IMPACT OF SWING CONDITIONS ON TURBINE DESIGN

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ABSTRACT

Frequently, large swings in operating conditions are specified when purchasing turbines. Although the objective of these operating conditions may be to insure conservatism, they also may result in turbine designs which yield higher initial turbine and plant costs, and in higher operating costs.

A handbook procedure for sizing a mechanical drive steam turbine is given which allows the turbine user to assess the impact of swing conditions on the turbine design. Extreme oversizing on the condensing section of extraction/induction turbines is a frequent occurrence; therefore, an example of this type is reviewed.

INTRODUCTION

When a mechanical drive steam turbine is purchased, the user must specify what operating conditions the turbine will see. Basic conditions are inlet presure, inlet temperature, exhaust pressure, horsepower and rpm. Normal practice is to define the guarantee point at the conditions where the turbine will operate for the majority of the time. The optimum turbine for the guarantee point would logically be designed strictly for guarantee point conditions; however, this is very infrequently the case. Ambient temperature changes affect exhaust pressure and require special start-up requirements and/or contingencies for boiler problems, and pipe pressure drops affect inlet conditions and contingencies for driven equipment. Varying operating conditions such as these affect horsepower and rpm. These are just a few of the items which cause a turbine buyer to specify swing conditions which are inconsistent with guarantee point conditions.

Table 1 demonstrates how swing conditions affect eight major turbine parameters at the guarantee point. The table indicates either an increase (I) or a decrease (D) in the parameter if the turbine is designed for the swing condition rather than the guarantee point condition. The "-S" indicates that the parameter has been affected in a secondary manner by one of the other parameters, rather than directly by the operating condition. A high inlet pressure may cause the first stage bucket width to increase, which increases the bearing span; therefore, the bearing span is affected in a secondary manner. It is worth noting that the bearing span is affected in a secondary manner by virtually all swing conditions.

Table 1. Turbine Parameters Affected by Oversizing.

CONDITIONS OVERSPECIFIED	FIRST STAGE BUCKET SIZE	FIRST STAGE EFF.	INLET COMPONENT SIZE	LAST STAGE BUCKET HEIGHT	EXH. CASING SIZE	LAST STAGE EFF.	BEARING SPAN	INLET THROTTLING LOSS
INLET PRESSURE								
HIGH	I	-	I	-	-	-	I-S	-
LOW	I	D	I	-	-	-	I-S	I-S
INLET TEMPERATURE								
HIGH	I	-	I	-	-	-	I-S	-
LOW	-	-	-	-	-	-	I-S	-
EXHAUST PRESSURE								
HIGH	I	D	-	-	-	-	-	I-S
LOW	-	-	-	I	I	I	I-S	-
SHAFT HORSEPOWER								
HIGH	I	D	-	I	I	I	I-S	I-S
LOW	-	-	-	-	-	-	-	-
SHAFT RPM								
HIGH	-	-	-	-	-	-	-	-
LOW	I	-	-	-	-	-	I-S	-

In order to obtain the maximum benefit from Table 1, the increases or decreases in the parameter must be quantified. This entails the need for designing the turbine at guarantee conditions and swing conditions and assessing the change in turbine design. Turbine cost data are not presented; however, the direction of cost change should be evident from the change in the turbine parameter.

A STEAM TURBINE DESIGN HANDBOOK

There are a great many factors, both mechanical and thermodynamic, which affect the design of a multi-stage, axial impulse type, mechanical drive steam turbine for use in the process industry. They cannot all be covered in a simple handbook; however, based on empirical data, a handbook which is accurate to $\pm 15\%$ can be established. The intent of this handbook procedure is not to enable the user to design steam turbines in the absolute sense, since a large amount of simplification has been introduced, but rather to allow the user to establish accurate changes in design. It should be kept in mind that most steam turbine manufacturers allow for changes in parameters in discrete steps, rather than on a continuous basis, and that this handbook is based on the continuous method. Proper use of this handbook should include consultation with the turbine vendor to confirm the results obtained for specific cases.

Step 1—Select Base Diameter

A good assumption when designing a mechanical drive steam turbine is that the base diameter, as defined in Figure 1, is constant on all stages. With this in mind, the turbine base diameter (BD) can be selected from Figure 2 when only the rpm at guarantee is known. Overspecifying operating conditions will not change the base diameter, but the base diameter is needed to determine other parameters later.

