Tests on a 75 K. W. Terry Steam Turbine.

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THESIS

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OBJECT.

In this thesis, we propose to obtain the conditions existing in the turbine under different loads and nozzle openings. These conditions include the heat drops between the different nozzles and blades, the velocity of steam in the different stages and the quality of the steam as it passes through the turbine. The steam consumption will be determined for thirty different combinations of loading and nozzles in use so as to determine the most economical conditions for different loads.

HISTORICAL

The following sketches of the development of the steam turbine are extracts from current technical magazines and text books, much of the material being obtained from Moyer's "Steam Turbines" and "The Design and Construction of Steam Turbines" by Martin.

Although turbines have come into common use only within the last decade, their use for experimental purposes has been found to reach back over a period of two thousand years.

In a book written by Hero, about one hundred and fifty B. C., there is shown a sketch of a revolving hollow metal ball, into which steam enters through a trunion from a boiler beneath and escapes tangentially from the outer rim through two arms bent backward so that the reaction causes the ball to rotate in a direction opposite that of the escaping steam. This wheel is the prototype of the reaction turbine.

Perhaps the steam turbine of the simplest type is a wheel, similar to a water wheel, which is moved by a jet of steam impinging at high velocity on its blades. Such a wheel was designed by Branca an Italian, in 1629. This type of turbine, similar to a Belton water wheel, is known as the impulse type.

In most modern steam turbines both effects are used, jets of steam striking blades or buckets inserted in the rim of the wheel so as to give it a forward impulse and escaping from it in a reverse direction so as to react upon it. The name impulse wheel, however, is now generally given to one in which the pressure on both sides of the wheel containing the blades in the same, and the name reaction wheel to one in which the steam decreases in pressure while passing through the blades.

In 1577 the cause of the steam turbine received its next beneficial consideration. In that year a German mechanic is said to have used Hero's engine to rotate a mechanism for a turnspit.

In 1642, shortly after the Italian architect Branca, had made an advancement in the steam trubind field, a Jesuit named Kircher used Branca's wheel with two jets of vapor acting instead of one.

Over a hundred years later, in 1784, Wolfgang de Kempelen was granted a British patent for "Obtaining and Transmitting Motive Power" by means of a steam turbine. This turbine was composed of a steam boiler with a small horizontal cylinger mounted, at the center of its length, on a bearing on top. Near each end of the cylinder and on opposite sides were small apertures through which steam could escape when the turbine was in operation. The reaction on the escaping steam on the air caused the cylinder to rotate. One half of the rotating cylinder acted as a crank and moved a pump like mechanism through a connecting rod and lever arrangement.

In the same year James Watt secured patents on an experimental steam trubine which was enclosed in a vessel.

Soon afterward, in 1791, James Sadler, an engineer of the city of Oxford was granted a patent for an invention entitled, "An engine for lessening the consumption of steam and fuel in steam or fire engines and gaining a considerable effect in time and force". This turbine embodied many up-to-date principles.

Noble produced a wheel shortly afterward that was driven in somewhat the same manner as a Pelton water wheel.

Other turbines followed, advancement in their development being made by Ericson, Perkins, Pilbrow and others, Pilbrows machine involving the first principles of successive expansion stages. This principle was later developed by Robert Wilson in 1898, first in radical-flow, then in parallel-flow turbines.

From this time on rapid advancement was made in the perfection of the economical steam turbine, especially of the Parsons type, until today we have turbines of several pressure stages and many velocity stages and in capacities up to eighty thousands of kilowatts.

THEORY

Many engineers and authors have, at various times, advanced treatises on the theory of Steam Turbine operation, all of which are of some importance in drawing a consistent conclusion.

In discussing the Mechanical Theory, elemental examples will be considered the more complex being developed from these.

In an impulse turbine with a single disk containing blades at its rim, steam at high pressure enters the smaller end of a diverging nozzle and as it passes through the nozzle, is expanded adiabatically down to the pressure of the atmosphere in a non-condensing turbine, or to the pressure of the vacuum if the turbine exhausts into a condenser. The steam thus expanded has its volume and its velocity enormously increased, its potential energy being converted into

kinetic energy. It then strikes tangentically the concave surface of the curved blades and thus drives the wheel forward. In passing through the blades it has its direction reversed, and the reaction of the escaping steam also helps drive the wheel forward.

If it were possible for the direction of the jet to be completely reversed through 180° and if the velocity of the blades in the direction of the entering jet was one half the velocity of the jet, then all the kinetic energy due to the velocity of the jet would be converted into work on the blades and the velocity of the steam with reference to the earth would be zero. This complete reversal, however, is impossible since room has to be allowed between the blades for the passage of the steam and therefore the blades are curved through an arc considerably less than 180° so that the jet, on leaving the wheel, still has some kinetic energy which is lost. The velocity of the entering steam jet is also so great, that it is not practicable to give the wheel rim a velocity equal to one half that of the jet which is beyond a safe speed. The speed of the wheel being less than half that of the entering jet also causes the steam to leave the wheel with some of its energy unutilized.

The mechanical efficiency of the wheel, neglecting internal losses, is expressed by $E1 - E_2/E_1$, in which E_1 is the kinetic energy of the steam jet impinging on the wheel and E_2 that of the steam as it leaves the blades.

In multi-stage impulse turbines, the high velocity of the wheel is reduced by causing the steam to pass through two or more rows of blades, which rows are separated by stationary curved blades which direct the steam from the outlet of one row to the inlet of the next.

The wheel with two rows of movable blades runs at one half the velocity of the single stage turbine, and one with three rows at one third the velocity which causes the same total reduction in velocity of steam as the single stage wheel. A greater reduction in velocity of the wheel can be obtained by increasing the number of rows of blades. It is therefore possible, by having a sufficient number of rows of blades or velocity stages, to run a turbine at comparatively low speed and yet have the steam escape from the last set of blades at a lower absolute velocity than is possible with a single turbine.

In reaction turbines, the expansion of the steam does not take place in diverging nozzles but also in stationary expanding guide vanes and the reduction of pressure with its conversion into kinetic energy, or energy of velocity, takes place in the blades which are made of such shape as to allow the steam to expand while passing through them. The stationary blades also allow of expansion in volume, thus taking the places of nozzles.

In all turbines, whether of the impulse or reaction type or a combination of the two, the object is to admit steam at high pressure and discharge it into the atmosphere or into

a condenser at the lowest pressure and largest volume possible, and with the lowest possible absolute velocity, or velocity with reference to the earth, consistent with getting the steam away from the wheel with the least loss of energy in the wheel due to friction of the steam through the passages, to shock due to incorrect shape or position of the blades, to windage or to frictional resistance of the steam in contact with the wheel or to other causes.

The minimizing of these several losses is a problem of extreme difficulty which is being solved by costly experiments and experience.

APPARATUS

The 75 K. W. Terry Steam Turbine used in this thesis is of the latest type manufactured by the Terry Steam Turbine Company of Hartford, Conn. The machine was designed to run at 2400 revolutions per minute on 1100 pounds gage pressure and twenty inches of mercury vacuum. A good idea of the appearance of the machine can be obtained from the photographs which appear in this book. The guaranteed water rate of the turbine and the efficiency of the generator can be obtained from the drawing on page 30. The details of the nozzles and blades are shown on the drawing on page 29. The connection of the apparatus is shown on the drawing on page 28.

