Design of Steam Turbines

Session delivered by:

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Session Objectives

This session is intended to discuss the following:

- Classification of steam turbines
- Working principles
- Pressure and velocity compounding
- Losses in steam turbines
- Design guidelines of steam turbines

Steam

Steam is a vapour used as a working substance in the operation of steam turbine.

Is steam a perfect gas?

Steam possess properties like those of gases, namely pressure, volume, temperature, internal energy, enthalpy and entropy. But the pressure volume and temperature of steam as a vapour are not connected by any simple relationship such as is expressed by the characteristic equation for a perfect gas.

Sensible heat – The heat absorbed by water in attaining its boiling point

Latent heat – The heat absorbed to convert boiling water into steam

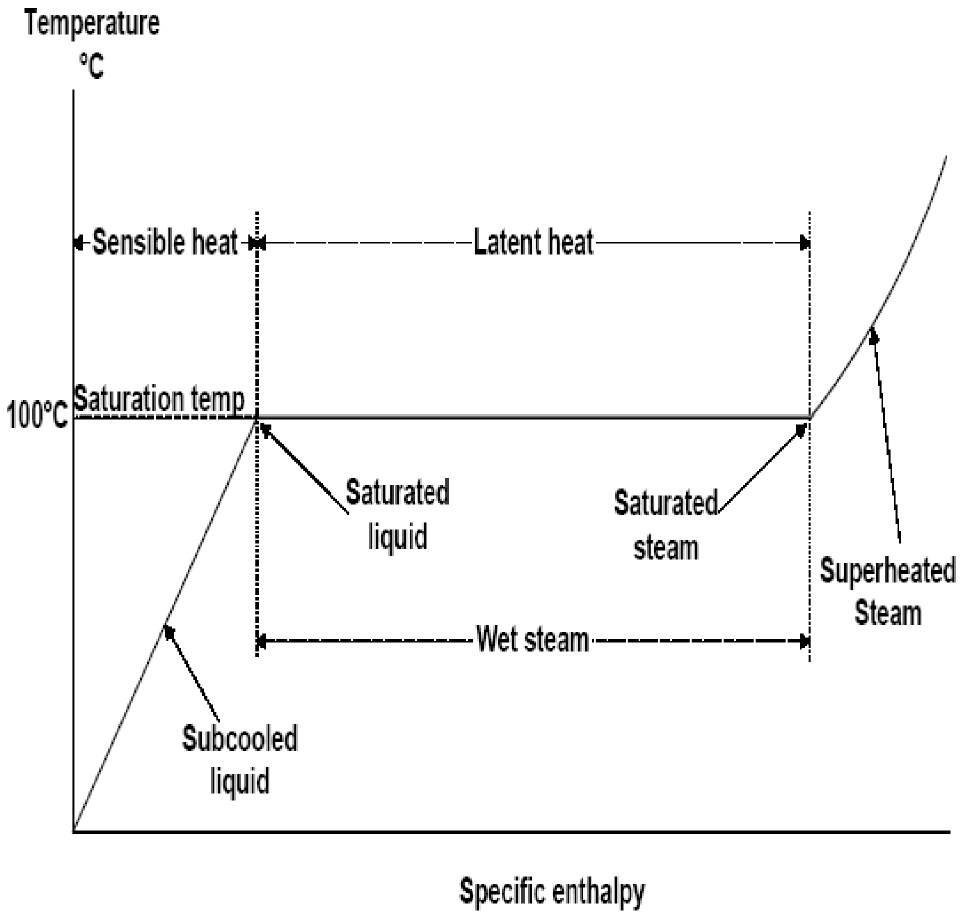
Wet steam – Steam containing some quantity of moisture content.

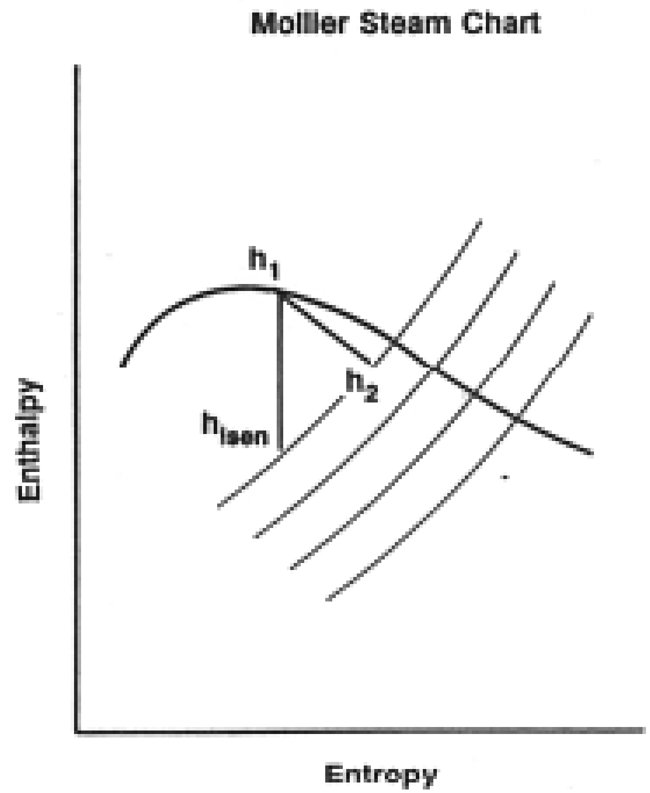
Dry steam – Steam that has no moisture content.

Superheated steam – Dry steam when heated at constant pressure attains superheat. Superheated steam behaves like perfect gas.

The properties of steam are dependent on its pressure.

Steam Properties





Enthalpy (H) kJ/kg
Entropy (s) kJ/kg-K
Density (ρ) kg/m^3

Internal energy (u) kJ/kg
Specific volume (v) m^3/kg
Isobaric heat capacity (c_p) kJ/kg-K

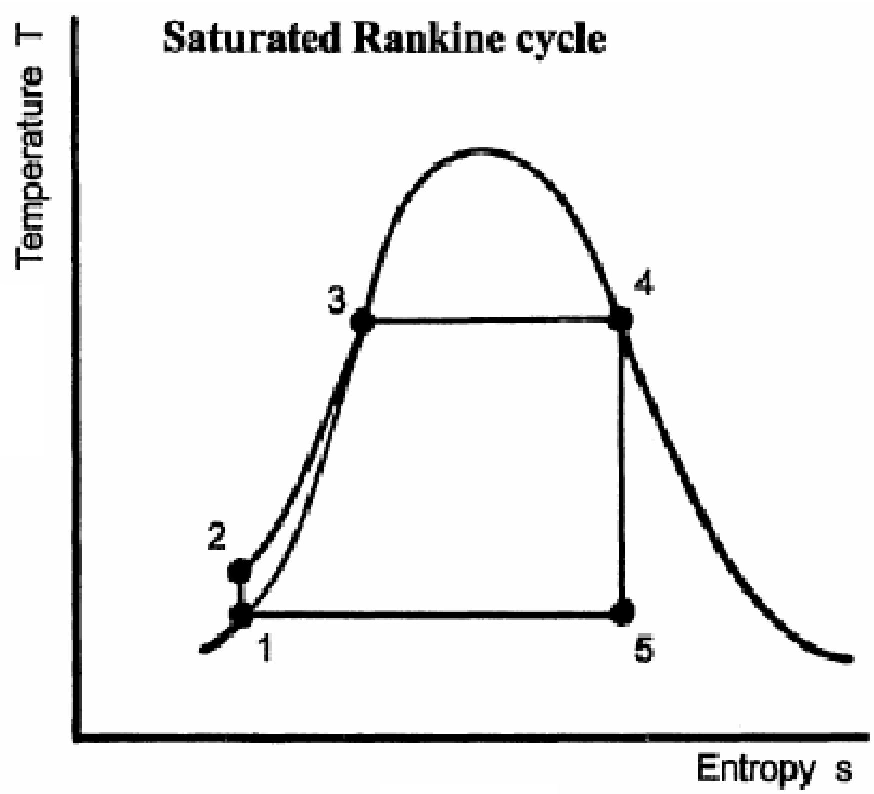
Steam Turbine

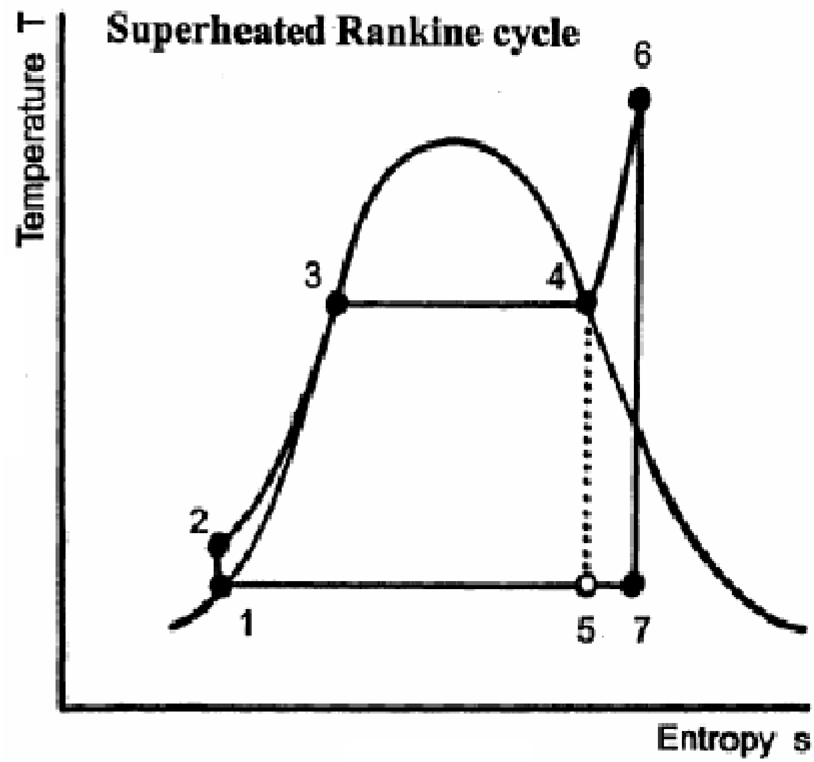
- Steam turbine converts a part of energy of the steam at high temperature and pressure into mechanical power, which may subsequently be converted into electrical power.
- The main components of a steam turbine are:
  - Feed water pump
  - Boiler
  - Turbine stages, comprising nozzle/stator and rotor blade rows
  - Condenser
- Steam from the boiler is expanded in the nozzle blade passages to produce high velocity jets, which impinge on the rotor blades mounted on a disc and shaft. The rate of change of momentum of steam flow across the rotor blades produces the required torque for the shaft to rotate.
- The conversion of energy across the blade rows takes place by **impulse, reaction or impulse reaction principle.**

Application of Steam Turbines

- Power generation
- Petrochemical refineries
- Pharmaceuticals
- Food processing
- Petroleum / gas processing
- Paper mills
- Sugar industry
- Waste-to-energy

Rankine Cycle





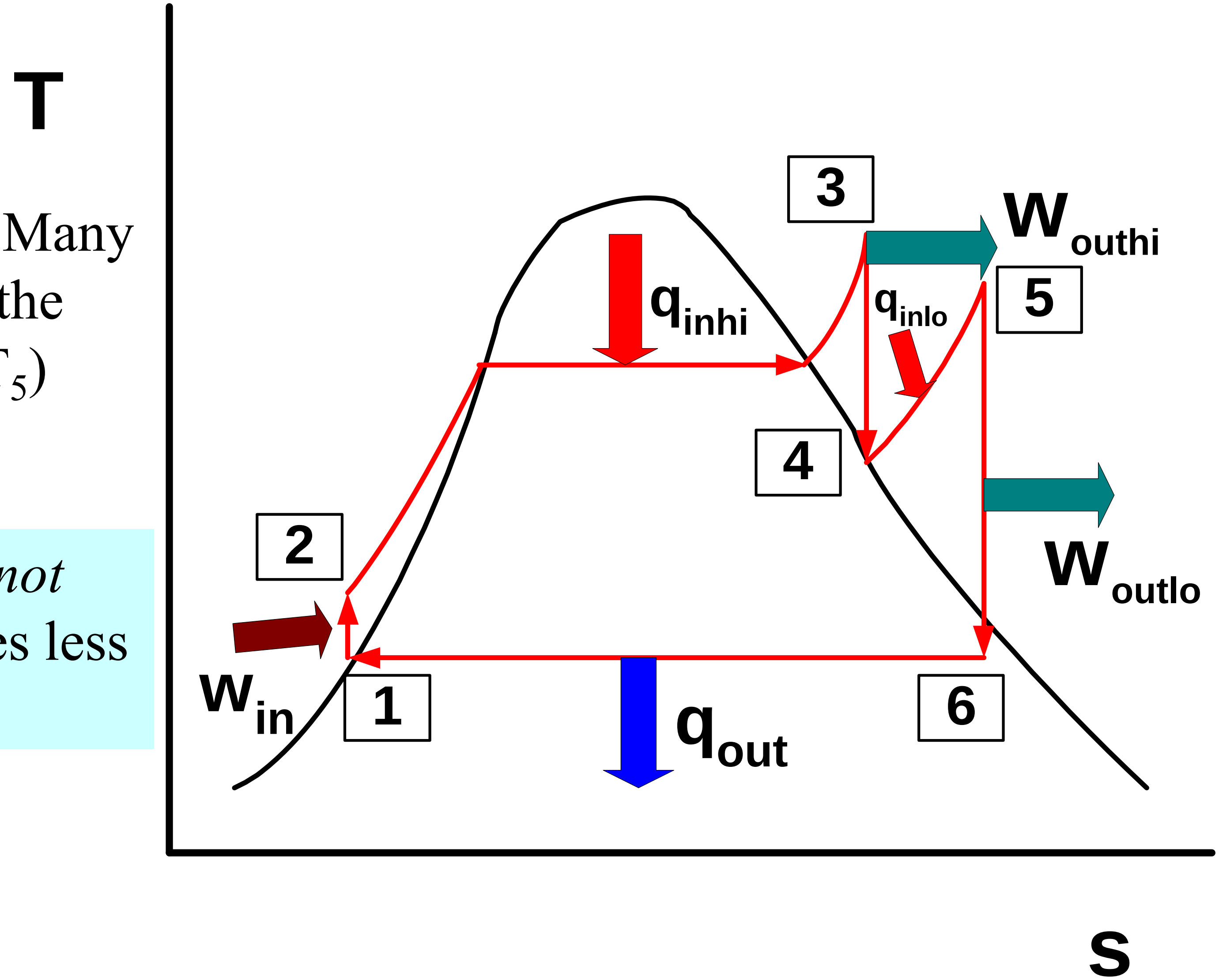
Saturated Rankine cycle

Superheated Rankine cycle

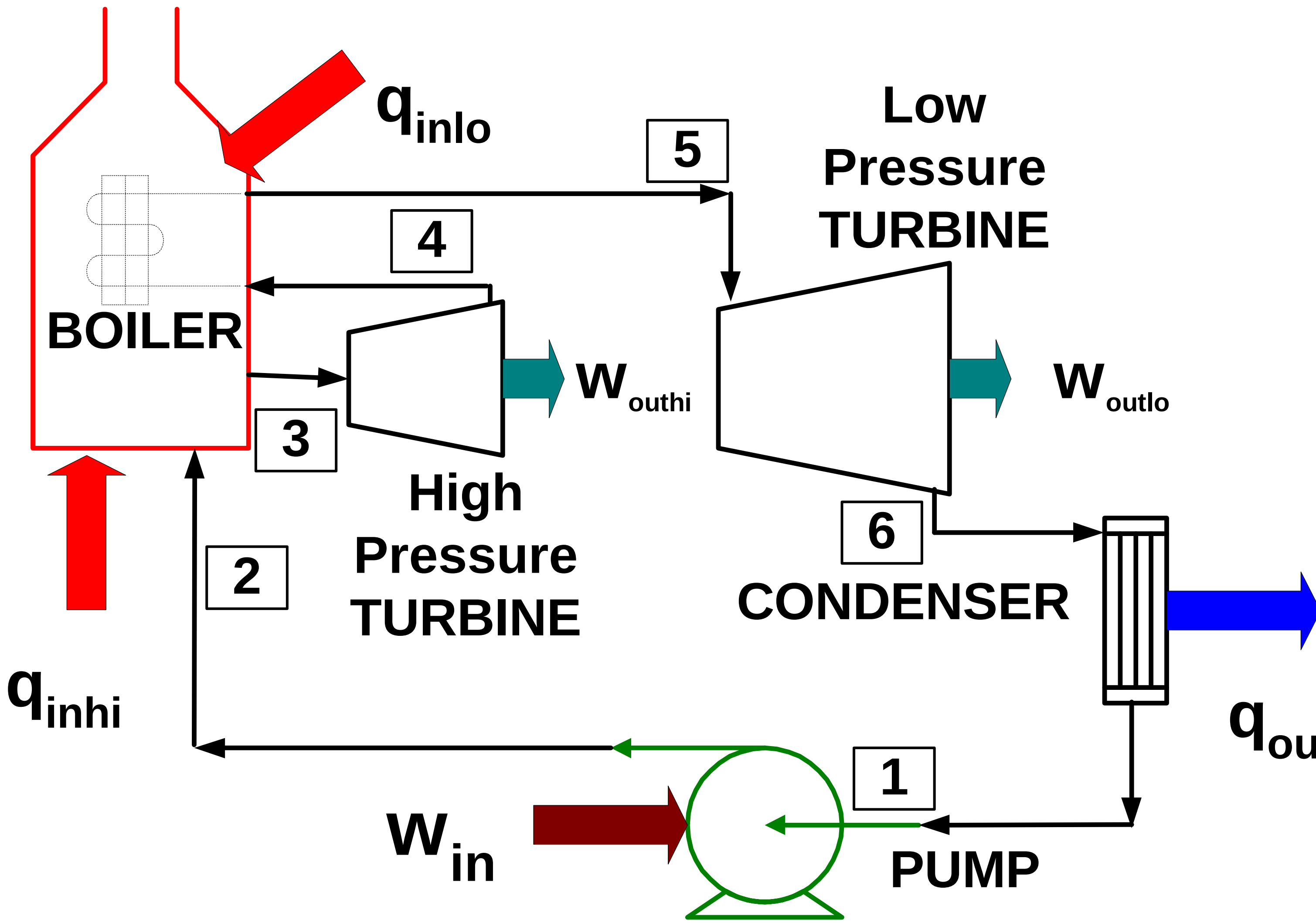
Reheat Cycle

Note that $T_5 < T_3$. Many systems reheat to the same temp ($T_3 = T_5$)