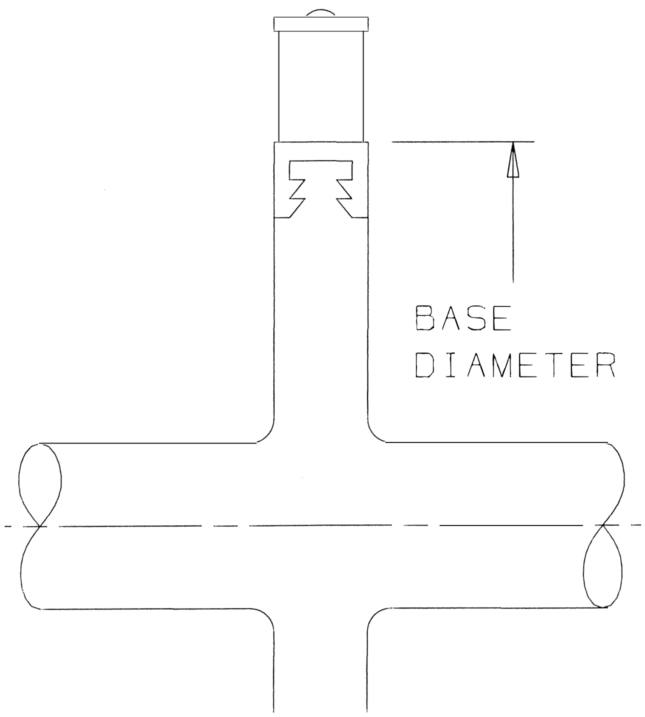

Figure 1. Base Diameter Definition.

Step 2—Establish Turbine Mass Flow

The turbine mass flow can be determined from Equation 1, when inlet and exhaust steam conditions and horsepower are known and an efficiency is assumed.

$$f = \frac{hp \times 2544.5}{\Delta h_{ti} \eta} \quad (1)$$

where

- f = mass flow (lb/hr)
- Δh_{ti} = isentropic available energy (BTU/lb) to turbine [1]
- η = assumed overall efficiency (.80)
- hp = shaft horsepower

An assumed efficiency of 80% is normally accurate to ±7% for a modern multi-stage steam turbine.

Step 3—Select the Number of Stages

An expression for the velocity ratio (u/c_o) of a stage is given by Equation 2.

$$u/c_o = \frac{(BD + BH) \times rpm}{229 \times 223.7 \times \sqrt{\Delta h_{si}}} \quad (2)$$

where

- BH = blade height (in.)
- Δh_{si} = isentropic available energy (BTU/lb) to stage

A detailed explanation of velocity ratio can be found in Reference 2.

Equation 3 can be used to calculate the number of stages (N) in a turbine.

$$N = \left[\frac{(u/c_o)\,(51{,}227)\sqrt{\Delta h_{ti}}}{(BD + AH) \times rpm}\right]^2 \quad (3)$$

where

- AH = average blade height in turbine (in.) (assume 1.5 initially)

On non-condensing turbines, Δh_{ti} can be used directly; but for condensing turbines, Δh_{ti} should be increased (a good value is 3%) to account for reheat [2]. A value for u/c_o which will yield reasonable results is 0.52. This will place all stages close to their most efficient operating point. If performance is of paramount importance, round the number of stages up; however, if initial capital investment is an overriding criterion, round the number of stages down.

Step 4—Select First Stage Size

Mass flow, in lb/hr, can be calculated using Equation 4.

$$f = .95\ P\ K\ A\ n \quad (4)$$

where

- P = total pressure upstream of diaphragm (psia)
- K = mass flow rate (from Figure 3) $\frac{lb/hr}{psia \times in. \times in.}$
- A = physical nozzle area of diaphragm (in.^{2})
- n = pressure ratio factor given by Equation 5:

$$n = 4.413 \left(\left(\frac{P_2}{P_1}\right)^{1.5385} - \left(\frac{P_2}{P_1}\right)^{1.7692} \right)^{\frac{1}{2}} \quad (5)$$

where

- n = 1 for $P_2/P_1 < .5464$

Equation 4 includes a stage flow coefficient of .95. Equation 6 can be used to obtain first stage P_1/P_2 for use in Equation 5 by assuming equal pressure ratios per stage.

