Erection experiences, unusual problems, and their solutions encountered in the installation of major heavy hydroelectric equipment
by Donald C Beckman

A THESIS Submitted to the Graduate Faculty in partial fulfillment of the requirements for the professional degree in Mechanical Engineering at Montana State College
Montana State University
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Abstract:
The installation of a third generating unit and all associated equipment in the Fort Peck Power Plant at Fort Peck Dam, Fort Peck, Montana, presented a number of unusual erection problems and necessitated immediate and correct solutions to insure the proper installation of the heavy hydroelectric equipment, A satisfactory Construction Progress Schedule, taking into consideration all possible reasons for delay, had first to be prepared to insure that the work was completed in the allotted time. When the actual erection of the penstock was started, difficulties were encountered in obtaining "tight" rivets. It was necessary also to analyze the penstock stresses to determine the best joint for the closure section.

The placement of the immense 82 ton butterfly valve was accomplished after considerable planning. Turbine erection problems included the proper alignment of the curb ring, the speed ring and the shaft.

The assembly of the rotor laminations, the installation of the keys, the assembly of the thrust collar assembly on the drive shaft, and the assembly of the rotor on the thrust collar all present problems that had to be satisfactorily solved during the process of the assembly and erection of the generator.

When the work was completed an overspeed and turbine efficiency test was performed. The satisfactory results obtained indicate that the erection problems encountered were solved satisfactorily.
ERECTION EXPERIENCES, UNUSUAL PROBLEMS, AND THEIR
SOLUTIONS ENCOUNTERED IN THE INSTALLATION OF MAJOR
HEAVY HYDROELECTRIC EQUIPMENT

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The installation of a third generating unit and all associated equipment in the Fort Peck Power Plant at Fort Peck Dam, Fort Peck, Montana, presented a number of unusual erection problems and necessitated immediate and correct solutions to insure the proper installation of the heavy hydroelectric equipment.

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INTRODUCTION

The problems presented in this treatise are unusual to those which normally occur in the erection of heavy hydroelectric equipment, and they are also distinctive because their solutions were primarily dependent upon the successful application of sound engineering knowledge and practices. All of the problems discussed in this thesis actually occurred during the installation of a 50,000 horsepower Francis type turbine, a 16-foot diameter butterfly valve, and a 35,000 kilowatt capacity generator, with the associated equipment, in the Fort Peck Power Plant. The writer was intimately associated in the problems and their solutions.

The Fort Peck Power Plant is located near the right abutment of the Fort Peck Dam. This dam is the largest hydraulic earth fill dam in existence in the world today, and may always be so, since the advent of improved heavy duty earth moving equipment makes hydraulic placement methods uneconomical. The Fort Peck Dam is located across the Missouri River at River Mile 1866.7, which is 1866.7 miles above its point of confluence with the Mississippi River, according to the 1941 adjusted river mileage established by the U. S. Coast and Geodetic Survey. The resultant impoundment created by this dam has a shoreline of approximately 1600 miles, is 169 miles long, has a maximum capacity of 19,412,000 acre-feet of water when filled to the elevation of 2250 mean sea level, and develops a total head for power generation of 210 feet at this elevation. As is commonly known, the power development of this project was accomplished with Federal funds, authorized by Act of Congress dated 18 May 1938, with the Corps of Engineers designated as the constructing
agency. While the power development of this project is of major importance, other multiple purposes of this project are concerned with flood control, navigation, irrigation and recreation features.

The work of installing the hydroelectric equipment mentioned above began in June of 1950, by means of a construction contract issued by the Department of the Army, Corps of Engineers. The successful bidder to the contract was the E. V. Lane Company, Engineers and Contractors, of Palo Alto, California.
THE FORMATION OF A CONSTRUCTION PROGRESS SCHEDULE

Since the terms of the contract required all of the work to be completed by 15 December 1952, and because that firm date had been established with the power marketing agency as the date upon which the production of power from the newly installed unit would be available, it was necessary that a carefully prepared installation schedule be established. The preparation of this schedule, while in outward appearances was simple, was full of troublesome problems. Unexpected work conditions and erection difficulties with resultant delays required consideration in the formulation of an accurate construction schedule. The problem was approached by considering all of the salient features of the work, and those mentioned above, and allowing for reasonable contingencies, then plotting the order of work on a graph. The greatest difficulty encountered was the bewildering number of nebulous variables introduced when the considerations stated above were tabulated, and the estimated working times were computed for each item of the contract. The sequence of the work was perhaps the easiest determination to make in the preparation of the schedule. The consolidated results of the estimates appear as Plate I, titled CONSTRUCTION PROGRESS CHART. The estimated scheduled progress of the entire amount of the work under the contract is plotted on the chart in a solid heavy line, and the actual progress of the work is plotted on the chart with a heavy broken line. It can be seen from the chart that the contract was successfully completed within the specified time. It is considered that this success was due to a carefully prepared schedule, and the diligent efforts of the contractor and the
supervising engineer to adhere to the schedule. It may also be stated that a minimum of overtime was worked on this contract, resulting in savings to the contractor. It is believed that a chart of this nature should be prepared for every contract.
PENSTOCK ERECTION PROBLEMS

The next problem which was of major importance occurred during the erection of the penstock, which in this case appears as item numbered 9 on the Construction Progress Schedule. The work item designated as "Penstock and Riser" involved the installation of approximately 790,000 pounds of shop fabricated steel plate. The penstock consisted of 7/8 inch thick carbon steel plates rolled so as to form a 14 foot diameter tube. These plates were joined by a single riveted butt joint which formed the circumferential joints. Each row of the 1 inch diameter button head rivets forming this circumferential joint was spaced 2-5/8 inches apart and pitched 3-3/4 inches. These hot driven rivets were installed in 1-1/16 inch diameter reamed holes in the single 5/8 inch thick exterior buttstrap and the 7/8 inch thick penstock plate. The longitudinal joints in the plates were made up of double riveted butt joints employing 1 inch diameter rivets driven in 1-1/16 inch diameter reamed holes pitched 3-3/4 inches in the 11/16 inch thick interior splice plate, the 7/8 inch thick penstock plate, and the 5/8 inch thick exterior splice plate. The interior splice plate was fabricated large enough to accommodate an extra row of 1 inch diameter rivets pitched 9 inches. Light caulk welds were utilized in making the joints watertight where the longitudinal plates met the circumferential joints.

