POWER PLANT SYSTEM ENGINEERING

Lec 10: Performance Analysis of Steam Turbines

Dear learners, greetings from IIT, Guwahati. We are in the MOOCs course Power Plant System Engineering, Module 2, Vapor Power Systems Part 2. Till this point of time we have covered the thermodynamic aspects of a vapour power plant followed by the expansion of steam in steam turbines. So, there we had two important concepts of steam turbines, one is through impulse principle, other is through reaction principles. So, based on that we have classified impulse turbines and reaction turbines. So, in this lecture we are now going to see some of the performance indicators for steam turbines.

There is one particular concept what you call as axial thrust. Normally this axial thrust is not recommended, but it has to be there and it has some practical significance as far as handling or fixing the rotor is concerned. Then we will see some of the design aspects of steam turbine blades. We have impulse turbines, we have reaction turbines.

Necessarily the design features of both the turbine blades are different because impulse turbines are mainly intended for high pressure steams and reaction turbines mainly deal with moderate steam velocities. Then having said this we will now look into some of the losses that is normally encountered in the steam turbines. Although losses cannot be completely ignored, but these losses needs to be minimized. So, how to minimize the loss that is another feature for steam turbines. So, we will discuss them one by one.

So, let us briefly understand what steam turbines we have learnt so far. So, in our previous lectures we have excessively discussed about single stage impulse turbines in which we have the rotating blades or normally this we call as a wheel and these blades are mounted on a rotor and we have steam nozzle that feeds the steam to this rotor or blades. So, as a results it rotates. Then other categories of turbines includes velocity compounded impulse turbine and pressure compounded impulse turbines. So, many a times a single nozzle and a set of moving blades does not solve the purpose. So, we require compounding and this compounding can be done in two aspects either you can incorporate multiple nozzles for each set of moving blades and such a turbine is called as a pressure compounded turbine or Rateau turbine. The advantage here is that since each nozzle supply the same pressure, so moving blades are supplied with their individual nozzles. So, this is something like multi stage impulse turbine. Another way of looking at the compounding is the velocity compounded turbine which means that single nozzle supplies the steam to the set of moving blades and after the exit from the moving blades, the steam is again reoriented to another set of moving blades through a passage normally called as stationary blades or many a times you also call this as a guide vanes. So, through this process we can draw this pressure and velocity characteristics along the passage through which the fluid moves.

And the last turbine that we discussed is the reaction turbines and this reaction turbine is we call as a Parson turbines where the pressure drop of the fluid has happened in stages and in the initial category the pressure drop is higher and towards the later part the slope of this pressure curve is less. So, each of these turbines have different advantages and disadvantages. However, the main common feature for all kind of steam turbine is to find their performance indicator. For impulse turbine which is normally referred as a *de Laval* turbine, we have derived the following expressions.

Work done per unit time: $\dot{W} = \dot{m} V_B \left(V_{s1} \cos\theta - V_{s2} \cos\delta \right)$

Initial kinetic energy of jet : $KE = \frac{1}{2} \dot{m} V_{s1}^2$; Blade speed ratio: $\frac{V_B}{V_{s1}}$

Diagram / Blade efficiency : $\eta_b = \frac{\dot{W}}{KE} = 2\left[\left(\frac{V_B}{V_{s1}} \right) \cos\theta - \left(\frac{V_B}{V_{s1}} \right) \left(\frac{V_{s2}}{V_{s1}} \right) \cos\delta \right]$

Optimum plate velocity for maximum power:

$V_{B,opt} = \frac{V_{s1} \cos\theta}{2}$; $\dot{W}_{\max} = 2 \dot{m} V_{B,opt}^2$; $\eta_{b,\max} = \frac{\dot{W}_{\max}}{KE} = \left(\cos\theta \right)^2$

Velocity coefficient: $k_v = \frac{V_{r2}}{V_{r1}}$; Stage efficiency : $\eta_s = \frac{\dot{W}}{\Delta H_s} = \frac{\dot{W}}{\dot{m}\left(\Delta h_s \right)}$

ΔH_s : Total enthalpy drop of the fluid for whole stage

First one is how much work is being done per unit time. We have also calculated the kinetic energy of the jet that is impinged from this nozzle, then we can find out the diagram/ blade efficiency and in the single stage turbine, for a given velocity and angle θ at which the fluid jet impinges, we have derived a particular speed at which blade must operate, that is optimum blade speed and normally all the turbines operate at their optimum speed, because of that we get the maximum work and then we can find out the blade efficiency.