The turbine has five stages which operates on both impulse and reaction principles. Five diverging nozzles of

circular cross section are distributed around the periphery of the first high pressure stage. The nozzles make an angle with the disc of about 19 degrees. All nozzles but one are fitted with valves so that any number from one to five may be open at one time. The stationary reversing blades for the second high pressure stage do not continue all around the disc but only for intervals of about five inches in front of each nozzle.

After leaving the second high pressure stage, the steam is carried to the generator end of the turbine and alternately passes through three sets of stationary and moving low pressure blades. The mean diameter of both high and low pressure stages is about 26 1/2 inches. The steam main diameter is 4 inches and the exhaust main 10 inches.

The turbine is built with a throttle govenor which operates from enclosed flyballs located directly on the turbine shaft. An overspeed govenor is operated by an entirely different mechanism which trips a butterfly valve in the steam main and also breaks the vacuum. This govenor is set to operate when the speed exceeds four percent of the normal running speed.

The turbine is fitted with tubes for obtaining the pressure, temperatures and qualities at all points shown on the apparatus diagram. Those used were: P_0 throttle main pressure; P_1 throttle pressure; C_1 throttle quality; P_2 interstage pressure; T_2 interstage temperature; C_2 inter-

stage quality; P_3 third stage stationary exit pressure; C_3 third stage stationary lost quality; P_5 fourth stage stationary exit pressure; P_7 fifth stage stationary exit pressure. P_2, P_3, P_5, P_7 were obtained by mercury manometers.

The turbine is fitted with labyrinth seal glands and water was supplied from a tank located on an Atlas scales.

The turbine is direct ly connected to an Allis-Chalmers 220 volt, D. C. generator of 75 K. W. capacity. The field of the generator was regulated by a switchboard rheostat so as to keep the voltage constant. The load was secured by an adjustable water rheostat with running water. Weston instruments were used to measure the voltage and amperage generated.

A Schaeffer and Budenberg tachometer was used to determine the speed of the macine. It was belted to the generator shaft and had a constant of three, that is, the readings were multiplied by three to get the revolutions per minute.

A wheeler surface condenser was used having a cooling surface of 298 square feet approximately. The cooling water was supplied by a Worthington Duplex pump and discharged into an orifice tank having a gauge glass and four orifice holes of .01 square foot area each. The number of holes in service could be varied from one to four.

The condensate was removed by a Wheeler single acting pump and was discharged into either of two tanks locat-

ed on Atlas scales by a deflecting pipe. These tanks were provided with large valves so that they could be emptied rapidly.

Thermometers were located as follows: T_a inlet cooling pipe; T_b outlet cooling pipe; T_c condensate pipe; Td seal gland tank; T_2 interstage; C_2 interstage calorimeter; C_3 third stage stationary exit calorimeter. A separating calorimeter was placed at P1 and throttling calorimeters at P_2 and P_3.

PRELIMINARY TESTS.

In making preliminary tests on the turbine and the apparatus, many difficulties were encountered which required changing the original plans.

The Wheeler surface condenser was tested for leaks. This was done by removing the cooling water discharge pipe and bolting a flange to the discharge opening in which was screwed a four foot length of pipe. The circulating pump was started and enough water run in to fill the stand pipe so formed. The pump was then stopped and the intake valve closed tight. The air pump was then started and after an hours running, the water failed to recede which showed that the condenser had no internal leaks.

The seal gland tank was placed on a scales which gave the water in the tank a two foot head. A test showed this to be sufficient as the water was acted on by the suction of the glands and just enough head was necessary to carry it to the glands.

The turbine was started and the manometer tubes tested for range. The mercury columns were all well within the limits of the tubes. The two separating calorimeters at P_2 and P_3 were observed to fill up quite rapidly. The seal gland tank emptied very rapidly on no load which showed that the seal gland water would alter the readings for total steam consumption materially. The two throttling calorimeters at P_0 and P_1 read 213^oF which indicated very wet steam, too wet for throttling camotimeters. Two loads were applied, one of 50 amperes and the other of 400 amperes. The calorimeter at P_2 failed to separate on the heavy load and it was concluded that the steam was either superheated or the calorimeter was not of order. All readings were taken for these two loads.

A separator was inserted in the steam main. This was placed directly below the throttle valve. The calorimeter piping was logged with cloth and the exhaust piping and condenser were logged with magnesia covering.

The first manometer tube had collected condensate. This was removed by drawing off the mercury and drying it with a cloth. The separating calorimeter at P_3 was removed and placed at P_0. A sampling tube was placed in the main. The calorimeter showed a quality of 94 percent which showed that a separating calorimeter would be necessary. The separating calorimeter at P2 was placed at P_0 which indicated a quality of 71 percent. The instrument was removed, examined and replaced. Another test gave similar results, so

it was decided not to use this instrument until thoroughly examined. The low pressure side of the three calorimeters at P_1, P_2 and P_3 were then piped to the condenser.

The diameter of the separating calorimeters orifices were determined on a dividing engine at the Physics Department. The thermometers were calibrated at the same department. The cooling water inlet thermometer was calibrated at 40°F, the outlet thermometers at 85°F, the condensate thermometer at 100°F and the two calorimeter thermometers at 225°F. They were all calibrated by means of a standard, the lower temperatures being obtained with water and the higher one with oil. The thermometer connections varied from C°F to 3°F. An extra thermometer was calibrated for 100°F and 225°F.

A complete series of readings at five minute intervals were taken for three loads of 100, 200, and 300 amperes. Five nozzles were used and each test was conducted for twenty minutes. It was found practical to take readings every five minutes and at intervals of fifteen minutes between tests showed consistent results for the following readings. From these results, data sheets and plans were made for the final tests.

FINAL TESTS.

The procedure for the thirty hours of final tests was as follows:-

Half an hour before the test began, the condenser and circulating pumps were started and steam was by passed to the turbine to warm it up. Not enough steam was admitted, however, to cause the motor to turn.

Twenty minutes before starting the test, the machine was slowly brought up to speed and the overspeed govenor tried. This was done by forcing the throttle open by hand and noting the maximum reading on the tachometer. The steam main valve was then closed, the butterfly and vacuum valves reset and the machine slowly brought back to speed.

The load to be carried by the macine for the test was then applied. The thermometers were placed in their respective positions. The seal gland tank was filled and the gland water regulated.

The man controling the condensate tanks was in charge of test. At the exact instant of each five minute period, he blew a whistle and took the following readings in the order named, the numbers referring to the blue print page 31 , 21, 11, 4, 3, 19, 20, 12, 13, 14, 25, 26, 27.

At the same instant the second man took the following readings in the order named, 7, 6, 5, 8, 9, 10.

At the same time the third man made the following observations: 22, 23, 24, 16, 18, 17, 15.

Each test was run for a period of 45 minutes and 15 minutes were allowed between tests. Loads were used as found on page 27 . The daily barometer readings are also given on the same page.

The following tests were made on the days specified.

April	Nozzles	Load
1	5	0, 1/4
2	5	1/2.3/4, 1, 400A.
3	4	0, 1/4, 1/2, 3/4
4	4	1, 1-1/4
4	3	0, 1/4
5	3	1/2, 3/4, 1, 1-1/4
8	2	0, 1/4
9	2	1/2, 3/4, 1, 1-1/4
16	1	0, 1/4, 1/2, 3/4
20	1	1, 1-1/4

The steam main pressure varied considerably and several tests were discontinued until better pressures were obtained.

COMPUTATIONS.

The data on pages 31, 32, 33, 34, 35 were obtained from averaging the results for each 45 minute test. The complete results from computations are also given on these pages. These results were obtained as follows:

Computation of Quality at C_1.