The riser group consisted of a "tee" that was riveted to the penstock, and a 11 foot diameter tube which connected the tee to the 50 foot diameter surge tank. The tee was fabricated from welded and rolled steel plate, the
completed assembly of which weighed approximately 25 tons. This tee was formed on the order of a saddle, having an outlet at the top of the tee which was connected by rivets to the 11 foot diameter steel plate tube connecting the penstock to the surge tank bowl. The plates and joints connecting the riser sections together were of the same type as those in the penstock. Within the 11 foot diameter opening which formed the top of the tee or riser, was a welded steel gridwork designed to act as an orifice and in some part restrict the surges of water that would result whenever the turbine wicket gates were closed rapidly. This steel gridwork consisted of a weldment of 4-1/2 inch thick plates. The gridwork was welded to the body of the tee by full strength welds. The body of the tee was of 2-3/4 inch thick rolled steel plate, and was provided with full length machined bearing surfaces for steel columns designed to support the entire weight of the riser, tee, and the portion of the penstock load which was applicable. Plate II is a photograph of this massive member, and shows a top view of the tee as it was being moved into position. The riser was approximately two weeks late in delivery to the project due to the fact that difficulty was experienced in fabricating the heavy member. There was but one steel fabricator in the west coast area who had rolls sufficiently large enough to roll the heavy 2-3/4 inch thick steel plate body, necessitating shipping the tee to the fabricator and returning it when done. The extremely heavy welds employed in joining the 4-1/2 inch gridwork caused distortion and induced internal stresses of great
PLATE II - 25 TON TEE
magnitudes. This condition required a special backstepping welding procedure\textsuperscript{1} to be followed in welding, and final stress relieving of the entire weldment in an annealing furnace.

Upon performing the work of riveting the platework and assembled structures together many loose rivets were found during the first riveting operations. The rivets were tested for looseness in the following manner:

a. First a light hammer blow was struck the side of the cooled button head of the rivet to be tested.

b. Next, a 1/2 inch diameter steel ball was held against the plate and the button head of the cold rivet, and a heavy hammer blow struck the rivet head from the side opposite the steel ball and opposite the direction previously struck.

If there is slight looseness in the rivet, the steel ball will not respond to the hammer blow; conversely, if the rivet is tight, the steel ball will rebound from the rivet and the plate. Figure 1 is a sketch of the application of this testing method.

It was found that there were many loose rivets when this test method was applied, and it became necessary to determine the trouble before proceeding.

The rivets initially were being heated to the proper driving temperature by means of a gas fired furnace. Upon analysis, it was discovered

Steel ball brazed on 1/8" rod — Hammer blow

Penstock plate

Splice plate

Hot driven rivet

Not drawn to scale

RIVET TESTING METHOD

FIGURE 1
that the sulphur and other impurities in the gas were causing a glaze
to form on the surface of the heated rivet, and when driven, the glaze
crumbled into a hard but porous type scale. While the rivet was cooling,
this scale allowed the contraction of the rivet to be taken up in crush-
ing the scale. When finally the rivet was cold, it was loose, and the
plates through which it had been driven were not gripped. The elimination
of the gas fired rivet heater was the obvious solution to this troublesome
problem. When the rivets were heated with a coke fired oven, the diffi-
culty was entirely eliminated, verifying that the analysis of the trouble
was correct.

During the process of installing the penstock and riser, the question
arose as to which of the many circumferential joints should be selected
for the closure section. As shown in Figure 2, the penstock and riser
both were restrained at each end. It was initially considered that a
large temperature differential occurring after the work of installing the
penstock and riser was complete, would probably have undesirable effects
on the completed assembly.

Since there were no expansion joints in either the penstock or riser
assembly to provide for expansion or contraction, it was necessary to
analyze the conditions that would affect the final closure. The plans
and specifications were not complete in this respect, therefore the
analysis of where to make the closure joints, and under what conditions,
was one of the decisions that had to be made in the field during construc-
tion. The calculation of the stresses due to the temperature variations
in the penstock and riser sections were simple applications of engineering practice and led to the determination of the closure section, and the conditions under which it was deemed safe to make the closure. The analysis appear below:

**General Design Data**

- Normal water temperature range: 45 to 55 degrees F.
- Ambient temperature range: 40 to 85 degrees F.
- Design maximum head exerted on penstock: 210 feet.
- Maximum recorded surge pressure: 115 psi.
- Coefficient of expansion: 0.0000067
- Maximum allowable working stress in tension, compression and bending: 15,000 psi.
- Maximum allowable working stress in shearing: 12,000 psi.
- Maximum allowable working stress in bearing: 20,000 psi.

**Basic Calculations**

- Area of penstock section: \( (14)^2 \times 0.785 \) which equals 154 square feet.
- Weight of water per foot of penstock: \( (62.4)(154) \) which equals 9,610 pounds.
- Weight of penstock plate per linear foot of penstock not including buttstraps and splice plates: \( (14 \text{ ft.})(3.14)(35.7 \text{ lbs. per square foot}) \) equals 1,570 pounds.

---

1. American Institute of Steel Construction, 1941 STEEL CONSTRUCTION, P. 336, American Institute of Steel Construction, New York, N. Y.
Basic Calculations (Cont'd)

Area of riser section................. \((11 \text{ ft. diam.})^2(0.785)\)
equals 95 square feet.

