Study of Steam Turbine Blade to Improve Its Performance

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Abstract: **A steam turbine is an engine which is the use of energy of stem is transformed into work. First the steam heat diffused to a nozzle and is converted into kinetic energy. Then, that kinetic energy is converted into mechanical energy on rotating blades. The usual steam turbine has 4 main parts. The stater consists of a casing and cylinder within which the rotor turns. The turbine has a base or frame and finally there are a nozzle or flow passage which expands the flow the cylinder casing and frame are often combined. The rotor is the rotating part which carries the blades/ buckets. Other parts necessary for proper operations would include as a control system, piping, lubrication system, and separated condenser. There are many different types of turbine.**

Keywords: **Steam Turbine, Modernization, Performance and efficiency, Curve, Design Of steam turbine blade.**

1.Introduction

1.1 Mathematical Model

In a steam turbine blade, the centripetal force is the outer force expected to make a body follow a curved path. Any force (gravitational, electromagnetic, etc.) or combination of forces can task to supply a centripetal force. This steam force is inwards, towards the centre of curvature path of the turbine blade. Here are shown in figure of the blades under discussion.

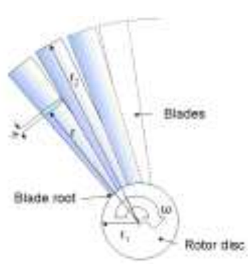


Fig1:- *Steam turbine* **Blade Force Diagram**

In a steam turbine equation for centripetal force is:

$$F = m\omega r^2 \ldots (1)$$

The mass of the moving blade, is the distance of the blade from to the centre of rotation (the radius of curvature) and mωr is the angular velocity of the turbine shaft. In the portion under consideration, we need to account for the fact that the mass of the blade distributed top of its length and the radius of curvature changes along the length of the blade.

Consider a short cross section area of mass δm, having width δr at a distance r from the centre. Then the above equation of the centripetal force δF on this cross section area is following by:

$$\delta F = \delta m\omega^2 r \ldots (2)$$

In practice, a turbine blade tapers in thickness towards its tip, the requirment of solution us assuming the blade to have a constant cross sectional area A (m^2) and material density ρ ($\mathrm{kg/m}^2$)), we can write:

$$\delta m = \rho A \delta r$$

and equation (2) becomes:

$$\delta F = \rho A \omega^2 r \delta r$$

or formally:

$$dF = \rho A \omega^2 r dr \ldots (3)$$

show here the radius of the rotor disc and the distance between the centre of the rotor disc and tip of the blade. Now, integrating equation (3) along to the total length of the blade, the total centripetal force acting on the blade is given by:

$$F = \rho A \omega^2 \int_{r1}^{r2} r dr$$

$$\text{So,}\quad F = \rho A \omega^2 \left(\frac{r2^2 - r1^2}{2}\right) \ldots (4)$$

We can convert the angular velocity, rpm (revolutions per minute) to (rps) radians per second using the following relationship:

$$\omega = \frac{rpm \times 2\pi}{60} \ldots (5)$$

$$\sigma = \frac{F}{Aroot} \ldots (6)$$

These fourmula are be used to find the solution of steam turbine, and rotor speed, the values of the cross-sectional area, density, angular velocity and radius of the rotor, then we calculate the force on a blade. A force calculated, the nominal stress σ on the turbine blade using the following relation.

5.2 Conversion of Kinetic Energy Of The Gas/Steam Into Blade Work

Consider a frictionless blade that turns the steam through 180° and exits with zero absolute velocity. This condition represents the greatest possible conversion of kinetic energy of the entering jet into blade work. We proceed to develop a relation between the absolute velocity of the jet entering the blade, V_1, and the blade speed, V_b. For a given blade speed, this relation will permit us to design a nozzle such that the exiting velocity will provide for maximum energy conversion, or, in different words, maximum efficiency. The positive direction is to the right. Here w, the velocity of the jet force relative to steam turbine blade.

$$V_1 = W_1 + V_b$$

$V_2 = W_2 + V_b$ Because the blade is frictionless, $W_2 = -W_1$.