The results of computations for quality at C_1 were not used as the qualities obtained were unreasonable and it was concluded that the calorimeter was defective. The formu-

la used was:

$$q = \frac{---W}{u + W}$$

q = quality of steam.
u = weight of dry steam discharged.
W = weight of moisture collected.

To find W, Napiers Rule was used as all high pressures were greater than 1.73 times the low pressures.

$$W = \frac{PA}{70}$$

W = pounds of steam passing through the orifice per second.
P = the absolute pressure in pounds per sq. inch.
A = the area of the orifice in square inches.

For five nozzles, full load.

$$W = \frac{PA}{70} = \frac{58.13 \times .00385}{70} = .0032 \text{ lbs. per second.}$$

Number of seconds to collect .2 pounds of moisture = 123.5

Pounds of steam through orifice in 123.5 seconds = 123.5 x .0032 = .395 pounds.

Quality of steam at C_1

$$q = \frac{W}{W+W} = \frac{.395}{.395 + .2} = .664 = 66.4\%$$

This quality is absurd because the steam would be dryer than at the main as it contains the same heat at a lower pressure.

Computation of Quality at C_2 and C_3.

The following equation for total heat of steam was used:

$$q + xr = H$$

q = Heat of the liquid of steam at the pressure existing in the steam main.

r = Heat of evaporization of steam at the pressure existing in the steam main.

x = the part of the steam that is in the form of vapor.

H = Total heat of steam at the temperature and pressure existing in the calorimeter.

The values of q and r are taken from Marks and Davis Saturated Steam Tables.

A sample computation of Q_2 at one half load with four nozzles open is given.

Calorimeter pressure = 4.48 pounds per square inch absolute.

Calorimater temperature = 172.3° F.

From Marks and Davis Steam Tables.

$$H = 1133.2 \text{ B.T.U.}$$
$$q = 163.0 \text{ B.T.U.}$$
$$r = 980.5 \text{ B.T.U.}$$

$$x = \frac{H - q}{r}$$

$$\frac{1133.2 - 163.0}{980.5} = \frac{970.2}{980.5} = .990 = 99.0\ \%\ \text{quality.}$$

Computation for Quality at C_8

The amount of cooling water used was obtained from the formula:

$$Q = .61A\sqrt{2GH} \times 60 = 2.935\sqrt{H}$$

A = area of circular opening in square feet = .01

H = head of water in feet.

g = Acceleration of gravity = 32.2

Q = Cubic feet per minute.

For the same conditions as stated above,

$Q = 2.935\sqrt{3} = 5.08$ cubic feet per minute per nozzle.

$5.08 \times 60 \times 3 \times 62.4 = 57200$ pounds of circulating water per hour.

Temperature raise of cooling water = $91.8^\circ - 48^\circ = 43.8^\circ$F.

B.T.U. per hour absorbed by cooling water = 57200×43.8 = 2, 505,000 B.T.U.

Temperature of condensate above 32.2 = 102.1 - 32 = 70.1°F.

Weight of condensate per hour = 2920 pounds.

B.T.U. of condensate above 32° F = 70.1×2920 = 204,900 B.T.U.

Weight of seal gland water used per hour = 291 pounds.

Temperature of gland water above 32° F = $57.6^\circ - 32^\circ = 25.6$

Total B.T.U. per hour above 32° F in steam = 2,505,000 plus 204,900 - 8272 = 2,701,180 B. T. U.

Weight of steam used per hour = 2630 pounds.

Total heat per pound of steam = $\frac{2,701,180}{2630}$ = 1027 B.T.U.

Quality of steam at the condenser

$$X = \frac{H - q}{r} = \frac{1027 - 131.9}{999.3} = .896 = 89.6\%$$

It has been stated that theoretically the fewer impulse nozzles open, the lower the steam consumption for a given load up to the capacity of the nozzles. This is due to the fact that the heat drop is greater with fewer nozzles open on account of a higher initial nozzle pressure and constant exhaust pressure, thus getting more heat out of each pound of steam. This theory was verified in this thesis in every case accepted that under loads on five nozzles the steam consumption was lower than for four nozzles, but this was probably due to the difference in exhaust pressure which the condenser maintained.

Bibliography.

In 1913 Longmans Greenand Co. of London published a book on the "Design and Construction of Steam Turbines" by H. M. Martin. This book is one of the best of its kind and deals with all details concerned in designing and constructing steam turbines of various types, both condensing and non-condensing. Bearing and governing problems are considered and methods of their design discussed. The volume also contains many drawings and diagrams with tables useful to the steam turbine engineer.

In 1911, McGraw Hill Co., published a book, "Power Plant Testing" by Moyer. This book describes various methods of testing turbines as well as giving an analysis of their points of advantage. He sums up the losses in turbines as

follows: Nozzle losses, rotation or windage losses due to friction of the steam on the blades, bearing kinetic energy in the steam leaving the turbine, and electrical output.

In 1917 John Wiley and Sons published "Steam Trubines" by Moyer which covers the turbine field quite thoroughly, going into considerable detail as to construction and testing. In this book some striking contrasts are brought out relative to comparative operation of machines under different conditions. There the statement is made that under comparative tests a steam turbine should give the following results: for a 100 # change in admission pressure the water rate changes 5# K.W. per hour, for 100° F change in superheat the water rate changes 2# per K.W. per hour, and for each inch of mercury change in exhaust pressure a change of 1# per K.W. per hour in the water rate should be effected. These corrections ofcourse, should be applied in the right direction as will be seen with a little thought. Here also, the thermodynamic principles are treated at some length and their practical application is shown. The design of nozzles, guide vanes and moving blades for impulse and reaction trubines are treated separately and problems of design are solved for each case. The velocities of the steam are determined for the different stages and a discussion and solution of the theoretical quality of the steam in different stages is given in much detail. Many commercial types of turbines are dissected and discussed at length and compared in their different details. The treatments are very clear and self-explaining and very up-to-date.

In 1902 Longmans Green and Co. published "Steam Turbines" by R. M. Neilson, which outlines the history and development of the steam turbine from its earliest experimental stage. He gives credit to many foreigners for the perfection of the present day machine saying that the Englishman Pilbrow devised the first turbine involving successive expansion stages. This principle was later developed by Robert Wilson in 1849. This work describes the development of the Parsons Turbine and compares steam turbines to the other motors and engines as to their relative merits. The questions of vanes and velocities, thermodynamic considerations, and nozzles are briefly discussed. Cuts of a number of De Laval, Parsons, and Rateau turbines are given. A short discussion of the application of turbines to marine work is also included.

In 1917 John Wiley and Sons published "Mechanical Equipment of Buildings" Vol II by Harding and Willard, in which the authors give comparison between the steam turbine and the reciprocating engine and show in what details the turbine excels in medium and large sizes with the use of superheated steam and low exhaust pressures. It deals with the factors governing economy and gives curves for the extimation of consumption for any size of unit under almost any condition. This book goes into the elementary theory of the turbine, the features of construction of the De Laval turbine, impulse turbines with velocity stages, velocity diagrams, the use of the Mollier diagram for steam, impulse and

reaction turbines and their principles. Some data is also given on low pressure and mixed pressure turbines and gives tables and data on many sizes and makes of turbines. Here it is stated that a turbine is most economical under high vacuum and that under a vacuum of 24" - 26" is only as economical as a high grade reciprocating engine. The heat drop in different stages is discussed both in the theoretical and practical cases and velocity diagrams are shown of the steam in different stages. The difference between impulse and reaction turbines is made clear and it shows how the economy of a plant can be increased by the use of reciprocating engines of the non-condensing type and low pressure turbines with high vacuum.

