large and the lever R is overhung on one shaft end, it causes a considerable pressure upon the adjacent trunnion bearing in addition to the load P from the wicket. Therefore, in such case an outboard bearing is provided. If the lever R is of the double-arm type, then the bending force is practically eliminated.

This is also accomplished by using a rotating piston placed directly upon the wicket shaft in a vertical position. This type of operating mechanism has been used with the 168 in. diameter butterfly valves of the turbines at Boulder, Nevada-Arizona. The piston 1 (Fig. 22) fits in a cylinder 2 with two separating walls, thereby forming four cylinder spaces, I, I', II, and II', to which oil under high pressure, produced by a motor-driven pump, is admitted to two chambers, the other two being connected to the drain. The design of these valves was carefully detailed by the U. S. Reclamation engineers and is very expensive.

European export turbine manufacturers have advertised a design of shut-off valve which is supposed to combine the advantages of both the gate valve and the butterfly valve design. From the first is embodied the filler-ring for guiding the water in open position of the valve and for protecting the seat surfaces. From the second the rotary or pivot feature is retained. Fig. 23 shows a section through valve in closed position and a section in wide open position. The valve housing 1 has two trunnion bearings for supporting the trunnions 2a of the rotary element 2. The center portion of the housing is a hollow sphere matching the machined spherical surfaces of the rotary element, one side of which contains a seal ring 3 which is pressed against the spherical seat surface in the housing. Between seal ring 3 and adjacent guide in rotary element 2 a space 3a is formed which can be put under penstock pressure alternately for sealing in closed position of the valve, or for release when the valve is moved from one position to the other.

Thus the valve can be made as tight-closing as a gate valve, and with as good a hydraulic flow condition as a filler ring type gate valve. The principal disadvantage of this design is that very severe flow conditions exist at intermediate position of valve, causing vibration and noise. In case of any emergency, closing may become critical mechanically unless a very rugged design is adopted. A valve of this type, properly designed, involves at least as much labor cost in our country as does a filler ring type gate valve.

The somewhat lengthy discussion of butterfly valve characteristics shall serve to emphasize the importance of proper design and construction of this type and to warn against adoption of mere standard market articles.

Tailrace Gates

To permit access to the draft tube of a turbine or the wheel pit of an impulse wheel, it is advisable to make provision for shutting the water off completely. Tailrace gates, or at least stop-logs, should be provided to prevent the water in the tailrace (or river) from "backing." The pit can then be unwatered by means of either a permanently installed or a temporarily used sump pump.

A very interesting case has recently been taken care of in an impulse wheel plant, where, because of abnormal seasonal floods, the tailwater would rise materially above the powerhouse floor. Reliable tailrace gates were provided, and sufficient advance notice of such a flood permits shutting the openings in the two wheel pits. Without these gates, to avoid the flood water's drowning the impulse wheels, the turbines would have to be placed at a materially higher elevation, resulting in a loss of head and subsequent loss of output capacity.

Since these floods are not frequent and are of short duration, it paid well to spend the additional money for such tailrace flood gates, and thus operate at a higher head during the major part of the time, thereby materially increasing the output available from the large storage reservoir.

In concluding this series of articles on turbines and accessories, attention is once more directed to the fact that Allis-Chalmers has a staff of specialists in this particular line of work with long years of experience in the entire field of hydraulic prime movers and their accessories, ready to assist prospective buyers to select the best-suited equipment. Best-suited equipment may not be of lowest initial cost but will always prove the best investment over a long period of years of use; in other words, it is not the value of the initially invested dollar that counts, but its value in the long run.

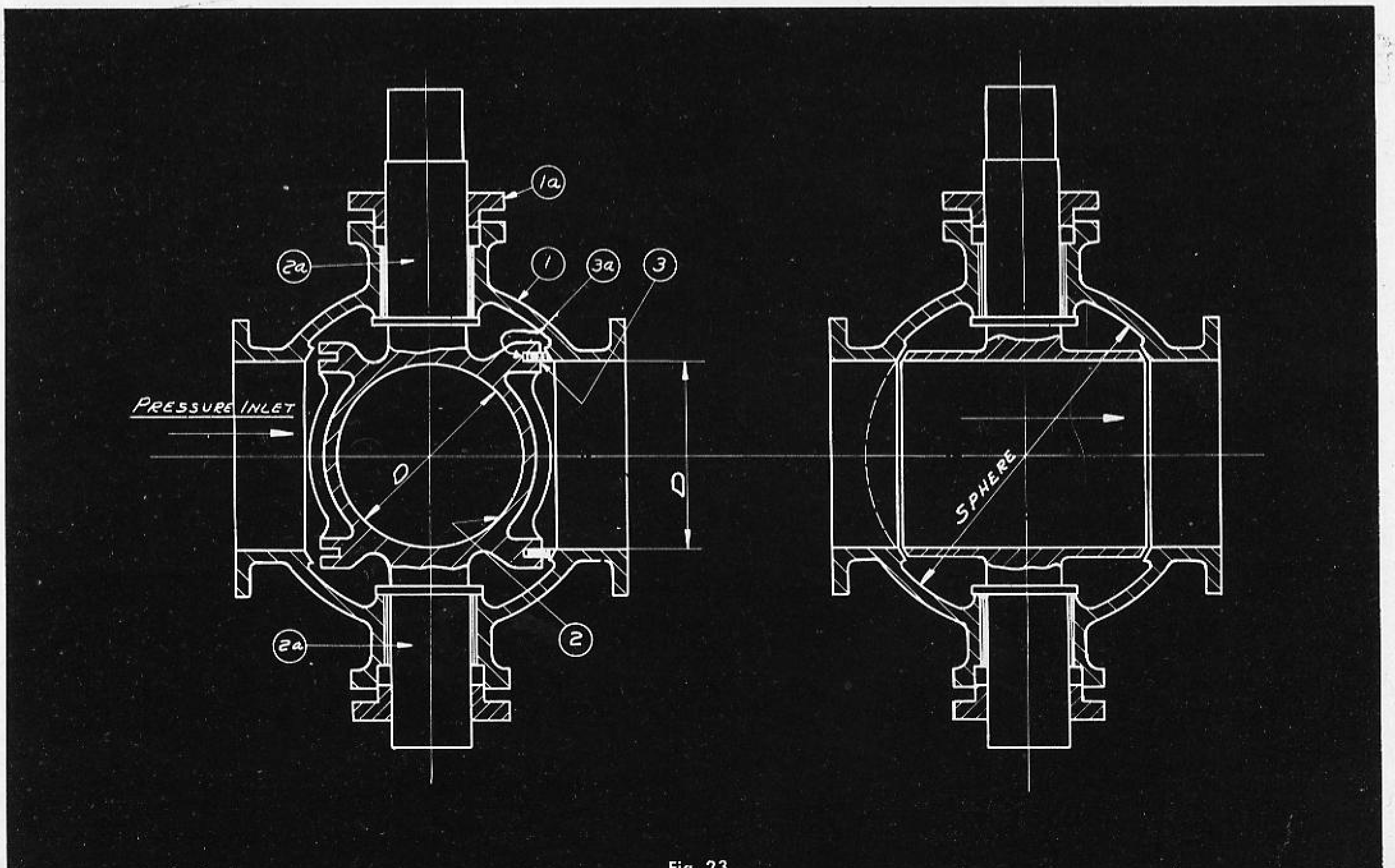


Fig. 23

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