Furthermore, because energy conversion in the blade is complete, $V_2 = 0$ Substituting and combining equations, we get:

$$V_1 + \cancel{V_2} = \cancel{W_1} + \cancel{W_2} + 2V_b$$

$$V_1 = 2V_b \qquad (1)$$

As we shall see later, the centrifugal force of rotation and the strength of the blade material limit the blade speed. Given the blade speed, however, we can determine the ideal absolute velocity entering the blade.

Actual Nozzle Angle

We must now modify this result to account for the geometry restrictions of a real turbine. In our derivation, the acute angle between V_1 and the tangential direction, called the nozzle angle, is zero. In an actual turbine, because of physical constraints, the nozzle angle must be greater than zero but not so great as to cause an appreciable loss in efficiency. Nor should the angle be so small as to cause an excessively long nozzle that would increase friction and decrease efficiency. "The values used in practice range from 10 to 30 deg., 12 to 20 deg. being common. The larger angles are used only when necessary and usually at the low-pressure end of large turbines."[1] Equation (1), corrected for a finite nozzle angle, α, becomes:

$$V_1 \cos\alpha = 2V_b \qquad (2)$$

Because of disk friction and fanning losses, V_1, is usually increased somewhat, say 10%, over the theoretical value.

Blade Work and Power

First write the Reynolds transport theorem for angular momentum:

$$\frac{D(\vec{r}xm\vec{v})_{Sys}}{Dt} = \frac{\partial(\vec{r}xm\vec{v})_{CV}}{\partial t} + \int_{CS}\left((\vec{r}x\vec{V})\rho\vec{V}\cdot d\vec{A}\right) = \sum\left(\vec{r}x\vec{F}\right) = T_{Shaft}$$

Assuming steady state and steady flow with one entrance (1) and one exit (2) , the equation reduces to:

$$T_{Shaft} = \dot{m}\left[\left(\vec{r}_2 x\vec{V}_2\right) - \left(\vec{r}_1 x\vec{V}_1\right)\right]$$

For the turbine blade, the mean radius is constant between entrance and exit. Furthermore, the tangential component of velocity is the only contributor to torque. The radial and axial components affect bearing loads but have no effect on torque, thus:

$$T_{Shaft} = \dot{m}\left(rV_{\theta 2} - rV_{\theta 1}\right)$$

The shaft work then is:

$$\dot{W}_{Shaft} = \omega T = \dot{m}\omega r\left(V_{\theta 2} - V_{\theta 1}\right)$$

But $V_b = \omega r$, therefore,

$$\dot{W}_{Shaft} = \dot{m}V_b\left(V_{\theta 2} - V_{\theta 1}\right) \qquad (3)$$

On a unit mass basis:

$$w_{Shaft} = V_b\left(V_{\theta 2} - V_{\theta 1}\right) \qquad (4)$$

This result is most easily visualized by constructing entering and leaving velocity triangles.

5.3 Impulse Bladeing Velocity Triangles and Blade Work

Having determined blade speed from strength considerations; nozzle angle from fabrication and efficiency considerations; and V_1 from equation (2); we proceed to construct the velocity triangles. From these triangles we can find the change in absolute tangential velocity and calculate the shaft work.

Entrance Triangle

We first draw a horizontal line representing the tangential direction. Then we construct a vector representing V_1 at angle α , after which we complete the entering triangle using the vector relation:

$$\vec{V}_1 = \vec{W}_1 + \vec{V}_b$$

The angle between the relative velocity and the tangential direction is designated β.

The Exit Triangle

Draw W_2 at angle γ to the tangent. Reducing γ somewhat from the calculated value for β will result in increased blade efficiency.