Reheat is usually *not* offered for turbines less than 50 MW



Schematic of Rankine Reheat Cycle



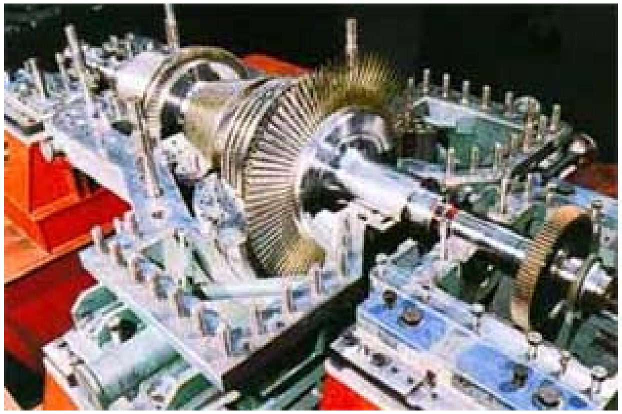
Steam Turbine Classification

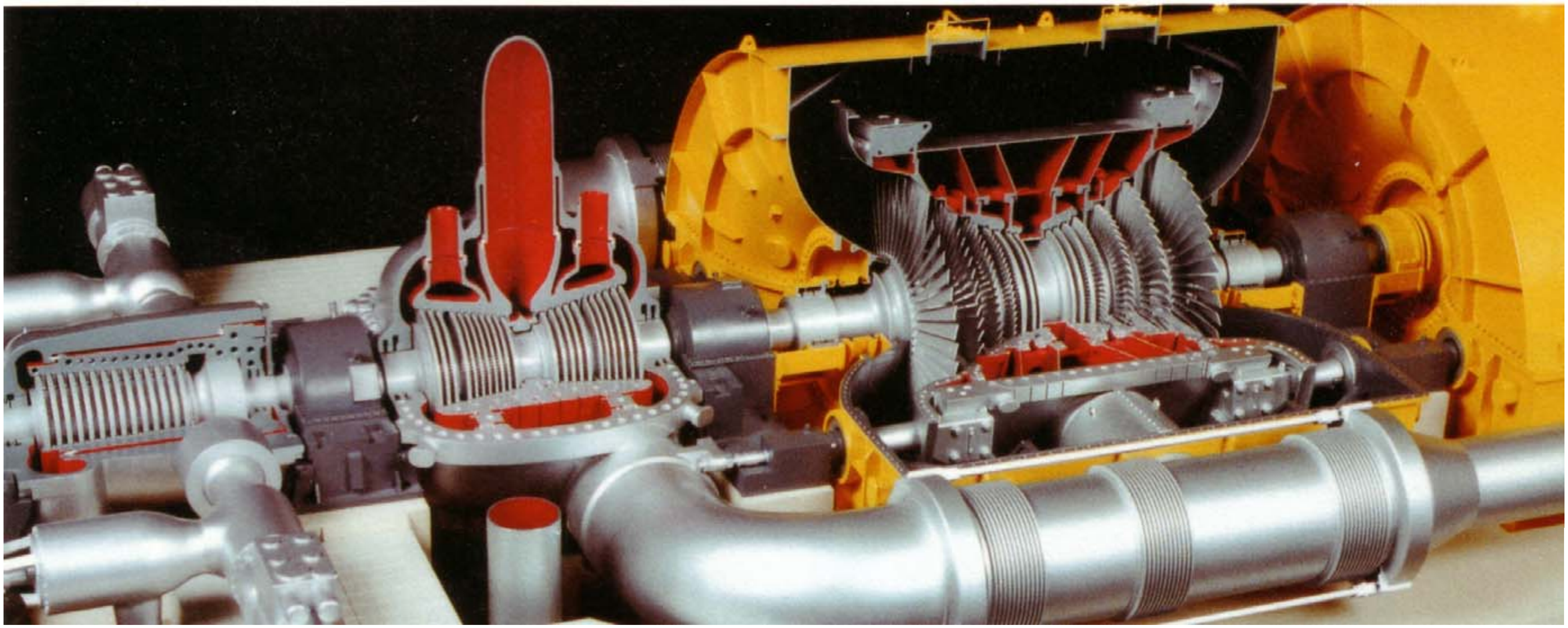
Steam turbines can be classified in several different ways:

1. By details of stage design
   - Impulse or reaction
2. By steam supply and exhaust conditions
   - Condensing or non-condensing
   - Automatic or controlled extraction
   - Mixed pressure
   - Reheat
3. By casing or shaft arrangement
   - Single casing, tandem compound or cross compound
4. By number of exhaust stages in parallel
   - Two flow, four flow or six flow
5. By direction of steam flow
   - Axial flow, radial flow or tangential flow
6. Single or multi-stage
7. By steam condition
   - Superheated or saturated

Turbine Designation

H - Single Flow HP Turbine
K - HP/IP Opposite flow
E - HP/LP Opposite flow
N - Double flow LP Turbine
M - Double flow IP Turbine





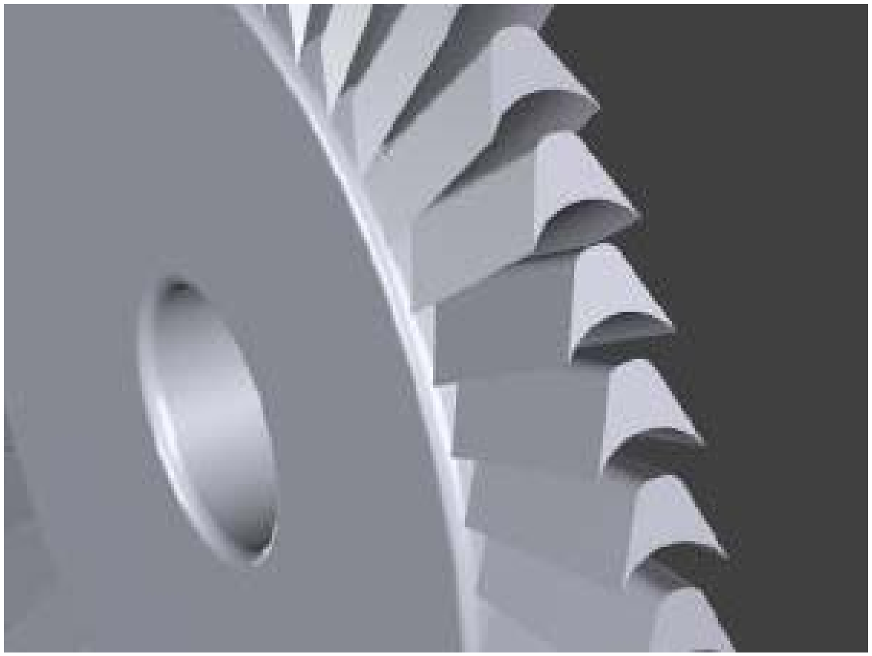
Steam Turbine Blade Rows

149 MW steam turbine rotor

Single cylinder type turbine casing

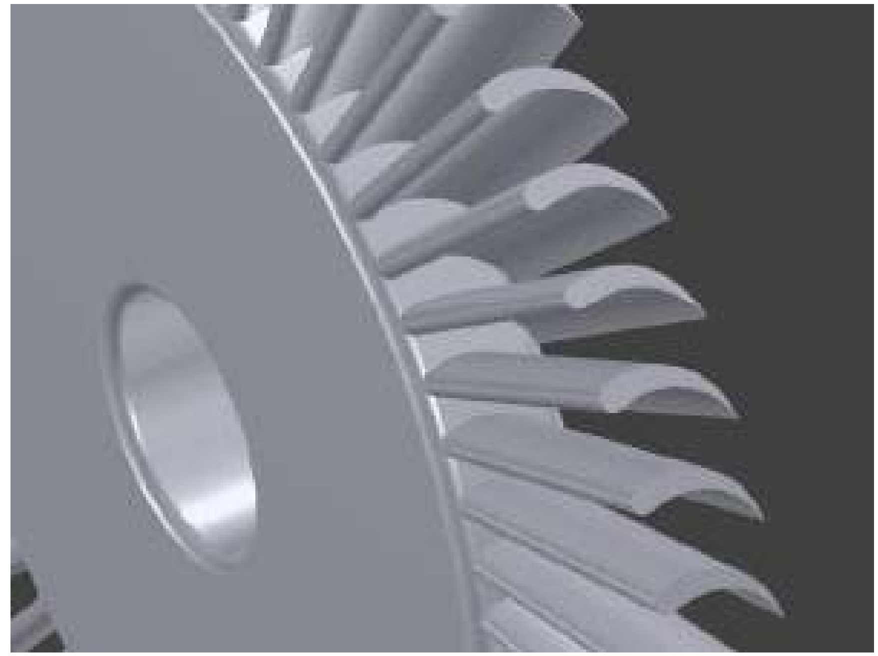
Types of Steam Turbine

Impulse Turbine



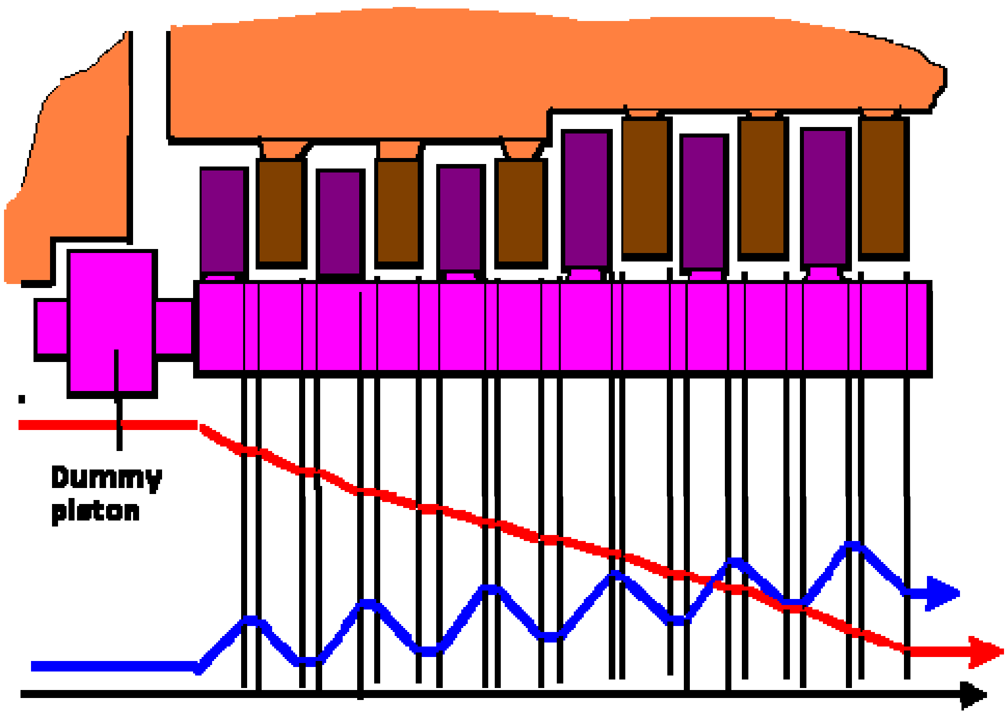
Complete expansion of steam takes place in stationary nozzle blades and the kinetic energy is converted into mechanical work in rotor blades

Reaction Turbine

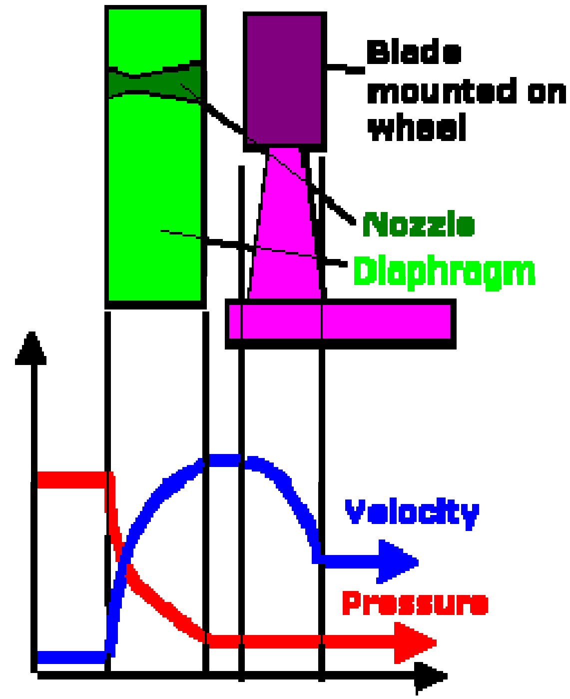


Expansion of steam takes place partly in nozzle / stator and partly in rotor. However, conversion of kinetic energy to mechanical work takes place only in rotor blades

Types of Steam Turbine

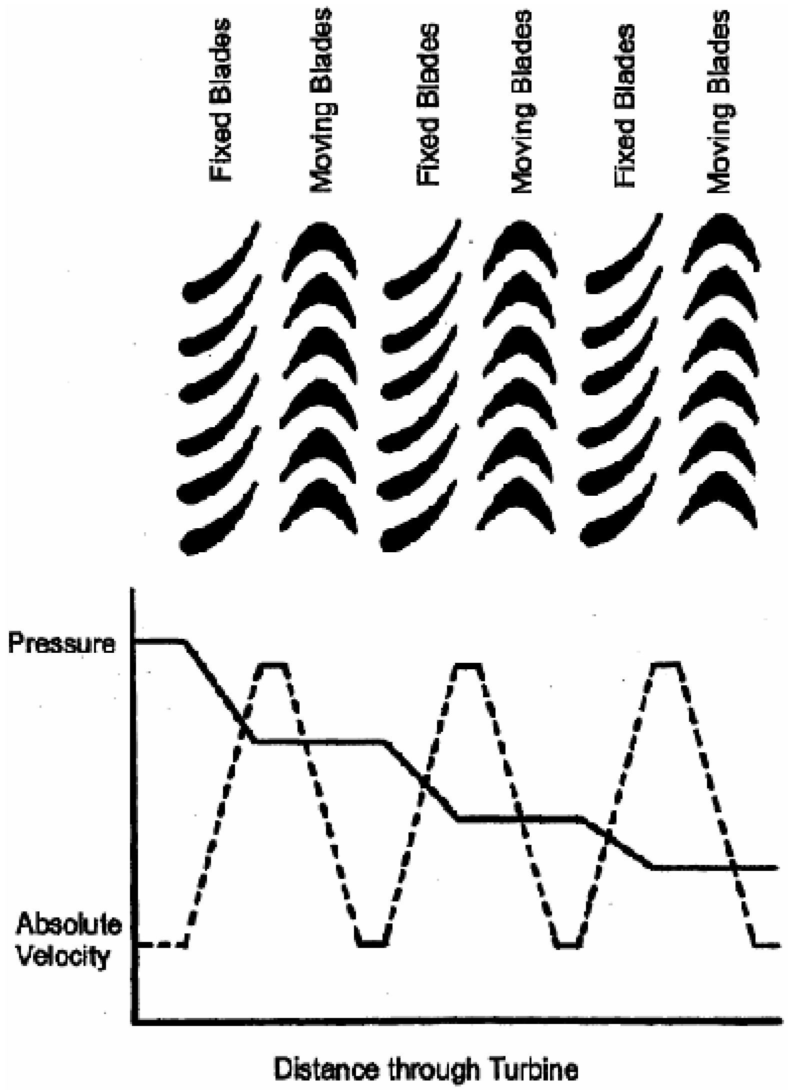


Parsons Reaction Turbine

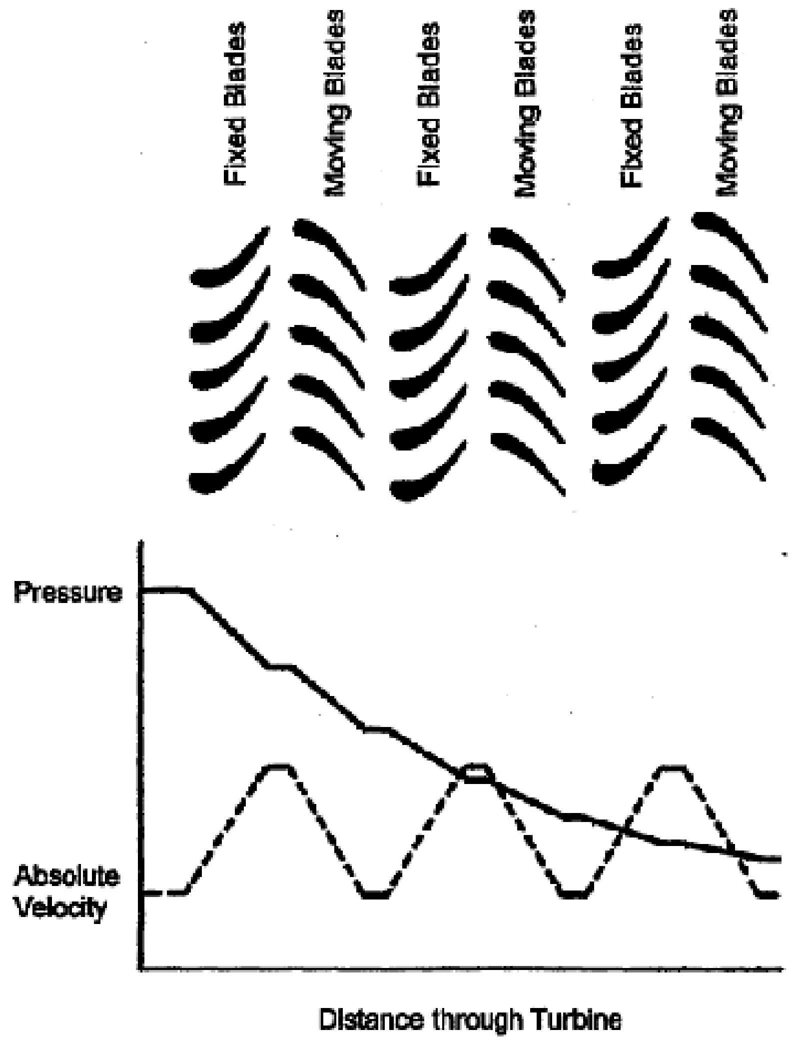


De Laval Impulse Turbine.

Flow through Multistage Steam Turbine



Impulse turbine blading and conditions



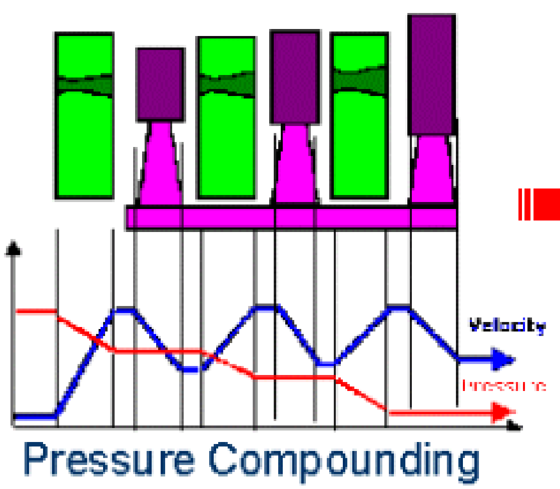
Reaction turbine blading and conditions

Compounding of Steam Turbines

- Compounding is done to reduce the rotational speed of the impulse turbine to practical limits.
- Compounding is achieved by using more than one set of nozzle and rotor blade rows, in series, so that either the steam pressure or the velocity (after expansion) is absorbed by the turbine in stages.

Three main types of compounded impulse turbines are:

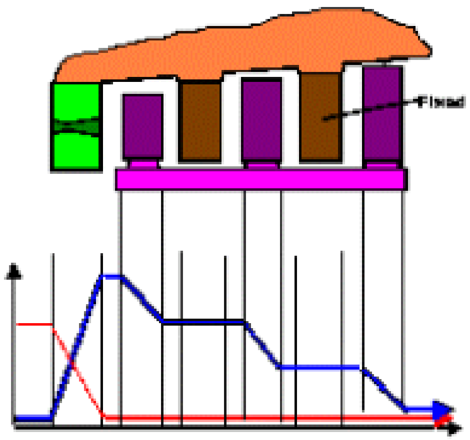
a. Pressure compounded
b. Velocity compounded
c. Pressure and velocity compounded impulse turbines



Pressure Compounding

Involves splitting of the whole pressure drop into a series of smaller pressure drops across several stages of impulse turbine. The nozzles are fitted into a diaphragm locked in the casing that separates one wheel chamber from another. All rotors are mounted on the same shaft.

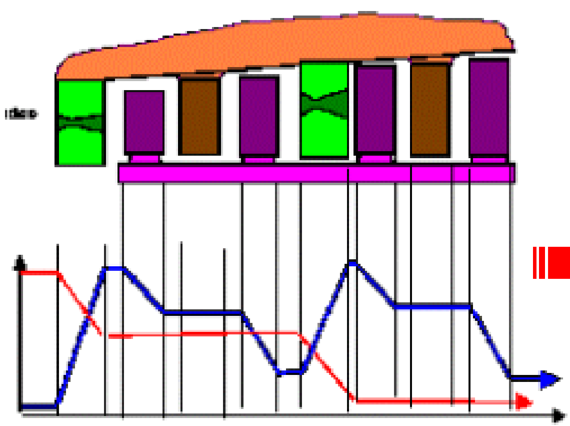
Compounding of Steam Turbines



Velocity Compounded

Velocity Compounding

Entire pressure drop is achieved across the first nozzle. The high velocity is then reduced stage by stage across the following rotor blade rows. There is no expansion across the stator rows.



Pressure-Velocity Compound

Pressure-Velocity Compounding

Pressure-velocity compounding is combination of pressure and velocity compounding. It gives the advantage of producing a shortened rotor compared to pure velocity compounding. In this design steam velocity at exit to the nozzles is kept reasonable and thus the blade speed is reduced.

Comparison of Impulse and Reaction Turbines

Impulse Turbines

- An impulse turbine has fixed nozzles that expand steam flow to produce high velocity jets
- Rotor blade profile is symmetrical as no pressure drop takes place across these blades
- The design is suitable for efficiently absorbing high velocity and high pressure
- Steam pressure is constant across the blades and therefore fine tip clearances are not necessary
- Efficiency is not maintained in the low pressure stages (high steam velocity cannot be achieved in the low pressure stages)

Reaction Turbines

- Reaction turbine makes use of the reaction force produced as the steam accelerates through the rotor blade passages
- Rotor blades have convergent passages allowing pressure drop to occur partly through them
- Efficient at the low pressure stages
- Fine blade tip clearances are necessary due to the pressure leakages
- Lower efficiencies in high pressure stages due to the leakage losses around the blade tips
- Fine tip clearances can cause damage to the blade tips

Losses in Steam Turbines

Profile Loss: Due to development of boundary layers on blade surfaces. It is influenced by the factors like Reynolds number, surface roughness, exit Mach number and trailing edge thickness.