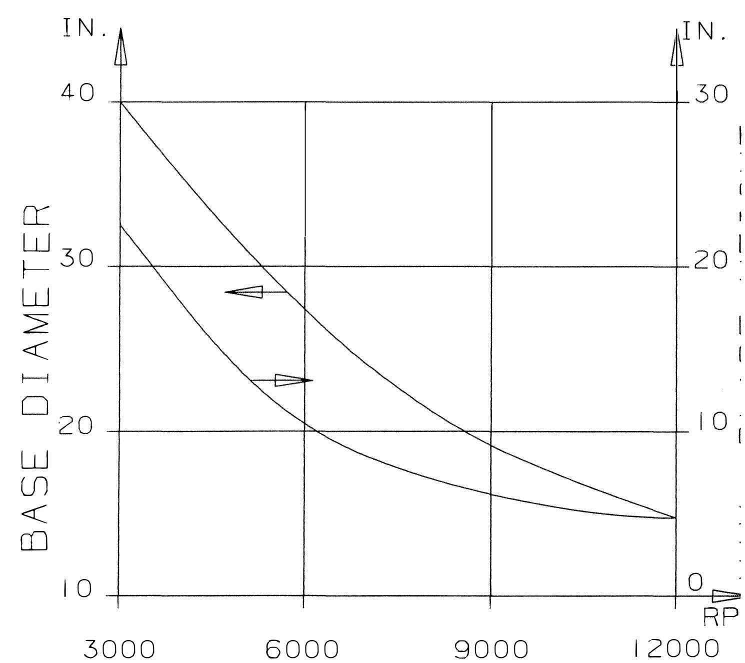

Figure 2. Guarantee rpm vs. Base Diameter and Maximum Blade Height.

$$P_1/P_2 = (P_{in}/P_{ex})^{\frac{1}{N}} \qquad (6)$$

where

P_{in} = inlet pressure (psia)

P_{ex} = exhaust pressure (psia)

Equation 7 will now yield the first stage nozzle area.

$$A = \frac{f}{.95 \times .95 \times P_{in} \times K \times n} \; ; (\text{in.}^2) \qquad (7)$$

Five percent throttling loss has been added to Equation 7. First stage height can now be related to the nozzle area using Equation 8.

$$A/X = .785\,[(BD+2H)^2 - BD^2]\,\sin 12° \qquad (8)$$

where

X = fraction of 360° which has nozzles present

H = first stage nozzle height (in.)

Equation 8 assumes a nozzle angle of 12°. X is normally either 0.5 or 1.0 for first stages.

There are three basic ways to modify the first stage design in order to assure structural integrity. Increasing either BD or X has the effect of lowering the height for a required area, thereby reducing stresses. The third way is to change the actual bucket design. The most influential parameter toward lowering stresses when changing a design is an increase in width. Only the change in width will be explained in detail. An increase in either BD or X is handled by simply increasing the values in Equation 8 and then following the same procedure as an increase in width.

There are many ways to determine the required first stage bucket width. One commonly used method is to combine alternating and steady state stresses to determine a Goodman Factor. Maximum alternating stress is calculated using the familiar form shown in Equation 9.

$$\sigma = \frac{M}{Z} \; ; (\text{psi}) \qquad (9)$$

where

M = moment (in. × lbs)

Z = section modulus (in.^{3})

Equation 10 provides a means to estimate section modulus for a given bucket width. It applies to low reaction stages and to the base sections of reaction stages which have low reaction at the hub.

$$Z = .006 \left(\frac{BW}{.75}\right)^3 \; ; (\text{in.}^3) \qquad (10)$$

where

BW = bucket axial width (in.)

The number of buckets per 360° can be found from Equation 11.

$$Q = \frac{\pi \times BD}{BW \times .62} \qquad (11)$$

where

Q = number of buckets

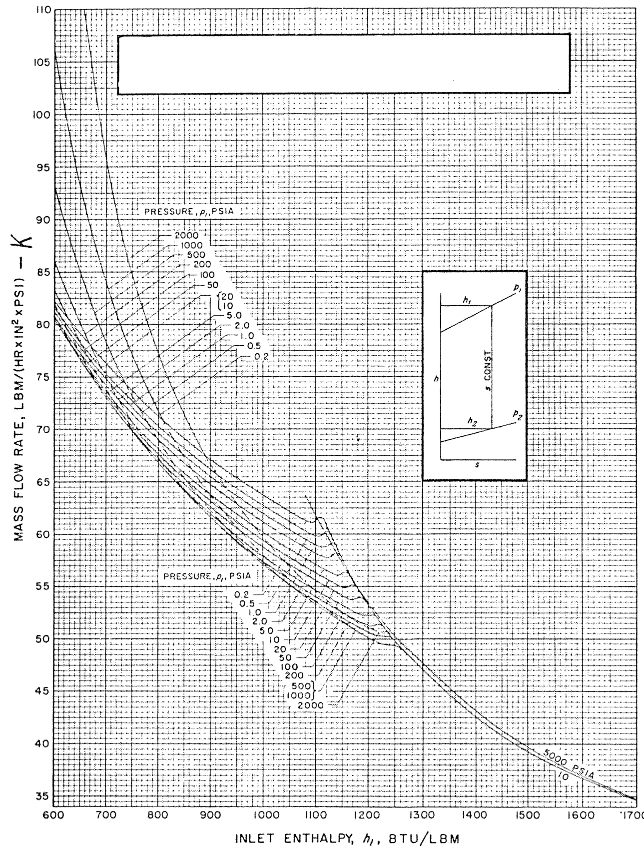

Figure 3. Mass Flow Rate (K) vs. Inlet Enthalpy (from ASME Steam Tables, Third Edition, by Courtesy of ASME).