"Values of γ in use vary from 15 to 30 deg. at high and intermediate pressures and from 30 to 40 deg. at the low-pressure end of the turbine, sometimes reaching 40 to 50 deg. in large turbines where maximum flow area is needed."[2] W_2 is found by

multiplying W_1 by the *velocity coefficient*, k_b, which accounts for friction and turbulence.
The velocity coefficient is a function of the total change of direction of the steam in the blade $\left[180^\circ - (\beta + \gamma)\right]$; the blade width to radius ratio; and the relative velocity and density at blade entrance. Because sufficient data are not available at the beginning of the design, the following empirical formula, adapted from Church for a one inch blade width, is suggested.

$$k_b = \left(0.892 - 6.00x10^{-5}W_1\right)^{1/2}$$

The triangles are easily solved for needed values as follows:

$$V_{\theta 1} = V_1 \cos\alpha$$

$$W_2 = k_b W_1$$

$$V_{a1} = W_{a1} = V_1 \sin\alpha \quad \text{(Axial component)}$$

$$V_{a2} = W_2 \sin\gamma$$

$$W_{\theta 1} = V_{\theta 1} - V_b$$

$$W_{\theta 2} = W_2 \cos\gamma$$

$$W_1 = \sqrt{W_{a1}^2 + W_{\theta 1}^2}$$

$$V_{\theta 2} = V_b + W_{\theta 2}$$

$$\beta = \tan^{-1}\frac{W_{a1}}{W_{\theta 1}}$$

$$V_2 = \sqrt{V_{a2}^2 + V_{\theta 2}^2}$$

5.4 The Reheat Factor and the Condition Curve
Only a portion of the available energy to a stage is turned into work. The remainder, termed *reheat* (q_r), shows up as an increase in the enthalpy of the steam. Because the constant pressure lines on an *h-s* chart (Mollier chart) diverge, the summation of the individual isentropic drops for the total stages is greater than the isentropic drop between the initial and final steam conditions. We account for this variation using a reheat factor, *R*, which has been pre-calculated by various investigators.

$$R = \frac{\sum_i (\Delta h_s)_i}{(\Delta h_s)_{total}}$$

For preliminary design, *R* can be estimated from the following chart taken from Church.

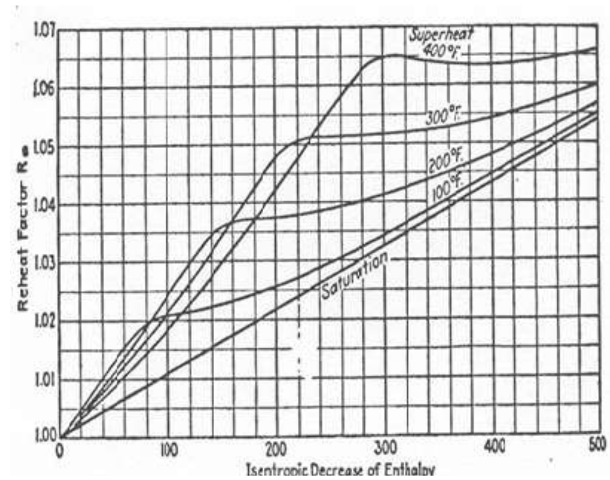


fig2:- Reheat factors for various enthelpy drops and initial superheats, and for an infinite number of stages of 80% stage efficiency

The value from the chart must be corrected for the actual number of stages and stage efficiency.

$$R_n = 1 + (R-1)\left(1 - \frac{1}{n}\right)\left(\frac{1-\eta_s}{0.2}\right) \quad (4)$$

A line connecting the initial and final states, plus the intervening states found by adding the reheat at constant pressure, is called the *condition line.*