Another important feature is that if we draw the velocity triangles from inlet to outlet, there may be a possibility that the relative velocities are not equal. So, in that case we define a parameter velocity coefficient. In fact, this is nothing, but the loss that is encountered when the fluid flows through this passage. Then from each one stage whatever work we derived and if you divide with respect to the total enthalpy drop then we will find out the stage efficiency.

And in similar philosophy when you say reaction turbine, we specifically referred one particular turbine which is known as Parson Turbine. Since complete reaction mode of operating turbine is impossible, we say this Parsons turbines is 50% reaction turbine, which essentially means to us is that a 50% enthalpy drop takes place in the fixed blade and rest 50% drop takes place in the moving blades. So the basic difference is that we do not have a nozzle rather this fixed blades or we call as guide vanes serve the nozzle action for the fluid, that is directed to the moving blade. So, with this philosophy we can have a full steam admission around the rotor periphery and of course, these blades are curved in opposite directions so that the fluid passage can be aligned in a particular direction. We can trace the path of steam in the picture where we can see that it goes along the fixed and moving blade that is the steam enters from the set of fixed blade and then leaves from the moving blade. So, accordingly the pressure drop takes place and absolute velocity changes in each of the blades.

So, here also in same sense, we can find out the following expressions.

Rate of work or power, $\dot{W} = \dot{m}V_B(2V_{s1}\cos\theta - V_B)$

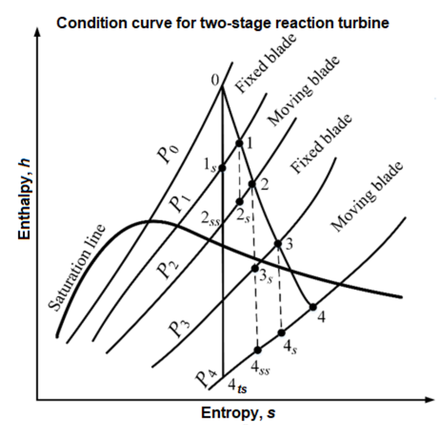
V_B:Bladevelocity;V_{s1}: Absolute fluid velocity leaving the nozzle

Optimum blade velocity, $V_{B,opt} = V_{s1}\cos\theta$;

Maximum work: $\dot{W} = \dot{m}V_{B,opt}^2 = \dot{m}(V_{s1}\cos\theta)^2$

Velocity coefficient: $k_v = \dfrac{V_{r2}}{V_{r1}}$; Blade speed ratio: $\dfrac{V_B}{V_{s1}}$

Since in a reaction turbine we have moving blades as well as fixed blades, so we define the efficiency in two parts one is fixed blade efficiency other is the moving blade efficiency. Here we can see that moving blade efficiency is nothing but the work that is being developed to the energy which is associated with kinetic energy of the jet plus the enthalpy drop in the moving blades. In same sense also we can calculate the nozzle efficiency then we also can get the stage efficiency. We can refer this conditioning curve for a two stage turbine where we can plot the enthalpy and entropy data and from this we can find all these enthalpy values and accordingly all these efficiency parameters can be calculated.



Moving blade efficiency: $\eta_B = \dfrac{\dot{W}}{\dot{m}\left[\left(\frac{V_{s1}^2}{2}\right)+\Delta h_{ms}\right]} = \dfrac{\dot{W}}{\dot{m}\left[\left(\frac{V_{s1}^2}{2}\right)+(h_1-h_{2s})\right]}$

Fixed blade/Nozzle efficiency: $\eta_N = \dfrac{\left(\frac{1}{2}\right)(V_{s1}^2 - V_{s0}^2)}{\Delta h_{fs}} = \dfrac{h_0 - h_1}{h_0 - h_{1s}}$

Stage efficiency, $\eta_{stage} = \dfrac{\dot{W}}{\dot{m}\Delta h_s} = \dfrac{\dot{W}}{\dot{m}(h_0 - h_{2ss})}$

Degree of reaction: $\Lambda = \dfrac{\text{Enthalpy drop in moving blade}}{\text{Enthalpy drop in a stage}} = \dfrac{h_1 - h_2}{h_0 - h_2}$

Isentropic enthalpy drop across fixed blade: $\Delta h_{fs} = h_0 - h_{1s}$ (first stage)

Isentropic enthalpy drop across moving blade: $\Delta h_{ms} = h_1 - h_{2s}$ (first stage)

Isentropic enthalpy drop across entire stage: $\Delta h_s = h_0 - h_{2ss}$ (both stages)

One more important factor, normally we have not referred so far is the reheat factor. I have also mentioned in my previous lectures, if you can recall, this conditioning curve or normally the Molyer diagram as shown in the figure. We have seen that when you go along these constant pressure lines, these constant pressure lines diverge when you are going for higher enthalpy. Since they diverge, we can take the advantage of higher enthalpy drops at higher pressure stages. So, for that reason from the design point of view, we operate this turbine at much higher pressure and temperatures and mostly we operate in the superheated regions. Since we take this advantage of divergence in the pressures at higher stages, we can find a ratio of the individual enthalpy drops to the enthalpy drop of the turbine section or for the entire stage and that is referred as the reheat factor (R_h).