Weight of water per foot of riser.............. \((11)(62.4)\)
equals 686 pounds.

Weight of riser plate per lineal foot of riser, not including
buttomseals and splice plates.............. \((11 ft.)(3.14)(35.7 \text{ lbs. per square foot})\)
equals 1,233 pounds.

Assuming that the worst possible condition would prevail, the stress-
es in tension resultant from the contraction of the penstock and riser
are calculated for the condition occurring when the assumption is made
that the closure section would be made when the ambient temperature was
85 degrees Farenheit and the penstock was dewatered, followed by filling
the penstock and riser with \(45\) degree Farenheit water under the full
design head of 210 feet and a surge of 115 psi occurred. The increase
in unit tensional stress for the above described condition follows:

\[ S_{\text{tension}} = \text{Unit elongation} \times \text{Modulus of Elasticity} \]

In this case, the unit elongation is 0.0000067 inches per inch multiplied by 40 degrees
F. and equals 0.000268 inch per inch. The Modulus of elasticity for the
steel plate is 29,000,000. Therefore it follows that the unit tension-
al stress developed in the plates as a result of the temperature change
would be 7,772 pounds per square inch, in both the riser and penstock.

1. American Institute of Steel Construction, 1941, STEEL CONSTRUCTION,
P. 337, American Institute of Steel Construction, New York, N. Y.
In addition to the stresses developed due to the temperature differential, the longitudinal stress in the plates from the hydrostatic pressure was calculated, using the basic relationship, Stress = $P/A$, and appears below:

$$S_1 = \frac{115 \text{ psi} \times (14 \text{ ft} \times 12)^2 \times 0.785}{7/8 \text{ in} \times 14 \text{ ft} \times 12 \times 3.14}$$

$$S_1 = 5,520 \text{ psi.}$$

In addition to the induced temperature stresses, and those developed by the hydrostatic pressure from the maximum recorded surge pressure of 115 psi, the approximate stresses due to the uniform loading between penstock supports was calculated. If the penstock were analyzed throughout its length, as shown in Figure 2 of this thesis, the diagrammatical loading of the penstock, using beam analogy would appear as shown in Figure 3. The complete analysis of a member of this type was considered difficult due to the inability to solve for all the reactions, a condition created by the unknown effect of the induced temperature stresses on the riser group which was connected to the penstock. To facilitate the analysis, the stress in the penstock due to the water loading was calculated for the longest restrained span, omitting one intermediate column support which gave calculated values higher than the actual stress, but for this purpose was considered as an added safety factor. The analysis of the penstock section using the above consideration is shown in Figure 4.

The maximum moment equals $wL^2/12$ for this condition, where $w$ is the
ELEVATION OF PENSTOCK - RISER LAYOUT

No scale

FIGURE 2
SKETCH OF PENSTOCK—RISER LAYOUT

FIGURE 3
Tee and riser group

Intermediate support omitted

\[ w = 11,180 \text{ lbs per foot} \]

\[ R_1 = 449,500 \text{ lbs} \]

\[ R_2 = R_1 \]

\[ \text{Maximum moment} = \frac{wl^2}{12} \]

FIGURE 4
-21-
unit load in pounds per lineal foot, and \( l \) is the distance between
supports in inches, which in this case is calculated as follows:

\[
M = \frac{(11.180)(965)^2}{12}
\]

\( M = 86,700,000 \) lbs. in.

The moment of inertia about the central axis of the penstock is
found from the relationship

\[
I_0 = 0.049087 (d^4 - d_i^4) \frac{c}{I}
\]

and is computed below:

\[
I_0 = 0.049087 (69.25)^4 - (168)^4 \frac{c}{I} = 1,658,000 \text{ in}^4
\]

The unit stress in bending can then be determined from the relation-
ship

\[
S = \frac{M \times c}{I}
\]

where \( M \) is the bending moment in pound inches, \( c \) is
the distance in inches from the center of the section to the extreme outer
fiber, and \( I \) is the moment of inertia about the central axis of the member.

Solving the equation,

\[
S = \frac{56.7 \times 10^6 \times 69}{1,658 \times 10^5}
\]

which yields 4,400 psi.

By adding the induced temperature stresses, the hydrostatic stresses
and the bending stresses, a maximum unit stress of 17,692 is obtained.

This value is slightly in excess of the maximum allowable working stress
for tension, and it was therefore considered necessary to reduce the
stresses created by the temperature differential as much as possible when
the final closure section was made in both the penstock and riser. The
work was scheduled in the winter, and the contractor was instructed that
the maximum temperature that the connection could be made was 70 degrees
Fahrenheit, although in actuality the actual temperature of the steel was
the same as that of the water when the closure was made. The analysis indicated that any section of the riser would be satisfactory for a closure section, and the joint nearest the surge tank was selected merely because of its convenience. The analysis indicated that the best joint for the closure section of the penstock would be a joint nearest one of the fixed supports, and in this case the joint nearest the butterfly valve was selected because of its accessability. Since the penstock and riser were designed with equivalent unit stresses in bending, compression, bearing and shear, consistent with the allowable stresses stated previously, it is not considered necessary to present them in this paper, as it is felt that to do so would be unnecessarily repetitious.

After the closure sections were made and the penstock and riser watered up, it was found that only a few rivets required tool caulking to make them water tight, which indicated that the closure had been made under conditions which had not overly stressed the joints, or there would have been many rivets that would have leaked due to their being disturbed. The improper closure procedure or selection of closure temperatures could have resulted in considerable delay and loss of time in repairing leaks and replacing rivets.
Erection of the 16-Foot Diameter Butterfly Valve

An interesting problem occurred when the preparations were being made for the erection of the huge 16-foot diameter butterfly valve. The disc portion of this valve was made of cast semi-steel, while the trunnions consisted of 24 inch diameter solid machined shafts pressed into the cast body, the aggregate of which weighed 52 tons. The valve body was fabricated of 2-1/4 inch thick rolled steel plates, reinforced with heavy welded gusset plates, and was carefully machined on the inside surface to close tolerances. The valve body was fabricated in two sections, one of which formed the top half of the valve, and provided a support platform for the bearing housing and operating mechanism. The lower half of the valve body provided a support for the bottom bearing housing and supports for the assembled valve.

