MOHAMED SATHAK A.J. COLLEGE OF ENGINEERING
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Department of Mechanical Engineering

Steam Turbines & Gas Turbine

Mr. Vigneshwaran V,
Assistant Professor,
Mechanical Engineerin

STEAM TURBINES

Types, Impulse and reaction principles, Velocity diagrams, Work done and efficiency – optimal operating conditions. Multi-staging, compounding and governing.

Classification

Steam turbines are classified according to :

Principle of action of steam

- Impulse turbine
- Reaction turbine

Direction of steam flow

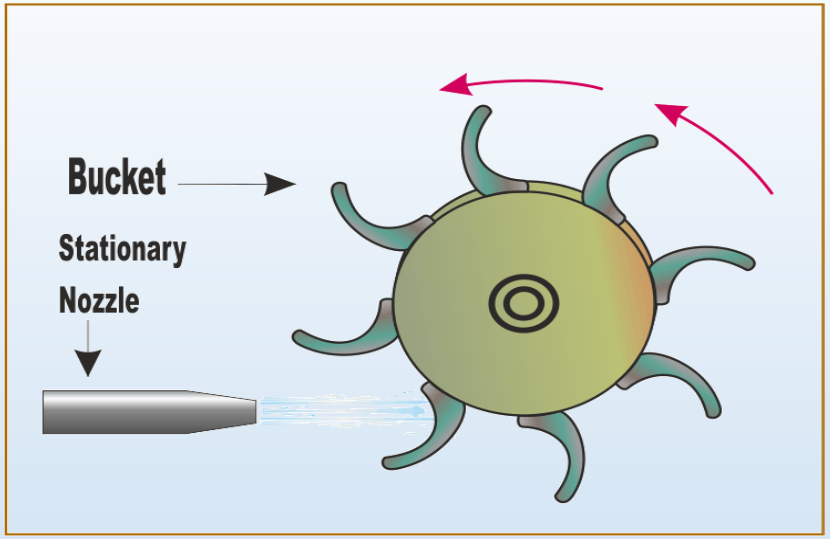
- Axial -flow occurs along the axis of rotation of the impeller - Kaplan
- Radial - flow occurs along the radius of the impeller, (i.e) in radial direction
- Tangential - flow is tangential to the circumference of the impeller - Pelton
- Mixed flow- flow occurs in both radial and axial directions - Francis