So, referring to the conditioning curve if you want to find out the reheat factor and your initial state is 0 and final state is P_4 that is h_{4ts} and if the process has to take place in an isentropic manner, the enthalpy drop would be $h_0 - h_{4ts}$. Now, we are going in stages so, one fixed plate and moving plate gives one stage, another set of fixed plate and moving plate gives another stage. So, for each stage we can find out the isentropic enthalpy drops. So, if you keep on calculating them then we will find out the reheat factor.
Two stages: $\eta_{s2} = \dfrac{h_0 - h_4}{h_0 - h_{4ts}}$; One stages: $\eta_{s1} = \dfrac{h_2 - h_4}{h_2 - h_{4ss}}$

Since, $\left(\dfrac{h_0 - h_4}{h_0 - h_{4ts}}\right) > \left(\dfrac{h_2 - h_4}{h_2 - h_{4ss}}\right) \Rightarrow \eta_{s2} > \eta_{s1}$

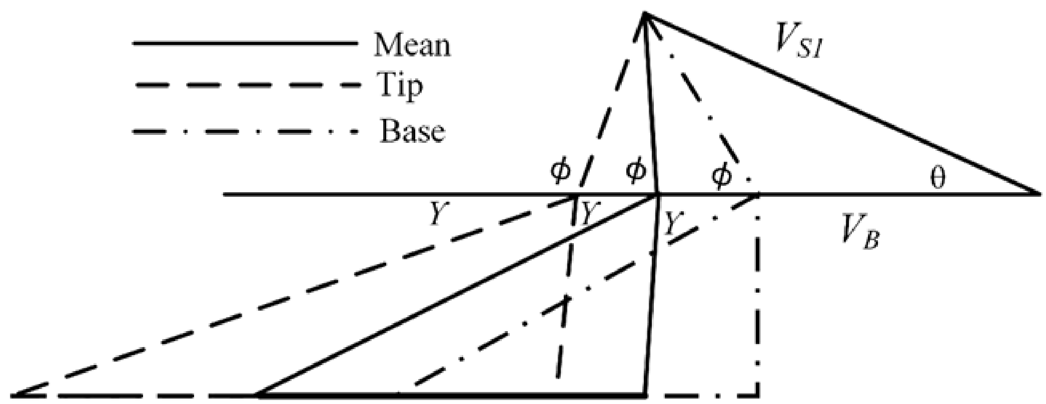
Reheat factor: $R_h = \dfrac{(h_0 - h_{2ss}) + (h_2 - h_{4ss})}{(h_0 - h_{4ts})}$; $R_h = 1\text{to}1.065$

So, this total enthalpy drop that happens in each stage which gets added and it is divided by had this process been isentropic, what would have been the enthalpy drop that is $h_0 - h_{4ts}$.

Then we will move on to next important concept i.e., axial thrust. So, normally what we say in our analysis is that if you refer a simple velocity diagram of a single stage impulse turbine, we see that we have a blade velocity V_B and this blade velocity is achieved when your steam jet impinges the blade at an angle θ and leaves at an angle δ with velocity V_{s2}. So, as a result we have this V_B velocity which is nothing, but the total whirl velocities equal to the combination of inlet and outlet velocity triangles. So, there are two parts - one is whirl velocity V_w and other is axial velocity V_a and this V_a velocity direction is almost perpendicular to V_w, but only difference is that if relative velocities are not equal that is $V_{r1} \neq V_{r2}$ then this axial velocity will come into pictures. So, this part we will discuss while solving some problem- how that axial component arises. However, in pure impulse turbine blades there is no axial thrust, so due to this axial thrust there is no resulting work. So, the entire work comes by virtue of this whirl velocity V_w.