In 1914 McGraw Hill Co. published "Heat Engines" by Allen and Bursley in which some of the elementary principles of the steam turbine are discussed, distinguishing clearly between impulse and reaction turbines. They dwell at some length on nozzle design and shape the subjects of action of steam in the blades, and nozzles, speed of blades and velocity of jets are discussed briefly. Many commercial types of machines are described and compared in their salient features.

In 1914 Henry Holt and Co. published "Thermodynamics" by Goodenough in which all the thermodynamic principles of steam turbines are discussed. Mention is made of the fact that the loss of efficiency due to energy transformation in a turbine is a considerable factor. This book points out the differences in the thermal problem of reciprocating engines and steam turbines. There is developed a formula for the work

of a jet and a complete solution for a single stage velocity turbine. The multiple turbine and the multiple pressure turbine are also discussed at length in their thermodynamic principle.

Many good facts concerning steam turbines may be found in Kent's "Mechanical Engineers Handbook" published by John Wiley in 1914. The work covered in this reference describes briefly the different types and makes of steam turbines, the mechanical and heat theory of the steam turbine, methods of calculating the velocity of the steam in the nozzles and the speed of the blades, losses in the turbine, factors governing efficiency, results of tests on large machines, and reduction gears for lowering speed. Kent also states that the mechanical efficiency of a turbine is the difference between the kinetic energy of the jet of steam impringing on the wheel and the kinetic energy of the steam leaving the blades, divided by the former factor. Some data is also given on design of turbines, nozzles, blades, allowances for windage, friction, blade leakage, rotation losses, and residual velocity losses.

In the A.S.M.E. Journal vol. 39 No. 12 for Dec. 1917, an article appeared by M.J.A. London, a member of the society. The article was devoted to a discussion of various commercial types of turbines and their status in power plants in general. The author advances many new ideas concerning the commercial field, perhaps the most interesting being that practically every builder has resorted to the composite type, that

is a multi-velocity staging in the low pressure end.

In the A.S.M.E. Journal vol. 38 No. 11 for Nov. 1916 an article appeared by Samuel Insull. It is another analysis of the commercial problems of power plants. The author advances the idea that the maximum practical capacity for turbines has been reached and laments the building of more machines of 60,000 and 70,000 K. W. capacity.

In the reprint No. 167 of bulletin vol. 7, No. 4 of the Bureau of Stds. was published an article on the steam turbine in which there is a good discussion on the calculation of the reheat factor and the form of the expansion line on the Mollier diagram.

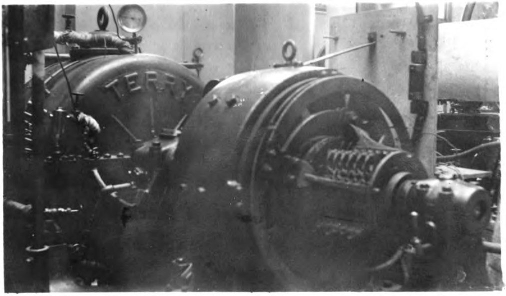
In 1912 the Van Nostrant Co. published "The Design and Construction of Steam Turbines" by H.M. Martin in which is given a short historical sketch of the steam turbine and its development, some valuable data on the reheat factor, a chapter on the design of nozzles, guide vanes and rotors. A problem of designing an impulse turbine is solved. In this book also there is given a chapter on high speed bearings in which it is stated that in well lubricated high speed bearings the jownal never comes in contact with the brasses so that there is practically no wear on either jownal or brasses. There is also a good discussion and description of numerous turbines of commercial importance. The marine turbine is also given some consideration.

In Mechanical Engineers Handbook by L. S. Marks the steam turbine is discussed by L. C. Lowenstein. He first

classifies turbines and tells how to recognize each class, discusses the application of steam turbines, the utilization of vacuum, the effect of superheat, steam consumption, flow of steam through nozzles, bucket velocity and coefficients, and bearing and rotation losses. The details of impulse turbines, nozzles, rotors, packings and bearings are considered. Reaction turbines are considered as to degree of reaction, grouping of stages, leakage losses, and thrust and balancing pistons. Problems of the design of an impulse and reaction turbine are worked out at length. Low pressure mixed pressure and extraction turbines are all described. The adaptation of turbines to the propulsion of ships is considered and the series arrangement, velocity stages, gear reductions, hydraulic transmission, reversing turbines and combination drives are all discussed. The section closes with general turbine data on foundations, expansion joints, and water rates.

In 1900 the McGraw Hill Co. published "High Speed Steam Engines" by W. Norris and B. H. Morgan in which is given a short discussion of early Parsons and De Laval turbines but not much that is of practical advantage today except historically.

In 1907 the Van Nostrant Co. published "The Steam Engine" by Heck in which there is a discussion of the action of steam in various types of turbines and a chapter is devoted to the theory of the design of nozzles and blades in different types. There is also a good discussion of the principles and practice used in the governing of turbines and also of bearing packing.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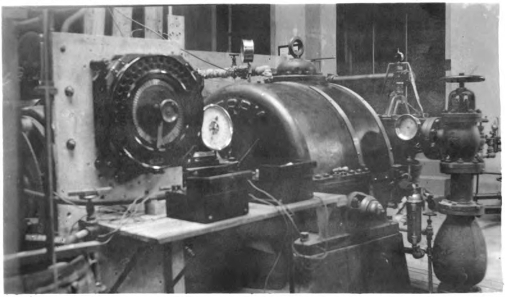


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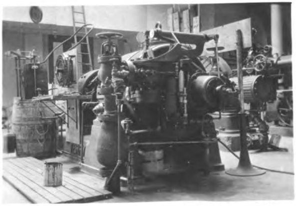

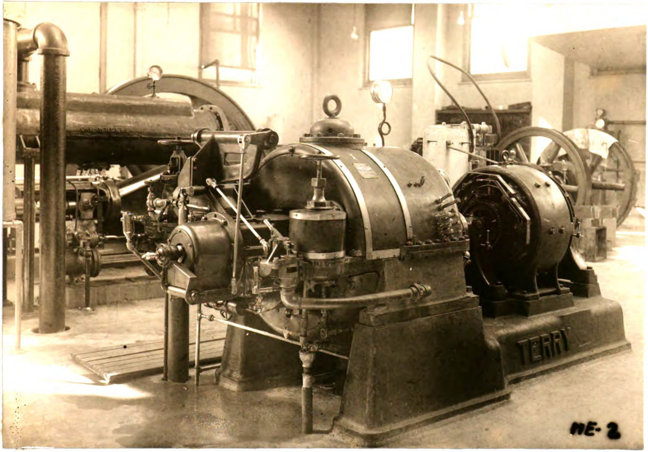
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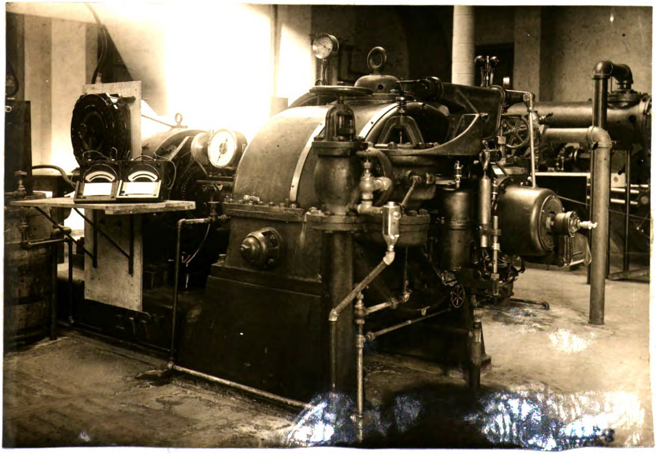