Secondary Loss: Due to development of boundary layers on the casing and hub walls. The influence factors are similar to those for the profile loss.

Tip Leakage Loss: Due to clearance between rotor blades and casing wall as well as between stator blades and rotating hub. The extent of tip leakage depends on whether the turbine is impulse or reaction. Due to pressure drop across the moving blades of reaction turbine, they are more prone to tip leakages.

Disc Windage Loss: Due to fluid friction on the turbine disc surfaces as they rotate in a steam atmosphere. The result is a reduction in shaft power and an increase in kinetic energy and heat energy of steam.

Losses in Steam Turbine

Lacing Wire Loss: Due to flow blockage created by the presence of lacing wires in long blade of LP stages.

Wetness Loss: Due to moisture entrained in the low pressure steam at the exit of LP turbine. The loss manifests in: firstly, a reduction in turbine efficiency due to energy absorption by the water droplets, and secondly, erosion of rotor blade leading edges in last stages.

Annulus Loss: Due to significant amount of diffusion between adjacent stages or where wall cavities occur between the fixed and moving blades. The extent of loss is greatly reduced at high annulus area ratios if the expansion of steam is controlled by a flared casing wall.

Leaving Loss: Due to kinetic energy of steam leaving the last stage of LP turbine. In practice, the steam slows down a bit after leaving the last blade, due to frictional losses.

Partial Admission Loss: Due to partial filling of steam, the flow between the blades is considerably accelerated causing a loss in power.

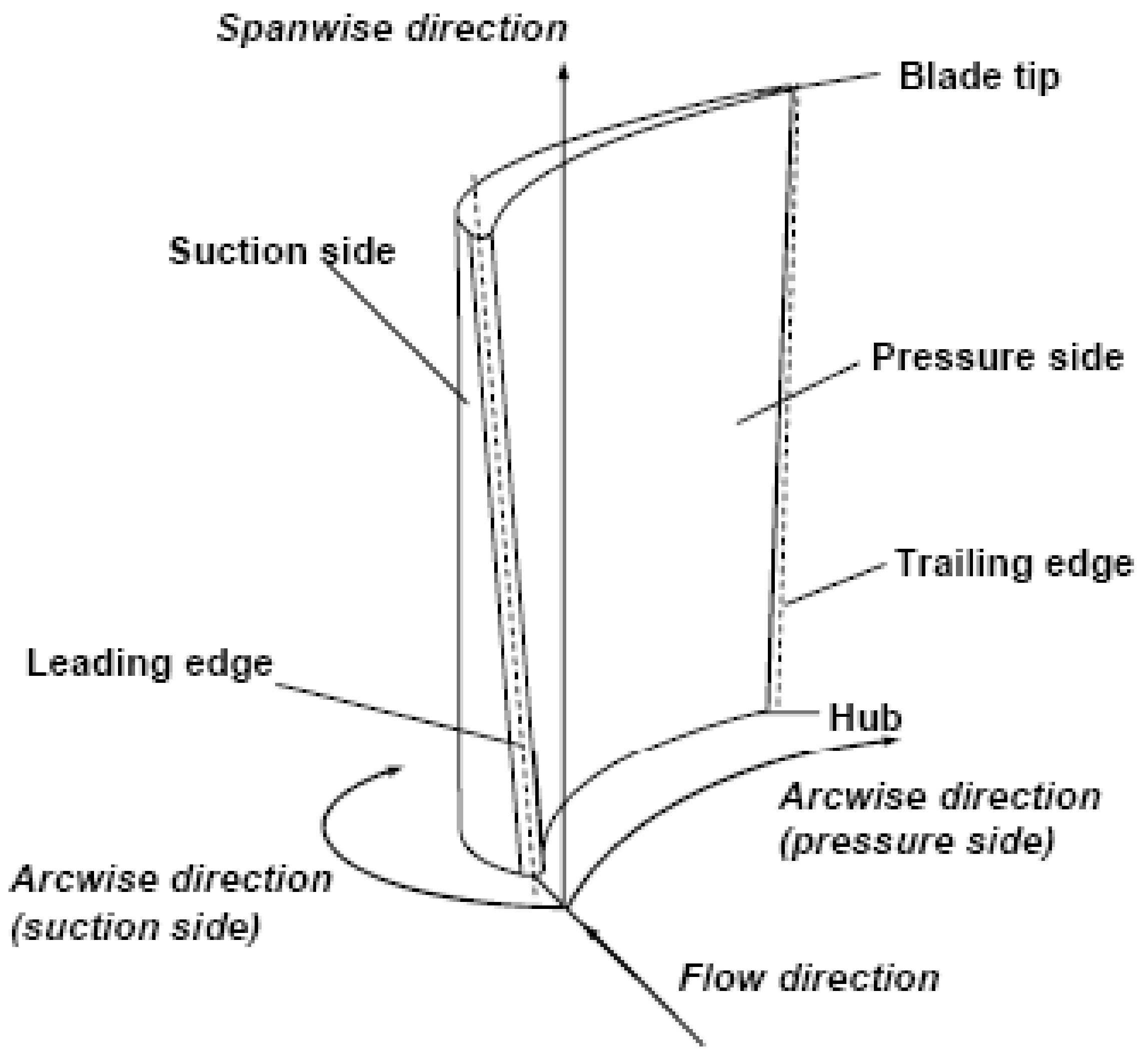
Nomenclature

V	Absolute velocity of steam
U	Blade velocity
W	Relative velocity of steam
$V_a = V_f = V_m$	Axial component or flow velocity
V_w	Whirl or tangential component
α	Nozzle angle
β	Blade angle
h	enthalpy

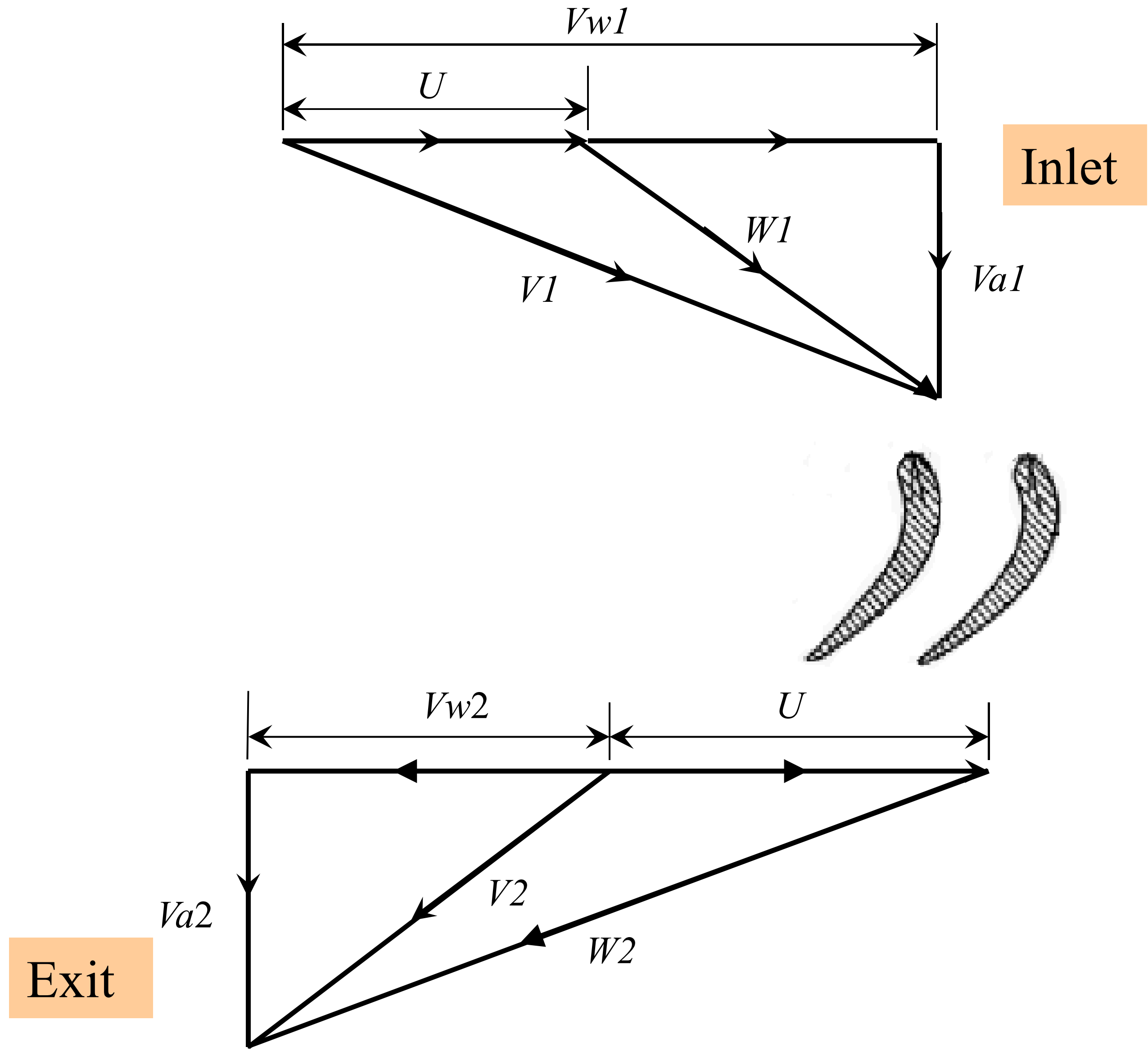
<u>Suffix</u>

1	Inlet
2	Outlet

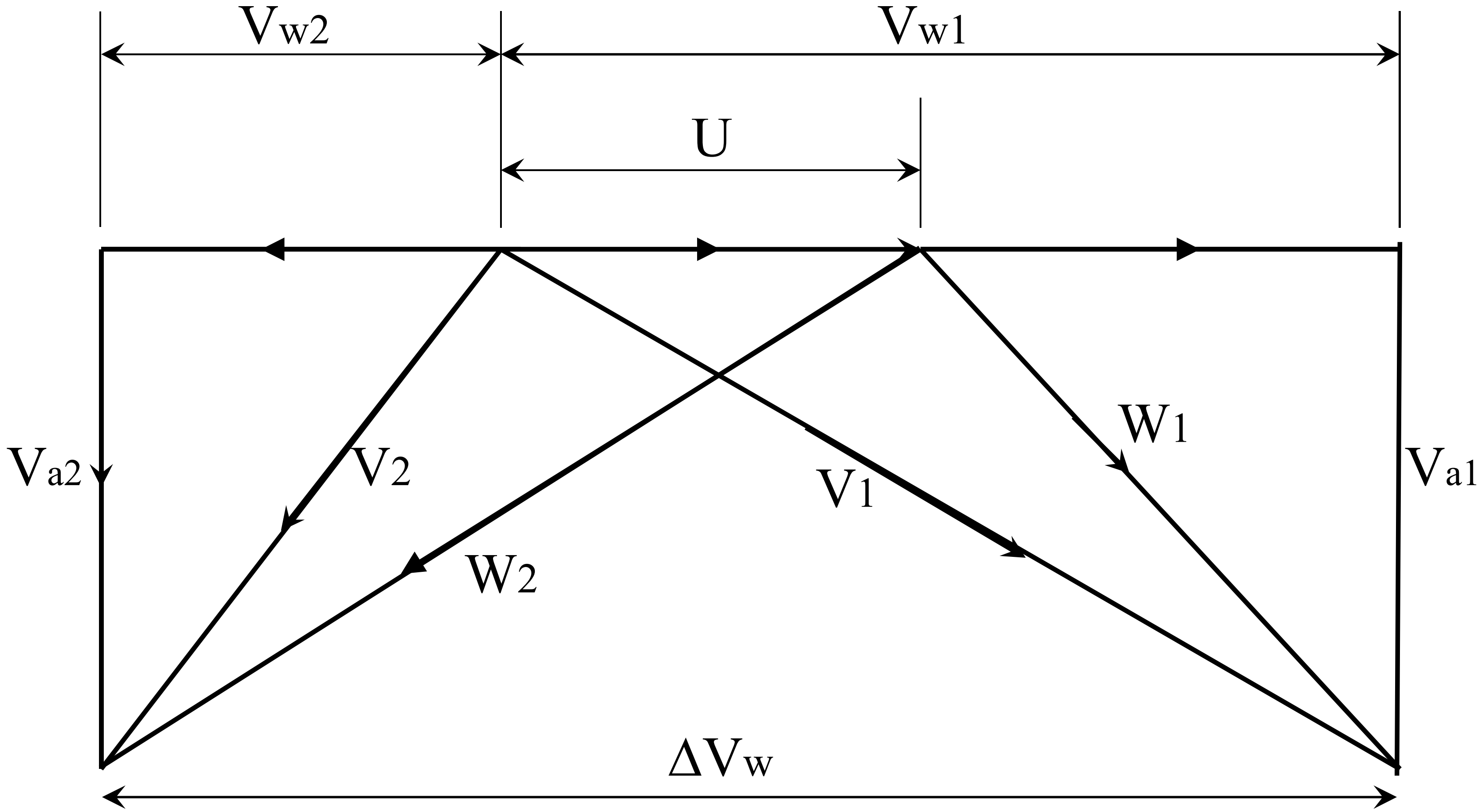
Steam Turbine Blade Terminology



Velocity Triangles



Combined Velocity Triangles



For 50% reaction design

Work Done – Impulse Steam Turbine

If the blade is symmetrical, then $\beta_1 = \beta_2$ and neglecting frictional effects, $W_1 = W_2$.

In actual case, the relative velocity is reduced by friction and expressed by a blade velocity coefficient k. Thus, $k = W_2/W_1$

The Euler work done by the steam is given by;

$$W_t = U(V_{w1}+V_{w2}) \tag{1}$$

Since V_{w2} is in negative direction, the work done per unit mass flow is given by,

If $V_{a1} = V_{a2}$, then,

$$Wt = U\ Va\ (\tan\alpha_1 + \tan\alpha_2) \text{ or } Wt = U\ V_a\ (\tan\beta_1 + \tan\beta_2) \tag{2}$$

Equation (2) is often referred to as the diagram work per unit mass flow and hence the diagram efficiency is defined as:

$$\eta_d = \frac{\text{Diagram work done per unit mass flow}}{\text{Work available per unit mass flow}}$$

It can be shown that $\eta_d = \dfrac{4U}{V_1}\left(\cos\alpha_1 - \dfrac{U}{V_1}\right)$

U/V_1 is called the blade speed ratio.

and max. $\eta_d = 4\cos^2\alpha_1$

Degree of Reaction

- **Degree of reaction** is a parameter that describes the relation between the energy transfer due to the static pressure change and the energy transfer due to dynamic pressure change.
- **Degree of reaction** is defined as the ratio of static pressure drop in the rotor to the static pressure drop in the stage. It is also defined as the ratio of static enthalpy drop in the rotor to the static enthalpy drop in the stage

$$\Lambda = \text{Degree of reaction} = \frac{\text{Static enthalpy change in rotor}}{\text{Static enthalpy change in stage}} = \frac{h_1 - h_2}{h_0 - h_2} \qquad (16)$$

The static enthalpy at the inlet to the fixed blade in terms of stagnation enthalpy and velocity at the inlet to the fixed blades is given by

$$h_0 = h_{00} - \frac{V_0^2}{2C_p} \quad \text{similarly} \quad h_2 = h_{02} - \frac{V_2^2}{2C_p}$$

Substituting $$\Lambda = \frac{(h_1 - h_2)}{\left(h_{00} - \frac{V_0^2}{2C_p}\right) - \left(h_{02} - \frac{V_2^2}{2C_p}\right)}$$

Degree of Reaction (… contd.)

But for a normal stage, $V_0 = V_2$ and since $h_{00} = h_{01}$ in the nozzle, then;

$$\Lambda = \frac{(h_1 - h_2)}{(h_{01} - h_{02})} \tag{17}$$

We know that $(h_{01} - h_{02}) = \quad (h_1 - h_2) + \dfrac{\left(V_{w1}^2 - V_{w2}^2\right)}{2} = 0$

Substituting for (h_1- h_2) in equation (17),

$$\Lambda = \frac{\left(V_{w2}^2 - V_{w1}^2\right)}{\left[2(h_{01} - h_{02})\right]} \quad = \frac{\left(V_{w2}^2 - V_{w1}^2\right)}{\left[2U(V_{w1} - V_{w2})\right]} \tag{18}$$

Assuming the axial velocity is constant through out the stage, then

$$\Lambda = \frac{\left(V_{w2}^2 - V_{w1}^2\right)}{\left[2U(U + V_{w1} + V_{w2} - U)\right]}$$

$$\Lambda = \frac{(V_{w2} - V_{w1})(V_{w2} + V_{w1})}{\left[2U(V_{w1} + V_{w2})\right]} \tag{19}$$

Degree of Reaction (… contd.)

$$\Lambda = \frac{V_a\left(\tan\beta_2 + \tan\beta_1\right)}{2U} \qquad (20)$$

From the velocity triangles it is seen that

$$V_{w1} = U + V_{w1} \qquad\qquad V_{w2} = V_{w2} - U$$

Therefore equation (20) can be arranged into a second form:

$$\Lambda = \frac{1}{2} + \frac{V_a}{2U}\left(\tan\beta_2 + \tan\alpha_2\right) \qquad (21)$$

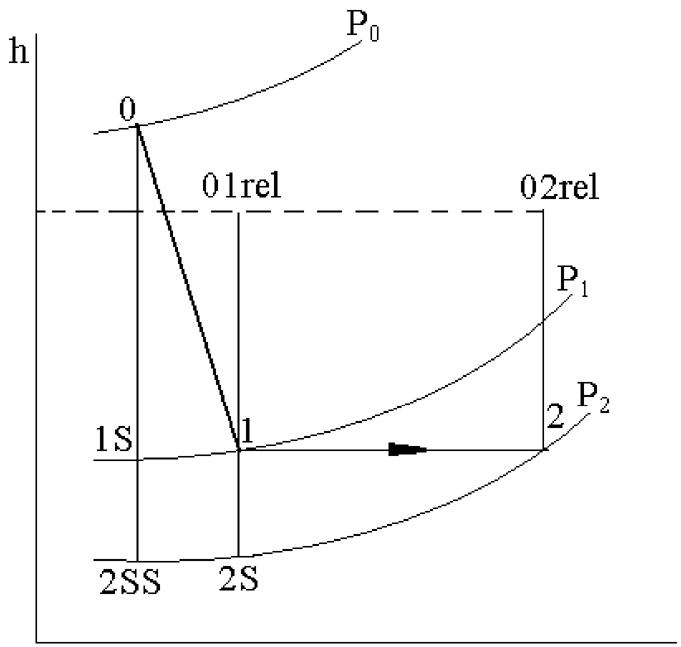
Putting $\Lambda = 0$ in equation (20), we get

$\left(\beta_2 = \beta_1\right)$ And $V_1 = V_2$ and for $\Lambda = 0.5$, $\left(\beta_2 = \alpha_1\right)$

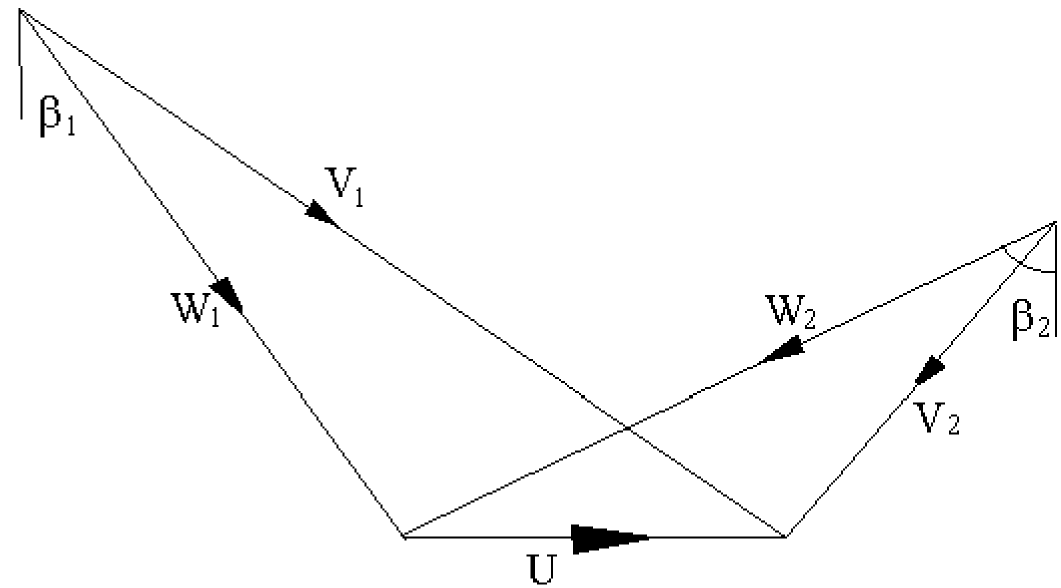
Zero Reaction Stage

Let us first discuss the special case of zero reaction. According to the definition of reaction, When $\Lambda = 0$, equation (16) reveals that $h_1 = h_2$ and equation (20) that $\beta_1 = \beta_2$.