Maximum alternating stress can now be calculated from Equation 12.

$$\sigma_a = \frac{BH \times hp \times 63{,}024}{Z \times Q \times X(BD + BH) \times rpm} \; ; (\text{psi}) \qquad (12)$$

where

σ_a = alternating stress (psi)

Equation 13 can be used to estimate the centrifugal stress in the base of the bucket vane.

$$\sigma_c = \frac{BH\,(BD + BH + .13)}{249{,}048} \times rpm^2 ; (\text{psi}) \qquad (13)$$

where

σ_c = centrifugal stress (psi)

It applies to constant cross section buckets only.

Figure 4 is a Goodman Diagram. It is a plot of steady state stress vs. alternating stress, with a line drawn between the material's ultimate strength and endurance limit ÷ 1.7 (notch factor) at temperature.

A Goodman Factor equals an allowable alternating stress divided by σ_a, and it should be greater than ten for reliable first stage operation. The process for obtaining the Goodman Factor is as follows:

1. Calculate a steady state stress equal to $\sigma_a + \sigma_c$.
2. Enter Figure 4 on the abscissa at the steady state stress calculated in Equation 1.
3. Project vertically up to the line.

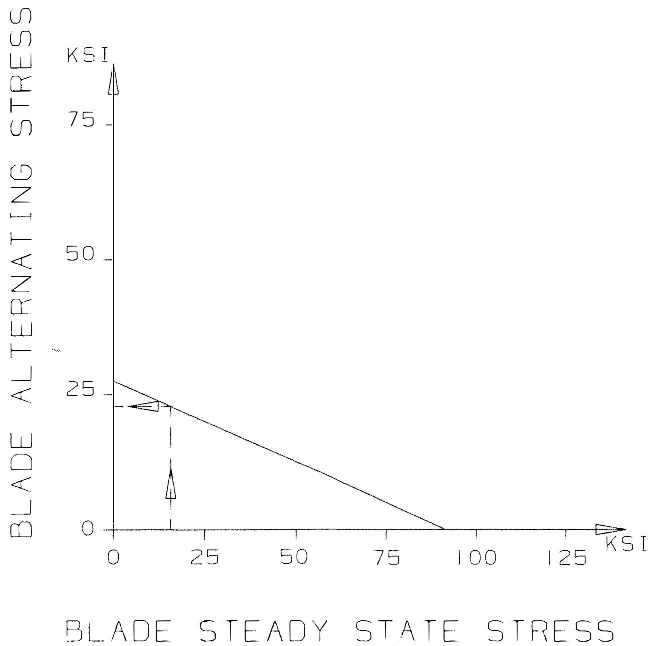

Figure 4. Goodman Diagram.

4. Project horizontally over to the ordinate. This is the allowable alternating stress.
5. Divide the allowable alternating stress by σ_a to obtain the Goodman Factor.

If a bucket does not meet the Goodman Factor criterion of 10, either BD, X or the bucket design must be changed.

Step 5—Select Last Stage Height

Estimate the last stage available energy using Equation 14.

$$\Delta h_{si} = \left[\frac{(BD+BH)\times rpm}{229\times 223.7\times u/c_o}\right]^2 \tag{14}$$

For non-condensing turbines, use .50 for u/c_o initially and 1.5 in. for BH. For condensing turbines, use .56 for u/c_o initially and 7.5 in. for BH.

Obtain the P_1/P_2 for the last stage from Reference 1 and the results of Equation 14. Assume P_1 is accompanied by an enthalpy of 1200 BTU/lb for non-condensing turbines, and 1100 BTU/lb for condensing turbines. P_2 is exhaust pressure.

Last stage area is obtained from Equation 7, using P_1 in place of P_{in}. Last stage height is obtained from Equation 8 for non-condensing turbines, with 16° used in place of 12° for condensing turbines. X is normally 1.0 for last stages.

To size a stage for swing conditions, use the same procedure as above, but set u/c_o to .35 for non-condensing turbines and .42 for condensing turbines.

Step 6—Check Steps 3-5 for Consistency

The average blade height assumed in Step 3 should equal the first plus the last stage heights, divided by 2 for non-condensing turbines and by 2.8 for condensing turbines. Steps 3-5 should be repeated to obtain reasonable agreement.

