erosion occurs on these faces because the runner acts as an impulse rather than a reaction wheel at small gate openings.

It was gratifying to note that even the seal rings did not show any wear. The manufacturer showed good judgment in applying different grades of stainless steel to the stationary and movable seal rings, since it was found that they had actually touched in a few spots without appreciable damage. It was found that the contact between the two surfaces was due to warpage of the inserts, which probably resulted from internal stresses which had not been relieved. This was corrected by grinding the stationary seal-ring inserts to a true circle by means of a field-constructed jig, and by buffing the movable rings. All of the surfaces were trued up and refinished without changing the original clearances appreciably.

The use of stainless steel has greatly reduced the amount of maintenance work which would have been required to keep the unit in a safe and efficient operating condition. By maintaining smooth surfaces it undoubtedly greatly increases the average long-term efficiency of the unit and removes some of the previous limitations that had been imposed to reduce wear on the unit.

**Conclusion**

During the more than 3 years' operation of this unit, all the difficulties encountered have been corrected. It is believed that the moderate initial cost (as compared with other types), the very high efficiency, and the absence of serious operating difficulties have amply justified the selection of the Francis type for the conditions under which this turbine operates, and that it will have a long life, with only moderate renewals and repairs.

**Discussion**

G. D. Johnson. The use of Francis instead of impulse turbines under head of 1000 ft and over has again been vindicated by the Nantahala turbine.

Because of the high efficiencies attainable and the economies in first cost due to high-speed equipment with small space requirements for the amount of power developed, there is a natural inclination to use Francis turbines for higher and higher heads. However, because of operating difficulties experienced with previous high-head installations, there are today only a few Francis turbines in the entire world operating under heads approximating 1000 ft. Apparently use of stainless steel and improved welding techniques will now assure satisfactory, reliable operation of units of this type without excessive maintenance.

The writer was privileged to witness a vivid illustration of this modern trend, when in November, 1945, he visited the Lages Plant of the Rio de Janeiro Tramway, Light, and Power Company (The Lights) in Brazil. At present this plant contains eight four-jet vertical-shaft impulse turbines and two vertical-shaft Francis turbines, all of European manufacture, operating under a rated head of 310 m (1018 ft). The small space occupied by the 50,000-jet vertical-shaft impulse turbines and two vertical-shaft Francis types for the conditions for which this turbine operates, and that it will have a long life, with only moderate renewals and repairs.

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S. J. Needs. The paper was particularly interesting to the writer, who happened to be at Nantahala on several occasions while the machine was being installed. It is also timely in that it discusses a most unusual machine after it has been proved successful in service. Comparison with larger machines makes it difficult to realize that such a small turbine can develop so much power. Its high output is due, to the exceedingly high head and this is one of the features which make the installation so outstanding.

The authors mention the spherical thrust bearing selected for this machine, and since the bearing is also quite unusual and interesting, a few words about it are in order. Instead of the flat thrust and cylindrical upper guide bearing, common in vertical machines of this type, a spherical thrust bearing is used. This bearing is capable of supporting radial as well as vertical loads, hence it eliminates the upper generator guide bearing.

The spherical thrust bearing for the Nantahala unit is shown in Fig. 10 of this discussion. It is located at the top of the machine and it runs in an oil bath with a cooler like most Kingsbury bearings in hydroelectric service. It is 51 in. diam and to date is the largest spherical bearing that has been built. The rubbing surface of the runner A is a zone of a sphere, the center of which is on the shaft axis. Six segmental spherical shoes B are fitted to the runner and pivoted in the usual Kingsbury manner. The bearing angle of the shoes is 39 deg with respect to the shaft center line. Total vertical thrust load is 575,000 lb or about 96,000 lb per shoe. Due to the bearing angle the normal load per shoe is about 123,000 lb and its horizontal component is approximately 78,000 lb. The shaft is held concentrically by the horizontal components of the thrust load acting radially at each shoe. The babbitt surface area of each shoe is 261 sq in. and loading per unit area is 472 psi.