Degree of Reaction (… contd.)



Mollier diagram



Velocity Diagram

Now $h_{01r01} = h_{02r02}$ and $h_1 = h_2$ for $\Lambda = 0$. Then $W_1 = W_2$. In the ideal case, there is no pressure drop in the rotor and points 1 2 and 2s on the mollier chart should coincide. But due to irreversibility, there is a pressure drop through the rotor. The zero reaction in the impulse stage by definition, means there is no pressure drop through the rotor. The Mollier diagram for an impulse stage is shown in Fig. 1.a (next slide), where it can be observed that the enthalpy increases through the rotor.

Degree of Reaction (… contd.)

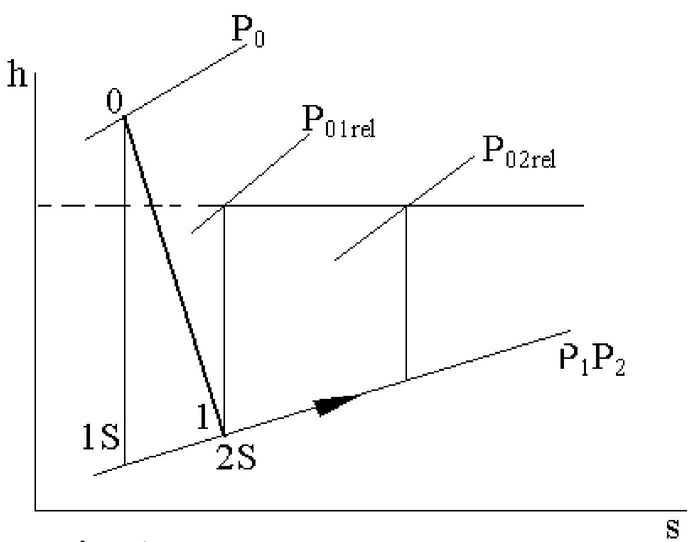


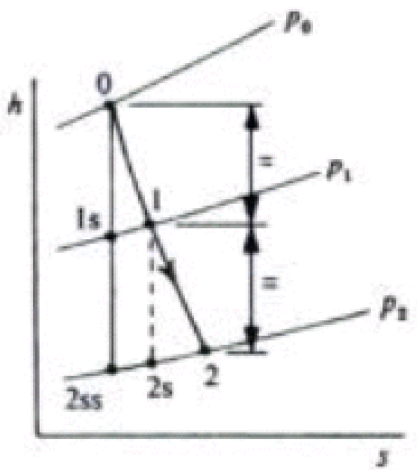
Fig.1.a

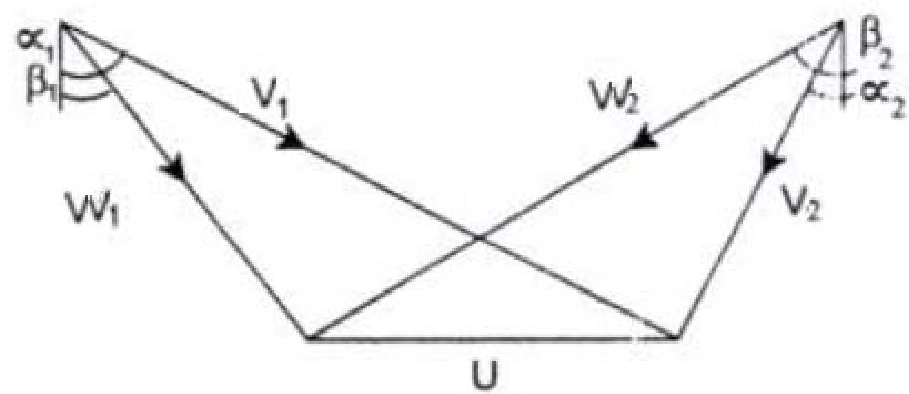
From equation (16) it is clear that the reaction is negative for the impulse turbine stage when irreversibility is taken into account.

Fifty Percent Reaction Stage

From equation (16) for $\Lambda = 0.5$, $\alpha_1 = \beta_2$ and the velocity diagram is symmetrical. Because of symmetry, it is also clear that $\alpha_2 = \beta_1$. For $\Lambda = 1/2$, the enthalpy drop in the nozzle row equals the enthalpy drop in the rotor. That is

$$h_0 - h_1 = h_1 - h_2$$



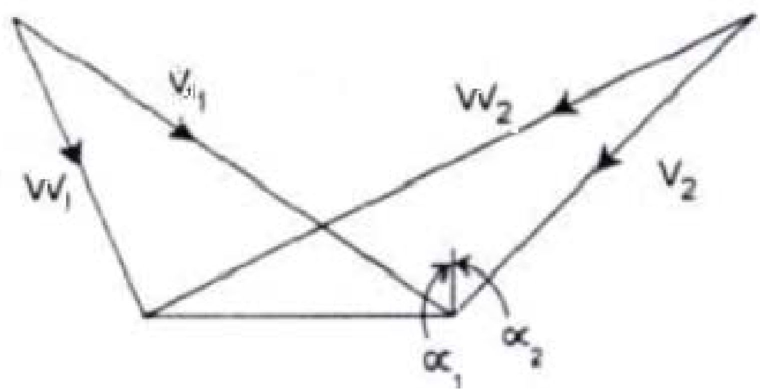


Degree of Reaction (… contd.)

Unity Reaction Stage

Substituting $\beta_2 = \tan\alpha_2 + \dfrac{U}{V_a}$ into equation (21)

$$\Lambda = 1 + \frac{V_a}{2U}\left(\tan\alpha_2 - \tan\alpha_1\right) \tag{22}$$



Thus when $\alpha_2 = \alpha_1$, the reaction is unity (also $V_1 = V_2$). The velocity diagram for $\Lambda = 1$ is shown in figure with the same values of V_a, U and W used for $\Lambda = 0$ and $\Lambda = ½$. It is obvious that if Λ exceeds unity, then $V_1 < V_0$ (i.e., nozzle flow diffusion).

Choice of Reaction and Effect on Efficiency

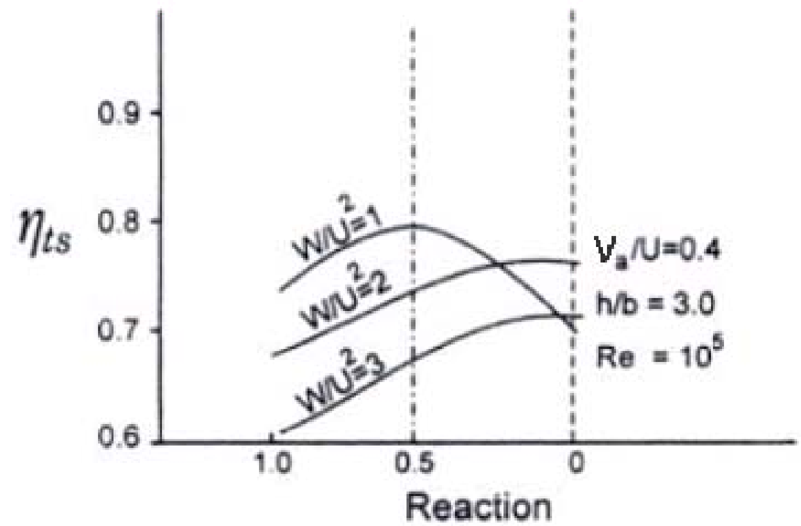
Equation (17) can be rewritten as $\Lambda = 1 + \dfrac{V_{w2} - V_{w1}}{2U}$

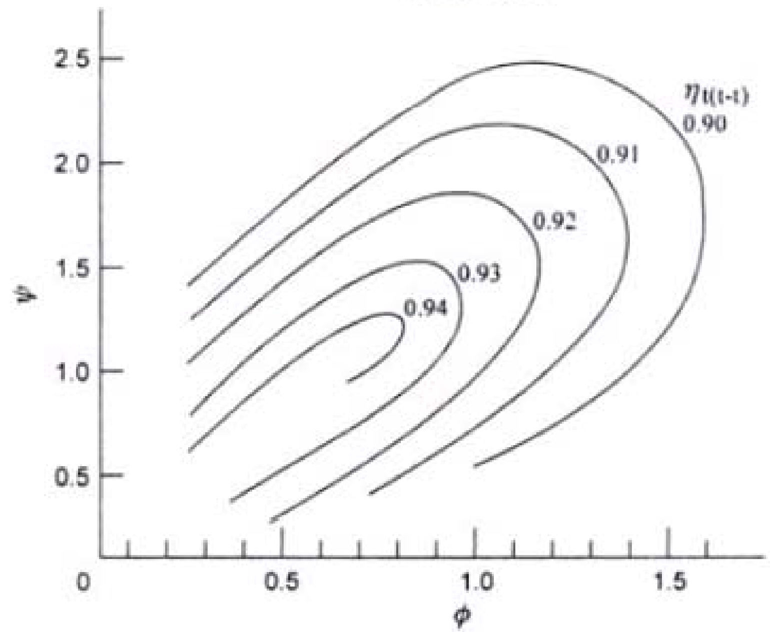
V_{w2} can be eliminated by using the relation

$$V_{w2} = \frac{W}{U} - V_{w1}, \quad \text{yielding} \quad \Lambda = 1 + \frac{W}{2U^2} - \frac{V_{w1}}{U}$$

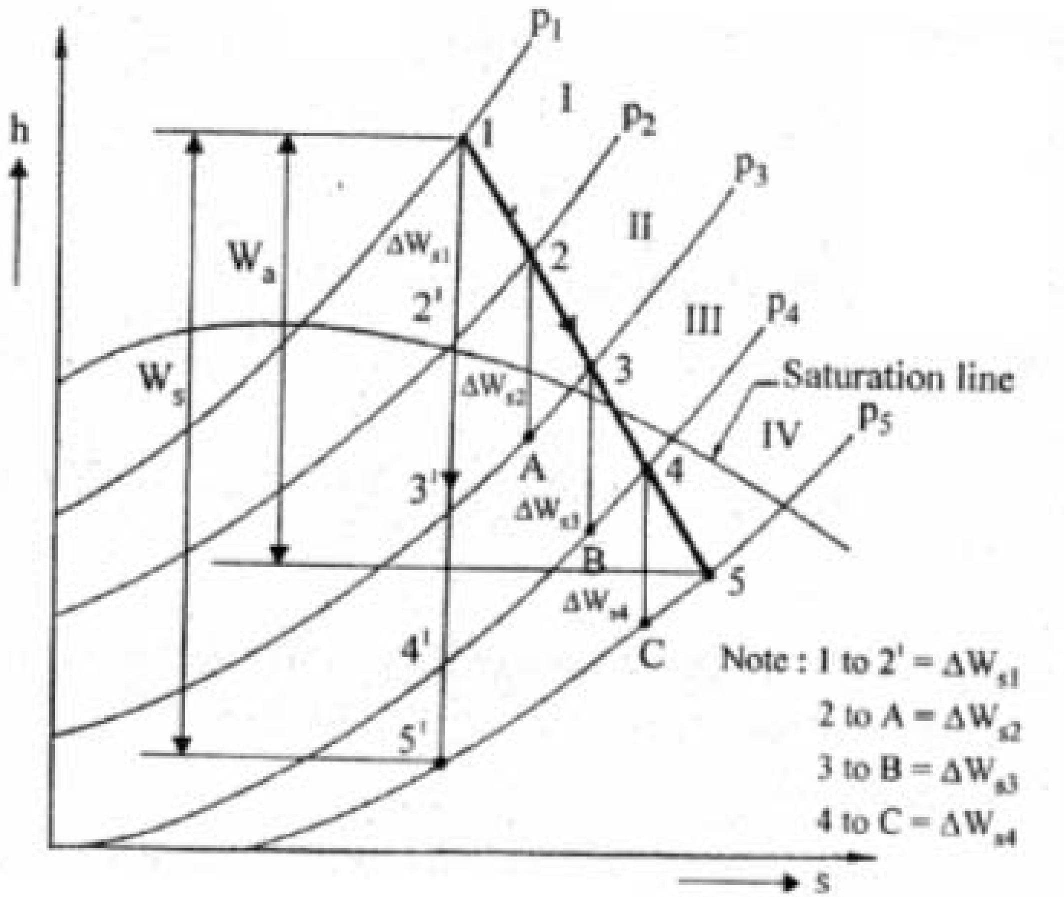
Degree of Reaction (… contd.)

In the figure, the total to static efficiencies are shown plotted against the degree of reaction.
When $W/U^2 = 2$, η_{ts} is maximum at $\Lambda = 0$. With lower loading, the optimum η_{ts} is obtained with higher reaction ratios. As shown in the figure, for a high total to total efficiency, the blade loading factor should be as small as possible, which implies the highest possible value of blade speed is consistent with blade stress limitations. It means that the total to static efficiency is heavily dependent upon the reaction ratio and η_{ts} can be optimized by choosing a suitable value of reaction.





Effect of Reheat Factor and Stage Efficiency



The thermodynamic effect on the turbine efficiency can be best understood by considering a number of stages, say 4, between states 1 and 5 as shown in the figure. Total expansion is divided into four stages having the same stage efficiency and pressure ratio.

Effect of Reheat Factor and Stage Efficiency

$$i.e., \frac{P_1}{P_2} = \frac{P_2}{P_3} = \frac{P_3}{P_4} = \frac{P_4}{P_5}$$

Let η_o is the overall efficiency of expansion and is defined as the ratio of actual work done per kg of steam to the isentropic work done per kg of steam between 1 and 5.

$$i.e.\ \eta_0 = \left(\frac{W_a}{W_s}\right) \quad \text{or} \quad \eta_0 = \frac{h_1 - h_5}{h_1 - h_5'}$$

The actual work done per kg of steam $W_a = \eta_o W_s$ (22)

Isentropic or ideal values in each stages are $\Delta W_{s1}, \Delta W_{s2}, \Delta W_{s3}, \Delta W_{s4}$.

Therefore the total value of the actual work done in these stages is,

$W_a = \Sigma(1\text{-}2)+(2\text{-}3)+(3\text{-}4)+(4\text{-}5)$

Also stage efficiency for each stage is given by

$$\eta_s = \frac{\text{actual work done/kg of steam}}{\text{Isentropic work done in stage}} = \frac{W_{a1}}{W_{s1}}$$

Effect of Reheat Factor and Stage Efficiency

For stage 1

$$i.e., \eta_{s1} = \frac{W_{a1}}{W_{s1}} = \frac{h_1 - h_2}{h_1 - h_2^{'}} = \frac{W_{a1}}{\Delta W_{s1}} or \Delta W_{a1} = \eta_{s1} \Delta W_{s1}$$

$$\therefore \Delta W_a = \Sigma \Delta W_a = \Sigma\left[\eta_{s1}\Delta W_{s1} + \eta_{s2}\Delta W_{s2} + \eta_{s3}\Delta W_{s3} + \eta_{s4}\Delta W_{s4}\right]$$

For same stage efficiency in each stage $\eta_{s1} = \eta_{s2} = \eta_{s3} = \eta_{s4}$

$$W_a = \eta_s \Sigma\left[\Delta W_{s1} + \Delta W_{s2} + \Delta W_{s3} + \Delta W_{s4}\right] = \eta_s \Sigma \Delta W_s \qquad (23)$$

From equation (22) and (23),

$$\eta_0 W_s = \eta_s \Sigma \Delta W_s$$

$$\therefore \eta_0 = \eta_s \frac{\Sigma \Delta W_s}{W_s} \qquad (24)$$

The slope of constant pressure lines on h-s plane is given by

$$\left(\frac{\partial h}{\partial s}\right)_p = T$$

Effect of Reheat Factor and Stage Efficiency

This shows that the constant pressure lines must diverge towards the right. Therefore $\frac{\Sigma \Delta W_s}{W_s} > 1$

.

For expansion process. It is obvious that the enthalpy increases when we move towards right along the constant pressure line. Hence the summation of ΔW_{s1} ΔW_{s1} etc., is more than the total isentropic enthalpy drop Ws

The ratio of summation of isentropic enthalpy drop for individual stage to the total isentropic enthalpy drop as a whole is called Reheat factor. Thus

$$RF = \frac{\Sigma\left[\Delta W_{s1} + \Delta W_{s2} + \Delta W_{s3} + \Delta W_{s4}\right]}{W_s} = \frac{\Sigma\left[(1-2') + (2-a') + (3-b') + (4-c')\right]}{(1-5)}$$

$$RF = \frac{\Sigma \Delta W_s}{W_s} \qquad (25)$$

Therefore the overall efficiency of the expansion process,

$$\eta_0 = \eta_{stage} \times RF \qquad (26)$$

Since $RF = (\Sigma \Delta W_s / W_s) > 1$

the overall efficiency of the turbine η_0 is greater than stage efficiencies η_s

$$i.e., \eta_0 > \eta_s \text{ for turbines} \qquad (27)$$

Merits and Demerits of Reheating

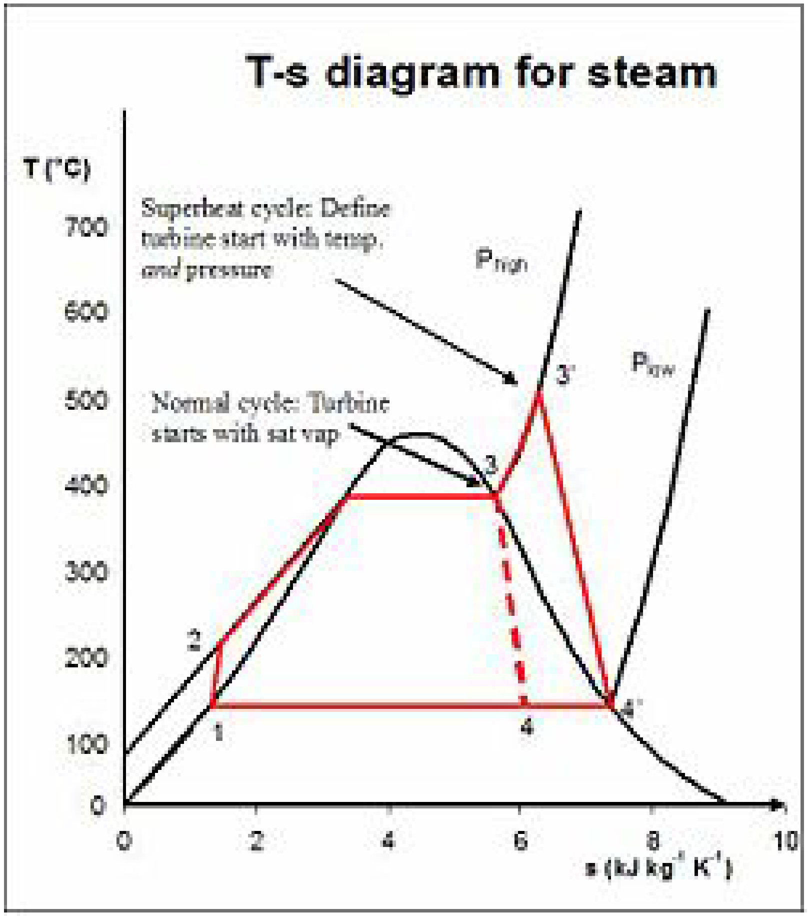
Advantages of Reheating

1. There is an increase in the overall output of turbine.
2. Erosion and corrosion problems in steam turbines are reduced.
3. There is an improvement in overall thermal efficiency of the turbine.
4. Condition of steam in last stage is improved.