In a similar sense also for reaction turbines the blades are designed in such a way that means, this blade passage, fixed & moving blades are designed in such a way that the fluid always passes through this sequence of fixed and moving blades in an organized manner as a results we try to ensure that there is no axial thrust due to change in the momentum of the fluid stream. However what happens, since the reaction turbines are operated at different pressure stages, due to which some axial thrust component arises in the low pressure stages. Another important point is that, this moving blades are kept in the rotor and to keep this rotor in running condition we also require some kind of balancing force, so, for that reason in some sense, axial thrust value should be as minimum as possible depending on the design of the turbo machine components.

Then we will try to see some design aspects of steam turbine blades. So, till this point of time we have been talking about the shape of the blades and velocity triangles. So, one of the critical features is that we have the angle θ, angle δ which relates the steam velocity at the inlet and exit of the blade. We also have relative velocities at the inlet and exit of the steam and we also have another important angles, $\emptyset$ & γ. And all these angles has to be synchronized in such a way that we require a fixed blade velocity V_B. And to achieve this fixed blade velocity and minimize this axial thrust, the angles are designed in such a way that we have highest velocity coefficients, at the same time we have fixed blade velocity. And to do that, these angles are adjusted.



For example, let's look at one particular reaction turbine & for which this particular velocity triangle is drawn here. The steam at velocity V_{s1} is impinging the blade at an angle θ , for this θ , we already have fixed δ & γ . Now in this particular case, V_{s1}, V_B & V_{r1} forms the inlet triangle. The line representing the relative velocity, V_{r1} can vary with respect to different values of $\emptyset$. So, that means it is possible to control the relative velocity at the exit velocity triangle with different $\emptyset$ values. So, this is one aspect. The second aspect is that now across this blade we may have different velocities that means if you take along the mean line of the blade or at the tip of the blade or at the base of the blade, we may have different corresponding exit relative velocities as shown in the figure. So, in other words we can say that whatever velocity triangles have been drawn so far, those are drawn for mean blade speed. However, in reality the blade speed varies from the tip to the base. And because of these reasons we have issues like partial admission of the fluid and diameter is also not constant because the diameter varies from the base to tip. So, because of this reason we require the need of the blade designs and this velocity diagram has to be drawn with respect to optimum conditions. So, considering these aspects what we all need to do is that we have to design our blades and then the shape of the blade has to change from tip to the base. When you put these diagrams in real scenarios, then the generated configuration is known as twisted blade or wrapped blade or vortex blades. Because the blade velocity varies from tip to base and for our analysis we always assume the mean value (shown in velocity triangle diagram) for angle $\emptyset$ and based on this all the designs, all the thermodynamic calculations are made.

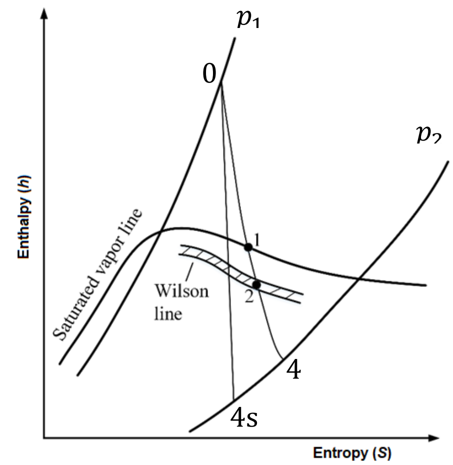
Then we will move on to next important topic which is losses in the turbines. The arrangement of a particular turbine to shoot a specific purpose requires many important consideration & that includes the losses. Sometimes losses are inevitable, but at the same time as a designer we must ensure that losses are minimized.

And the thermodynamic sense, one of the important loss is due to an irreversible phenomena known as super saturation. We will come back to this theory of the super saturation. Apart from this thermodynamic loss, the other internal losses that can happen are-

1. Friction losses in the nozzles, blades, discs which rotates the fluid.
2. Then we have leakage loss during fluid admissions in stages, seals & glands.
3. Then we have moisture losses,
4. Then we have residual velocity losses
5. In addition to that we also have some miscellaneous losses that occurs due to heat transfer; there could be mechanical and electrical losses.

But our main important losses are super saturation which is a thermodynamic loss and other losses are losses due to friction, leakage, moisture, residual velocity. And in fact, we will try to see that some of the losses we have to retain for a better performance of the plant. So, let us see the first loss concept which is super saturation.

So, for super saturation let us understand this Mollier diagram in which we have plotted two pressure lines- one is high pressure, p_1 and other is low pressure p_2 and the steam is being expanded starting at point 0, then encounters many stages, finally reaches to point 4s and 4. But during this expansion process, the fluid encounters various enthalpies and on this diagram if you superimpose the saturated vapour line, we can see that 0- 4s is the enthalpy isentropic drop and 0-4 is the actual drop. And during this passage the fluid encounters some intermediate stages- one on the saturated vapor line at point 1, which says that in the vicinity before the point 1 the fluid is vapor and after this point 1the fluid is liquid.