Step 7—Estimate Inlet Component Size

The inlet flange diameter can be calculated from Equation 15.

$$D = \left(\frac{f\times \bar{v}\times .051}{V}\right)^{\frac{1}{2}};\ (\text{in.}) \tag{15}$$

where

D = inlet diameter (in.)
$\bar{v}$ = inlet specific volume (ft^3/lb)
V = inlet velocity (ft/sec)

A good inlet velocity for sizing purposes is 150 ft/sec.

Step 8—Size Exhaust Casing

The exhaust casing can be sized using Equation 9, with the exhaust specific volume and the exhaust velocity. Use 200 ft/sec to size a non-condensing exhaust casing, and 350 ft/sec to size a condensing exhaust casing.

Step 9—Estimate First Stage Efficiency

The absolute value of stage efficiency is dependent on factors too numerous to be included in the scope of this paper. Calculate the stage u/c_o using Equation 2, and assume that the stage efficiency is represented in Figure 5.

Step 10—Estimate Last Stage Efficiency

The same comments which apply to the first stage efficiency also apply to the last stage efficiency.

Step 11—Estimate Inlet Throttling Losses

Throttling losses can account for an overly large performance penalty when there is a large excess of first stage nozzle area over the guarantee point. Figure 6 is a means for estimating the throttling loss of a turbine with 90 BTU/lb of available energy. Throttling losses on non-condensing turbines are especially severe. For turbines with higher available energies, the throttling loss should be decreased by the ratio of the available energies. Five percent throttling loss is assumed for turbines operating in the valve(s) wide open condition.

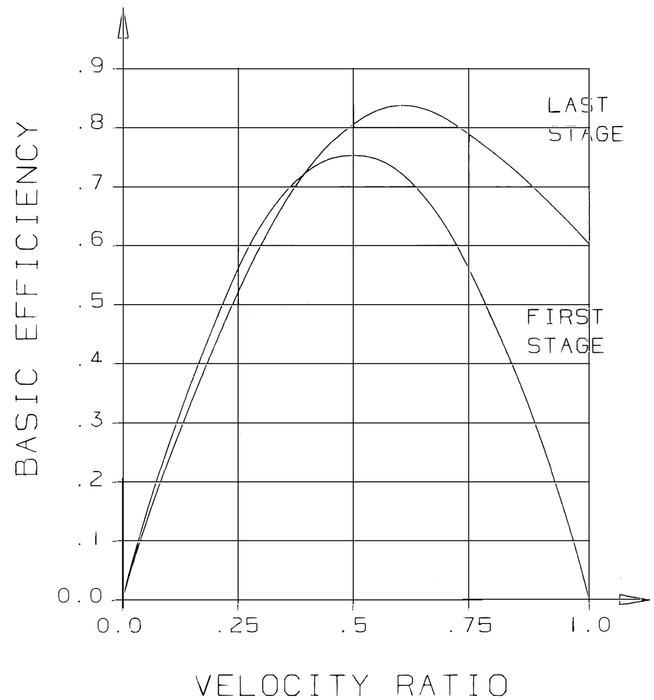

Figure 5. First and Last Stage Basic Efficiency vs. Velocity Ratio.

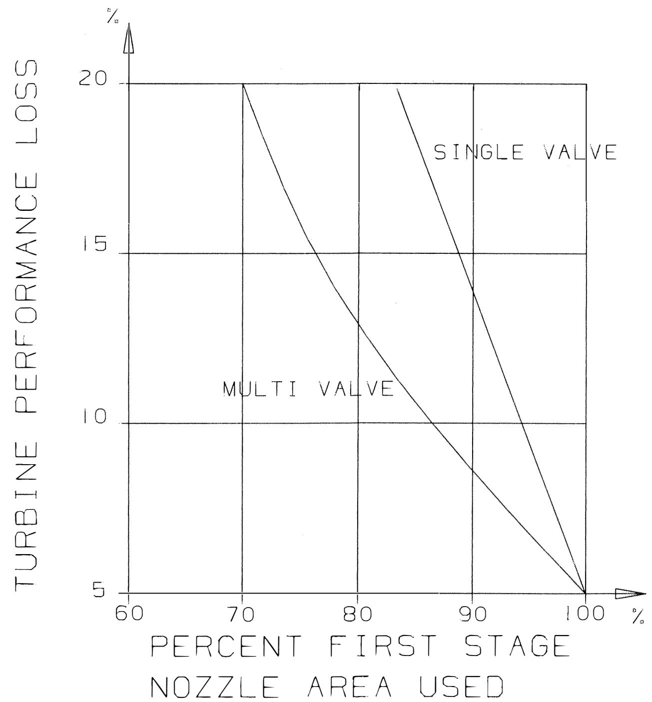

Figure 6. Turbine Performance Loss vs. Percent Nozzle Area Used at Guarantee Point (for Turbine With 90 BTU/lb of Available Energy).