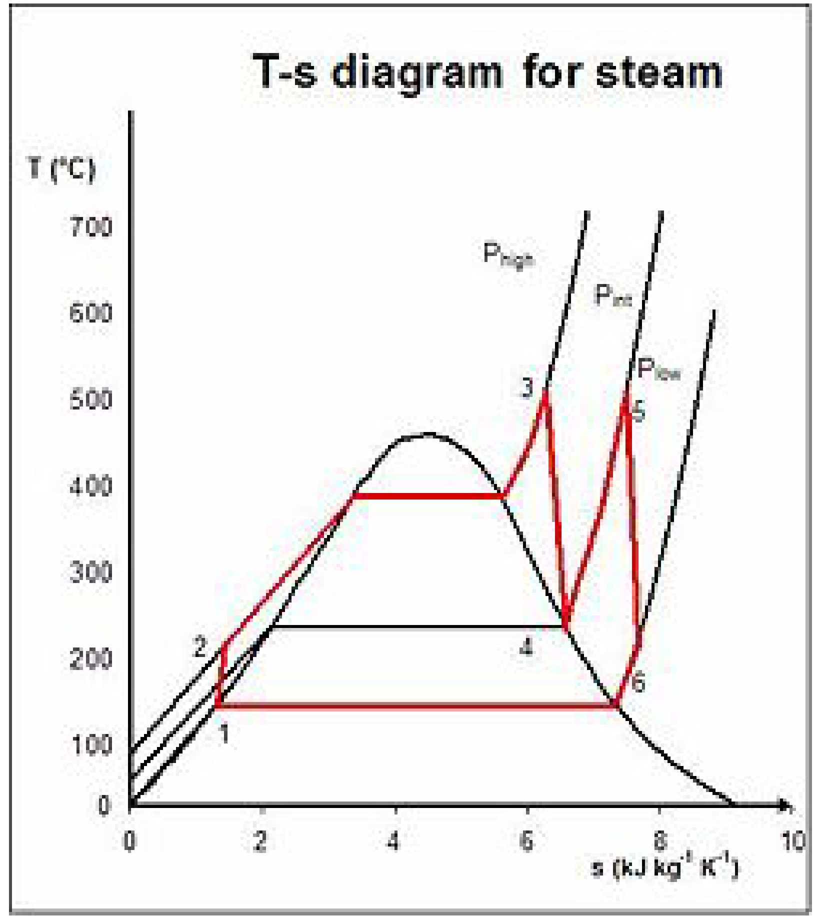
Demerits

1. Capital cost required for reheating
2. Increase in thermal efficiency is not appreciable compared to expenditure incurred in reheating for smaller capacity turbines.

Steam Turbine Cycle (Subcritical)

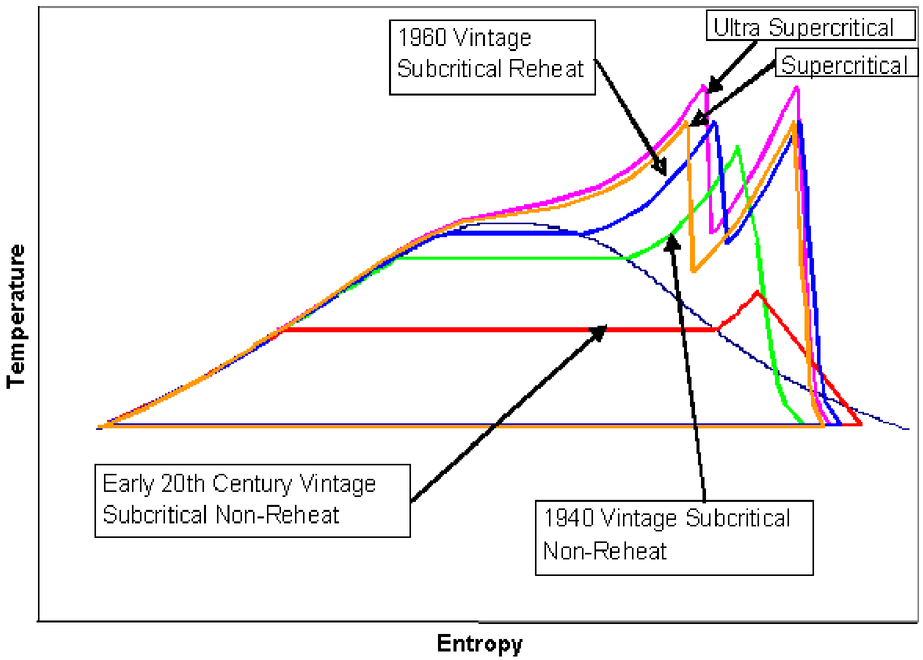


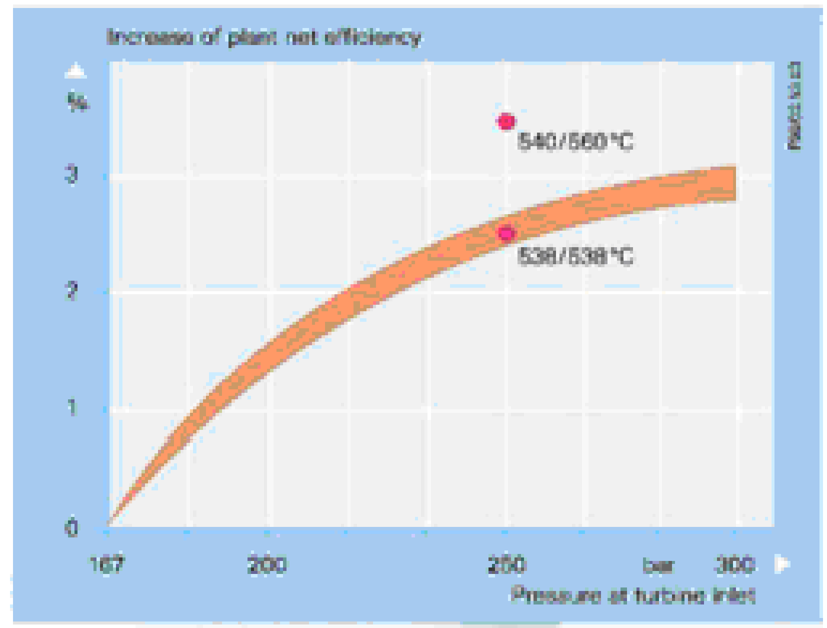
Rankine cycle with superheat



Rankine cycle with reheat

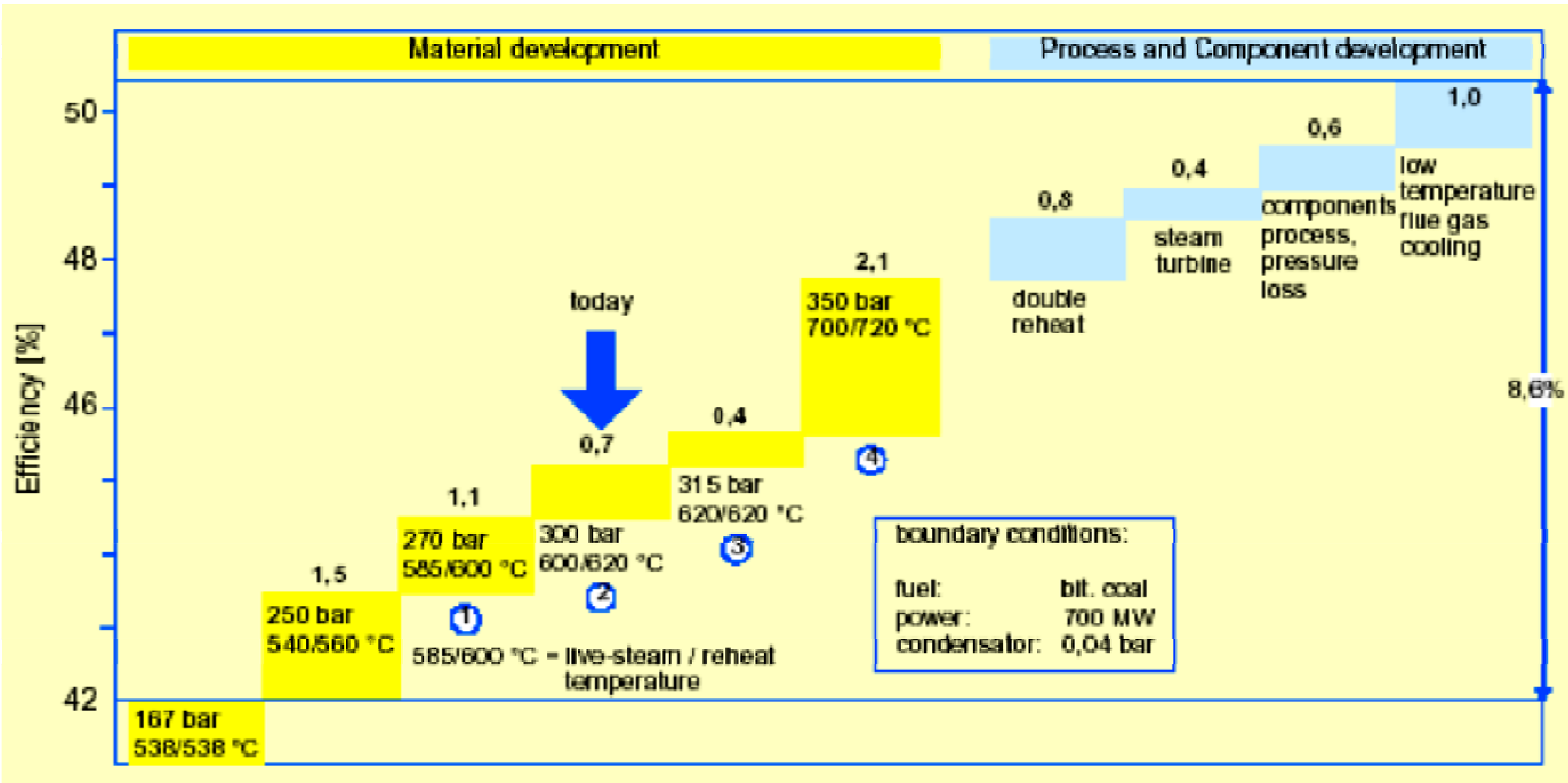
Steam Turbine Cycle (Supercritical)



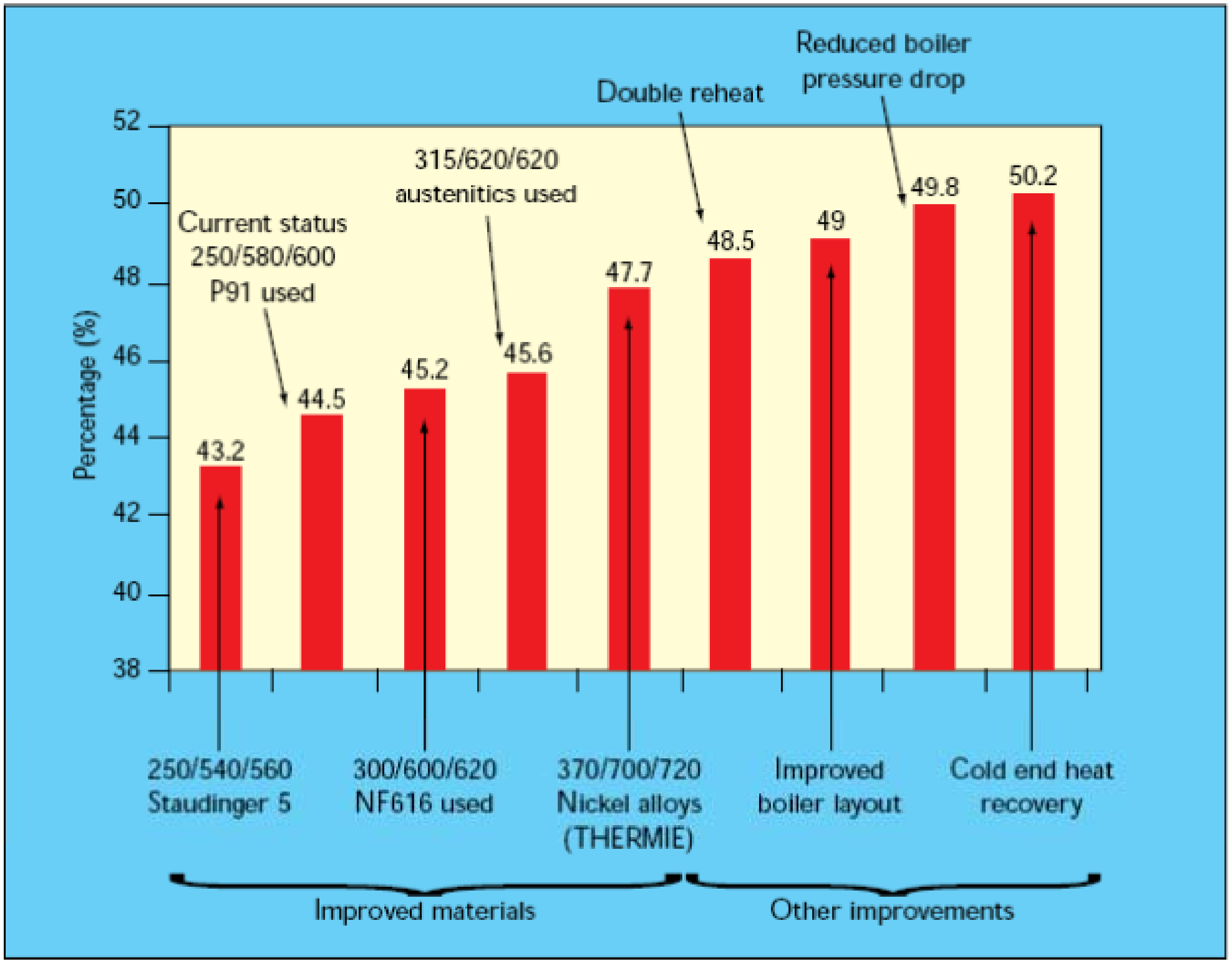


There are inherent advantages of supercritical steam cycle:
High pressures and temperatures enhance thermal efficiency; avoid use of steam dryers and steam separators; and reduce CO_2 emissions.

Development of Conventional Coal Fired Steam Power Plants

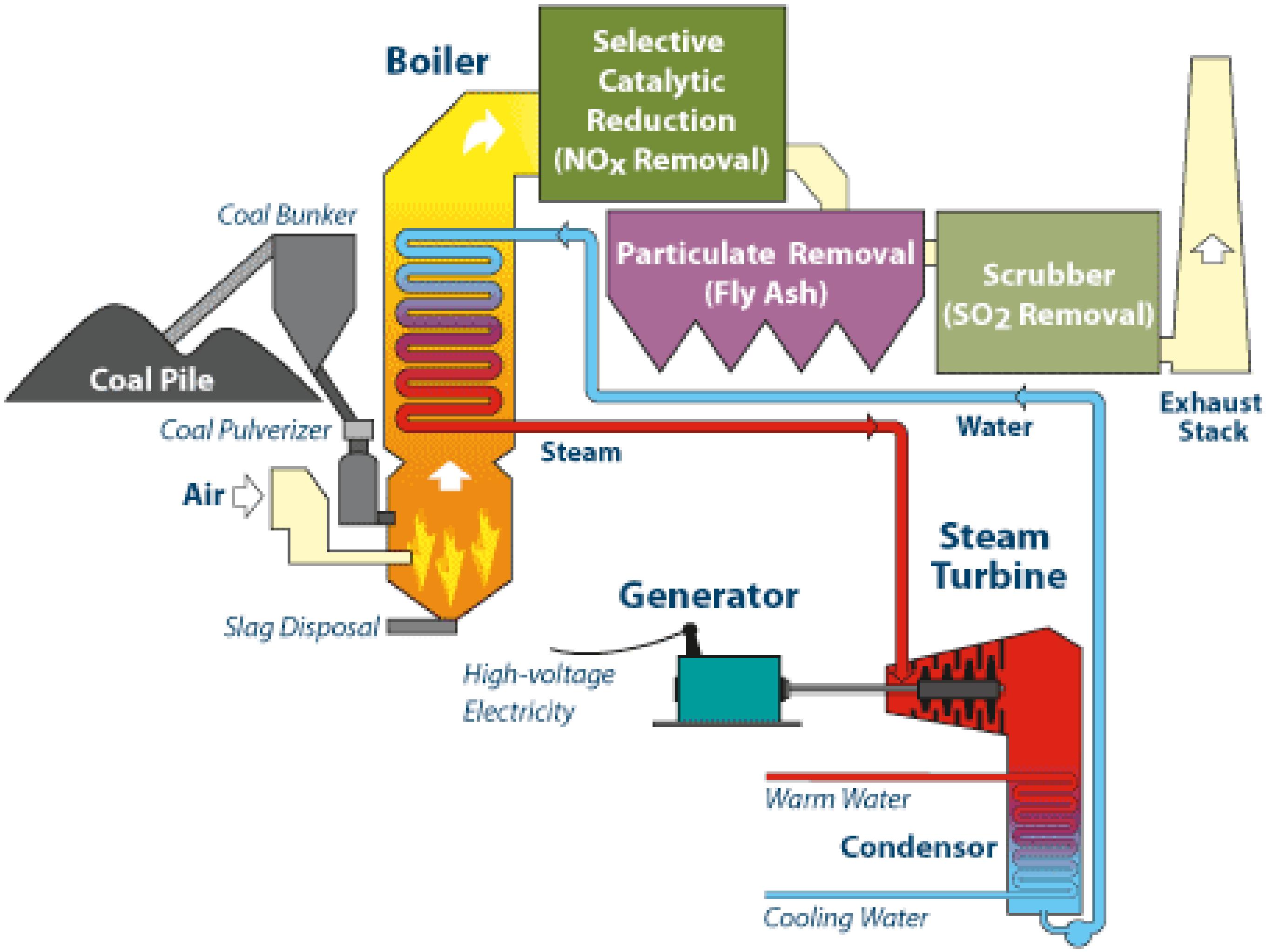


Potential Efficiency Improvements

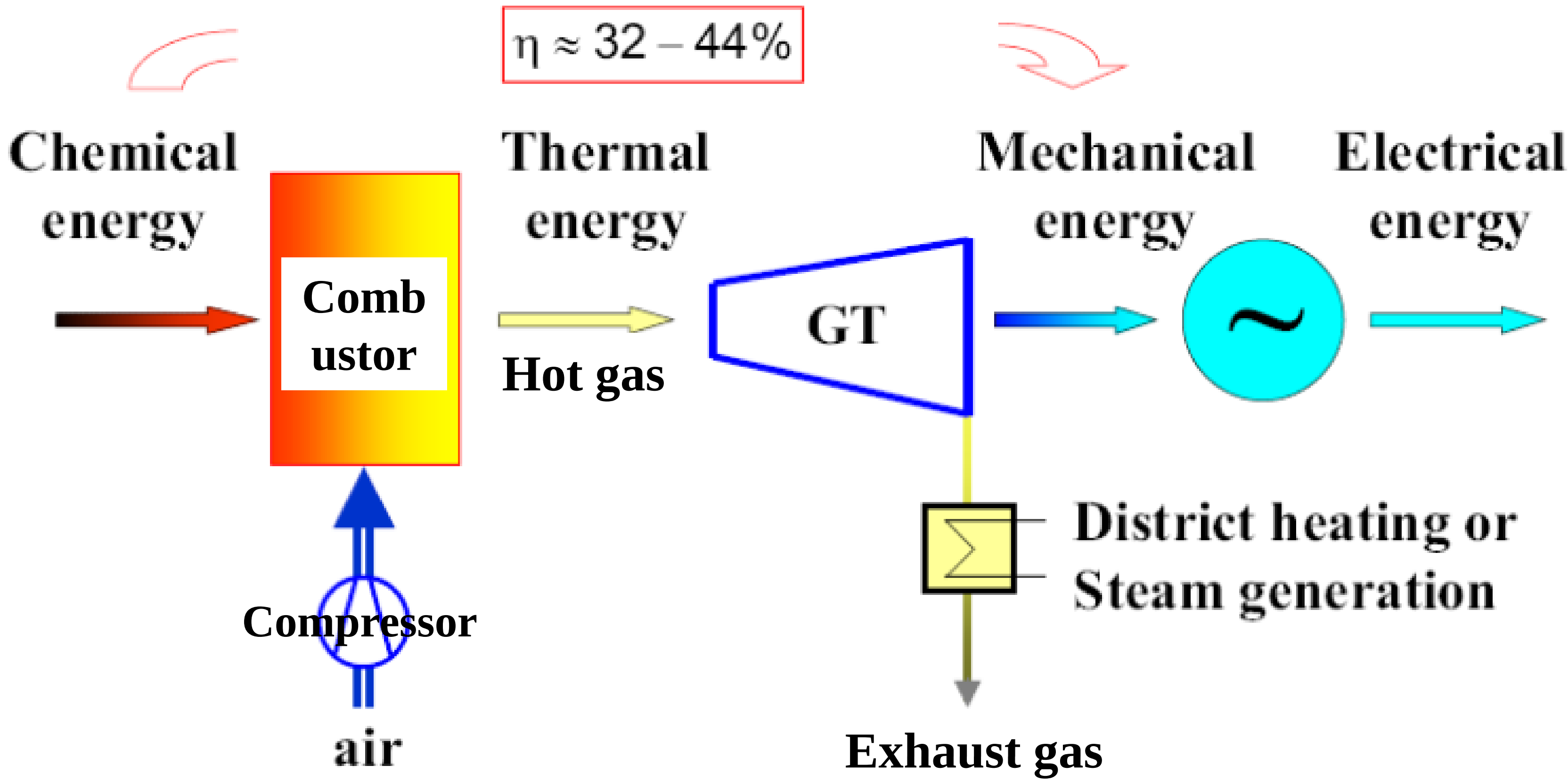


(Based on a 700MW bituminous coal fired plant, with a 40mbar condenser pressure)