5.5 Example Steam Turbine Design
"The design of a steam turbine, like that of any other important machine, involves a judicious combination of theory with the results of experience, governed to a great extent by the commercial element, cost. The progress of a particular design involves a continuous series of compromises between what is most efficient, what will operate most reliably, and what will cost the least."
The following example design is for a pressure-stage impulse turbine. This is one of the most simple and straightforward to design. All of the steam expansion for this turbine takes place in the fixed nozzles, not in the passages with the moving turbine blades.
The turbine is multi-stage. Each stage has a chamber with a single impulse turbine in it, with all wheels on the same shaft. Each individual chamber or stage receives the steam through groups of nozzles. The pressure drop for the turbine is divided into as many steps as there are chambers, and each is considered to be a pressure stage. The last stage of the turbine discharges to the condenser.
The client will usually specify steam conditions, condenser vacuum, rotational speed and capacity in kilowatts or horsepower. The client may also specify a maximum cost and minimum efficiency.

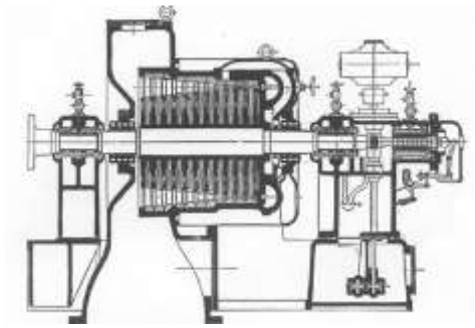


fig3:- Pressure-stage Impulse turbine

C s and blading of a turbine given the following specifications:

Power delivered at the shaft coupling	5000 kW
Revolutions per minute	2400 rpm
Maximum blade speed	570 ft/s
Initial steam pressure	150 psia
Initial steam temperature	540° F
Condenser pressure	1 in Hg

Select a nozzle angle equal to 20°, then:

$$\left(V_1\right)_{ideal} = \frac{2V_b}{\cos 20^\circ} = 1213\, ft/s$$

Increase this value by about 10% (the example uses 11.86% to follow Church) to account for disk friction and fanning.

$V_1 = 1357\, ft/s$

The Entrance Triangle

$$V_{\theta 1} = V_1 \cos\alpha = 1357 \cos 20^o = 1275\, ft/s$$
$$W_{\theta 1} = V_{\theta 1} - V_b = 1275 - 570 = 705\, ft/s$$
$$W_{a1} = V_{a1} = V_1 \sin\alpha = 1357 \sin 20^\circ = 464\, ft/s$$
$$W_1 = \sqrt{W_{a1}{}^2 + W_{\theta 1}{}^2} = \sqrt{464^2 + 705^2} = 844\, ft/s$$
$$\beta = \tan^{-1}\frac{W_{a1}}{W_{\theta 1}} = \tan^{-1}\frac{464}{705} = 33.35^\circ$$

The Exit Triangle

$$k_b = \left(0.892 - 6x10^{-5} W_1\right)^{1/2} = \left(0.892 - 6x10^{-5} x844\right)^{1/2} = 0.917$$
$$W_2 = k_b W_1 = (0.917)(844) = 774\, ft/s$$

Assume that $\gamma = \beta = 33.35°$

$$W_{\theta 2} = W_2 \cos\gamma = -774 \cos 33.35^0 = -647\, ft/s$$
$$V_{\theta 2} = W_{\theta 2} + V_b = -647 + 570 = -76.5\, ft/s$$
$$W_{a2} = V_{a2} = W_2 \sin\gamma = 774 \sin 33.35^\circ = 426\, ft/s$$
$$\delta = \tan^{-1}\frac{V_{a2}}{-V_{\theta 2}} = an^{-1}\frac{426}{-(-76.5)} = 79.8^0$$

Blade Work per Unit Mass

$$w_b = -V_b\left(V_{\theta 2} - V_{\theta 1}\right) = -\frac{(570)\cancel{ft}}{\cancel{s}}\frac{(-76.5-1275)\cancel{ft}}{\cancel{s}}\frac{Btu}{(778)\cancel{ft}\cdot \cancel{lb_f}}\frac{\cancel{lb_f}\cdot \cancel{s}^2}{lb_m (32.2)\cancel{ft}}$$
$$w_b = 30.75 Btu/lb_m$$