So, ideally this is what the thermodynamic analysis says, but in reality it does not happen. That means, if you carry out such a high pressure or superheated steam and try to expand it in this manner; then instantly the fluid does not come back or steam does not come back from vapor stage to liquid stage immediately. So, there is a condition that exist we call it as Metastable equilibrium. That means instantly it does not change its state rather it takes some time to respond and while doing that the coordinates are changed. So, in actual scenario the fluid becomes liquid at point 2. So, instead of the vicinity of point 1 it now becomes liquid at point 2. So, essentially this means that there is a loss in the enthalpy values between this point 1 to 2.

That means, the steam is actually becomes liquid at state 2, but it should have become liquid at state 1 and this is a metastable situation and this happens in real sense. So, likewise we can draw different pressure lines, we can create a band which is called as Wilson line for each enthalpy drop. This Wilson line will tell you the locus of all source curves which says that in reality where the steam becomes liquid and this steam is called as super saturated or undercooled steam. The locus of point 2 at varied pressure, forms a band or zone which is called as Wilson line. Now, to counter this losses due to super saturation we define a term which is called as a degree of super saturation or degree of undercooling. The ratio of actual pressure to the pressure corresponding to lower temperature is called as the degree of super saturation. Of course, these two points are at two different pressures and we can take this ratio and that is called as a degree of super saturation or degree of undercooling.

And after point 2 the thermodynamic equilibrium is established, which is detected by the pressure volume diagram at point 2. And this is the phenomena where initial condensation results in liquid droplets of small diameters with large curvature and subsequently they grow to a larger one. Since this particular expansion process is across the Wilson line, it accounts for the sudden condensation with release of enthalpy of vaporization, but at the same time it results sudden pressure rise with reduction in the specific volume and velocity. So, in steam power plant point of view such rapid expansion in turbine and nozzles is normally referred as condensation shock. Shock is nothing, but a phenomena across which the changes are very drastic and it is an instantaneous changes and whatever changes that happens in an irreversible manner. So, more on the details of the super saturation, I will cover again when we discuss about the steam nozzles.

Then moving on to other losses, we have another type of losses which is called fluid friction. Fluid friction is very common and whether you say it is impulse or reaction turbines, I have already emphasized that the fluid has to follow certain path or the enclosed passage between the fixed and moving blades in a successive manner. And since it is encountering different fixed and moving blades, it is expected that there has to be a losses. And since the purpose at which the velocity triangles are designed, same quantity of the fluid may not enter into the subsequent stages and that also happens in either in a compounding mode for pressure as well as velocity compounded turbines. Since they are entering in different stages and there could be a possibility of admission loss as the steam passes from the nozzle to wheel. And again in in case of rotating blades, after it crosses the moving blades, not necessarily same quantity of the fluid again enters to the fixed blades. That means the fluid moving blade churns the fluid, hence we call this as a fanning losses. So, basically for fixed blade we have admission losses and for moving blades we have fanning losses.

And the net effect or net loss of this combined things like compounding either pressure or velocity compounding, that comes out from the fixed and then moving blades is called as blade windage loss. The other type of situation that may arise is the nature of the flow. So, what happens is that normally when you go for high pressure steam or high temperature steam and then this expansion takes place, there is a possibility that fluid is no longer becomes near laminar. It becomes turbulent. This turbulence is also available because the entrance angles of the steams may not suit to the design loads. So, when we have laminar or turbulent flows that passes through the blades then we can encounter the boundary layers. So, we may have a laminar or turbulent boundary layers. If turbulent boundary layers are encountered then we can say there is a loss that happens due to the nature of the flow. However, looking at the kind of expansion that takes place from high pressure steam, the fluid friction losses is almost close to 10% from all available energy sources.

Another important phenomena is leakage. So, leakage is very common when you expand the steam from high pressure sides. And of course, except single stage, for all other things like moving blades, nozzles, reaction turbines, array of fixed and moving blades, there is possibility that the steam leaks out, either from the fixed blades or from the nozzles. As this leak that can happen from shaft bearings or this leak can happens also from the outside of the turbine, so we require proper sealing from them. And because of these reasons, when you are looking at very high pressure steams, whether it is a reaction or impulse turbine, there is a possibility that leak can happen from any of the stages. And in fact, higher is the pressure drop (greater the ratio of tip clearance to the blade height), greater is the leakage. And this leakage loss, from a designer point of view, normally contributes 1% of the total energy available to the turbine.