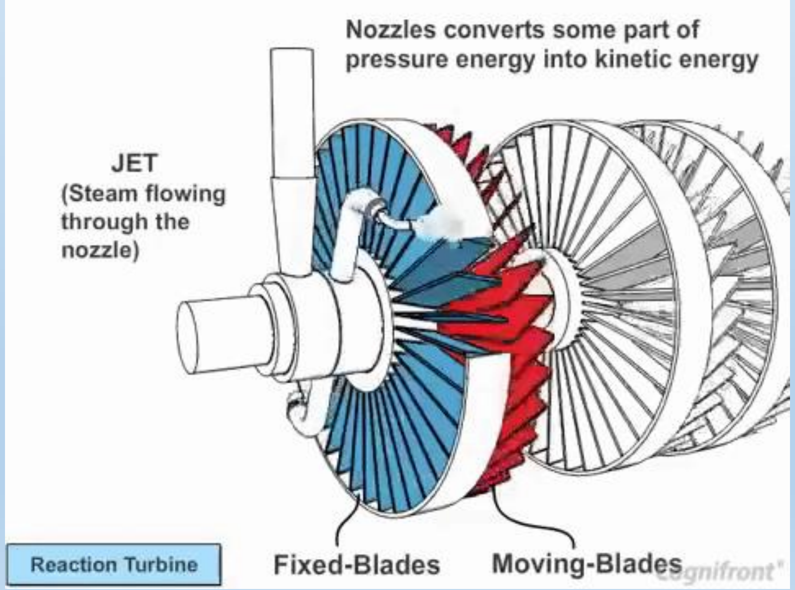
Number of pressure stages

- Single stage
- Multi stage

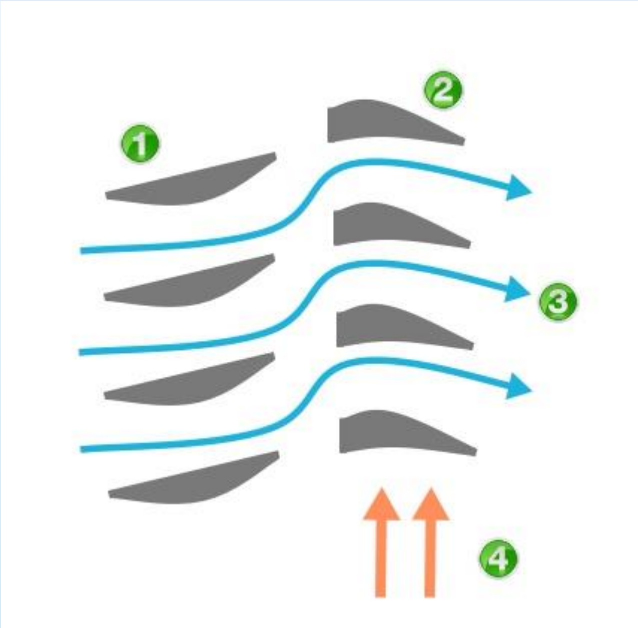
Method of governing

- Throttle
- Nozzle
- By-pass
- Combination of throttle , nozzle by pass





A **reaction turbine** is a type of steam turbine that works on the principle that the rotor spins, as the name suggests, from a reaction force rather than an impact or impulse force.



Notice from the diagram of the reaction turbine above that:

(1) The steam enters through a section of curved blades in a fixed position.

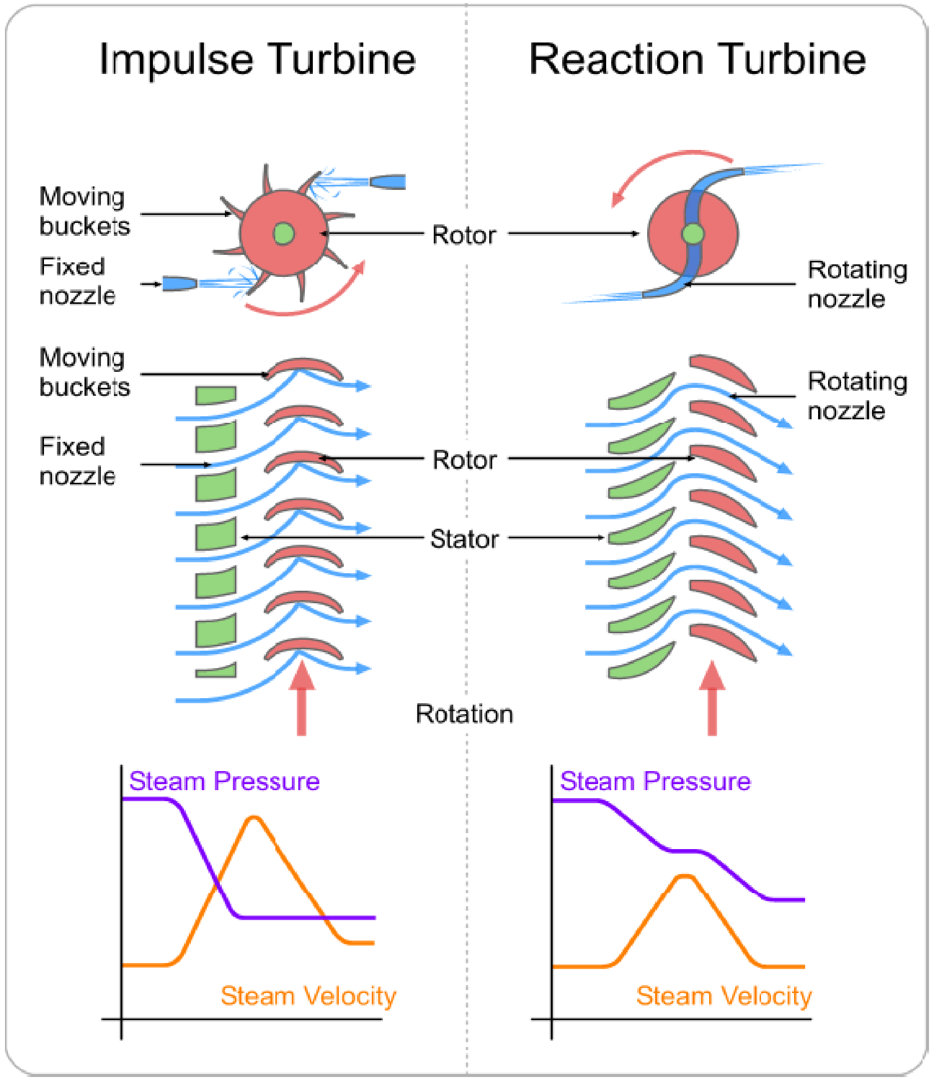
(2) The steam then enters the set of moving blades and creates enough reactive force to rotate them,

(3) The steam exits the section of rotating blades.