At the normal operating speed of 450 rpm the average minimum film thickness is 2.25 in. along the trailing edges of the shoes is 0.0025 in. under the loading cited. When a horizontal load comes on the bearing the shaft can move sidewise due to the slight decrease in oil-film thickness at the more heavily loaded shoes. A side load of 10 per cent of the vertical thrust will decrease the minimum film thickness 0.0004 in., or 15 per cent. Twenty per cent side load will decrease the minimum film thickness 0.0006 in., or 24 per cent, and 40 per cent side load will cause the normal minimum film thickness to decrease somewhat less than 0.001 in. or about 36 per cent. Sidewise movement of the shaft is approximately the same as the change in minimum film thickness, from which it follows that the shaft is held very close to a central running position, even under appreciable side loads.

Total frictional power loss at full load and speed due to oil-film shear is 175 hp. This is about the same as the combined frictions of the corresponding flat thrust and cylindrical guide bearings. The frictional heat is removed by a cooling coil consisting of approximately 1000 ft of 1½-in-diam copper tubing. Oil is circulated around the cooler by means of a centrifugal pump consisting merely of two opposite holes drilled in the runner at an angle. At 450 rpm the peripheral speed of the


runner is 101 fps or about 69 mph. Static pressure at the discharge ends of the pumping holes is 43 psi and about 35 gpm is circulated around the cooler tubes. This flow is increased by the centrifugal action of the runner surface on the oil in the spaces between the shoes.

Clearance between the air-seal ring C and the removable thrust block D is so adjusted that oil is maintained in this space, thus sealing the bath against entrance of air. The inner air-seal ring at the runner bore prevents air from entering the oil at that point. Due to these seal rings the top of the bath is clear and practically motionless.

In the unlikely event that the shaft should ever rise, a cylindrical bronze guide bearing E is provided. Radial clearance between the outside diameter of the thrust runner and the guide bearing is 1/16 in., hence this emergency bearing is practically frictionless in normal operation.

Beneath the solid base ring F the bearing is provided with double insulation against shaft currents. A lead from the inter-base G makes it possible to test the insulation at any time.

The principal advantages claimed for the spherical thrust bearing are as follows:

1. The shaft is held in a concentric running position as if in a guide bearing of zero clearance.
2. Any inaccuracies of the thrust-block and ring key are compensated for by the permissible rolling action of the spherical runner. Movement so induced at right angles to the direction of motion would be of small amplitude and practically resisted by the fluid oil film. Hence galling between thrust block and shaft and shaft throwout at the turbine bearing due to thrust-block inaccuracies are eliminated.
3. The spherical bearing simplifies design of the thrust bracket by eliminating the cylindrical upper generator guide bearing and the special oil pump and piping usually required for its lubrication.
4. The spherical bearing is so constructed that at zero speed the babbitt faces of the shoes are segments of a true spherical zone and will bear equally against the runner. Thus equal shoe...
loading is inherent, no field adjustments of loading are necessary.

5 The vertical center lines of bearing and shaft will intersect at the center of the runner sphere, but it is not essential that these center lines coincide. Hence equal face loading will obtain despite shifting of foundations and appreciable departure of the shaft from plumb. To obtain this self-alignment with a flat bearing requires the use of leveling-plate equalizing construction.

J. F. Robertson. This 60,000-hp Francis-type turbine, operating under 925 ft head, has set a new mark in American practice for high-head reaction-type turbines.

One point which strikes us as particularly important is the marked improvement which stainless steel shows as compared with either cast steel or rolled steel in resisting cavitation and scour erosion due to the high velocities under these relatively high heads. This further confirms data gathered in regard to cavitation of Kaplan and propeller-type wheels. Possibly the Europeans are right in their conclusions that it is well worth while to make the runners and some other vital parts entirely of stainless steel. In this particular case, had the runner, guide vanes, and facing plates been made of stainless steel, possibly all of the repairs required at the end of 2 years of operation might have been avoided. Until recently it has been impossible to secure a stainless-steel runner in this country, but during the war several of the steel foundries have enlarged their stainless-steel casting capacity and several stainless-steel runners are now being manufactured. It will be interesting to see whether or not these outlast their carbon-steel predecessors.

The fact that stainless steel was used in the butterfly valve both as the seal ring on the disk and seat in the housing with excellent results further confirms the data regarding its greater resistance to wear and erosion. While discussing the butterfly valve, the excellent streamlining on the downstream side of the disk so as to maintain practically uniform velocity as the water flows from the valve into the spiral casing deserves mention. The housing contracts rapidly so that there is no decrease in velocity as the water leaves the area surrounding the disk, which in this case probably occupies from 25 to 30 per cent of the effective area in the valve.