Actual Energy Available to Blade

$$\left(A.E.\right)_{b-actual} = \frac{V_1^2}{2} = \frac{(1357)^2 \cancel{ft}^2}{2\cdot \cancel{s}^2}\frac{Btu}{(778)\cancel{ft}\cdot \cancel{lb_f}}\frac{\cancel{lb_f}\cdot \cancel{s}^2}{lb_m (32.2)\cancel{ft}} = 36.75 Btu/lb_m$$

Blade Efficiency

$$\eta_b = \frac{w_b}{\left(A.E.\right)_{b-actual}} = \frac{30.75}{36.75} = 0.836$$

Nozzle Velocity Coefficient, k_n

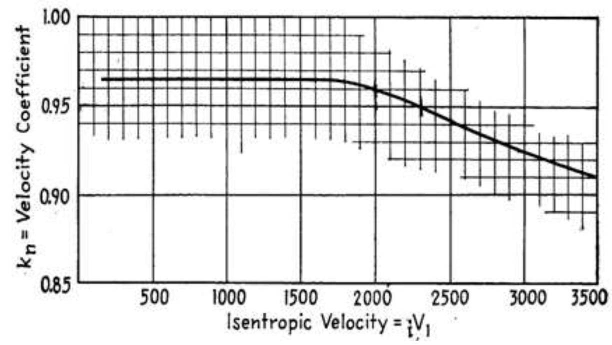


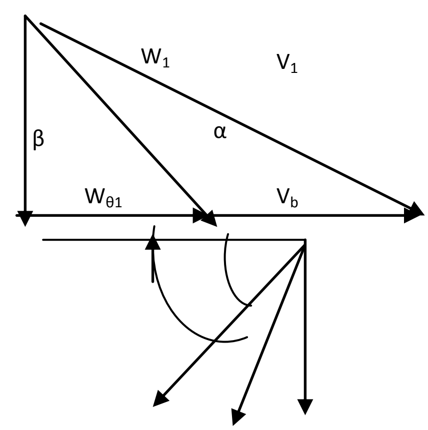
fig4:- Nozzle velocity coefficient for superheated steam

experimental results was adapted from Church.

$$k_n = 1.021 - 0.164x + 0.165x^2 - 0.0671x^3 + 0.0088x^4$$

Where $x = V_{s1}/1000$

Examining Figure 3-20[3] above, it can be seen that a good starting value for k_n is 0.965, which is confirmed by the calculation below.



Ideal (Isentropic) Nozzle Exit/Blade Entrance Velocity

$$V_{s1} = \frac{V_1}{k_n} = \frac{1357}{0.965} = 1406\, ft/s$$

Nozzle Efficiency

$$\eta_n = \frac{V_1^2/2}{V_{s1}^2/2} = \frac{k_n^2 V_{s1}^2/2}{V_{s1}^2/2} = k_n^2 = (0.965)^2 = 0.931$$

Combined Nozzle and Blade Efficiency

$$\eta_{nb} = \eta_n \eta_b = (0.931)(0.836) = 0.778$$

Stage Efficiency

Assume an average loss from disk friction and fanning of 4% and from leakage of 1.5 %. Correcting for these effects gives:

$$\eta_{st} = \eta_{nb}\left(1-\left(Friction/Fanning + Leakage\right)\right) = (0.778)\left[1-(0.04+0.015)\right] = 0.735$$

Let $\eta_{st} = 0.73$.

This is a provisional value. It will be modified as the design proceeds and more precise information becomes available.