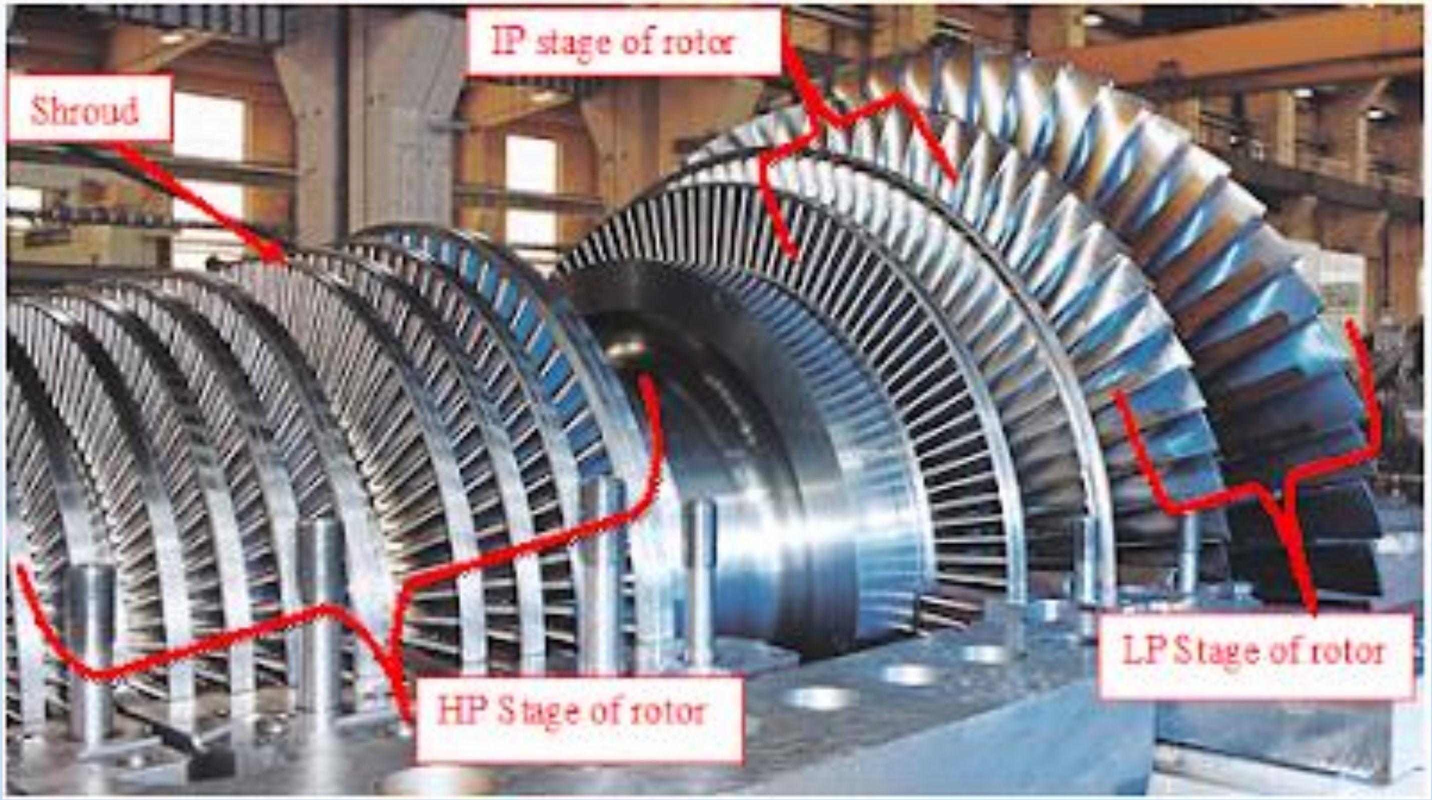
(4) The direction of rotation.