Moving further another leakage that can happen through the external glands, then leakage loss can also be part of the blade tips and casing. And the leakage are higher when the pressure is high. So, that is the reason, it is mainly advisable to use impulse stage at high pressure region and reaction stages at low pressure region. So, at low pressure the friction losses becomes more important than the leakage.

Another vital feature of steam turbines, irrespective of the stages or impulse or reactions or corticestage or reactive turbine whatever may be process you use, if you look at the thermodynamic viewpoint, at the end of expansion, the minimum steam quality should be 88%. So, that means, the fluid must pass through the turbines when it is with minimum of 88% quality. And more and more it encounters liquid droplets in the turbine expansion it creates the thermal endurance and we call this as a moisture loss. Because normally all types of blades are designed for handling only steam not water droplet. And another important aspect of this high pressure moisture content is that they cause blade erosions as a result of impingement of water droplets on the blades. So, this is an unavoidable affair.

The last losses is miscellaneous losses. So, this miscellaneous losses comes into the picture, where we take the consideration of leaving losses. For a power plant to operate at its best capability, we must have a residual velocity in the range of 300 m/s. And we also expect that the steam or the liquid that leaves after condensation, will have a velocity close to 300 m/s. And if you do that then there could be losses and they add up to the losses close to 3%. Too low velocity can also lead to inappropriate design of the blades and too high velocity is also not advisable, because we can encounter many reaction stages. The possibilities of this thing that either you improve the number of reaction stages or you leave this fluid velocity at a higher level.

But many a times the infrastructure difficulty restricts us that we should not go for a higher number of stages. Rather we have to leave with some residual velocity of the steam which may be higher, but it is the need of the design.

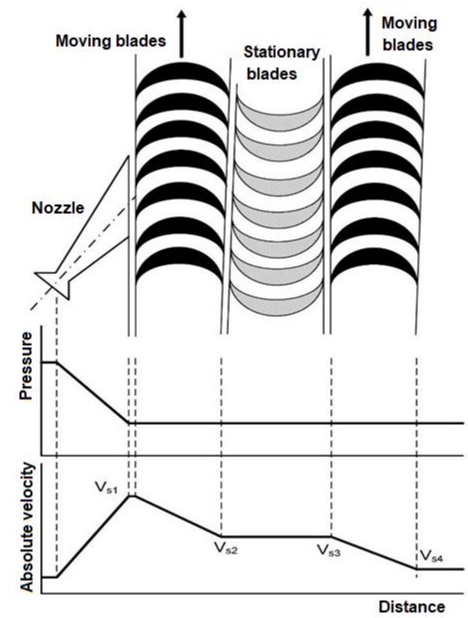
Apart from this, since we are looking at very high pressure and high temperature steam, so there is a possibility that heat transfer can takes place in any of the modes: conduction, convection and radiation. Of course, these losses are too small because most of the cases the protections or insulations are provided.

Next and last one is not with respect to the steam point of view rather with respect to how you are extracting the power from the turbines, either in an electrical way or in mechanical way. So, it is like a generator or any kind of rotor which extract power in rotating the blades and is usually 99% efficient. And these rotating velocity is normally kept at 3000 rpm and this accounts for about 1% of the turbine losses and to the best possible extent we have to minimize this.

So, these are the losses that we have seen in a steam turbines. So, with this we conclude this. Now, before we leave I will try to solve a numerical problems and mainly emphasize the fact that how the whirl velocity or axial velocity that gets generated in a turbine. So, far we have not accounted for any kind of axial velocity while solving the previous problems. Here we will try to emphasize the concept of the whirl velocity and axial velocity and how that propagates for transformation of thrust and power. So, corresponding thrust that we get is either a tangential thrust which is caused due to whirl velocity or axial thrust which is caused due to axial velocity and all these things we can find out from the velocity diagrams.

Q1. In two-row velocity-compounded impulse turbine, the stream velocity at the nozzle exit is 600 m/s and at an angle of 16°. The mean blade velocity is 120 m/s and the steam flow rate is 5 kg/s. The exit angles for first row of moving blade, fixed guide blade and second row moving blade are 18°, 22° and 36°, respectively. The blade velocity coefficient is 0.85. Calculate, (a) tangential thrust; (b) axial thrust; (c) power developed by the turbine; (d) blade efficiency.