Supercritical Steam Turbine Plant

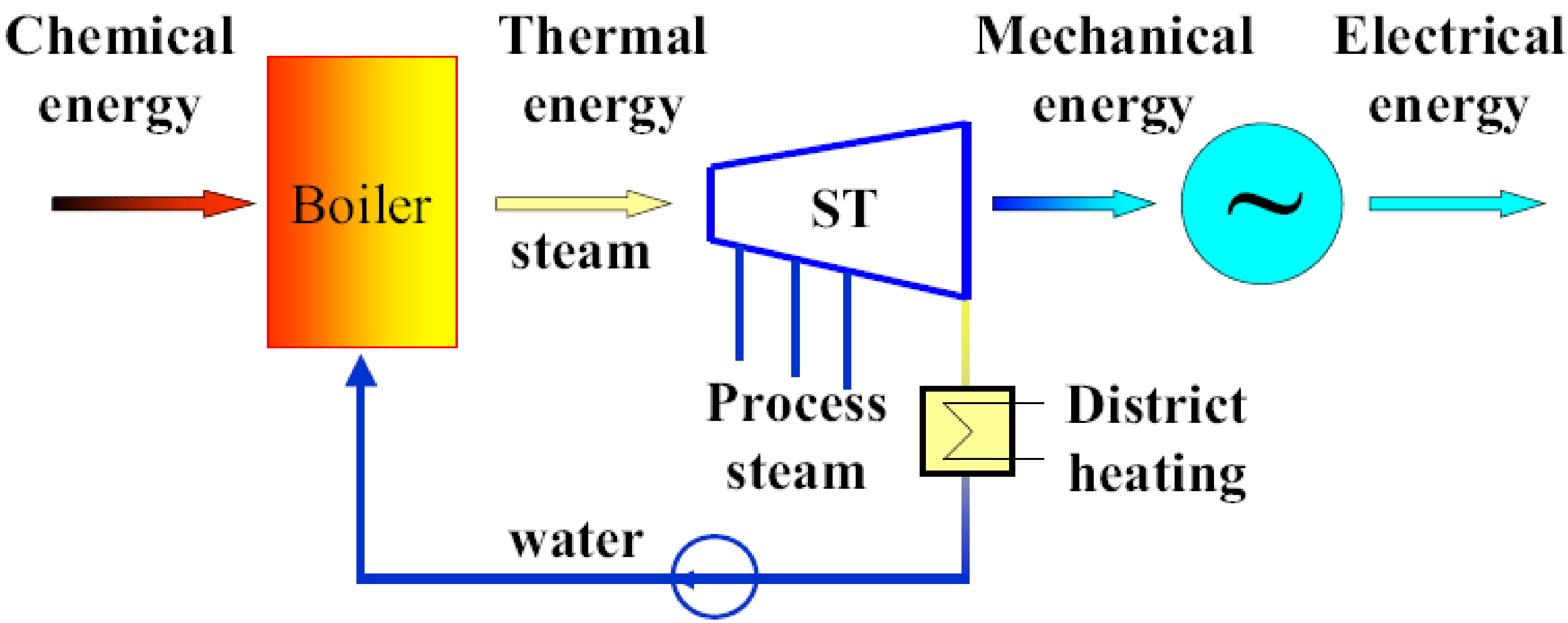


Gas Turbine Power Plant



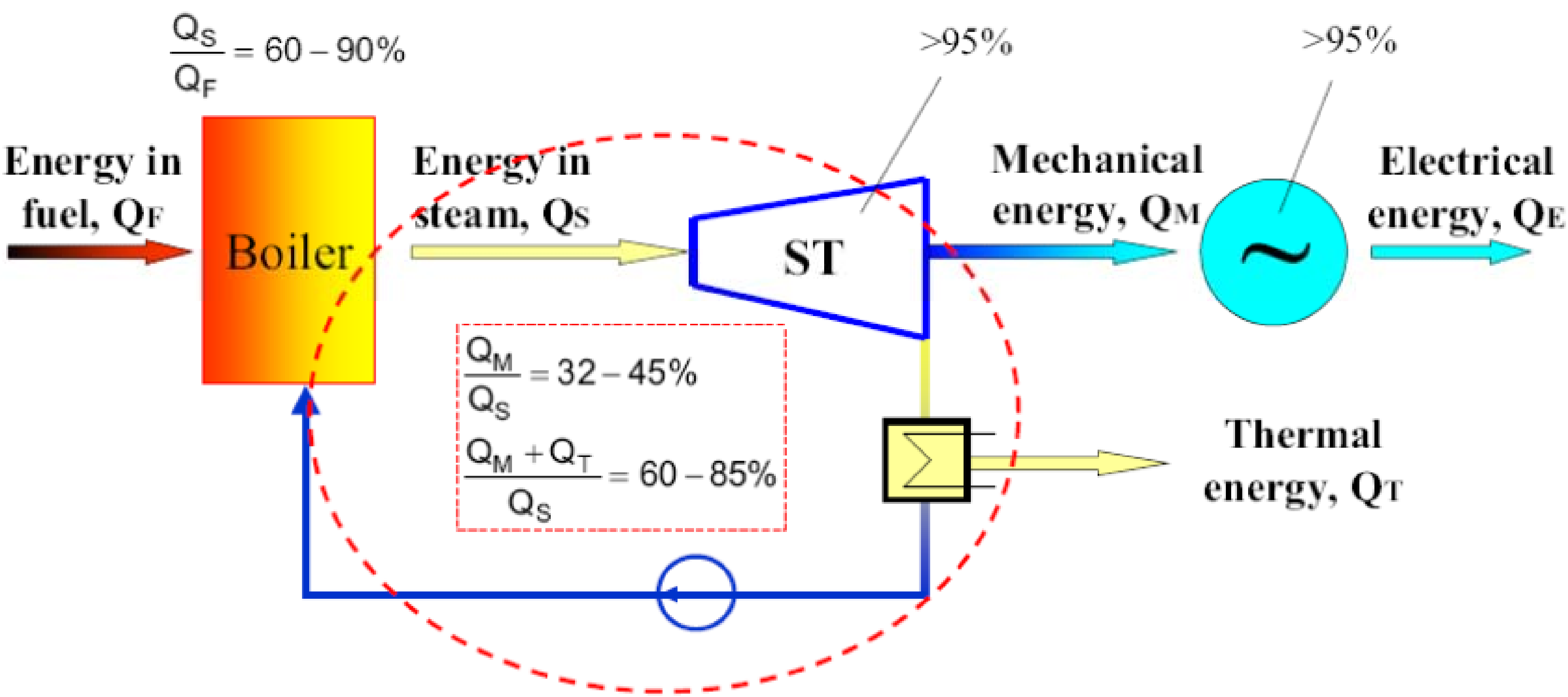
17-$Boiler_control

Steam Turbine Power Plant



Steam Turbine Power Plant
Power Generation

Efficiency = energy out / energy in



Overall efficiency = (QE+QT)/QF = (60-90%) * (35-85%) * 95% = 20-72%

Influence Parameters in Steam Turbine Design

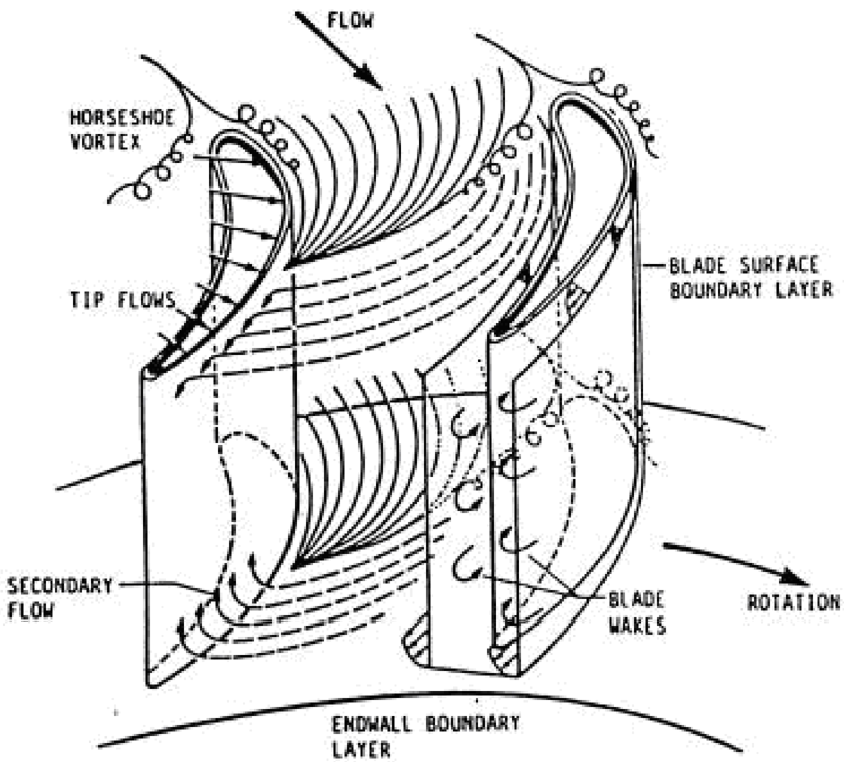
Aerodynamic Parameters

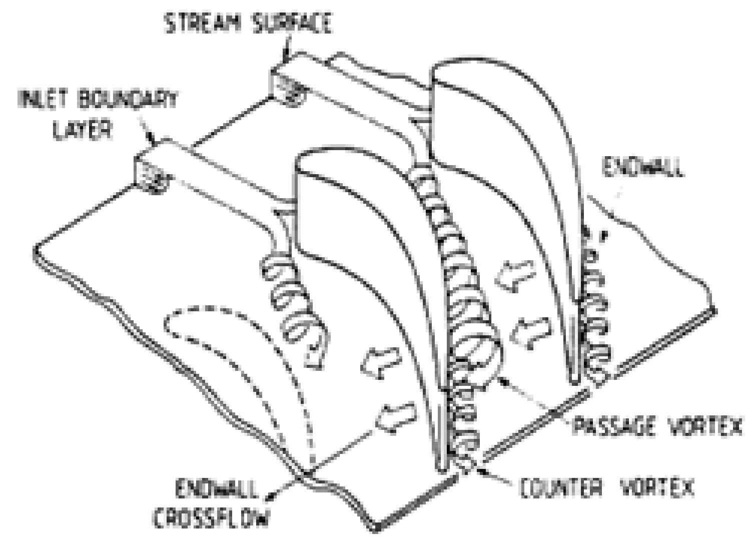
- Incidence angle
- Stator – rotor interaction
- Degree of reaction
- Whirl distribution

Geometric Parameters

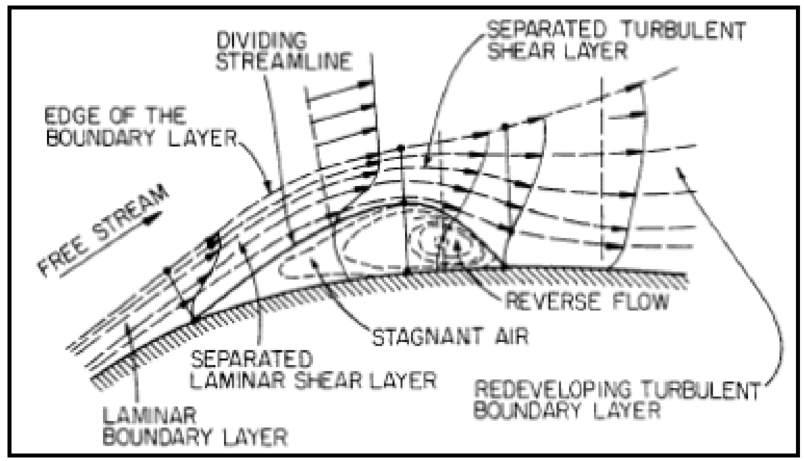
- Blade profile
- Blade solidity
- Blade aspect ratio
- Blade stacking

Loss Mechanisms



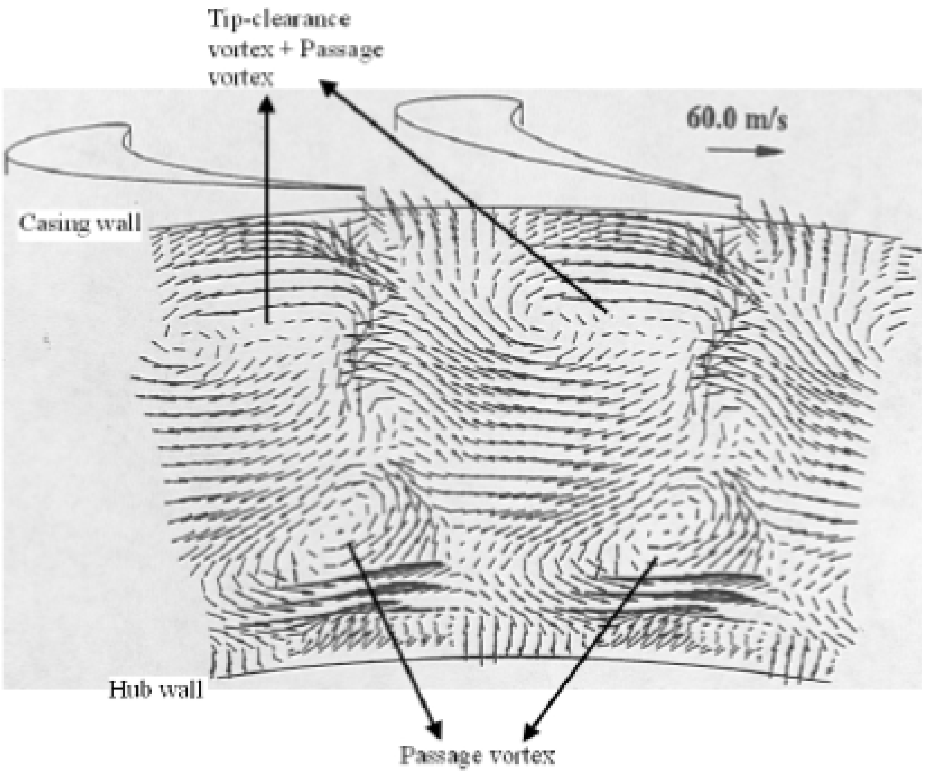


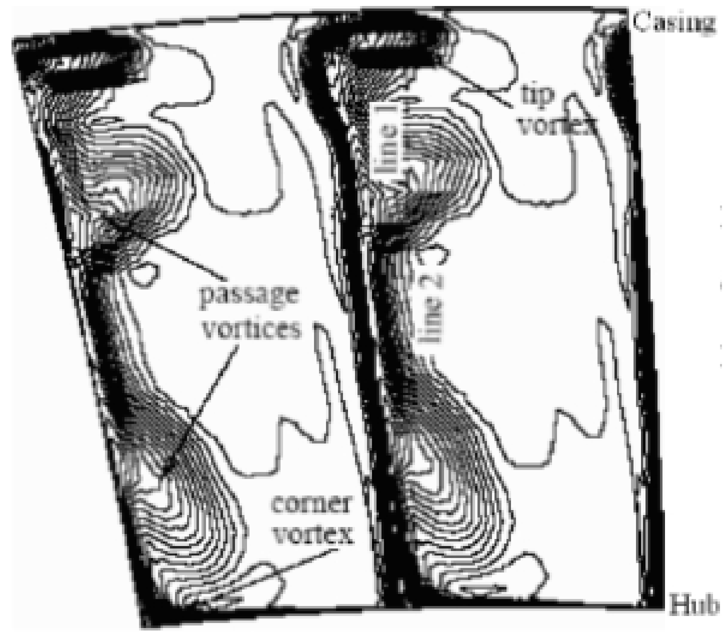
Flow through turbine blade rows is viscous, unsteady and three dimensional.
Losses comprise: profile loss; end wall loss; secondary flow loss; tip clearance loss.