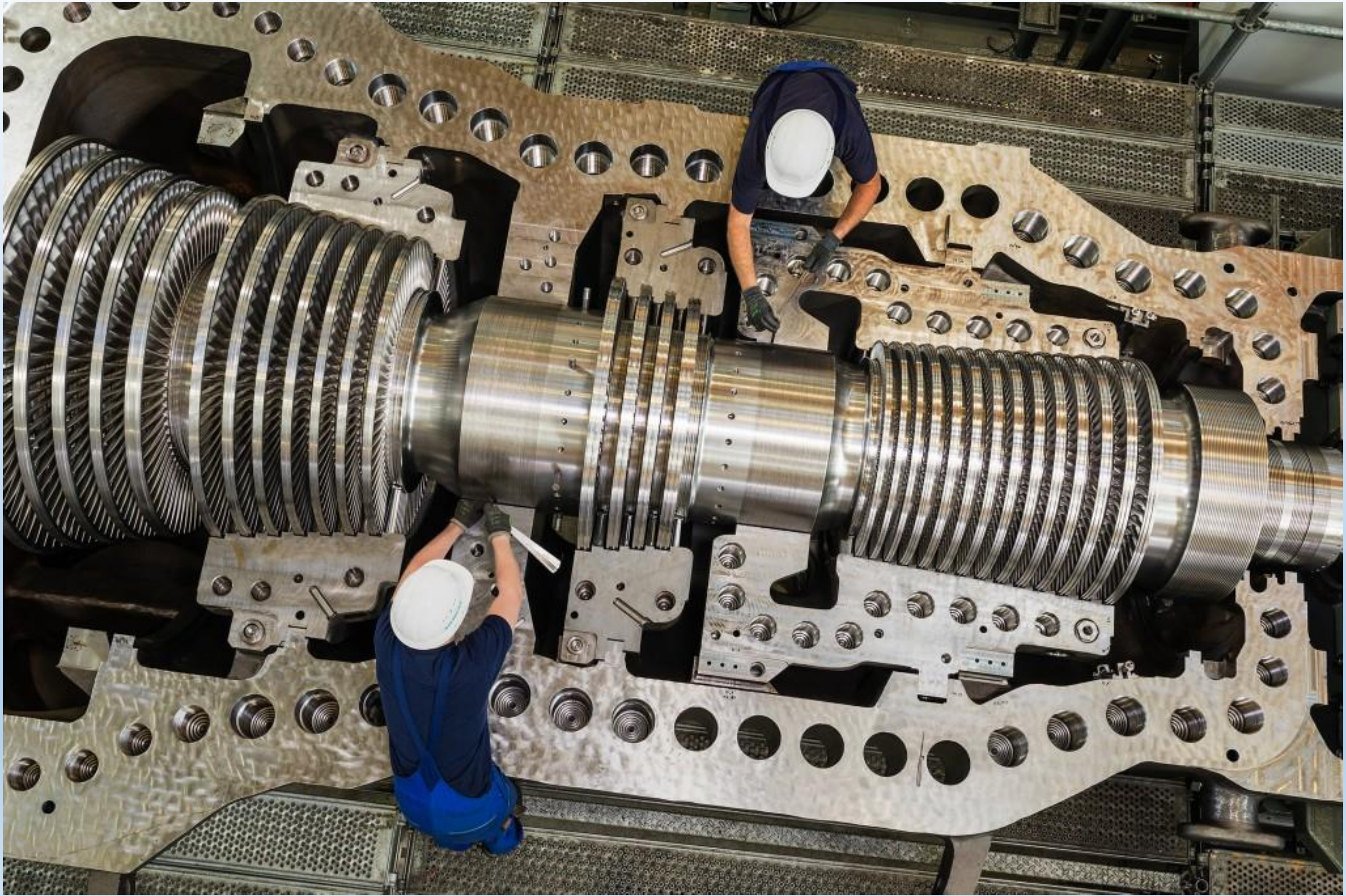
Comparison between Impulse and Reaction Turbine

1. In impulse turbine, there are nozzle and moving blades are in series while there are fixed blades and moving blades are present in Reaction turbine (No nozzle is present in reaction turbine).
2. In impulse turbine pressure falls in nozzle while in reaction turbine in fixed blade boiler pressure falls.
3. In impulse turbine velocity (or kinetic energy) of steam increases in nozzle while this work is to be done by fixed blades in the reaction turbine.
4. Compounding is to be done for impulse turbines to increase their efficiency while no compounding is necessary in reaction turbine.
5. In impulse turbine pressure drop per stage is more than reaction turbine.
6. Not much power can be developed in impulse turbine than reaction turbine.
7. Efficiency of impulse turbine is lower than reaction turbine.
8. Impulse turbine requires less space than reaction turbine.
9. Blade manufacturing of impulse turbine is not difficult as in reaction turbine it is difficult.