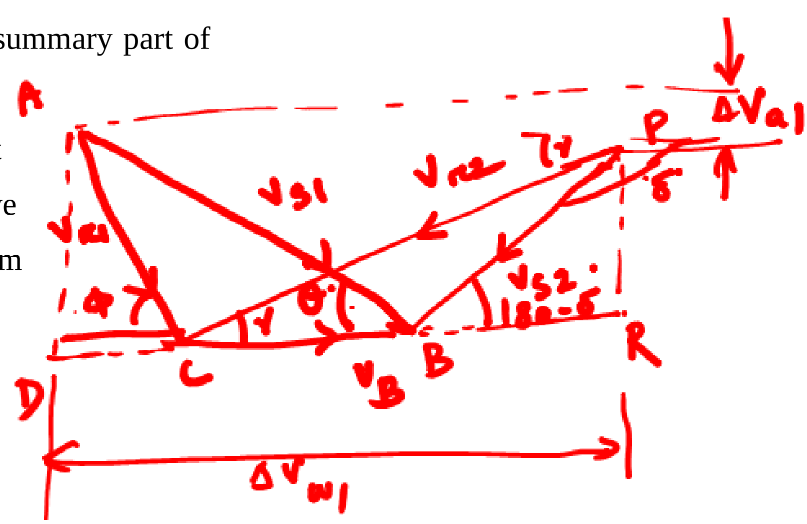
So, the problem statement says that we have a two row velocity compounded impulse turbine which is shown here. The stream from the nozzle exit is 600 m/s and the nozzle hits this blade at an angle 16° and because of these, we have this moving blade velocity is 20m/s and the steam is supplied at a rate 5 kg/s.



Now, here the exit angles for the first row of moving blade and fixed blade then fixed blade then second row of moving blades. So, we have two rows of moving blades which is organized by one set of stationary blades. And these angles are 18°, 22°and 36°. So, we can clarify these angles, once we draw the velocity diagrams properly for each of these moving blades as well as the stationary blades.

So, basically there are four diagrams- 1) for inlet of moving blade, then 2) inlet of second set of moving blades and 3) outlet of second set of moving blades. And for each velocity cases the blade velocity coefficient is 85% and because of this 85% we can see that there is an axial thrust that always exist. Because complete velocity transmission normally does not take place. So, we will try to solve the problem, but before that we have to draw these velocity diagrams clearly. Similar problems we have already solved in our previous lectures.

Here we will try to focus on the final or summary part of this velocity diagrams. I will not derive these formulae because they are already part of previous velocity diagrams. So, here we will try to see how this velocity diagram should look like.



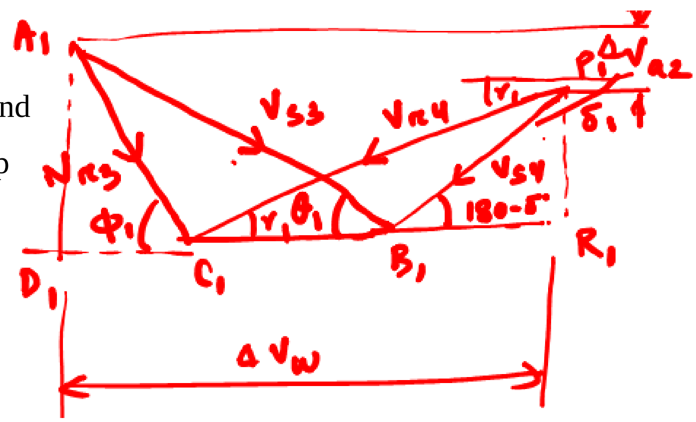
So, for this first moving blade, yellow one required?

we have velocity V_B, with corresponding stream velocity V_{s1} and the angle between them is θ and this angle we say as ϕ, and you drop a vertical then for this outlet we have this V_{r2}, V_{s2} and we can drop a vertical here, and after drawing this velocity diagrams you name this as a-b-c. this naming was done in in same manner as we did in the in our previous problem.

We have drawn in the diagram the whirl velocity, ΔV_{w1}. And if you see, the difference in the velocity in the axial direction is ΔV_{a1}.

Similar triangles we can draw for the fixed blade.

Now, here also for same velocity V_B we have V_{s3}, and the corresponding angles are θ_1, ϕ_1 . So, you can drop a vertical and from here also we can draw like this outlet where we can say it is V_{r4}, V_{s4} this angles can be rewritten as δ_1.



So, the velocity triangles can be named as $A_1B_1C_1D_1P_1R_1$. So, here also we have this net whirl velocity is ΔV_W which is the net effect of velocity in the tangential direction which gives the thrust and the net axial velocity we can find out the difference as V_{a2} .