The contract plans and specifications provided that the lower half of the valve body was to be imbedded in concrete which formed a foundation pad. A lifting pin was provided in the ceiling of the butterfly valve corridor which was intended for use in lifting the heavy parts into position, but upon careful analysis, it was discovered that the lifting pin and the supporting structure was of insufficient strength to support the loads of hoisting the valve disc and trunnion into position. The problem was further complicated by the restricted space for maneuvering the valve and its parts. It was initially planned to use the generator bay crane to hoist the valve parts, move them from the erection bay floor, and lower
them into the penstock area. These parts then had to be moved down the penstock corridor, and finally hoisted into position into the butterfly valve corridor. It was questioned whether the slot in the floor separating the penstock area from the butterfly valve corridor was large enough to accommodate the disc and trunnion, and whether there was room in the areas mentioned for the required maneuvering. The final determination was made by constructing a paper scale model of the valve parts and maneuvering them on the cross sectional drawing of the areas affected, from which the best method of assembly was derived. The hoisting of the valve parts into position into the butterfly valve corridor was the bottleneck to the problem. The contractor proposed to use the lifting lug mentioned above to hoist the operating mechanism and the top half of the valve body into position, then utilize the top half of the valve body itself for the anchor and support for hoisting the disc and trunnion into position. This method was rejected because the valve manufacturer stated that the valve body would be permanently deformed, and the guaranteed leakage minimum could not be met. After several other methods of hoisting the valve into position were considered, one method became obvious as the best solution to the problem. The top half of the valve and the operating mechanism were hoisted into position using the lifting lug, as they each weighed less than 30 tons, and hoisted singly would not overload the lifting pin. Then steel beams were used to bridge the opening in the floor, and the top half of the valve body was supported by the beams. Inside the valve body a temporary "A" frame was built, which was
supported by the steel beams. Finally, a jacking bar was built, and hydraulic jacks were used to lift the disc and trunnion into position. The remainder of the valve was assembled without further difficulty, and when the seals were finally adjusted and tested, the total leakage was less than 5 gallons per minute, the guaranteed leakage. This indicated that the valve body was not deformed or abused in the difficult assembly. Plate III is a photograph of the valve while being assembled and Plate IV is a photograph showing the jacking arrangement.
PLATE III - PARTIAL ASSEMBLY OF VALVE
PLATE IV - JACKING ARRANGEMENT
ERECTON LAYOUT PROBLEMS

One of the problems causing the most concern during the installation of this generating unit was the matter of layout and accurate control of elevations, and alignments. The contract plans were complete in the establishment of elevations, dimensions and alignments, but to transfer and control these points in the field was a matter of constant concern. It may lend emphasis to the point somewhat if we consider the matter of the elevation of the curb ring, which forms the foundation upon which the turbine is ultimately erected. Since an error of 1/64 inch is about all that the tolerance in length of the shaft coupling the turbine to the generator will permit, the elevation of all the composite parts of the turbine must be carefully controlled. To accomplish this, control points were carefully established in convenient locations throughout the work area, both for critical elevations and for alignment. These control points were laid in with the transit and level, but even by careful closing and checking of all centerlines and elevations, it became necessary to design instruments more accurate than these to accurately establish alignments and elevations. The curb ring was the first portion of the turbine that caused this concern. The ring was set to elevation by means of a dumpy level and checked for level with a machinist's level placed on the machined top surface of the curb ring. To our astonishment, neither agreed with the other level, for when the curb ring was leveled with the engineer's level, the machinist's level would not confirm the ring as being level. Therefore it was considered that the
engineer's level was causing the inaccuracy due to the inability of the reader to read the rod closely enough. A "micrometer rod" with a light target was then constructed, and the level was equipped with a short reading eyepiece. The resultant work with this equipment gave better accuracy than that indicated by a machinist's level. Refer to Plate numbered V which is a photograph of the "micrometer rod". It is considered that without the use of this specially designed tool that the subsequent erection of the turbine would not have been accurate. This fact is borne out by the fact that the No. 1 unit in this power plant, which is similar to the No. 3 unit discussed herein, has always been slightly out of plumb, and consequently the thrust and guide bearing in unit No. 1 constantly requires readjustment because the unit is leaning into one side of the bearings. No such difficulties have been experienced in the No. 3 unit, and are not expected. It is considered that the above described method of leveling represents one solution to major leveling problems in the installation of heavy hydroelectric equipment.

Coupled with the leveling problem of the curb ring mentioned in the preceding paragraph, the alignment of the speed ring became somewhat critical when it was discovered that after the scroll case had been riveted to the speed ring, and although careful precautions and concrete placement procedures were followed, the speed ring was found to be out of round approximately 0.050 of an inch on the inner bore. It was found that the heavy caulk welds made in joining the riveted scroll case plates
to the speed ring caused this out of round condition, but the problem was that with the speed ring out of round, the top and bottom plates would no longer fit properly and alignment of the turbine wicket gate stems and gates would be most difficult. By reference to the true centerline established by the control point the amount of out-of-round was determined, and the "eggled" portion was then trued up by grinding by hand; a very time consuming, costly, and painstaking job. It has since been concluded that a more accurate and expeditious method is to erect a grinding arm by setting up a spider in the true centerline of the unit, then setting the grinder on a machined arm from this center and grind or machine out a new true radius for the top and bottom plates consistent with the allowable tolerances. It has been determined impractical to attempt to prevent the misalignment or "eggling" due to welding because the caulk welds are necessary to prevent leakage past the scroll case plates. In any case, it is necessary to carefully establish a true centerline of unit reference points in a suitable location in order that it may be used throughout the entire erection of the unit, and it is also concluded that the latter method of truing up the machined bores of the speed ring will result in a better and more accurate method of maintaining the installation of the unit on a true centerline.
Along with the alignment of the turbine distributor or "speed ring" on a true centerline was the problem of aligning the turbine shaft with the turbine guide bearing and at the same time setting up the generator guide bearing alignment, and the thrust bearing adjustment. This turbine-generator unit has three major bearings, two of which are guide bearings and one of which is a thrust bearing. The generator guide bearing consists of a series of adjustable shoes, babbitt lined and adjustable by means of radial jackscrews set into the bearing housing. This bearing maintains vertical shaft alignment.