Schematic diagram outlining the difference between an impulse and a reaction turbine


IP stage of rotor
Shroud
HP Stage of rotor
LP Stage of rotor



compounding

1. In simple impulse turbine , the steam is expanded from the boiler pr to condenser pr in one stage.
2. The speed of the rotor becomes tremendously high this leads practical complications.
3. There are several methods of reducing this speed to lower value.
4. All these methods utilize a multiple system of rotor in series .
5. This is known as compounding

    1.Velocity compounding

    2. Pressure compounding

    3.Pressure velocity compounding

1. Velocity compounding

i) Steam is expanded through a stationary nozzle from the boiler or inlet pressure to condenser pressure.

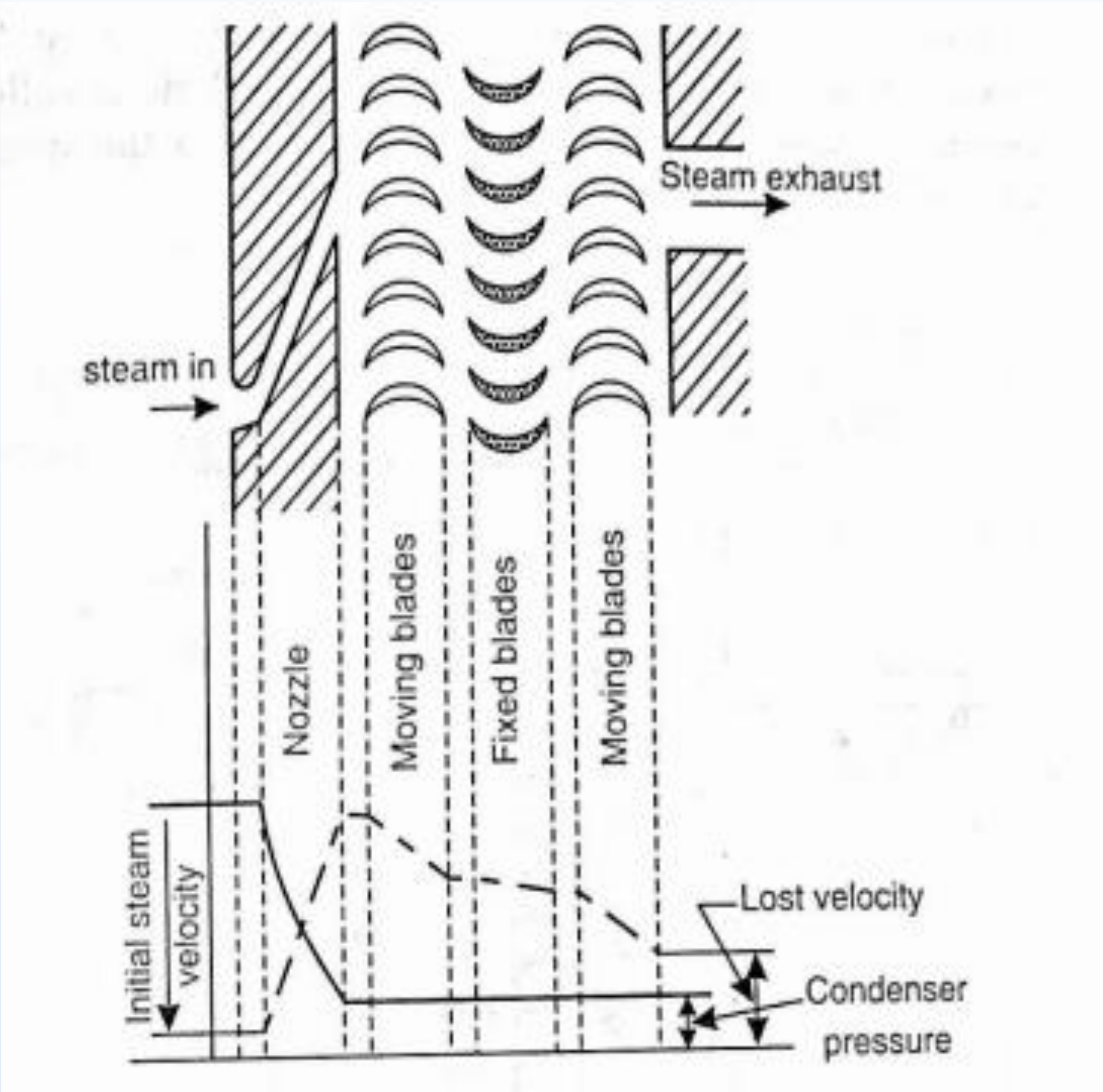
ii) So the pressure in the nozzle drops, the kinetic energy of the steam increases due to increase in velocity.