Laminar separation bubble

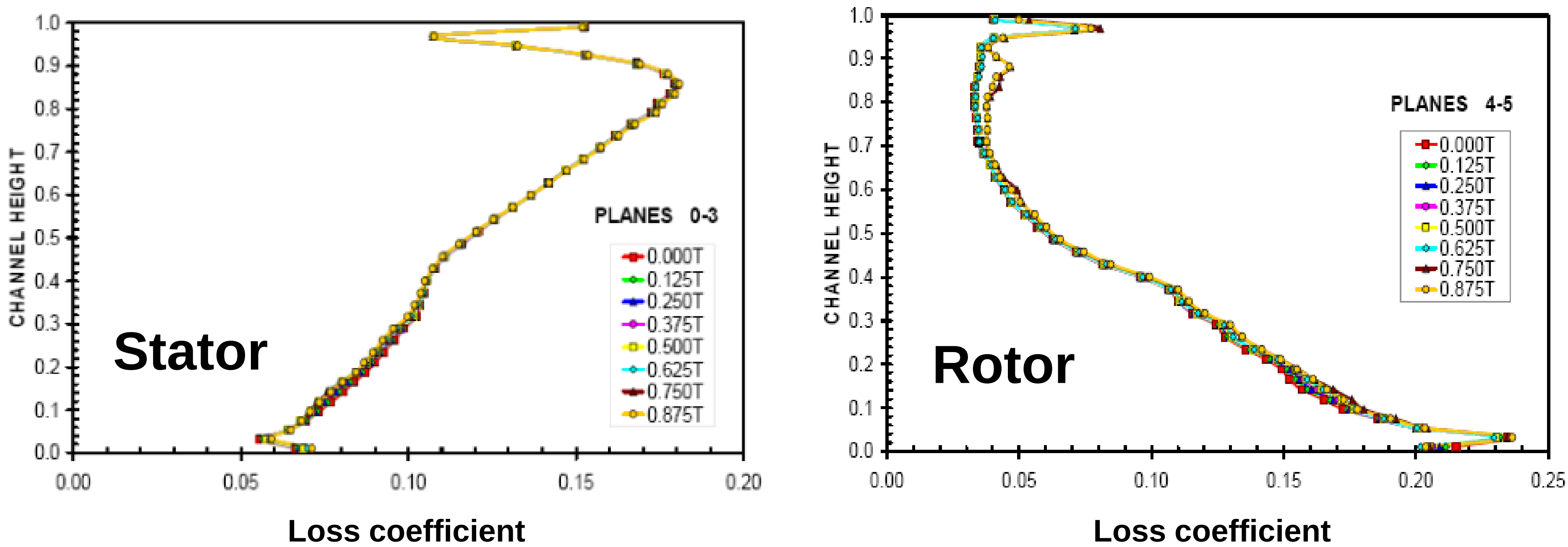
Loss Mechanisms





Tip clearance and passage vortices

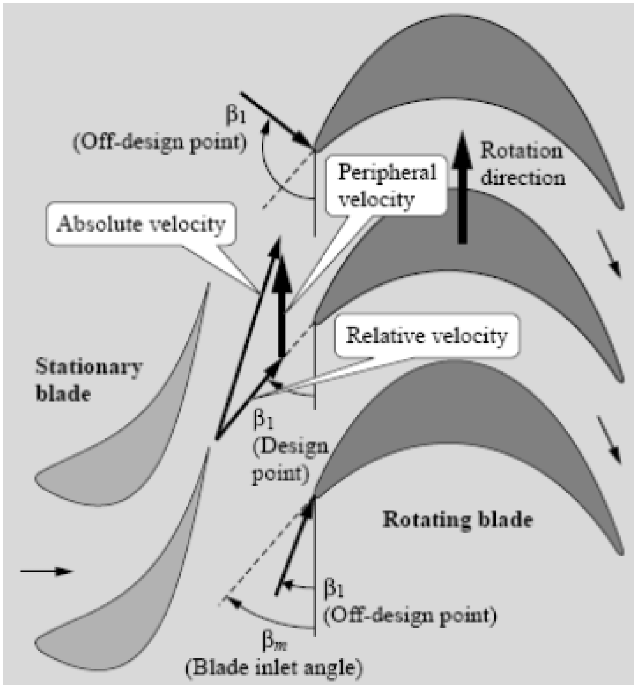
Spanwise Variation of Loss Coefficient

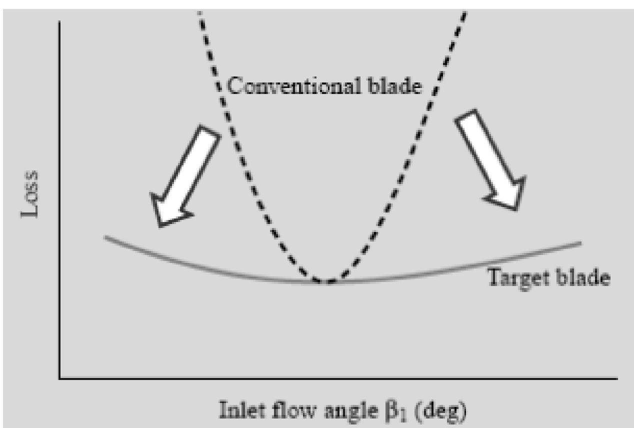


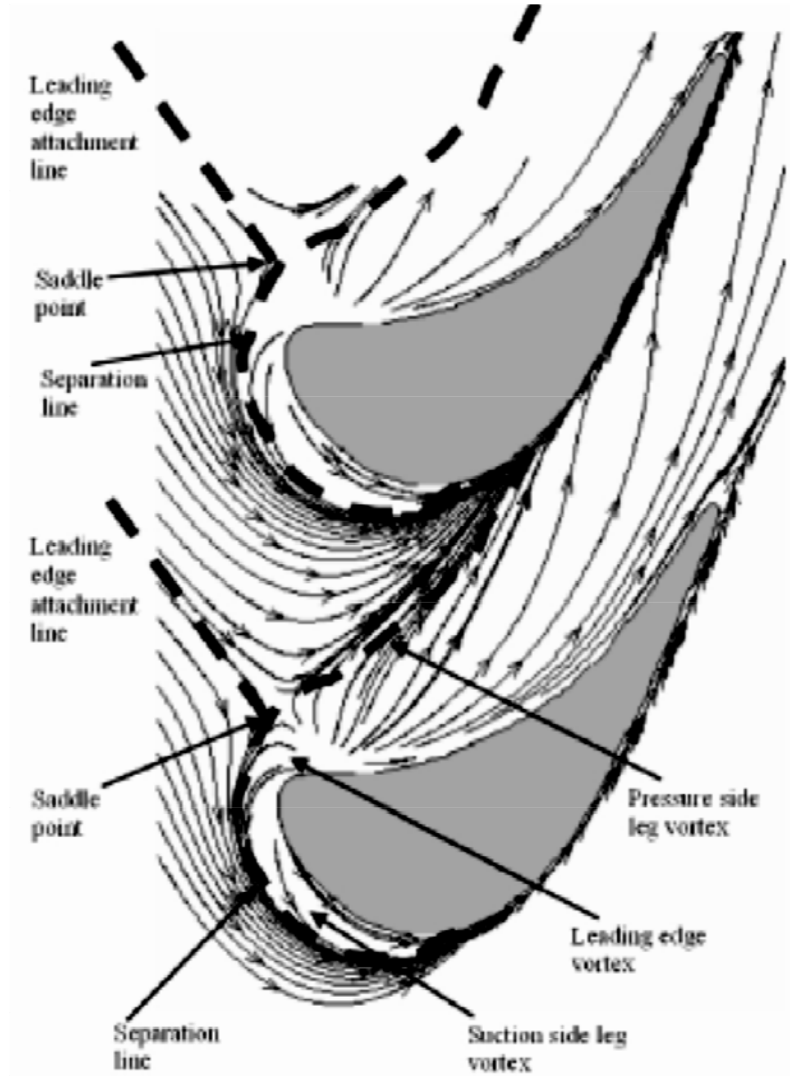
Loss coefficient across the last stage stator and rotor of a large steam turbine (Power = 1000 MW ; Speed = 3000 rpm ; length of rotor blade = 1.085 m)

[Ref: M. Stastny, et al, Czech Technical University, Praha]

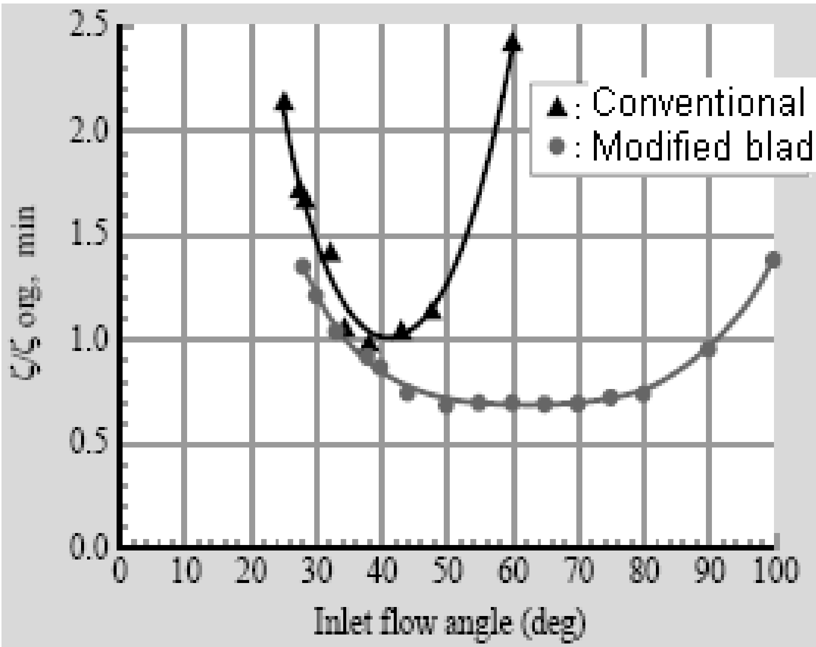
Effect of Incidence Angle



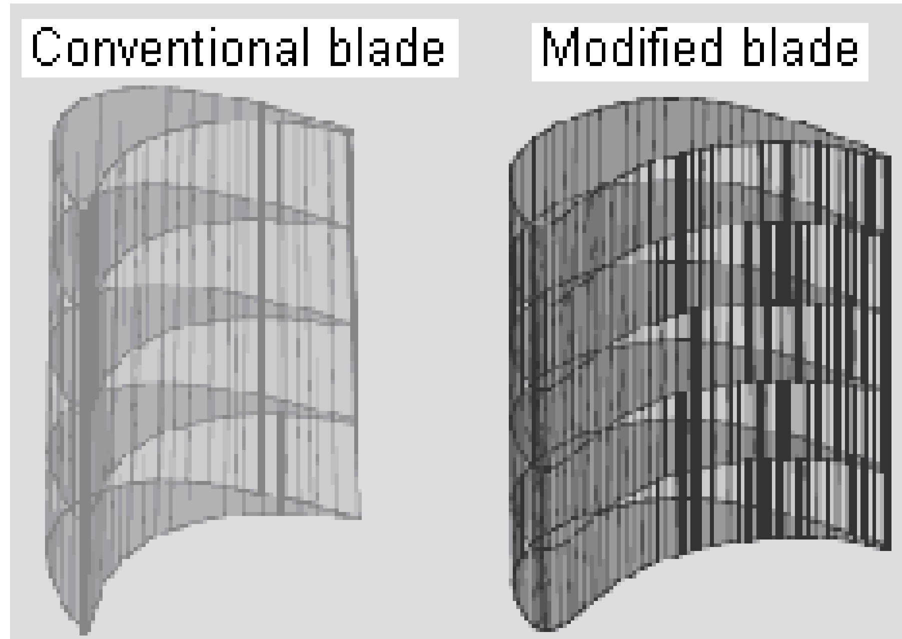




Effect of Incidence Angle



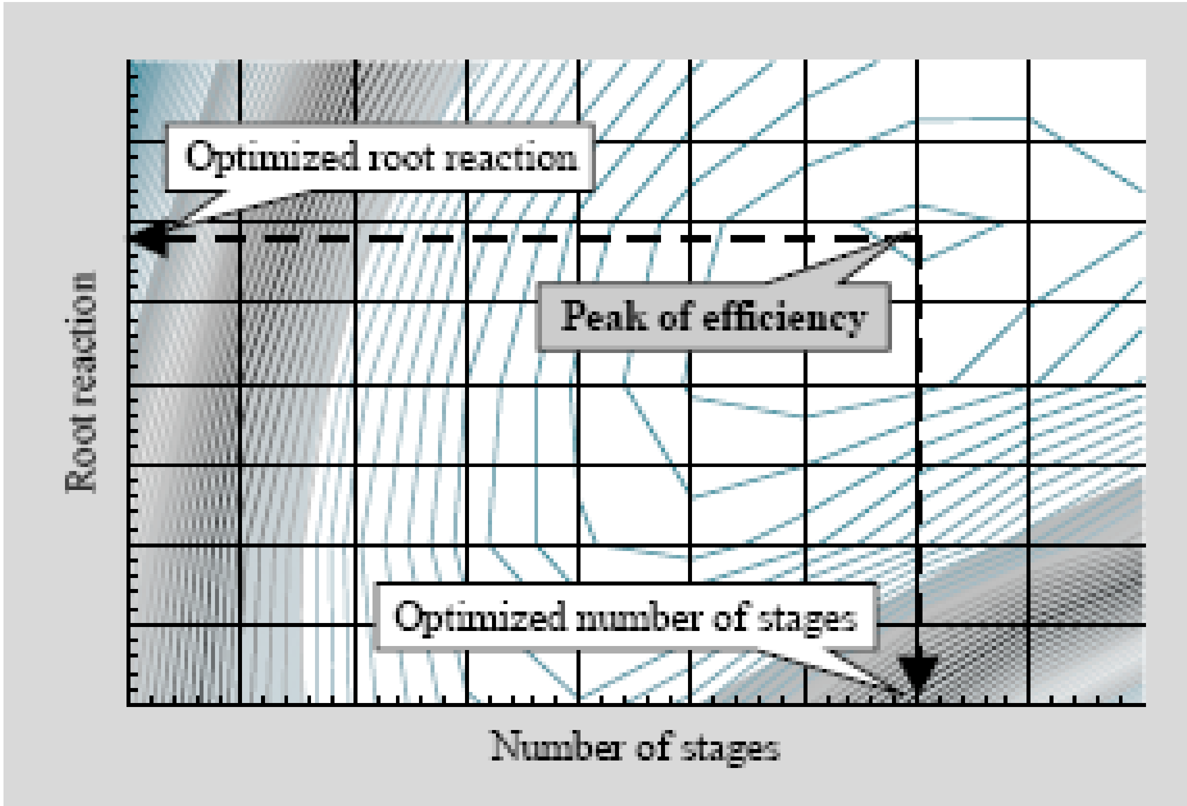
Variation of profile loss coefficient with inlet flow angle



Conventional and modified low loss turbine blades

The blade sections should be designed for low loss incidence, both under design and off-design conditions. This requires extensive cascade testing of blade sections to arrive at acceptable low loss profiles.

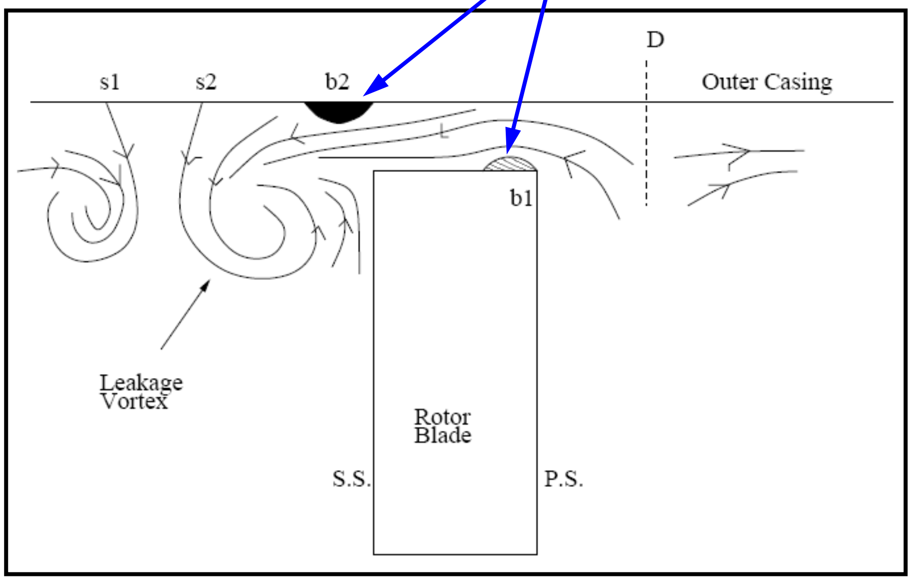
Degree of Reaction



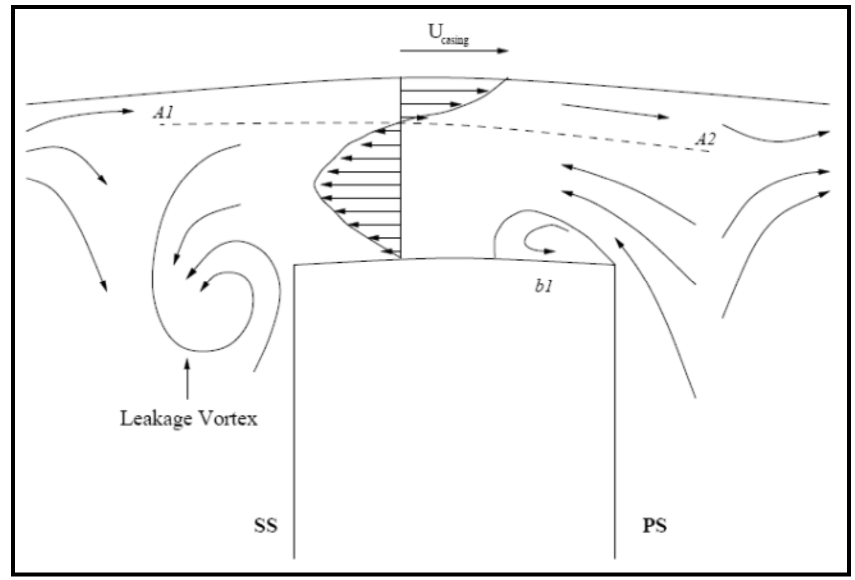
Improvement in the efficiency of turbine stages can be achieved by smaller root diameter and more number of stages. Optimization studies need to be made taking into account the blade root diameter, number of stages and the **degree of reaction**.

Schematic of Tip Leakage Flow

Separation Bubble



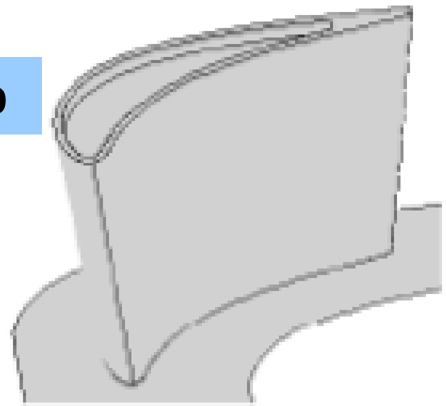
Linear Cascade

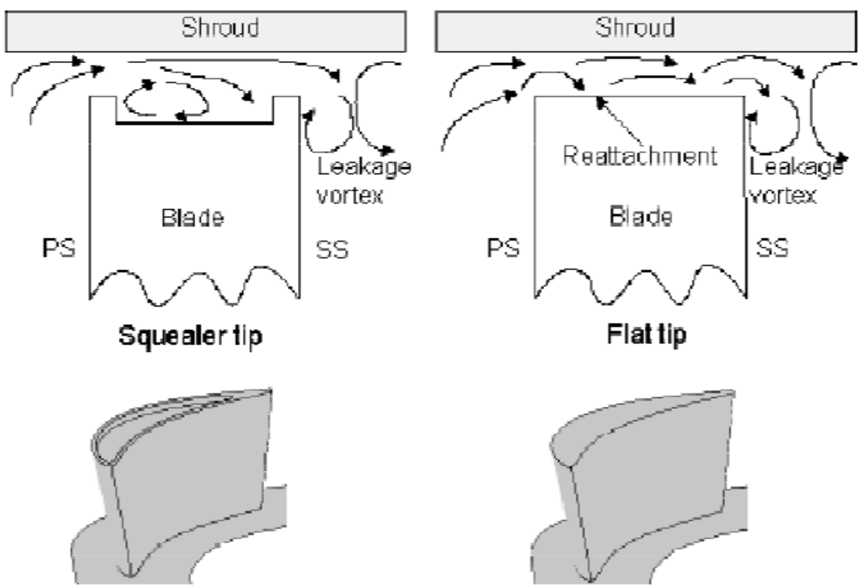


Turbine Rotor

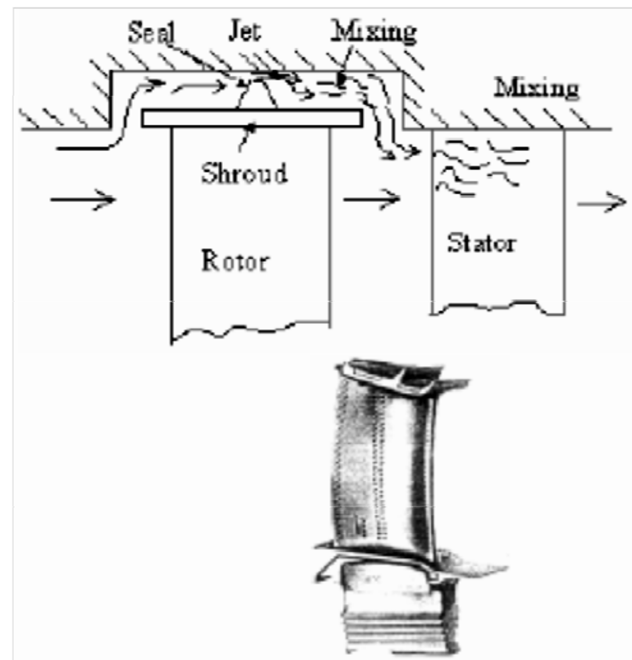
Squealer and Tip Leakage Flow

Squealer tip





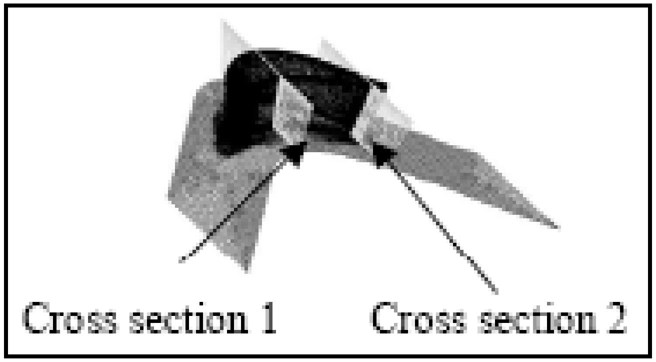
Tip leakage flow for unshrouded blade

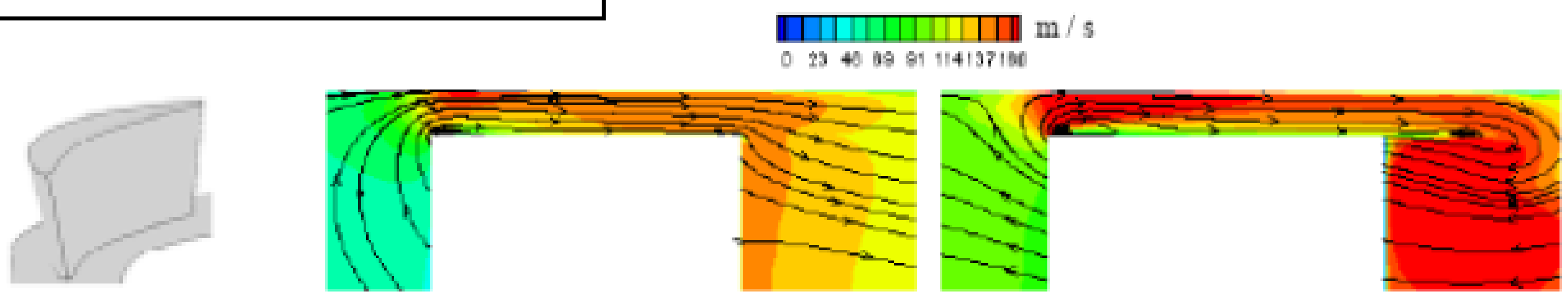


Tip leakage flow over a shrouded tip seal

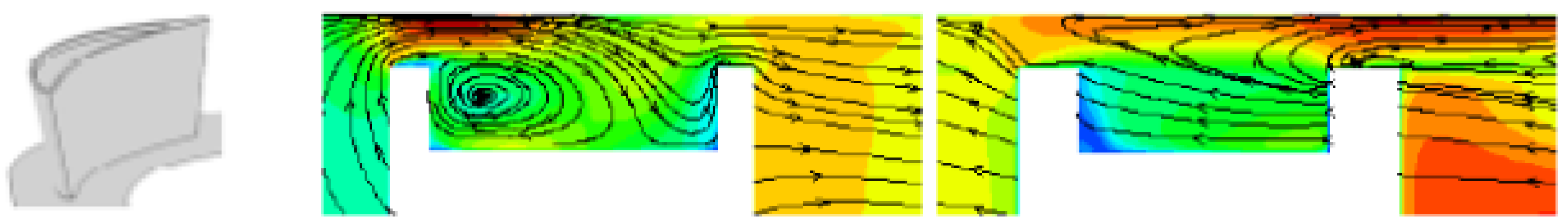
Squealer and Tip Leakage Flow

(Streamlines and Velocity Magnitude Contours)