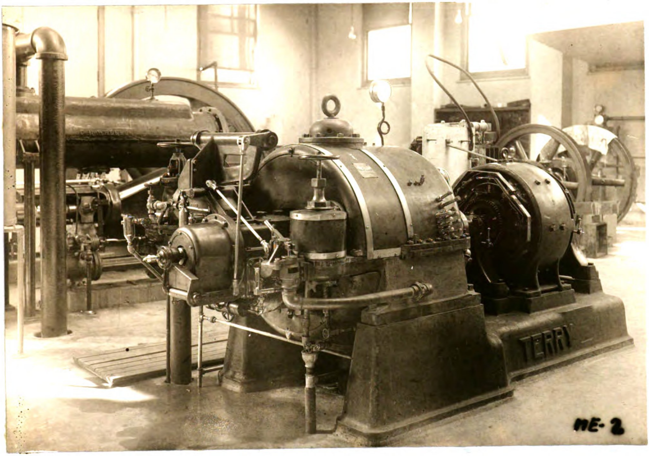
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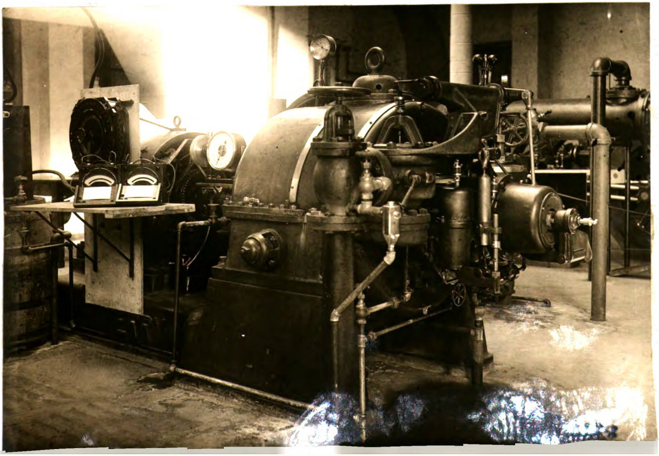
ME-2



LOADS in AMPERES

based on

FULL LOAD – 5 NOZZLES – 340 AMPERES

$340 \times 220 = 74.8$ K.W.

Nozzles \ Load	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{4}$
1	17	34	51	68	85
2	34	78	112	146	180
3	51	102	153	204	255
4	68	136	204	272	340
5	85	170	255	340	425 (400 USED)

BAROMETER

U.S.W.B. E.LANSING, MICH.

3:00 P.M.

DATE	inches-hg.	lbs. per in.^{2}
APRIL 1	28.55	14.03
" 2	28.94	14.18
" 3	29.12	14.30
" 4	29.32	14.40
" 5	29.43	14.46
" 8	29.50	14.50
" 9	29.53	14.51
" 16	28.90	14.20
" 20	28.02	14.25

THERMOMETER READINGS

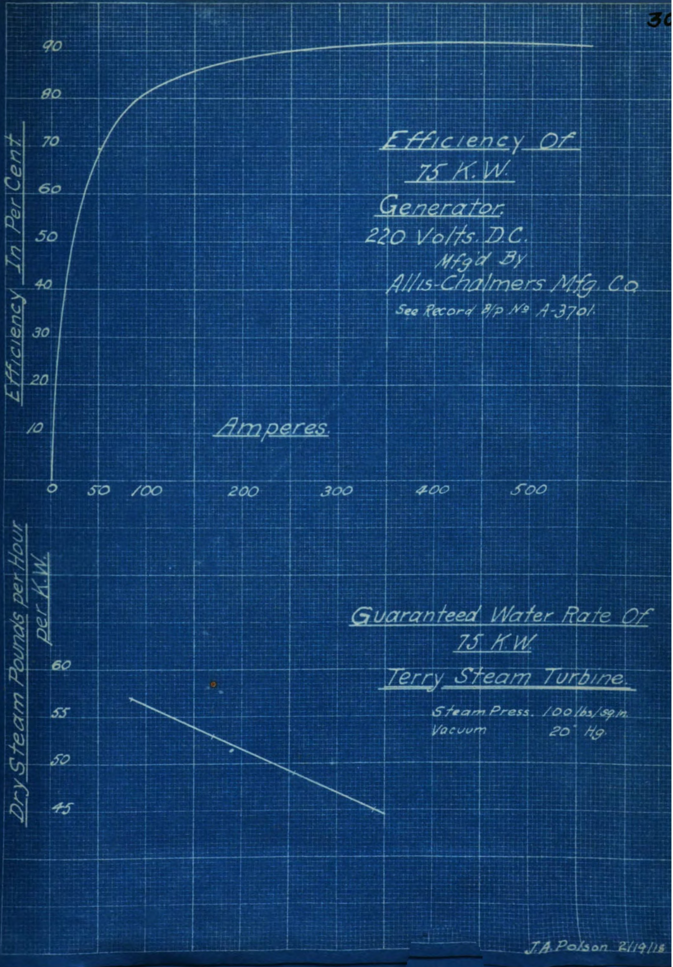
TEMP. T_2 – 2° LOW

TEMP. C_2 – 3° LOW

TEMP. C_3 – 2.5° LOW

H.M.S.

W.E.H.



Efficiency Of
75 K.W.
Generator.
220 Volts. D.C.
Mfg'd By
Allis-Chalmers Mfg. Co.
See Record B/p No. A-3701.
Efficiency In Per Cent
90
80
70
60
50
40
30
20
10
Amperes.
0
50
100
200
300
400
500
Guaranteed Water Rate Of
75 K.W.
Terry Steam Turbine.
Steam Press. 100 lbs/sq.in.
Vacuum 20" Hg.
Dry Steam Pounds per Hour per K.W.
60
55
50
45
J.A. Polson 2/19/15

W.E.H.

20	21	22	23	24	25	26	27
Number of orifices open	Total weight of Condensate - pounds	Weight of gland water used - pounds	C_1 Throttle calorimeter time in minutes.	C_1 Throttle calorimeter hundredths of pounds	Voltage	Amperage	Tachrometer
2	916	396	—	—	220	0	800
2	1052	390	32	12	220	17	800
2	1180	375	30	16	220	34	800
2	1299	354	27	20	220	51	799
2	1438	348	26	20	220	68	798
2	1537	324	18	20	220	85	797

47	48	49	50	51	52	53	54
TS							
P_3 Exhaust Total heat	P_2-P_3 Low pressure Re-heat	P_2-P_3 Low pressure Heat drop	Condenser cooling water pounds per hour	Steam used pounds per hour	Pounds of steam per K.W. hour	K.W. Generated	Revolutions per min.
—	—	—	38100	683	∞	0	2400
—	—	—	38100	883	235.5	3.76	2400
900	218	234	38100	1073	143.0	7.50	2400
910	205	225	38100	1260	112.0	11.25	2397
950	157	184	38100	1453	96.9	15.00	2394
1023	80	112	38100	1617	86.4	18.75	2381

H.M.G.

W.E.H.