AN EXAMPLE CALCULATION

Now that a method for estimating the change in turbine parameters is available, an example calculation will demonstrate quantitatively how overspecification of operating conditions affect a turbine's design and its performance. The example used will be the condensing end of an automatic extraction turbine with multiple extraction control valves. Table 2 gives the guarantee point conditions and a set of swing conditions to which the turbine will be designed.

OPERATING CONDITIONS	GUARANTEE POINT	SWING CONDITIONS
INLET PRESSURE (PSIA)	300	250
INLET TEMPERATURE (DEG F)	600	600
EXHAUST PRESSURE ("HGA)	6	2
HORSEPOWER	10000	13500
RPM	7500	7500

Table 2. Example Operating Conditions.

Step 1—Select Base Diameter

From Figure 2 at 7500 rpm, the base diameter is 23 in.

Step 2—Estimate Mass Flow

The isentropic available energy obtained from Reference 1, based on inlet and exhaust conditions at the guarantee point, is 346 BTU/lb. Mass flow at the guarantee point can then be calculated using Equation 2.

$$f_g = \frac{10{,}000 \times 2544.5}{346 \times 0.8} = 92{,}000 \text{ lb/hr}$$

where

f_g = mass flow at guarantee point

The largest mass flow will occur at 250 psia, 600°F, 6 in. and 13,500 hp. This is the operating point with maximum inlet volume flow.

The associated isentropic available energy is 338 BTU/lb. The mass flow at swing conditions can now be calculated using Equation 2.

$$f_s = \frac{13{,}500 \times 2544.5}{338 \times 0.8} = 127{,}040 \text{ lb/hr}$$

where

f_s = mass flow at swing point (lb/hr)

Step 3—Estimate Number of Stages

Using Equation 3:

$$N = \left[\frac{.52 \times 51{,}227 \times \sqrt{346 \times 1.03}}{(23 + 1.5) \times 7500)}\right]^2 = 7.5$$

Use 7 stages.

Step 4—Select First Stage Size

Using Equation 6 at the guarantee point,

$$P_1/P_2 = (300 \times .95/2.95)^{\frac{1}{7}} = 1.92$$

where

2.95 psia = 6 in. HgA

The first stage pressure ratio equals 1.92. The required nozzle area at the guarantee point can be calculated from Equation 7.

$$A_g = \frac{92{,}000}{.95 \times .95 \times 300 \times 46.5} = 7.31 \text{ in.}^2$$

where

A_g = first stage nozzle area at guarantee point
n = 1, from Equation 5.
K = 46.5, from Figure 3.

Assuming X = .5, Equation 8 can be used to obtain the first stage height, resulting in H_g = 0.935 in.

The next topic to be addressed is the required nozzle area to pass f_s, which must be approached in a different manner. The pressure between the first and second stages at the guarantee point was shown to be

$$300 \times .95/1.92 = 149 \text{ psia}$$

An accurate assumption is that the pressure ratio across the second stage remains constant, independent of flow. Therefore, the pressure between the first and second stages required to pass f_s through the second stage can be obtained by multiplying 149 psia by the ratio f_s/f_g. This equals 206 psia.

The pressure ratio across the first stage at the swing point is, therefore,

$$P_1/P_2 = \frac{250 \times .95}{206} = 1.15$$

If pressure ratio vs. n is plotted from Equation 5, it will be seen that n becomes very sensitive to pressure ratio at low pressure ratios. For this reason, a lower pressure ratio limit of approximately 1.3 is set by turbine designers. To do this, the second stage nozzle area must be increased by approximately $(1.3 \div 1.15 - 1) \times 100 = 13.0\%$. This will cause the first stage pressure ratio at the guarantee point (for the turbine designed for swing conditions) to increase from 1.9 to approximately $1.9 \times 1.13 = 2.15$.

The first stage area to pass the swing flow can now be calculated.

$n = .874$ (at $P_1/P_2 = 1.3$)

$K = 46.4$

$$A_s = \frac{127{,}040}{.95 \times .95 \times 250 \times 46.4 \times .874} = 13.88 \text{ in.}^2$$

The first stage height now becomes 1.719 in., assuming that X still equals 0.5 and BD still equals 23 in.