Now we can recall our previous analysis and try to frame the formulae for moving blades and the fixed blades.

For Moving Blades:

$$\tan \emptyset = \frac{V_{s1} \sin \theta}{V_{s1} \cos \theta - V_B}$$

$$V_{r1} \sin \emptyset = V_{s1} \sin \theta$$

$$\tan \delta = \frac{V_{r2} \sin \gamma}{V_B - V_{r2} \cos \gamma}$$

For Fixed Blades:

$$\tan \emptyset_1 = \frac{V_{s3} \sin \theta_1}{V_{s3} \cos \theta_1 - V_B}$$

$$V_{r3} \sin \emptyset_1 = V_{s3} \sin \theta_1$$

Now let us understand, what the data are given.

$V_B = 120$ m/s ; $V_{s1} = 600$m/s ; $\theta = 16°$; $\gamma = 18°$; $\theta_1 = 22°$; $\gamma = 36°$; $K_v = 0.85$

First thing we need to find; $\tan \emptyset = \frac{V_{s1} \sin \theta}{V_{s1} \cos \theta - V_B} = \frac{600 \sin 16°}{600 \cos 16 - 120} = 0.362 \Rightarrow \emptyset = 19.9°$

Now

$$V_{r1} = \frac{V_{s1} \sin \theta}{\sin \emptyset} = \frac{600 \sin 16°}{\sin 19.9°} = 485.8\ m/s$$

$$V_{r2} = V_{r1}(K_v) = 485.8 \times 0.85 = 413\ m/s$$

Now, the second part of analysis, we can find

$$\tan \delta = \frac{V_{r2} \sin \gamma}{V_{r2} \cos \gamma - V_B} = \frac{413 \sin 18}{413 \cos 18 - 120} = 0.47 \Rightarrow \delta = 25°$$

$$V_{s2} = \frac{V_{r2} \sin \gamma}{\sin \delta} = \frac{413 \sin 18}{\sin 25} = 301 m/s$$

$$V_{s3} = K_v(V_{s2}) = 255.9\ m/s$$

From this velocity diagram now we will be able to find out

$$(\Delta V_w)_1 = V_{s1} \cos \theta + V_{s2} \cos \delta = 600 \cos 16 + 301 \cos 25 = 850\ m/s$$

$$(\Delta V_a)_2 = V_{s1} \sin \theta - V_{s2} \sin \delta = 600 \sin 16 - 301 \sin 25 = 37.8\ m/s$$

So, in similar way we can find out

$$\tan \emptyset_1 = \frac{255.9 \sin 22}{255.9 \cos 22 - 120} = 0.81 \Rightarrow \emptyset_1 = 39.3°$$

$$V_{r3} = \frac{V_{s3} \sin \theta_1}{\sin \emptyset_1} = 151.5\ m/s$$

$V_{r4} = 0.85(V_{r3}) = 128.7\ m/s$

Now

$(\Delta V_w)_2 = V_{r3} \cos \emptyset_1 + V_{r4} \cos \gamma_1 = 151.5 \cos 39.3 + 128.7 \cos 36 = 220\ m/s$

$(\Delta V_a)_1 = V_{s3} \sin \theta_1 - V_{r4} \cos \gamma_1 = 20.2\ m/s$

So, we have all the numbers. Now, we will be able to find out what is tangential thrust.

Whirl velocity, $\Delta V_w = (\Delta V_w)_1 + (\Delta V_w)_2 = 850 + 220 = 1070\ m/s$

Axial Velocity, $\Delta V_a = (\Delta V_a)_1 + (\Delta V_a)_2 = 37.8 + 20.2 = 58\ m/s$

a) Tangential thrust, $P_t = \dot{m}\ (\Delta V_w) = 5.35\ kN$ as $\dot{m} = 5\ kg/s$
b) Axial thrust, $P_a = \dot{m}(\Delta V_a) = 0.29\ kN$
c) Power developed, $\dot{W} = P_t\ V_B = 5.35 \times 120 = 642\ KW$
d) Blade Efficiency, $\eta = \frac{2(\Delta V_w) V_B}{V_{S1}^2} = \frac{2\times1070\times120}{600\times600} = 71.3\%$

So, idea of this problem is to give some emphasize that for a steam turbine what is the concept of tangential thrust and axial thrust and now what is the concept of whirl velocity and axial velocity. And we say that this axial thrust has to be there because we need a balancing force since rotor is rotating we may also need a balancing thrust that has to come from the steam. So, it must provide some axial thrust to counterbalance the reactions developed by the rotor. With this I conclude. Thank you for your attention. Thank you.