iii) A portion of this available energy is absorbed by a row of moving blades.

iv) The steam (whose velocity has decreased while moving over the moving blades) then flows through the second row of blades which are fixed.

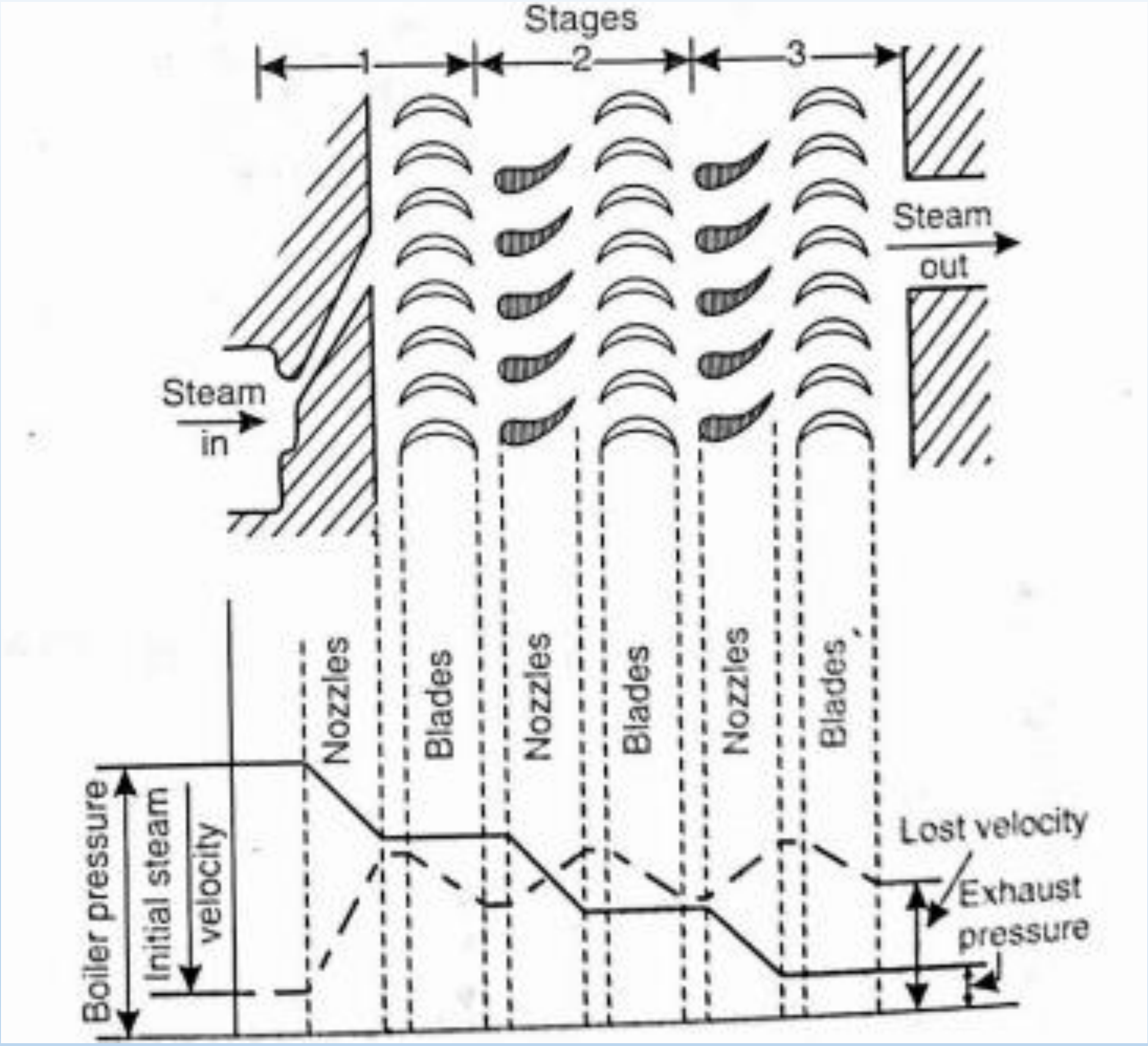
v) The function of these fixed blades is to re-direct the steam flow without altering its velocity to the following next row moving blades where again work is done on them and steam leaves the turbine with a low velocity.

vi) Though this method has the advantage that the initial cost is low due to lesser number of stages yet its efficiency is low.