The generator thrust bearing accepts the entire vertical loading of the weight of the rotor, shaft, turbine runner, and the hydraulic down-thrust. In this case the total combined weight of these parts is 130,000 pounds or 65 tons. The hydraulic downthrust at full load is 247,000 pounds. The bearing supporting this vertical loading is of the Kingsbury type, having adjustable thrust shoes. The problem is to adjust each of the thrust shoes in the Kingsbury thrust bearing, and each of the guide shoes in the generator guide bearing while maintaining the true alignment with the bore of the turbine guide bearing, the latter of which is not adjustable, while simultaneously maintaining the correct elevation at the lip of the runner. Mention is made of the latter consideration because a difference of 1/8 inch in elevation of the turbine runner with relation to the gate ring throat will result in major cavitation difficulties and greatly reduce the unit's efficiency. The problem of alignment of these three bearings, and elevational control was approached in the following
manner: First the Kingsbury thrust shoes were brought to bear on the thrust collar after the shaft had been roughly centered. By utilizing the "slugged arc" method, the jackscrews were gradually tightened until the shaft with the turbine runner attached was raised by the Kingsbury thrust shoes until the runner was within 1/64 inch of the desired elevation. By this time the threads on the jackscrews had "worn in" sufficiently to permit an adjustment to be maintained. The shaft was then centered up in the bore of the turbine bearing housing and the generator guide bearing thrust shoes were tightened up snugly. This operation permitted the shaft assembly to freely swing like a gigantic plumb bob.

Dial indicators were then fastened to the turbine guide bearing housing and the process of bringing the runner up to elevation while equalizing the pressure on each of the thrust shoes was begun. To very accurately center the shaft into the bore of the turbine guide bearing, to accurately determine the straightness of the shaft and to maintain plumb in the shaft at all times a special tool was constructed and used. Refer to Plate VI which is a view of this tool. Next .008 diameter piano wire was suspended from the generator bearing housing from adjustable points equidistant from the center of the generator guide bearing. These four wires suspended as described above were weighted with 20 pounds of lead, and these weights were immersed in lightweight lubricating oil to dampen any swing.

1. This tool was used in the following manner: Headset leads were attached to the tape as shown on the photograph. The vee block was placed against the shaft, and the micrometer adjusted until it contacted the wire. Upon contact a "click" was audible in the headset.
PLATE VI - SPECIAL ALIGNING TOOL
caused by vibration. Throughout the process of bringing the shaft into line, these wires were used to check the plumb of the shaft, and the centering. Prior to the final setting of all bearings, the shaft was checked for straightness by rotating it by quarter turns and checking the "run out" at several places along the wire. It was found that the bolts at the coupling had not been tightened uniformly, and that there was considerable misalignment of the shaft. By slugging up the nuts with a slugging wrench made for that purpose, the coupling bolts were tightened uniformly until the shaft ran true. It was discovered that the special tool used for this purpose was accurate to 0.00025 of an inch. Finally, the thrust shoes on the Kingsbury bearing were slugged up until a reading of 0.010 inch was recorded on the dial indicator rigged up in the bore of the turbine bearing housing opposite the position of the thrust shoe. Then the opposite thrust shoe was slugged up until its corresponding dial indicator read zero again, and this process was repeated around the entire circle of thrust shoes. When the proper elevation for the runner had been achieved, the thrust shoes were all equally loaded by this method, and the shaft rested in the true center of the nonadjustable turbine bearing and was perfectly plumb. The calculated clearance for the generator radial guide bearing shoes was 0.018 inch, which was obtained by backing off the retaining screws uniformly by calculating the pitch of thread in the jackscrews and turning the screws the calculated amount to give 0.018 inch clearance. When the unit was placed into operation, the runout of the shaft was measured by means of a dial
indicator placed at the turbine guide bearing housing and at the generator guide bearing bridge. The maximum runout measured with the indicator was less than 0.0005 inch, which is exceptionally good for this size shaft. The total bearing clearances at the turbine guide bearing were set up as six to eight thousandths of an inch. The temperature indicator for this bearing indicated it was properly aligned, since the temperatures were within the required limits. The total of two thousandths of an inch is allowed for the clearance necessary for the lubricating oil.

The true test of whether the generator bearing thrust shoes were perfectly loaded was the temperature recorded by the recording cells placed in each of the shoes. After the unit was placed into operation, each of the temperatures of the shoes were recorded as equal, indicating equal loading. It would have been very costly to stop the unit and re-adjust the bearings by the "cut and try" method employed by some erectors, and in this case may have delayed the placement of the unit into service.

A large unit recently installed at the Hungry Horse Project sustained bearing failure due to improper adjustment of the thrust bearing, and it is considered that the failure may have been due, in fact, must have been due to the one segment of the thrust bearing being required to sustain the entire load. While this method of shaft alignment and bearing loading was not in its entirety developed at this project, essentially the major principles and procedures were, together with the special tools, which it is believed, assisted greatly in the accurate alignment of the
bearings in the first attempt. This method of alignment was employed in realigning the existing similar unit at the Fort Peck Power Plant, and was found to correct most of the troubles experienced with this unit previously, even though the shaft could not be plumbed in its entirety, due to limitations in the adjustments. Therefore it is concluded that this method of bearing alignment is developed as one of the best methods of establishing true bearing loading and alignments for hydro units of this type.
During the process of the assembly and erection of the 35,000 kW generator, several problems developed, and in order of their occurrence they were as follows:

a. Assembly of the rotor laminations.

b. Installation of the keys binding the rotor spider to the laminations, and the keys binding the field poles to the laminations.

c. Assembly of the thrust collar assembly on the drive shaft.

d. Assembly of the rotor on the thrust collar and the installation of the keys that bind the shaft to the collar.