Number of Stages

Ideal Available Energy to Blade

$$\Delta h_s = \frac{V_{s1}^2}{2} = \frac{(1406\, \cancel{ft}/\cancel{s})^{\cancel{2}} \cdot \cancel{lb_f} \cdot \cancel{s}^2 \cdot Btu}{(2)(32.2\, \cancel{ft} \cdot lb_m)(778\, \cancel{ft} \cdot \cancel{lb_f})} = 39.5\, Btu/lb_m$$

Use the enthalpy at the given inlet conditions to the turbine and that of the inlet to the condenser to get the total isentropic drop in enthalpy:

$$(\Delta h_s)_{total} = (1293.7 - 902.9) = 390.8\, Btu/lb_m$$

$R_\infty = 1.0465$ (Reheat from above graph)

Trial Number of Stages, *n*

$$n = \frac{(\Delta h_s)_{total} R_\infty}{\Delta h_s} = \frac{(390.8)\, Btu/lb_m (1.0465)}{(39.5)\, Btu/lb_m} = 10.35$$

Select 10 stages; then correct for actual value of reheat:

$$R = 1 + (1.0465 - 1)\left(1 - \frac{1}{10}\right)\left(\frac{1-0.73}{0.2}\right) = 1.0565$$

Trial Isentropic Drop Per Stage

The enthalpy drop per stage is the total amount of actual enthalpy change in the turbine, including the reheat, divided by the number of stages.

$$\Delta h_s = \frac{(390.8)\, Btu/lb_m (1.0565)}{(10)} = 41.3\, Btu/lb_m$$

The trial value is 1.8 Btu/lb_m greater than 39.5 and will cause a slight increase in V_1 which will be accounted for later.

Stage Reheat, q_r

Here we determine ***the*** amount of heat leak at each stage.

$$q_r = \Delta h_s - \Delta h_s \eta_{st} = (41.3)(1-0.73) = 11.15\, Btu/lb_m$$

If the desired final pressure (0.491 psia) is not reached, a new trial Δh_s and corresponding q_r are found and additional iterations run as needed. To estimate the required change, divide the error in enthalpy by the number of stages and by the stage efficiency and then add or subtract this quantity from Δh_s to get a new trial value. After the correct value for Δh_s is found, the final velocity triangles can be constructed and corresponding values calculated.

The corrected value for V_1 is given by:

$$V_{s1} = \sqrt{2\Delta h_s} = \sqrt{\frac{2(41.3\, Btu/lb_m)(778\, ft \cdot lb_f)(32.2\, ft \cdot lb_m)}{Btu \cdot lb_f \cdot s^2}} = 1438.49\, ft/s$$

$$V_1 = k_n V_{s1} = (0.965) \cdot 1438.49 = 1387\, ft/s$$

From the velocity triangle, W_1 is 873 ft/s and the blade velocity coefficient is:

$$k_b = \left[0.892 - (6x10^{-5})(873)\right]^{1/2} = 0.916$$

This value is close enough to the previously calculated value of 0.836. Proceeding:

Total Internal Work per Pound of Steam

$$w_i = n \cdot (\Delta h_s - q_r) = (10) \cdot (41.3 - 11.15)\, Btu/lb_m = 301.5\, Btu/lb_m$$

Internal Efficiency of the Turbine

$$\eta_i = \frac{301.5\, Btu/lb_m}{390.8\, Btu/lb_m} = 0.7715$$

Mechanical Losses

Church states the following rough rule for total mechanical losses, including bearing friction and gland, pump and governor resistances:

Mechanical loss in per cent at normal rating

$$= \frac{4}{\sqrt{kw/1000}} = \frac{4}{\sqrt{5000/1000}} = 1.8$$

Assume a radiation loss of about 0.2 %, the combined mechanical and radiation losses amount to about 2 %.