20	21	22	23	24	25	26	27
Number of orifices open	Total weight of condensate - pounds	Weight of gland water used - pounds	C_1 Throttle calorimeter time in minutes.	C_1 Throttle calorimeter hundredths of pounds	Voltage	Amperage	Tachrometer
2	916	396	—	—	220	0	800
2	1052	390	32	12	220	17	800
2	1180	375	30	16	220	34	800
2	1299	354	27	20	220	51	799
2	1438	348	26	20	220	68	798
2	1537	324	18	20	220	85	797

47	48	49	50	51	52	53	54
TS							
P_3 Exhaust Total heat	P_2-P_3 Low pressure Re-heat	P_2-P_3 Low pressure Heat drop	Condenser cooling water pounds per hour	Steam used pounds per hour	Pounds of steam per K.W. hour	K.W. Generated	Revolutions per min.
—	—	—	38100	683	∞	0	2400
—	—	—	38100	883	235.5	3.75	2400
900	218	234	38100	1073	143.0	7.50	2400
910	205	225	38100	1260	112.0	11.25	2397
950	157	184	38100	1453	96.9	15.00	2394
1023	80	112	38100	1617	86.4	18.75	2391

H.M.G.

IE

20	21	22	23	24	25	26	27
Number of orifices open	Total weight of condensate - pounds	Weight of gland water used - pounds	C_1 Throttle calorimeter - Time in minutes	C_1 Throttle calorimeter hundredths of pounds	Voltage	Amperage	Tachrometer
2	989.0	419	—	—	220	0	800.6
2	1189.0	359	30	11	220	34	798.8
3	1533.0	353	—	—	220	78	795.0
3	1700.0	292	22.5	20	220	112	794.9
3	1913.0	270	13.42	20	220	146	791.2
3	2178.5	275	6.74	20	220	180	790.6

47	48	49	50	51	52	53	54
NITS							
P_8 Exhaust Total heat	P_2-P_8 Low pressure Re-heat	P_2-P_8 Low pressure Heat drop	Condenser cooling water pounds per hour	Steam used pounds per hour	Pounds of steam per K.W. hour	K.W. Generated	Revolutions per min.
942	182	193	38200	760	∞	0	2402
941	177	195	38200	1120	149.6	7.48	2396
906	204	227	57200	1573	91.7	17.15	2388
915	197	218	57200	1877	76.2	24.65	2385
977	118	157	57200	2191	68.2	32.15	2374
948	140	186	57200	2538	64.0	39.65	2372

H.H.S.

W.E.H.

20	21	22	23	24	25	26	27
Number of orifices open	Total weight of condensate - pounds	Weight of gland water used - pounds	C_1 Throttle calorimeter time in minutes	C_1 Throttle calorimeter hundredths of pounds	Voltage	Amperage	Tachometer.
2	1044	381	—	—	220	0	799.7
2	1489	360	30	8	220	51	793.5
3	1921	358	11	20	220	102	790.2
3	2264	300	4.546	20	220	153	788.8
3	2424	116	3.581	20	220	204	788.0
4	2575	0	6.146	20	220	255	786.0

47	48	49	50	51	52	53	54
ITS							
P_8 Exhaust Total heat	P_2-P_8 Low pressure Re-heat	P_2-P_8 Low pressure Heat drop	Condenser cooling water pounds per hour	Steam used pounds per hour	Pounds of steam per K.W. hour.	K.W. Generated	Revolutions per minute
952	170	183	38200	884	∞	0	2399
984	127	152	38200	1506	134.0	11.3	2380
1024	78	111	57200	2085	92.7	22.5	2371
1027	63	107	57200	2620	77.8	33.7	2366
1047	31	87	57200	3078	68.4	45.0	2364
1030	45	105	76200	3435	61.0	56.2	2358

H.M.S.

W.E.H.

seen by the increase in vel. at that

W.E.H.

	20	21	22	23	24	25	26	27
water in feet.	Total weight of condensate - pounds.	Number of orifices open.	Weight of gland water used - pounds.	C_1 Throttle calorimeter. Time in minutes.	C_1 Throttle calorimeter. Hundredths of pounds	Voltage	Amperage	Tachrometer
	1107.3	2	336	—	—	220	0	800.0
	1774.0	2	330	14.906	20	220	68	780.0
	2192.5	3	218	4.733	20	220	136	786.0
	2483.0	4	113	7.676	20	220	204	783.5
	2831.0	4	3	2.970	20	220	272	783.3
	3232.0	4	—	2.097	20	220	340	780.0

	47	48	49	50	51	52	53	54
NITS								
heat before reheat	P_8 Exhaust Total heat	P_2-P_8 Low Pressure Re-heat	P_8-P_8 Low Pressure Heat drop	Condenser cooling water pounds per hour	Steam used pounds per hour	Pounds of steam per K.W. hour	K.W. Generator	Revolutions per minute
	987	128	147	38100	1028	∞	0	2400
	1005	96	130	38100	1927	128.9	14.96	2370
7	1025	62	109	57200	2633	87.9	29.92	2358
8	1008	58	127	76200	3160	70.4	44.9	2331
3	1038	22	104	76200	3775	63.1	59.8	2350
7	1075	-48	60	76200	4310	57.6	74.8	2340

H.M.B.

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seen by the increase in vel. at that

W.E.

W.E.H.

	20	21	22	23	24	25	26	27
water in feet.	Total weight of condensate - pounds	Number of orifices open.	Weight of gland water used - pounds.	C_1 Throttle calorimeter. Time in minutes.	C_1 Throttle calorimeter. Hundredths of pounds	Voltage	Amperage	Tachrometer
	1107.3	2	336	—	—	220	0	800.0
	1774.0	2	330	14.906	20	220	68	780.0
	2192.5	3	218	4.733	20	220	136	786.0
	2483.0	4	113	7.676	20	220	204	783.5
	2831.0	4	3	2.970	20	220	272	783.3
	3232.0	4	—	2.097	20	220	340	780.0

	47	48	49	50	51	52	53	54
NITS								
heat before reheat	P_8 Exhaust Total heat	P_3-P_8 Low Pressure Re-heat	P_8-P_8 Low Pressure Heat Drop	Condenser cooling water pounds per hour	Steam used pounds per hour	Pounds of steam per K.W. hour	K.W. Generator	Revolutions per minute
	987	128	147	38100	1028	∞	0	2400
	1005	96	130	38100	1927	128.9	14.96	2370
7	1025	62	109	57200	2633	87.9	29.92	2358
6	1008	58	127	76200	3160	70.4	44.9	2351
3	1038	22	104	76200	3775	63.1	59.8	2350
7	1075	-48	60	76200	4310	57.6	74.8	2340

H.M.S.

seen by the increase in vel. at tha

W.E.

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20	21	22	23	24	25	26	27
Number of orifices open.	Total weight of condensate—pounds.	Weight of gland water used—pounds.	C_2 Throttle calorimeter time in minutes	C_1 Throttle calorimeter hundredths of pounds	Voltage	Amperage	Tachrometer.
2	1149.5	364.5	—	—	220	0	800.5
2	1797.0	363.0	23.00	20	220	85	790.0
3	2406.0	300.5	1.56	10	220	170	784.2
4	2881.5	263.0	1.85	10	220	225	780.0
4	3569.0	—	2.06	20	220	340	778.0
4	4083.0	—	1.39	20	220	400	774.8

	47	48	49	50	51	52	53	54
	P_8 Exhaust Total heat	P_2-P_8 Low pressure Re-heat	P_2-P_8 Low pressure Heat drop	Condenser cooling water pounds per hour	Steam used pounds per hour.	Pounds of steam per K.W. hour	K.W. Generated	Revolutions per minute
	1124	−9	10	38100	1047	∞	0	2402
	—	—	—	38100	1911	102.2	18.7	2370
7	—	—	—	57200	2805	75.0	37.4	2353
5	—	—	—	76200	3490	62.2	56.1	2340
6	1061	−35	76	76200	4760	63.6	74.8	2334
4	1072	−48	68	82300	5445	61.8	88.0	2324

H.M.S.

ft./sec

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seen by the increase in vel. at that

W.E.H.