The difficulties will be discussed in order of their occurrence.

The first indication of trouble in the assembly of the rotor laminations was experienced when an attempt was made to install the long slender 1 inch diameter bolts through the sleeved holes in the laminations which had been stacked on the rotor spider rim. It was discovered that there was a marked difference in the total thickness of the rotor laminations as the rotor was stacked. At first, it was considered that this condition was due to irregularities in the spaces between the laminations and the insertion of the binding bolts was thought to be the means whereby the laminations would be compressed to form a uniform homogeneous mass, but it was found that this was not the case. In the first place, the bolts were, according to the manufacturer's recommendation, to be tightened to a uniform tensile stress not greater than the normal
allowable working strength of the material, which in this case was mild steel, with an allowable stress of 20,000 psi. After calculating the amount of elongation necessary to produce this stress, a gage was made which measured the elongation accurately. The nuts were then tightened down on the bolts until the predetermined elongation was measured with the gage. It was found that after these bolts that were supposed to bind the rotor laminations together were tightened, there still remained considerable difference in the thickness of the stacked rotor laminations. Since the laminations were of carefully stamped stretcher leveled mild steel, and after careful measurements had been made with a micrometer, it appeared as if all laminations were one and the same, no reason could readily be established for the difference in the thickness of the completely stacked rotor. Plate VII is a photograph of the partially stacked rotor. The assembly of the generator was stopped until a reason for this variable thickness could be found, because it was known that too great a difference in rotor rim thickness would affect the balance of the machine. As mentioned before, the laminations were checked and rechecked for any significant dimensional differences without any satisfactory results. The laminations were flat stamped sheet steel 0.078 inch (1/4 gage) thick, approximately 16 inches wide and 5 feet long measured on an arc of approximately 22 feet radius. The total laminations as stacked on the rotor spider weighed 88,425 pounds. The solution to our problem became remarkably simple when in desperation, we compared the weights of various laminations. It was found that although the dimension of the laminations
were quite similar, their weights were variable, as much as 5 ounces. The laminations were then sorted into several piles, using 2 ounce increments as the means of sorting. These resorted laminations were then restacked on the rotor spider, taking care to stack a row of laminations of the same weight until all had been stacked. It was found that the laminations stacked by uniform weight selection gave a very uniform rotor thickness, and too, when the binding bolts were tightened down, all bolts accepted a uniform elongation by the gage without difficulty. Proof that the rotor was properly stacked and balanced became evident during the overspeed test. The vibrations measured during the overspeed tests were of less than normal magnitude. It can be pointed out that while some erection problems are simple in solution, an improper solution can have lasting effects, since in the assembly of previous units of this kind, no concern was given toward uniform lamination "stacking" and as a result, these generators were out of balance. Number 2 unit at this project was in such a state of unbalance when it was tested that it could not be run without excessive vibrations. The balancing of the rotor was accomplished by applying sandbags to the rotor rim and trying various weights in various locations until a condition was obtained where the vibrations occurring during rated speed and overspeed were reduced to a safe level. This balancing process required two weeks, and since no loss of time was incurred for balancing the number three unit, it appears as if the three days time lost while the laminations were sorted and restacked was well occupied. The manufacturer has adopted the procedure of weighing
and sorting the laminations as they are fabricated, and shipping them in marked crates as a result of this experience.

The next problem of importance experienced in the erection of the generator was the installation of the keys that bind or lock the rotor rim laminations to the spider. According to the procedure established by the manufacturer, the rotor pole windings, each weighing about 1,350 pounds, were installed in the dovetail slots in the rotor laminations, and all electrical connections made up to connect the field poles in series. Two heavy electric welders were then connected into the windings and a full output of 800 amperes was passed through the windings for a period of about 48 hours, or until the rotor windings and laminations were up to the temperature of 125 degrees Farenheit. According to the manufacturer's calculations, the temperature of 125 degrees Farenheit would cause the rotor laminations to expand and permit the oversize keys to be driven into the machined slots on the spider and the stamped slots in the laminations. It was soon found that this was not the case, as the laminated rotor rim did not expand according to the laws for a solid member, and the actual expansion was found to be much less than that originally calculated. Therefore, some means had to be found whereby the spider could be shrunk enough to permit the keys to be driven into their slots. "Dry ice" or carbon dioxide in solid form was a practicable means to accomplish this, and 2,300 pounds of it were procured and packed on the spider arms. The keys themselves were packed in this material and finally, when the rotor rim was as warm as it could safely be made by
passing the welding current through the field pole windings and the keys and rotor spider were as cold as the solid carbon dioxide packed around them would cool them, the attempt was made to drive the keys. A heavy jackhammer was used, and would sink the keys full length easily. It was discovered when the temperatures were allowed to equalize that the keys were firmly seized.

In previous machines of this type, and in particular the units at this power plant, difficulty has been experienced in the past of securing the proper fit of the keys that are inserted in the dovetail slots that hold the field poles in place. Usual practice is to file the key until it can be driven into place, and it is hoped that the vibration and temperature changes will not loosen it. It was considered that the successful method experienced in the installation of the keys used to bind the rotor laminations to the spider could be adapted to this assembly.

Accordingly, the keys were shrunk by packing them in "dry ice" and the space in the dovetail slot was made as large as possible by allowing the field poles to cool to the ambient. The keys, oversize by 0.010 inch were driven easily and temperatures allowed to equalize. The keys were effectively seized in their slots, and no loosening of the field poles has been experienced to date.