Engine Efficiency, η_e

$$\eta_e = (0.7715)(0.98) = 0.756$$

Ideal Steam Rate

The ideal steam rate represents the mass of steam required to produce a single kilowatt of power.

$$ISR = \frac{(3413)\, \cancel{Btu}}{kWh} \frac{lb_m}{(390.8)\, \cancel{Btu}} = 8.73\, lb_m/kWh$$

Brake Steam Rate

The brake steam rate corrects the ideal steam rate for the inefficiencies of the engine (turbine).

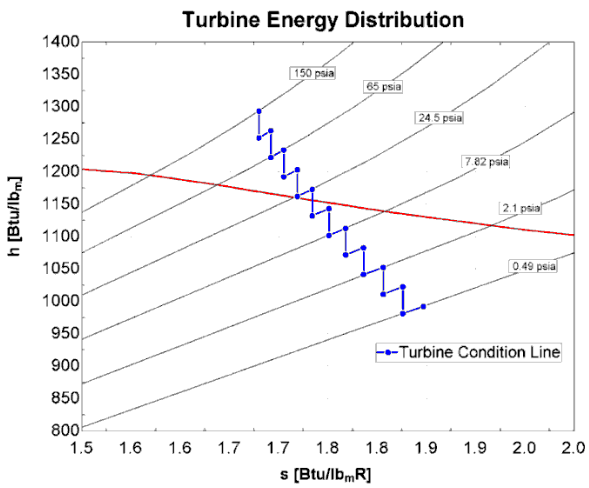
$$BSR = ISR/\eta_e = (8.73\, lb_m/kWh)/0.756 = 11.55\, lb_m/kWh$$

Turbine Mass Flow Rate

$$\dot{m} = \frac{(11.55)\, lb_m}{\cancel{KWh}} \frac{(5000)\, \cancel{KWh}}{(3600)\, s} = 16.04\, lb_m/s$$

5.6 Mollier Diagram

A Mollier chart, or an *h-s* diagram, offers a unique description of the thermodynamic interactions occurring within the turbine. The condition line details the thermodynamic state progression and is usually drawn superimposed on the Mollier chart. In order to construct this chart, it is useful to detail the state at each of the various stages. This is most easily achieved by constructing a table of the stage properties.



Mollier chart (h-s diagram) showing energy distribution within the turbine.

To construct the table, and from there the Mollier chart, one must understand how the thermodynamic state changes between stages. All the pertinent information has already been determined. It is simply a question of organization. The state table for the example in progress is given on the following page. Using a tabular format, we proceed in turn for each stage to subtract Δh_s from the entrance enthalpy and then add back the reheat to determine the stage end point. The needed thermodynamic state properties, including specific volume, are found as the process proceeds. Knowing the new value of the enthalpy and assuming isentropic expansion, it is possible to determine the pressure at the end of the nozzle.

Initially, the state is fixed and set by the inlet steam pressure and temperature. It is possible to determine the entropy of the steam directly. Then as the steam flows through the first stage nozzle, it goes through an isentropic expansion. It is here that the enthalpy of the fluid drops. The amount of enthalpy change at each stage is considered constant, and has already been determined.

Stage	**Part**	**Enthalpy (BTU/lb_m)**	**Entropy ($BTU/lb_m\cdot R$)**	**Pressure (psia)**
1	Nozzle	1293.70 -41.30 (□h_{trial})	1.680	150
	Wheel	1252.40 +11.15 (q)	…	100.5
2	N	1263.55 -41.30	1.692	…
	W	1222.23 +11.15	…	65.0
3	N	1233.40 -41.30	1.705	…
	W	1192.10 +11.15	…	40.9
4	N	1203.25 -41.30	1.719	…
	W	1161.95 +11.15	…	24.5
5	N	1173.10 -41.30	1.734	…
	W	1131.80 +11.15	…	14.0
6	N	1142.95 -41.30	1.751	…
	W	1101.65 +11.15	…	7.82
7	N	1112.80 -41.30	1.768	…
	W	1071.50 +11.15	…	4.07
8	N	1082.65 -41.30	1.786	…
	W	1041.35 +11.15	…	2.10
9	N	1052.50 -41.30	1.806	…
	W	1011.20 +11.15	…	1.02
10	N	1022.35 -41.30	1.826	…
	W	981.05 +11.15	…	1.00 (in-Hg)
End Point …		992.20	1.847	