TOTAL HEAT THROTTLE = 1152
TOTAL HEAT INTERSTAGE = 1135
TOTAL HEAT AT NOZZLE THROAT = 1054

1152 − 1054 = 98 B.t.u. drop in nozzle

$223.7\sqrt{98}$ = 2215 ft./sec. vel. steam at nozzle

$\frac{\pi \times 26.5}{12} \times \frac{2400}{60}$ = 278 ft./sec. vel. of rotor

From velocity diagram $\alpha = 20°$
$\beta_1 = 24° - 45'$
$V = 1415$ ft./sec.

$\left(\frac{2215}{223.7}\right)^2 - \left(\frac{1415}{223.7}\right)^2 = 58.5$ B.t.u. loss in nozzle

For vel. steam leaving nozzle = 1415
V_1 = vel. steam entering #1 moving blades = 1160 ft./sec

For measured angles of blades and guide

V_2 = 800 ft./sec. V_5 = 290 ft./sec.
V_3 = 550 ft./sec. V_6 = 280 ft./sec.
V_4 = 620 ft./sec. V_7 = 250 ft./sec

$\left(\frac{1160}{223.7}\right)^2 - \left(\frac{800}{223.7}\right)^2 = 14.2$ B.t.u. loss in #1 blades

$\left(\frac{550}{223.7}\right)^2 - \left(\frac{620}{223.7}\right)^2 = -1.7$ B.t.u. reheat in #1 guide

$\left(\frac{290}{223.7}\right)^2 - \left(\frac{280}{223.7}\right)^2 = .14$ B.t.u. loss in #2 blades

Residual vel. = 250 ft./sec.

$\left(\frac{250}{223.7}\right)^2 = 1.25$ B.t.u. residual heat

58.5 + 14.2 − 1.7 + .14 + 1.25 = 75.79 Total loss in High Pressure stage by diagram

Heat loss by calorimeters 1135 − 1038 = 97 B.t.u.

97 − 75.79 = 21.21 B.t.u. radiation and convection

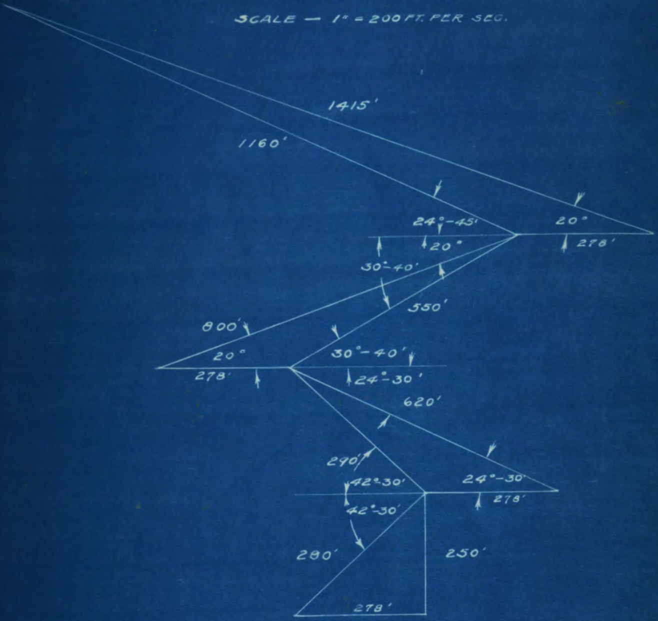
REMARKS.

The stationary guide is in nature an expanding nozzle due to difference in entrance and exit angles, consequently the pressure thru the stage is not absolutely constant as may be seen by the increase in vel. at that point.

W.E.H.

HIGH PRESSURE
VELOCITY DIAGRAM
4 NOZZLES FULL LOAD

SCALE — 1" = 200 FT. PER SEC.



TYPICAL VELOCITY DIAGRAM
CONSTRUCTED FROM MEASURED
ANGLES AND WITH RESULTS
HERE OBTAINED

WEH

CALCULATIONS for VELOCITIES IN LOW PRESSURE STAGE AND EQUIVALENT HEAT LOSS

Pressure in interstage = 13.54 lb.
Pressure 1st vel. stage = 9.47 lb.
Pressure 2nd vel. stage = 7.20 lb.
Pressure 3rd vel. stage = 5.28 lb.
Pressure in condenser = 3.95 lb.
Total heat interstage = 1135 B.t.u.
Total heat condenser = 1031 B.t.u.
1135 − 1031 = 104 B.t.u. drop thru 3 stages
104 ÷ 3 = 34.66 B.t.u. drop per stage
$223.7\sqrt{34.66} = 1318$ ft. per sec. theoretical velocity in each nozzle
From velocity diagram vel. #1 = 665 ft./sec.
1318 − 665 = 653 ft./sec. or
$\left(\frac{653}{223.7}\right)^2 = 8.53$ B.t.u. loss each nozzle
Also $\left(\frac{500}{223.7}\right)^2 - \left(\frac{315}{223.7}\right)^2 = 5.01 - 1.99 = 3.02$ B.t.u. loss per row of blades

From pressures above considering the heat loss uniformly distributed, reheat per stage as per H—T diagram is 4 B.t.u.
since the residual heat is
$\left(\frac{105}{223.7}\right)^2 = .221$ B.t.u.
$3 \times 8.53 + 3 \times 3.02 + .221 = 35.76$ B.t.u. loss by diag. as against 34.66 theoretical loss.

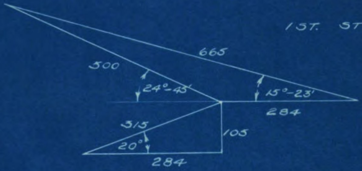
The nozzles were designed for pure impulse effect but the entire low pressure stage is impulse-reaction, as may be seen by diagram.
The pressures above fall according to a curve consequently it is slightly erroneous to consider the heat drop equally distributed but for general conclusions the result is sufficiently accurate.

velocity of rotor is found as before
$\frac{\pi \times 26\frac{15}{16}}{12} \times \frac{2400}{60} = 284$ ft. per sec.