During the assembly of the thrust collar on the shaft, a slight variation of this method was used. The thrust collar has an inside bore of 30.500 inches with tolerances of plus nothing and minus .001 inch, and has a total length of fitted surface of 18 inches at these dimensions.
Since both materials are similar metal, and are high carbon steel forgings, the normal procedure for forcing the thrust collar onto the shaft by means of heavy jackscrews was considered inefficient and involved some risk, since if there was a slight roughness on either fitted surface, it might cause galling, and the collar might become seized or stuck. The collar was heated to a moderately high temperature of 140 degrees Fahrenheit as calculated to give plenty of clearance, and a lubricant called "Molykote" which consisted of an molybdenum sulphide compound was employed at the fitted surfaces. The collar slipped on in a matter of minutes, saving much time and eliminating the risk of galling mentioned above. If the collar ever has to be removed, it is believed that the lubricant will assist the hydraulic jacks to remove it.

The next difficulty encountered occurred when the assembled rotor was lowered and assembled on the thrust collar shaft assembly. The rotor was intended to be secured to the thrust collar shaft assembly by means of three keys two inches square, and 36 inches long and cut on a diagonal. The slots or key seats in the thrust collar did not match those in the spider hub, although they were cut 120 degrees apart. It was out of the question to recut the seats, so the obvious solution was to alter the keys. Careful measurements were taken with a feeler gage and the keys were cut to fit the offset in the seats. This solution saved many days time, made a satisfactorily tight fit, and there is no danger of the keys loosening in the slots.
TESTING OF THE COMPLETED
WORK, OVERSPEED AND EFFICIENCY TESTS

There were two principle tests which the writer was associated with that were conducted at the completion of the work of installing the unit mentioned in this paper. One test was the overspeed test, and the other was the performance, or turbine efficiency test.

The overspeed test consisted of allowing the unit to operate at runaway speed with the gates blocked open for a minimum period of five minutes. The procedure followed in performing the overspeed test, in general, was as follows:

A sensitive tachometer was calibrated to agree with the known speed of the machine when operating at synchronous speed, no load conditions, which is 128.6 rpm for this machine. The tachometer used was actually an electric voltmeter connected into the governor electric tachometer magneto circuit. The tachometer was calibrated from the known speed and the characteristic curve of the tachometer magneto. The connection of a suitably accurate tachometer to the turbine-generator unit was one of the difficulties of the test. There was no practical way of connecting a tachometer by mechanical means to the turbine shaft. A stroboscope could have been used with very accurate results, but one was not easily obtainable. Therefore the method described above of utilizing the magneto circuit as the most practical, and was employed.

After the tachometer was calibrated and adjusted, the governor control for the unit was then blocked, the machine disconnected from the bus,
and the field current control brought to zero field. The manual gate control was then actuated and the turbine wicket gates opened to 100 percent, or the maximum opening, and held there for five minutes after the unit reached its maximum speed of 225 rpm which was 175 percent of the rated speed. During this time, the action of the governor overspeed trip was observed to determine that it functioned to trip at 130 percent of the rated speed. At the conclusion of the test the trip was checked to determine that it reset at 105 percent of the rated speed. The gross operating head on the unit at the time of the test was 185.9 feet. According to Marks' Handbook\(^1\), and the practices of hydraulic turbine manufacturers, the observed overspeed was within the acceptable range for this type of Francis turbine. Since the overspeed is related to the specific speed of the turbine, the specific speed for this unit at design conditions is calculated as follows:

The general formula for specific speed is \( n_s = n (P)^{1/2} / H^{5/4} \) where \( n_s \) is the specific speed, \( n \) is the rpm of the unit, \( P \) is the rated horsepower of the unit, and \( H \) is the net effective head in feet. For the unit considered herein, the rpm of the unit at synchronous speed is 128.6, the rated horsepower of the unit is 50,000, and the net effective design head is 170 feet. Applying this data to the formula, \( n_s = \frac{(128.6)(225.6)}{613.9} \) or 47.

As was stated earlier in this paper, the characteristics of the

machine regarding balance and stability were observed during this test, and it was noted that the machine ran exceptionally smooth during the overspeed condition, and was quite stable, as no "bumps" or other heavy vibrations were observed.

It was concluded as a result of the overspeed test that the characteristics of the machine were within specification requirements, and insofar as that particular phase of the testing was concerned, the machine was acceptable under the terms of the contract.

The turbine efficiency test was performed chiefly for the purpose of verifying the contractor's guaranteed efficiency for the turbine in lieu of performing an official acceptance test, and to verify the efficiency curve plotted as a result of the model test for the subject turbine. Actually, an acceptance test could not be conducted in accordance with the A.S.H.E. Power Test Codes, because the accurate measurement of the quantity of water passing the turbine gates by any of the accepted methods would be difficult and expensive in this plant, as the main penstock supplying water to the three units is approximately 5200 feet in length and is not accessible at the intake, or any intermediate point, until it terminates in a trifurcating joint, from which three separate penstocks originate and connect the individual turbines to the main penstock. As stated previously, these individual penstocks connect by means of a riser into three separate surge tanks, and the surge tanks are in turn interconnected by pairs of 5 foot diameter interconnecting tubes located at two levels. In addition, the main penstock is equipped
with a surge tank located near the intake structure. Between the intake structure which drafts water from the reservoir, and the turbine wicket gates, are located a pair of emergency tractor type vertical lift gates, a transition section, a surge tank or shaft, a trifurcating head-block, a riser connection to another surge tank which serves an individual unit, and a butterfly valve with suitable transition sections, all of which are connected by penstocks and form the water carrying system. Thus it can be noted that water measurements of one individual turbine connected into this system would be difficult. Coupled with these difficulties, was the problem that load conditions and power commitments were such that the operating conditions specified in the A.S.M.E. Power Test Code could not be maintained within the prescribed limits. Therefore, it was necessary that an adaptation of the Index Testing Method be used in testing the efficiency of the unit.