The first stage bucket width is selected based on a Goodman Factor of 10 at the guarantee point operating conditions. The first thing to do is estimate the first stage horsepower. For the guarantee point height (pressure ratio = 1.9) the available energy is 67 BTU/lb [1]. First stage horsepower can then be calculated as below.

$$hp = \frac{92{,}000 \times 67 \times .8}{2544.5} = 1937 \text{ hp}$$

The minimum bucket width in common use today is 0.5 in. This width yields a Goodman Factor of 20, well within the acceptable limits.

The available energy on the first stage at the guarantee point for the turbine designed for swing conditions (pressure ratio = 2.15) is 79 BTU/lb. The stage horsepower is then

$$hp = \frac{92{,}000 \times 79 \times .8}{2544.5} = 2285 \text{ hp}$$

The fraction of 360° which has nozzles present (X) at the guarantee point flow is reduced from .5 to

$$X = .5 \left(\frac{7.31}{13.88}\right) = .26$$

Iterating with Equations 10-13 and Figure 3 yields a bucket width of 0.75 inches to obtain a Goodman Factor of 10.

Step 5—Select Last Stage Height

Last stage available energy at the guarantee point is

$$\Delta h_{si} = \left(\frac{30.5 \times 7500}{229 \times 223.7 \times .56}\right)^2 = 64 \text{ BTU/lb}$$

From Reference 1, the results of the above calculation and $P_2 = 2.95$ psia (6 in. H_gA),

$$P_1/P_2 = \frac{8.33}{2.95} = 2.82$$

The last stage area can now be estimated using Equation 7.

$$A_g = \frac{92{,}000}{.95 \times 8.33 \times 56.7} = 205 \text{ in.}^2$$

where

$n = 1$

$K = 56.7$

The last stage height is found from Equation 8, using 16°, and is 7.7 in. Experience has shown that, for these operating conditions, this is an extremely tall height.

The swing conditions which must be used for sizing the last stage are 250 psia, 600°F, 2 in. HgA at 13,500 hp. This is the maximum exhaust volume flow point. The flow at this point is

$$f_s = \frac{13{,}500 \times 2544.5}{391 \times .8} = 109{,}820 \text{ lb/hr}$$

Assuming a PD of 32.5 gives

$$\Delta h_{si} = \left(\frac{32.5 \times 7500}{229 \times 223.7 \times .42}\right)^2 = 128 \text{ BTU/lb}$$

If an exhaust enthalpy of 972 BTU/lb is assumed at 0.98 psia (2 in. HgA), and 128 BTU/lb is added, the stage inlet pressure is 8.3 psia. This gives a last stage pressure ratio of $8.3 \div .98 = 8.5$.

Assuming an enthalpy upstream of the last stage of 1100 BTU/lb yields

$n = 1$

$K = 55.9$

The area for swing conditions is then

$$A_s = \frac{109{,}820}{.95 \times 8.3 \times 55.9} = 249 \text{ in.}^2$$

The accompanying height is 9.0 in. This height is extremely large for these operating conditions (see Figure 2), and this turbine would most likely become a double flow last stage design.

Step 6—Check Steps 3-5 for Consistency

The average blade height assumed in Step 3 was 1.5 in., while the actual height is

$$AH = \frac{.935 + 7.7}{2.8} = 3.1 \text{ in.}$$

The number of stages is then checked to be

$$N = \left[\frac{.52 \times 51{,}227 \ \times 346 \times 1.03}{(23 + 3.1) \times 7500)}\right]^{\frac{1}{2}} = 7.0$$

Step 7—Estimate Inlet Component Size and Material

At the guarantee point conditions, the inlet diameter is

$$D = \left(\frac{92{,}000 \times 2 \times .051}{150}\right)^{\frac{1}{2}} = 7.9 \text{ in.}$$

At the swing conditions the inlet diameter is

$$D = \left(\frac{127{,}040 \times 2.43 \times .051}{150}\right)^{\frac{1}{2}} = 10.2 \text{ in.}$$

Step 8—Size Exhaust Casing

At the design point the exhaust casing flow diameter is

$$D = \left(\frac{92{,}000 \times 117 \times .051}{350}\right)^{\frac{1}{2}} = 40 \text{ in.}$$

At the swing conditions, the exhaust diameter will be

$$D = \left(\frac{109{,}820 \times 167 \times .051}{350}\right)^{\frac{1}{2}} = 52 \text{ in.}$$

Step 9—Estimate First Stage Efficiency

At the guarantee conditions, the first stage velocity ratio can be found using Equation 15.

$$u/c_o = \frac{23.935 \times 7500}{229 \times 223.7 \times \sqrt{67}} = .43$$

The efficiency, from Figure 4, is 74%.