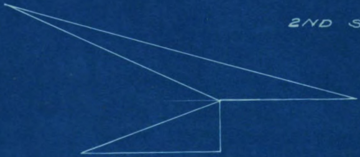
WEH

LOW PRESSURE VELOCITY DIAGRAM
4 NOZZLES – FULL LOAD
IMPULSE – REACTION

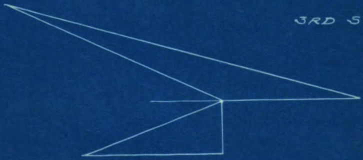
1ST. STAGE



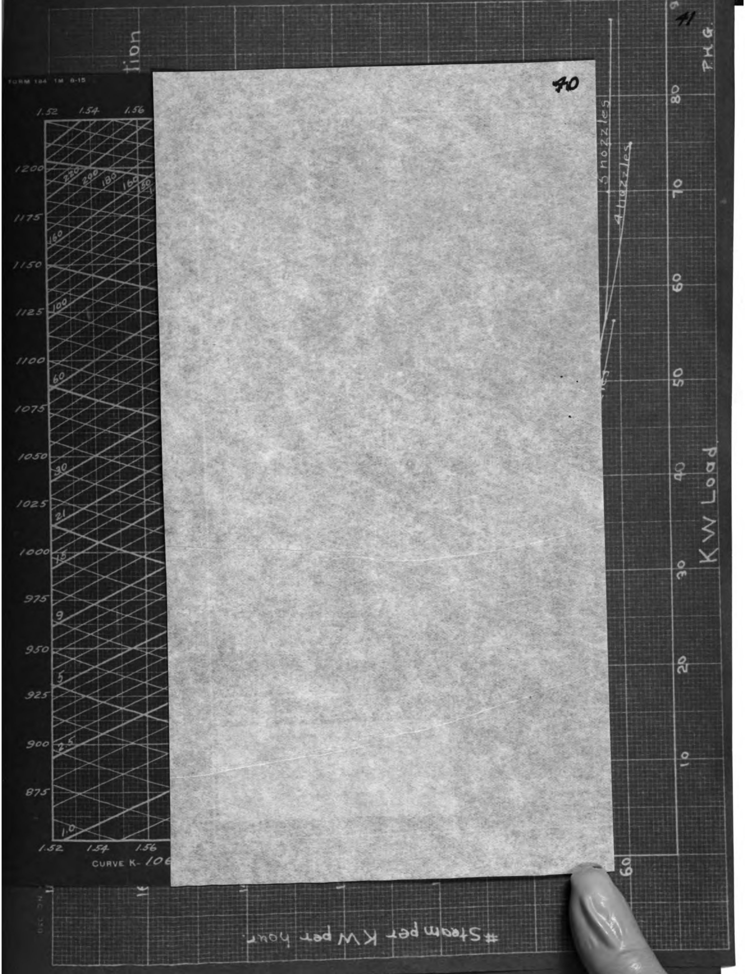
2ND STAGE



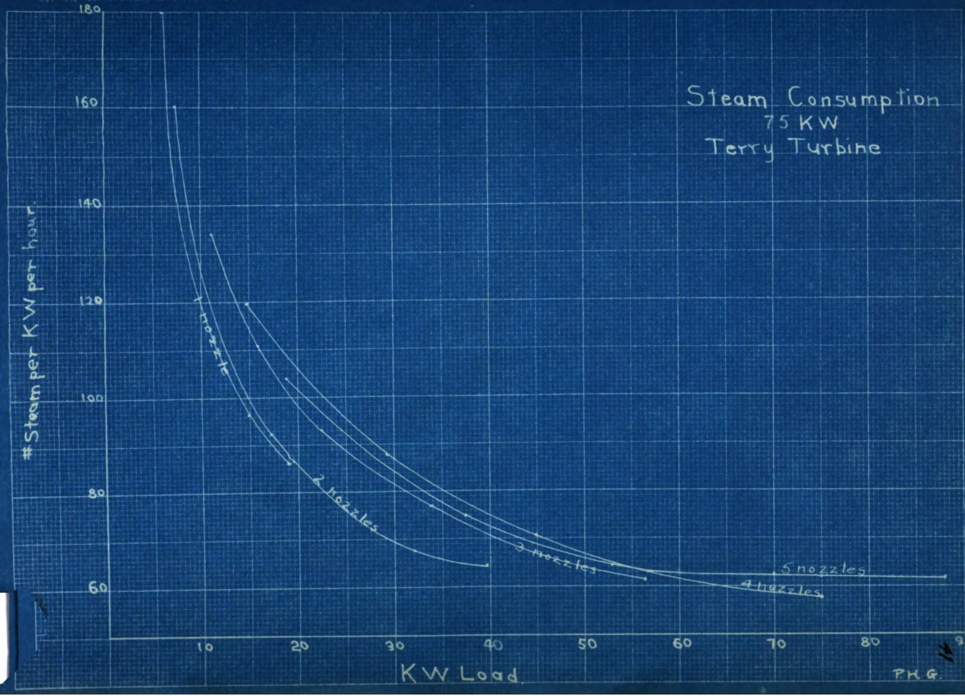
3RD STAGE

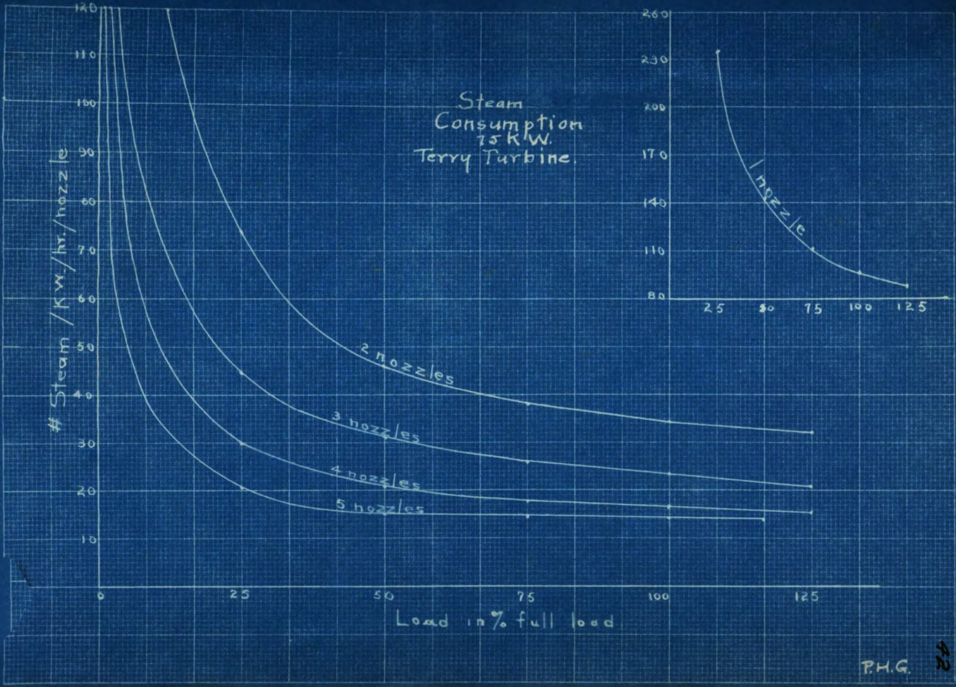


LOW PRESSURE NOZZLE ANGLES AND BLADE ENTRANCE AND EXIT ANGLES EQUAL FOR THREE VELOCITY STAGES



tion
FORM 184 1M 8-15
1.52 1.54 1.56
1200
1175
1150
1125
1100
1075
1050
1025
1000
975
950
925
900
875
CURVE K- 106
5 nozzles
4 nozzles
P.H.G.
KW Load
#Steam per KW per hour.


Steam Consumption
75 KW
Terry Turbine
#Steam per KW per hour.
KW Load
180
160
140
120
100
80
60
10
20
30
40
50
60
70
80
1 nozzle
2 nozzles
3 nozzles
4 nozzles
5 nozzles
P.H.G.
41



Steam
Consumption
75 K.W.
Terry Turbine.
Steam / K.W./hr./nozzle
Load in % full load
1 nozzle
2 nozzles
3 nozzles
4 nozzles
5 nozzles
P.H.G.