The general procedure of conducting the test was as stated below:

a. Basic Assumption. In order to obtain the necessary turbine water discharge quantities, it was assumed that the calibration of the Winter-Kennedy manometer tapes in the scroll case of Turbine No. 3 was the same as that for those in the scroll case of Turbine No. 1 since


both turbines have identical scroll cases with the Winter-Kennedy taps located at the same points. From previous water flow measurements for the calibration of the Winter-Kennedy taps in the scroll case of Unit No. 1, operating singly, the following calibration was obtained:

\[ Q = 1043d^{1/2} \]

where \( Q \) equals turbine discharge in cubic feet per second, and \( d \) equals the manometer deflection in inches of mercury, when connected between the high-pressure and inner-low-pressure Winter-Kennedy taps in the turbine scroll case.

b. **Measurements of Quantity of Water.** Readings were taken during each test on the mercury manometers connected to the Winter-Kennedy taps in the turbine scroll case, from which the quantity of water flowing through the turbine under test was computed from the relationship above.

c. **Measurement of Power Output.** The generator output was measured by the two-element, three-phase, watt-hour meter in each generator circuit. An integral number of revolutions of the watt-hour meter disc were timed with a stop watch and the generator output computed from the disc constant and current and potential transformer ratios. As a check on these values the output of the generator was also measured by means of three calibrated indicating wattmeters. The load currents and field current for the generator were also measured in order that the generator losses could be more accurately computed.

d. **Measurement of Head.** The pressure head at the intake of each turbine scroll case was measured by means of Bourdon gages located in the turbine pits at elevation 2052.0 and connected to piezometer taps located
just upstream from each turbine intake. Calibration corrections for these gages were obtained by closing the wicket gates on all units and checking the gage readings against the surge tank and reservoir water elevations. The tailrace water elevations for each test were taken from the tailrace water level recorder in the control room. This recorder indicates the water level in a stilling well which is connected to the tailrace at a point adjacent to the draft tube discharge of Turbine No. 1.

e. Computation of Power Output. The power output was computed from the relationship:

\[
\text{Turbine Output (HP)} = \frac{\text{Generator Output (KW)}}{0.746 \times \text{Generator Efficiency}}
\]

The generator manufacturer's guaranteed efficiencies were used in computing the turbine output for all tests, since the results of the generator efficiency tests were not available when these tests were made.

f. Computation of Turbine Net Head. From the corrected pressure head at the entrance to the turbine scroll case as measured by the Bourdon gage in the turbine pit and from the elevation of the tailrace water level, the net head on the turbine was computed using the relationship

\[
h = 2052.0 + h_p + \frac{V^2}{2g} - \text{(tailrace water elevation + } \frac{V^2}{2g})
\]

where:

- \( h \) equals net head on turbine in feet of water.
- \( h_p \) equals corrected pressure head in feet of water measured by the turbine pit gage at elevation 2052.0.
- \( V \) equals penstock velocity of the location of the turbine.
intake piezometer taps and is equivalent to \( q/A \) where \( A \) equals the cross-sectional area of the penstock at the piezometer taps and is 160 square feet.

\( V_r \) equals the residual tailrace velocity at the section where the tailrace water elevation was measured. (The value of \( V_r \) used was computed from the total plant load, assuming uniform flow in the tailrace as a whole, using approximate turbine and generator efficiencies, by the equation \( V_r = \frac{\text{Total Plant Net Output (Megawatts)}}{\text{Head}} \times 5.85. \) This was a very minor correction as the maximum values were about 0.3 feet.)

\( g. \) **Computation of Turbine Efficiency.** The turbine efficiency was computed from the equation:

\[
\text{Turbine Efficiency} = \frac{550 P_e}{Qh},
\]

where \( P_e \) is the turbine output in H. P., \( Q \) is the quantity of water used by the turbine in cubic feet per second, \( w \) is the weight of water per cubic foot and is 62.37, and \( h \) is the effective net head in feet.

The turbine discharge and output horsepower for each test were reduced to a constant effective head of 170 feet by the assumed relationships \( Q_{170} = \left(\frac{170}{h}\right)^{1/2} Q \), and \( P_{170} = P_e \left(\frac{170}{h}\right)^{3/2} \). These relationships are not strictly true unless the speed of the units is adjusted during the test to equal rated speed times \( \left(\frac{h}{170}\right)^{1/2} \), which was impractical, since the unit was in power production and operating at synchronous speed.
The values of $Q_{170}$ and $P_{170}$ were plotted separately and smooth curves drawn representing the mean of the observations. From these two curves the efficiency of the unit was plotted and compared with the model curve established by the manufacturer's model test of the unit. Refer to Plate VIII for the resultant curve. It was found they compared favorably, and it was therefore concluded that formal acceptance tests for the unit would be unnecessary when the expense of conducting the tests versus the value of the information received was compared.

In summary, it can be stated that while the problems, and their solutions discussed herein pertain to a particular unit in a particular hydroelectric plant and were peculiar only to this plant, the principles and methods used to solve these difficulties can be applied to similar installations of heavy hydroelectric equipment and are therefore believed to be suitable experiences to offer to the field of Mechanical Engineering.
PLATE VIII - EFFICIENCY CURVE OF TURBINE

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170 ft. net head efficiency curve computed from Model tests.

Efficiency at rated net head of 170 ft. from field tests.

TEST POINTS

TURBINE OUTPUT - HORSEPOWER

0 10,000 20,000 30,000 40,000 50,000 60,000
LITERATURE CITED AND CONSULTED

American Institute of Steel Construction, 1941, STEEL CONSTRUCTION, PP. 337-338, American Institute of Steel Construction, New York, N. Y.