The company with which the writer is connected built a 30,000-hp impulse turbine of the double-overhung type for this same power company at the time the Nantahala turbines were built. The difference in efficiency of an impulse turbine as compared with the reaction turbine was more than expected. While the impulse turbine gave about 1 per cent higher than the expected 88 per cent, or about 89 per cent, it still was over 4½ per cent below the 93.7 per cent obtained on the Nantahala turbine. Mr. Growdon and the other officials of the Nantahala Power Company, in discussing this difference, agree with the writer that if we had had as much experience with high-head reaction turbines at the time of building these plants, both plants, that is, Glenville as well as Nantahala would have been of the high-head reaction type.

The tabulation showing the comparison in efficiency between the 22-in. model and the 85-in. Nantahala turbine deserves consideration, especially the increase of 9 per cent at half load from the 22-in. model and the 85-in. Nantahala turbine deserves consideration, especially the increase of 9 per cent at half load from the 22-in. model and the 85-in. Nantahala turbine. The maximum efficiency obtained of 93.7 per cent is indeed a splendid record.

As pointed out in the paper, the successful use of a Francis-type unit instead of the impulse type results in higher efficiency, higher speed, and lower cost. The difficulties to be guarded against are unstable operation, pitting due to cavitation, and wear of internal parts, caused by the extremely high velocities and abrasive materials in the water, and loss of the original efficiencies due to wear at the runner seals.

An interesting comparison with the Nantahala turbine is the 39,000-hp unit built by the writer's company for the Ixtapantongo Development of the Comision Federal de Electricidad in Mexico. This unit was designed for a net head of 1028 ft at a speed of 500 rpm. The specific speed is 20.3, only slightly less than the value of 21.0 given in the paper.

In the Nantahala unit a radial clearance of about 6 in. was allowed between the wicket gates and entrance to the runner vanes to result in smooth flow from the gates into the runner. For the Ixtapantongo unit the corresponding clearance was about 2½ in. It is our experience that so large clearance is not necessary if wicket gates are so streamlined as to produce smooth flow. This smaller clearance results in decreased size of unit and lower cost.

The authors describe how the heavy upward hydraulic thrust on the bottom of the gate stems caused a deflection of the upper flange of the crown plate, which permitted the top of the gates to rub the upper wearing plate, and this friction prevented sufficient gate closure to stop the turbine completely.

This difficulty was avoided in the Ixtapantongo unit by extending the lower gate stems through stuffing boxes in the lower cover, thus eliminating all upward thrust.

The repairs made after 2 years of operation are of particular interest. It is noted that considerable wear occurred on the runner, throat ring, gates, curb, and crown-plate wearing rings and seal rings, and that these parts were repaired by welding the surfaces with 25-20 stainless-steel rods.

In the case of the Ixtapantongo unit, the same carbon-manganese steel (S.A.E. 1045) was used for the wearing rings and seal rings, and the same bronze inserts in the stationary seal rings. This unit, however, has been in operation only about 18 months, but the power company reported that an inspection made after the first year of operation showed no appreciable pitting or wear of the internal parts.

The authors speak of erosion or mechanical wear rather than pitting and the writer would like to know if this wear was caused by erosive action from sand or other abrasive materials in the water. It is noted that water is carried through an unlined pres-

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4 Manager, Hydraulic Department, Allis Chalmers Manufacturing Company, Milwaukee, Wis. Mem. A.S.M.E.

sure tunnel, which might result in foreign materials being present in the water.

It has been our practice to use 18-8 stainless steel on the runner vanes, gates, wearing rings and seal rings to resist pitting rather than the higher chrome-nickel materials. We understand that recent tests made at the Massachusetts Institute of Technology indicated that a 17-7 stainless steel showed greater resistance to pitting than the higher combination of chrome and nickel.

It is quite possible that the 25-20 stainless steel used at Nantahala has a higher Brinell hardness and will stand up better where sand is present under these extremely high velocities. We would like to have the authors' opinions on this matter.

The efficiencies obtained on the field test as compared to the model test are unusual, as at best efficiency the prototype shows 93.7 per cent and the model 89.5 per cent; a gain of 4.2 per cent. The well-known Moody formula would show for the runner sizes and heads stated, a gain of 0.25 per cent for a coefficient of \( n = 0.25 \), and a gain of 2\( \frac{1}{4} \) per cent for a coefficient of \( n = 0.20 \).