Table:- Thermodynamic state table for the stages of the steam turbine

Steam then flows across the vanes on the wheel where it is reheated due to friction. This process occurs under constant pressure, or isobaric conditions. Thus, the increase in energy due to the heating is added to the previous value of the enthalpy. The constant pressure assumption fixes the state, and the resulting value of entropy can be determined at the entrance to the nozzle for the next stage. This procedure is continued and the values are tabulated until the total number of stages has been completed.

Once the values of the enthalpy and the entropy are determined at each stage, it is possible to plot these values on the Mollier chart. These plotted values create the condition line and indicate the state progression of the steam through the various turbine stages. The appropriate Mollier chart for this example is given on page 14. EES will produce a property plot automatically. The tabulated enthalpy and entropy values can then be superimposed on the plot using the "Overlay Plot" option in the Plot menu.

After a presentation of the functionality of steam turbines of thermal power plants, we present the method of enthalpy difference used for the calculation of the steam turbine efficiency. This is why we present a stratification survey by circuit, by equipment and by organ. The objective is to determine the possible reasons for the deterioration of the heat rate of a thermal power plant at the level of a steam turbine. This survey concerns particularly the application of the causal analysis for determining the different losses at the level of a steam turbine.

Refrence

1. Zachary Stuck and Stanley Schurdak, "Steam Turbine Blade Design" Twelfth Annual Freshman Conference 2214United state of America, Conference Session B6, 14th APR 2012.
2. T. Mısek ,Z.Kubın "Static and Dynamic Analysis of 1220 mm Steel Last Stage Blade for Steam Turbine", Applied and Computational Mechanics 133–140,18th Nov 2008.
3. Tulsidas. D , Dr.Shantharaja. M, Dr. Kumar. K "Design modification for fillet stresses in steam turbine blade" International Journal of Advances in Engineering & Technology (IJAET) Vol.III, Issue I, January-March, 2012. Pp. 343-346.
4. A book of "Analysis of Fretting Fatigue Strength of Integral Shroud Blade for Steam Turbine" Proceedings of the Internati onal Conference on Power Engineering 2007.
5. B. Stanisaa, V. Ivusicb "Erosion behavior and mechanisms for steam turbine rotor blades" International Journal of Advances in Engineering & Technology (IJAET), Vol.II , Issue II, April-June, 2011, 110-117.
6. Anestis Kalfas "Turbine Blading Performance Evaluation Using Geometry Scanning and Flow field Prediction Tools" Journal of Power and Energy Systems, Volume 2, Issue 6, pp. 1345-1358 (2008).
7. M. Ahmad , M. Schatz, M.V. Casey "Experimental investigation of droplet size influence on low pressure steam turbine blade erosion "Volume 303, Issues 1–2, 15 June 2013, Pages 83–86
8. Sandeep Soni, P.Pandey "Erosion behaviour of steam turbine blades of glass-epoxy" International Journal of Advances in Engineering & Technology (IJAET),Vol.II, Issue II, April-June, 2011, pp.110-117.
9. E. K. Sevidova,V. M. Matsevityi,I. B. Kazak,K. V. Vakulenko "Evaluation of the protective properties of multilayer coatings for steam turbine blades" IJAET Volume 4 Issue 2.
10. Kiyoshi Segawa, Yoshio Shikano, Kuniyoshi Tsubouchi and Naoaki Shibashita. "Development of a Highly Loaded Rotor Blade for Steam
11. Turbines." Japan Society Mechanical Engineering (JSME) International Journal Series B, Vol. 45, No. 4 (2002),pp.881-890.