The velocity ratio for the turbine designed to swing conditions but operating at the guarantee point is

$$u/c_o = \frac{24.719 \times 7500}{229 \times 223.7 \times \sqrt{79}} = .41$$

The efficiency, from Figure 4, is 72.5%.

Step 10—Estimate Last Stage Efficiency

The velocity ratio of the last stage at guarantee conditions is .56. The efficiency, from Figure 4, is 84%.

The pressure upstream of the last stage required to pass guarantee flow through the stage designed for swing conditions is approximately

$$P = \frac{92{,}000}{56.2 \times 249 \times .95} = 6.9 \text{ psi}$$

The last stage pressure ratio is

$$P_1/P_2 = \frac{6.9}{2.95} = 2.3$$

The available energy is 53 BTU/lb.

The last stage velocity ratio is then

$$u/c_o = \frac{32.0 \times 7500}{229 \times 223.7 \times \sqrt{53}} = .65$$

The corresponding last stage efficiency is also 84%. The efficiency has not changed for this particular case; however, it is very common for it to do so.

Step 11—Estimate Inlet Throttling Losses

The percent first stage nozzle area used at the guarantee point is 7.31 ÷ 13.88 = 52%. This yields additional throttling losses well in excess of 15% on a turbine with 90 BTU/lb of available energy, or in excess of 15*90 ÷ 346 = 3.9% on this turbine.

DISCUSSION

Below is a comparison of the major parameters affected.

1.	Percent increase in first stage nozzle area.	90%
2.	Percent increase in first stage blade height.	84%
3.	Percent increase in first stage blade width.	50%
4.	Percent increase in last stage nozzle area.	21%
5.	Percent increase in last stage blade height.	17%
6.	Percent increase in inlet pipe diameter.	29%
7.	Percent increase in exhaust casing diameter.	30%
8.	Percent decrease in first stage efficiency.	2.0%
9.	Percent decrease in last stage efficiency.	--
10.	Percent decrease in overall turbine efficiency due to throttling losses.	3.9%+
11.	Bearing span will increase.	--

Items 1-4 are all reflected in the decreased first stage performance (Item 8). The actual drop in performance will be greater than 2.0%, due to higher windage caused by increased height, increased width and a smaller fraction of 360° passing steam flow at the guarantee point. Other effects will be less influential.

The additional throttling losses (Item 10) are a direct result of oversizing the first stage nozzle area (Item 1) to pass the required flow at the swing conditions. An increased inlet size, as in Item 6, carries with it not only a higher initial turbine cost but also a large increase in plant piping cost.

The increased last stage height does not carry a performance penalty for this particular case. A last stage performance drop is lost to the entire turbine, while a portion of a first stage loss is recoverable in the form of additional reheat. It is not uncommon for the last stage of an extraction/induction turbine to actually operate with no positive horsepower output at the guarantee, but rather a windage drag due to oversizing.

An increased exhaust casing size also carries with it higher turbine cost, higher expansion joint cost and higher duct work cost.

Items 3, 4, 5 and 7 will all have the effect of increasing bearing span, which adds cost and changes critical speeds. In addition, if the last stage of the turbine becomes a double flow design, the bearing span will increase by 15-25 in., and there will be a significant cost increase.

CONCLUSION

A method has been presented to estimate the differential impact on the design of major turbine parameters due to swings in turbine operating conditions. Impacts will be especially severe if swing conditions are assumed to occur simultaneously, as was done when sizing the last stage with minimum exhaust pressure, maximum horsepower and minimum inlet steam conditions.

Since the methods presented are simple and relatively accurate for estimating differential changes to a turbine design, it is suggested that they could be used to assist in the evaluation of various plant operational schemes before finalizing swing conditions.

Various items suggested for review are given below.

1. Reduced condenser cooling water flow during night and/or winter operation would eliminate the need for very low exhaust pressure swings. The cost of higher steam flows due to the reduced cooling water flow can be compared to the higher steam flows caused by poor performance during normal operation.
2. Reduced horsepower margins will minimize oversizing, which in turn will potentially reduce turbine size and cost and piping size and cost, and maximize turbine performance.
3. Direct piping, with proper valving, to extraction/induction openings may eliminate the need for extremely low pressure swings due to start up conditions.
4. The last and by far the most influential item which should be reviewed is the need for simultaneous swings on operating conditions. Unless otherwise specified, a turbine vendor will assume that swing conditions occur in such a manner as to insure the most conservative mechanical design possible. This is done to prevent today's high warranty costs. Unfortunately, this may result in a turbine which unduly penalizes the user, without his ever knowing.

REFERENCES

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