For this value to be homologous on the model it would be of interest.

One of the principal advantages in obtaining well-conducted field tests, to the turbine designer, is the comparison thereof with synchronous-condenser operation. The Nantahala field test is unusual in that the step up in efficiency is so great. The Moody formula on the basis of the losses varying inversely with the one-fourth root of the runner diameters would result in a field efficiency of 92.5 per cent. A field efficiency of 93.7 per cent corresponds with an exponent of about \( \frac{22}{20} \).

Possibly one reason for the large increase might be in the relatively large model seal clearance. For this value to be homologous on the model it would be

\[
0.015 \times \frac{22}{20} = 0.0039 \text{ in.}
\]

which is an impractically small value.

On the upper portion of Fig. 11 of this discussion is shown the model curve as compared with the field, as plotted from the data in the authors' paper. It is noted that the guarantees at part load are disproportionately higher than the model curve which falls off quite rapidly. Further information in regard to this point would be of interest.

On the upper portion of Fig. 11 a comparison is given of the Shipshaw field and model tests of the Smith unit tested. In this comparison amount to more than the sums of those included, it is believed that this comparison gives a fair indication of what efficiency might be expected from a Francis turbine of still lower specific speed for possible use with an appreciably greater head than 1000 ft, that is, something over 92 per cent for such conditions would be in order with a runner of about 85 in. diam. An account has been given of a unit of Escher Wyss design which attained from field test an efficiency of 91.4 per cent when developing 16,000 hp under a head of 918 ft, at a specific speed of 15.07. The diameter of this runner was only 70 in.

The introduction of 80 gpm of clear water into the seals is an interesting feature, but the small amount used leads the writer to question the advantage gained thereby as a means of protecting the seal rings from the passage of foreign matter. Possibly the original reason for the introduction of this water was in connection with future guarantees and designs. The Nantahala field test is unusual in that the step up in efficiency is so great. The Moody formula on the basis of the losses varying inversely with the one-fourth root of the runner diameters would result in a field efficiency of 92.5 per cent. A field efficiency of 93.7 per cent corresponds with an exponent of about \( \frac{22}{20} \).

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The Moody formula steps this up to 93.8 per cent field efficiency, which agrees quite closely with the actual value of 93.6 per cent. Both the Shipshaw and Nantahala field tests show satisfactorily high part-load efficiencies, with the Nantahala values somewhat higher if they are both brought to the same percentage of load for maximum efficiency.

The writers make no mention of presence or absence of foreign matter in the water, but it would appear that there is some abrasive matter present, and possibly some degree of acid content. The writer had occasion to examine the seals of Units N5 or N6 at Boulder after about a year of operation under a head above 500 ft, and found the clearances on the average slightly less than when the units were installed, due to the deposit, in some places, of verdigris on the white brass inserts in the stationary rings. The Boulder water was, during this period, exceptionally free from abrasive matter.

The substitution on the Nantahala turbine, of stainless steel on the stationary wearing ring for bronze, while showing gratifying results from the standpoint of maintenance low leakage loss, appears questionable to the writer from the standpoint of safety during possible runaway speed. Assuming a runaway speed of 725 rpm, which is believed to be conservative, calculations made indicate an expansion at this speed of about 0.008 in. on the radius, leaving 0.008 in. as the clearance under that condition, or a possibility of this being reduced to 0.003 in., if the shaft swings to its extreme position in the bearing. Even though the rotating and stationary seal surfaces are not subject to galling, the presence of two broad surfaces of hard metals under this admittedly abnormal condition might result in violent seizure due to heating after contact. In view of the excellent resistance of stainless steel for these surfaces, as brought out by the authors, labyrinth seals of this metal along the lines used in steam-turbine design might be a good solution from all standpoints.

On the Ixtapantongo unit mentioned the lower gate stems extended through stuffing boxes in the bottom, or curb plate, and were of the same diameter as the upper stems. This was for the purpose of avoiding the heavy upward thrust on the stems, and also of avoiding the loss of grease due to water passing under the lower ends of the stems and forcing the grease upward on the sides of the stems nearest the runner.

The writer wishes to congratulate Mr. Terry on the excellent design and performance of the Nantahala turbine, and all of the authors on the frank manner in which they discuss the difficulties encountered. This attitude is of distinct long-range benefit both to manufacturers and users.