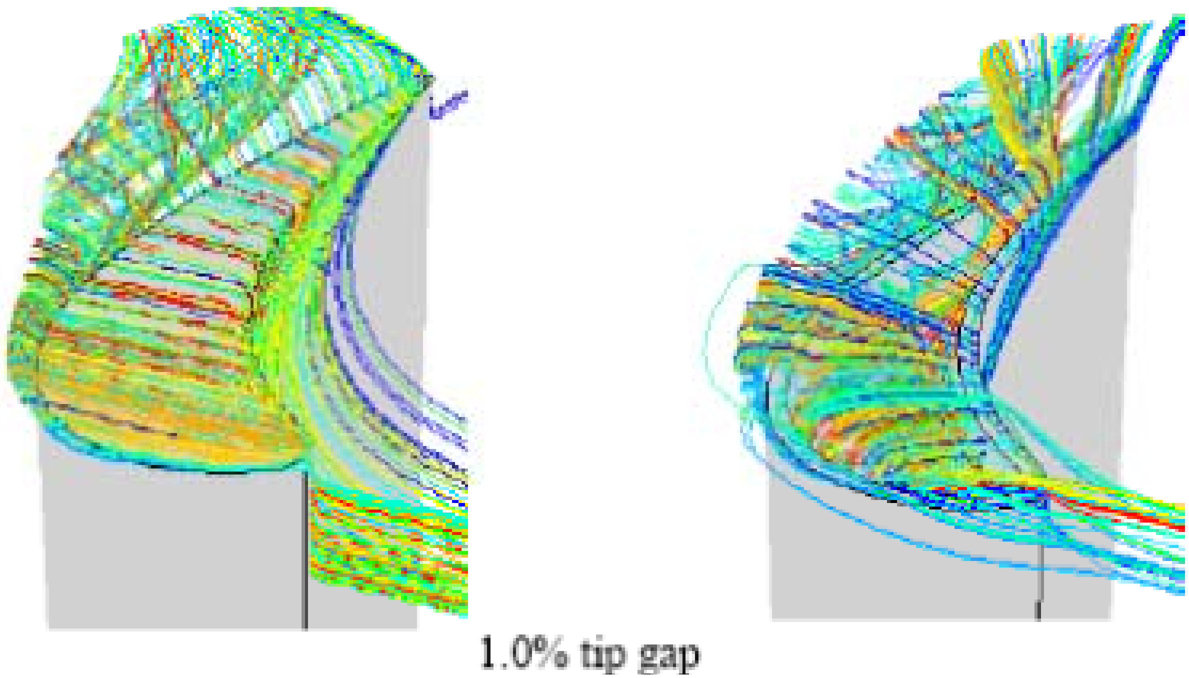


Flat blade tip, 20% and 60% axial chord distance



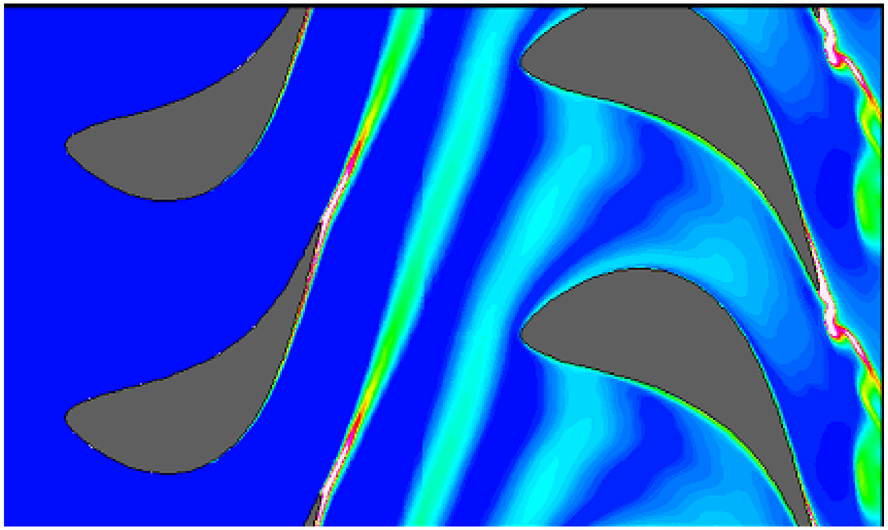
Squealer blade tip, 20% and 70% axial chord distance

Flow Streamlines over the Blade

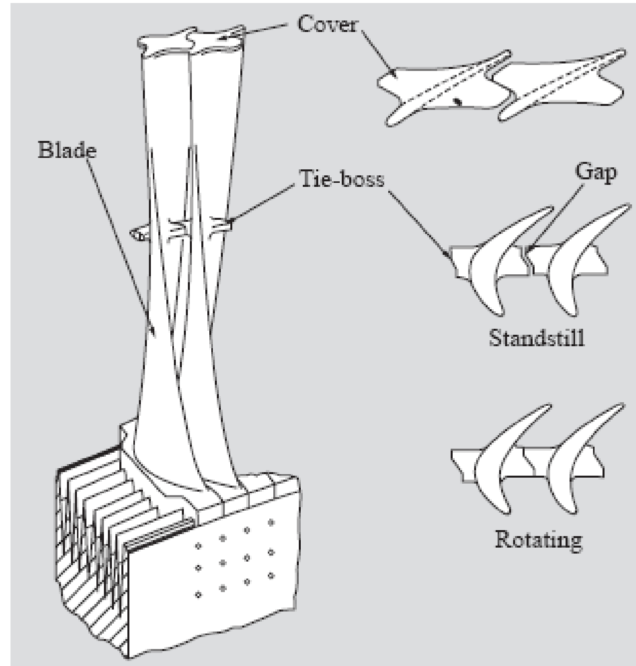


Flat tip **Squealer tip**

Unsteady Blade Row Interaction



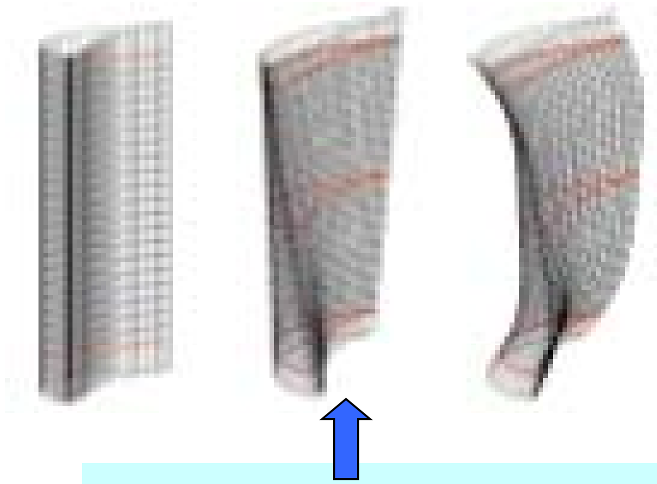
Blade-wake interaction

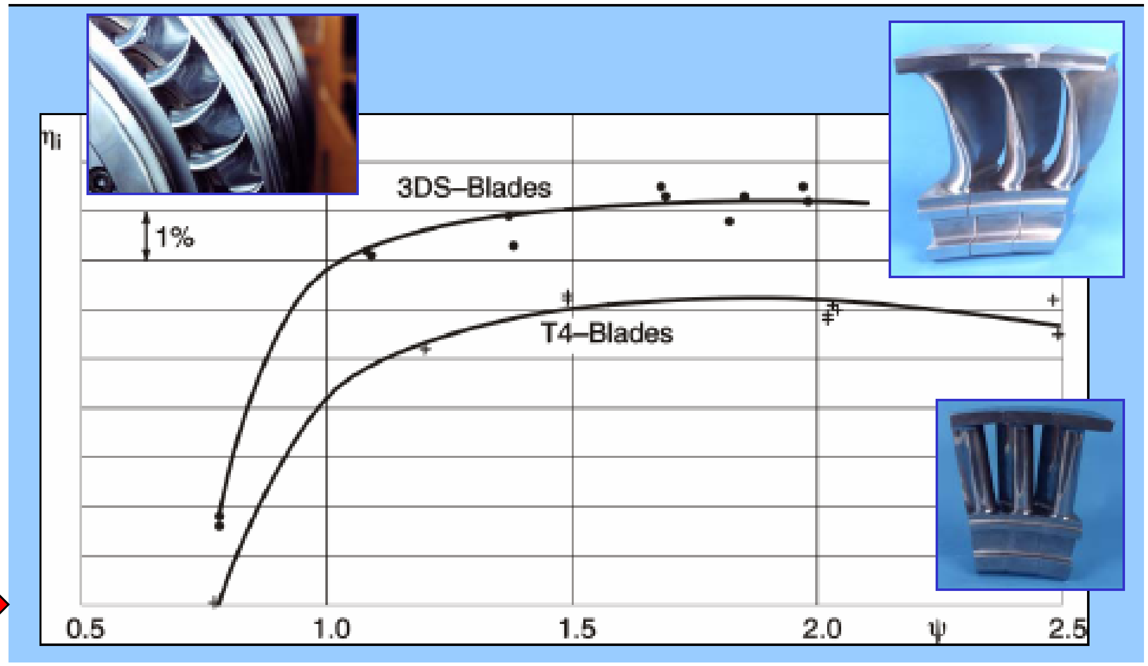


Stator wakes induce unsteady pressure distribution on the rotor blades, leading to unsteady forces, vibration and fatigue failure

Blades of last stages are inter-connected by shroud covers and tie-bosses. Covers interlock as blades begin to rotate, and tie-bosses interlock once rotation is under way.

3 D Blading

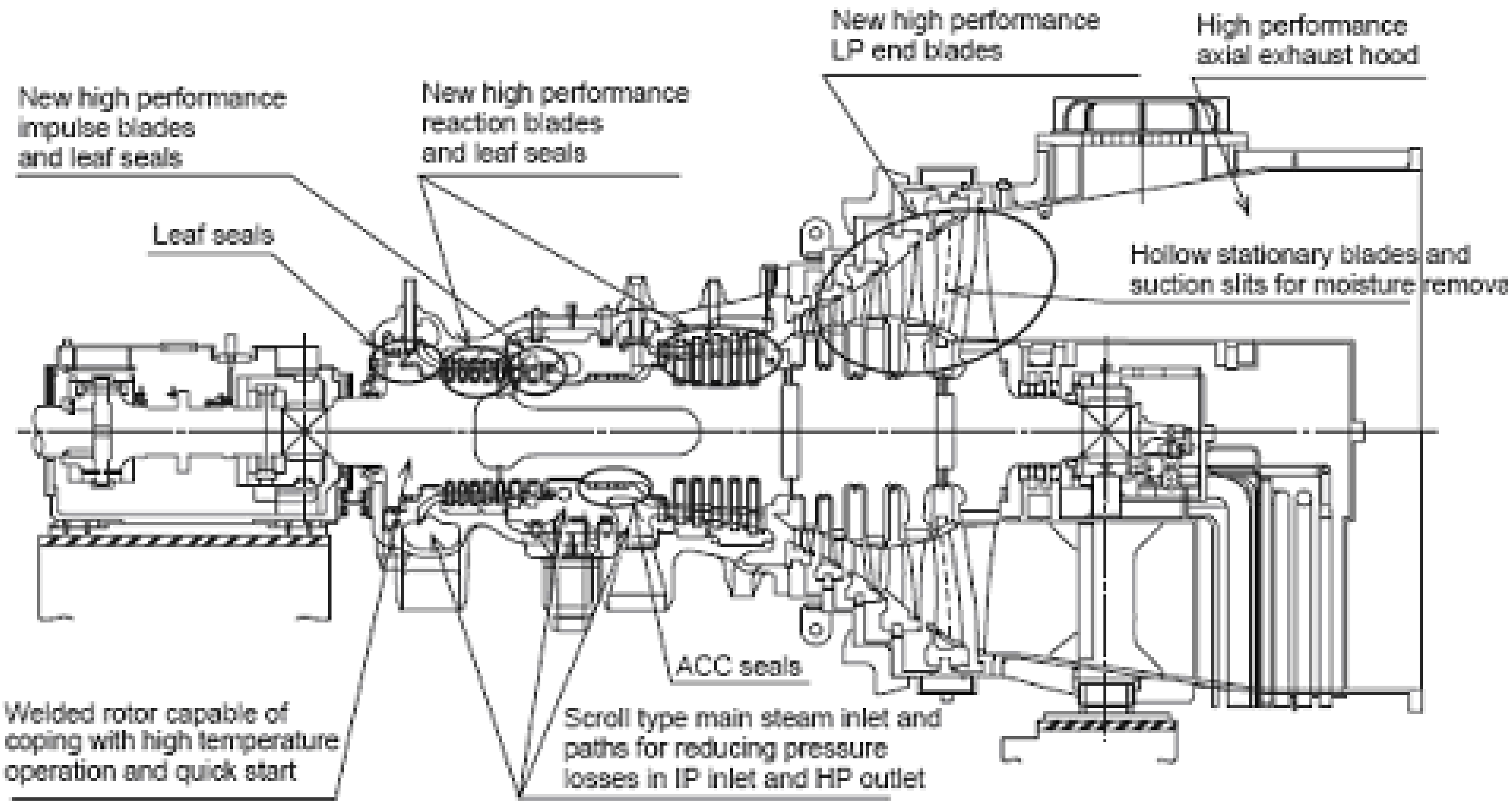




Conventional and 3D blades and comparison of thermal efficiencies

A bowed stacking line reduces the strength of the secondary passage vortices and the flow tends to shift towards more efficient mid-span region, thus increasing the turbine stage efficiency.

High Performance Steam Turbine

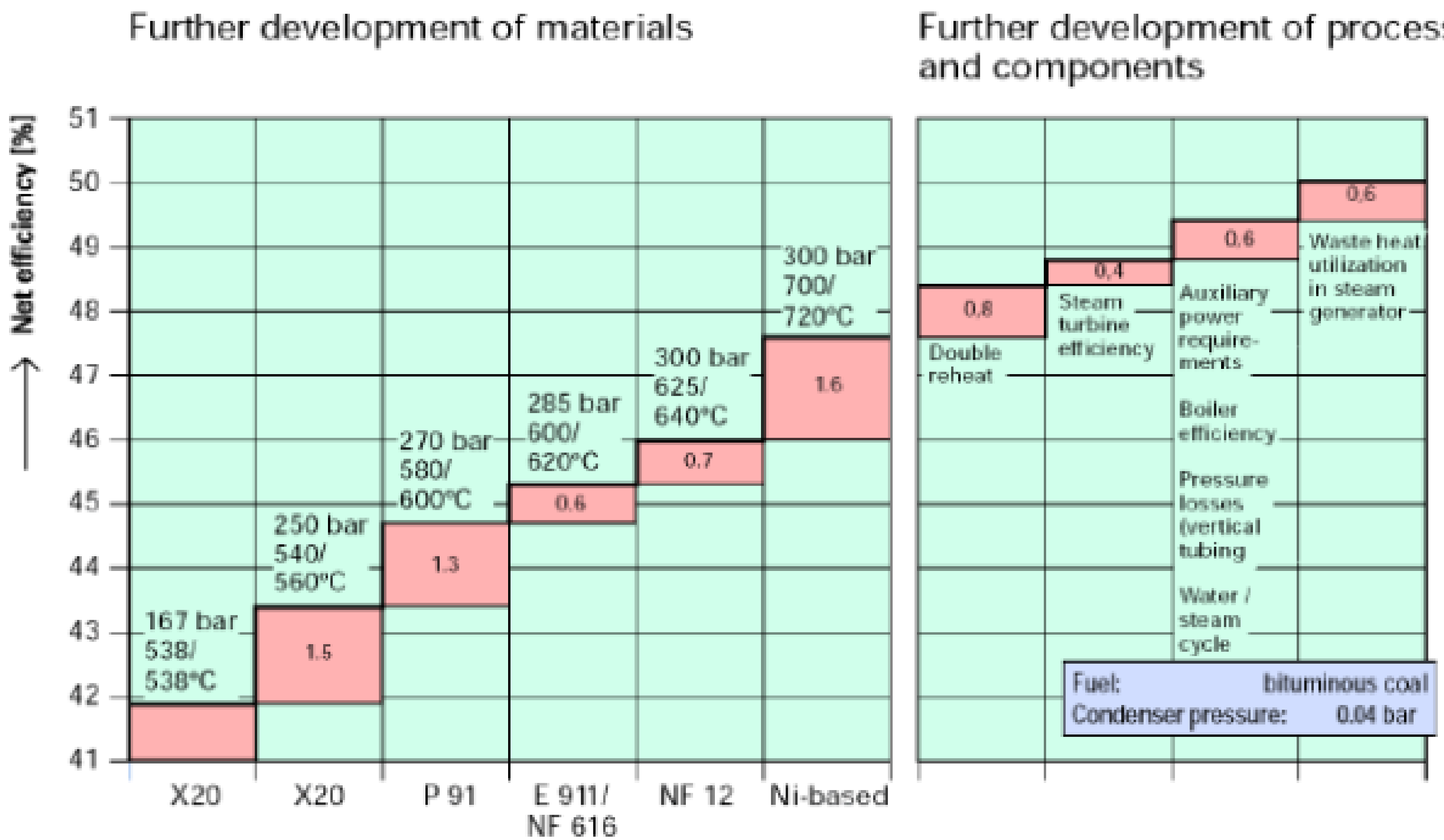


A Mitsubishi high performance steam turbine of 105 MW capacity showing the location of new technologies

High Performance Steam Turbine

High performance LP end blades (3600 rpm - 36 inches) of a typical Mitsubishi high performance steam turbine

Increasing Net Efficiency



Source: J. Franke, R. Kral, and E. Wittchow (1999): Steam generators for the next generation of power plants aspects of design and operating performance. VGB Power Tech 12/9.

Future Trends

- Ultra Supercritical (USC) steam turbines
- High cycle pressures (~350 bar) and temperatures (>700º C) by year 2013 !
- 3D blades having low loss aerodynamic shapes with bow and lean
- Introduction of rotor tip seal
- Ferritic steels with improved creep strength for forged turbine rotors and blades
- Improved boiler designs with vertical single pass tubes

Session Summary

In this session the following aspects of steam turbines have been discussed:

- Working principle
- Classification of steam turbines
- Types of compounding
- Work done and efficiency of steam turbine stages
- Broad design concepts
- Supercritical steam turbines
- Future trends in steam turbine design