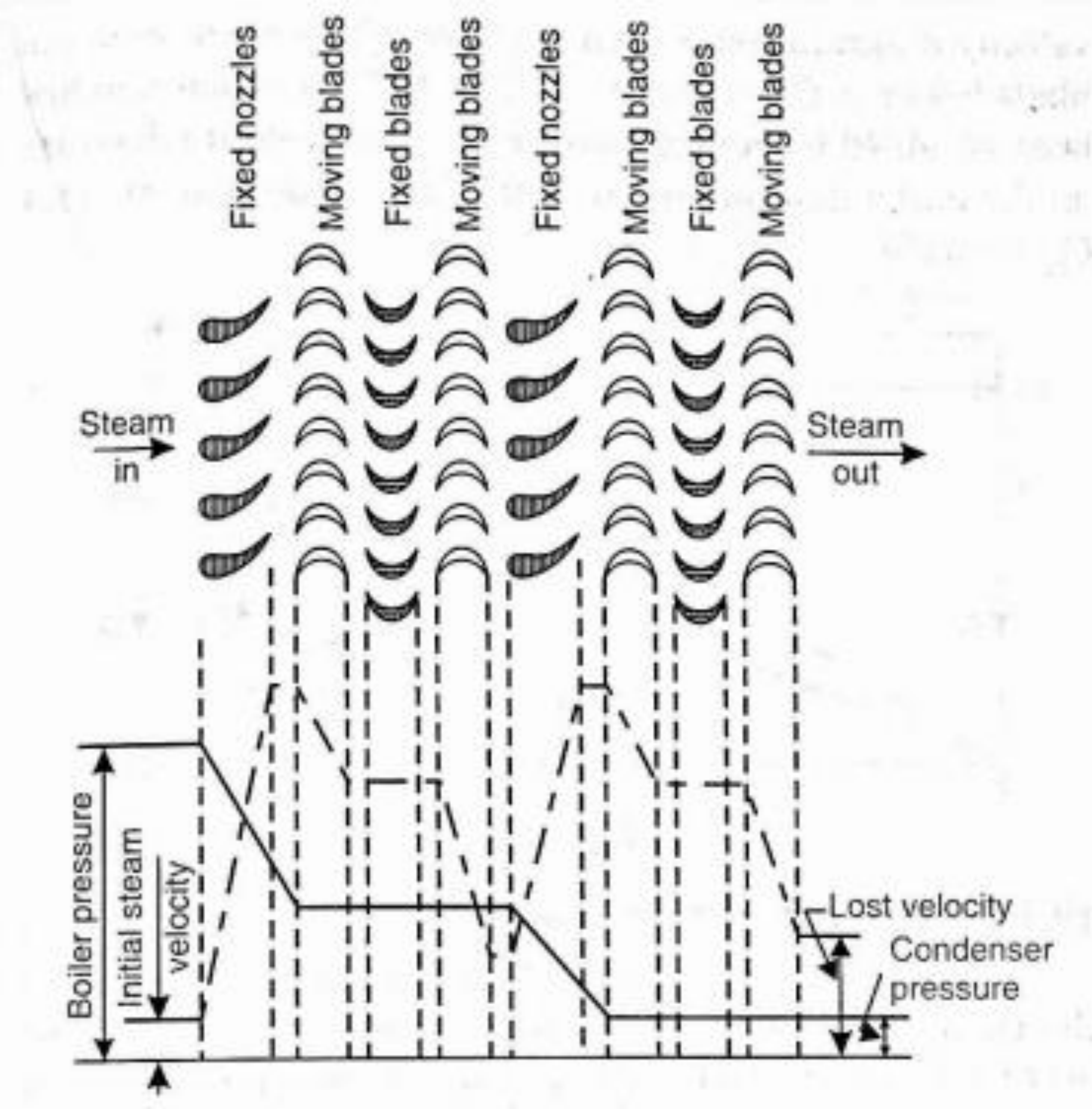
2. Pressure Compounding

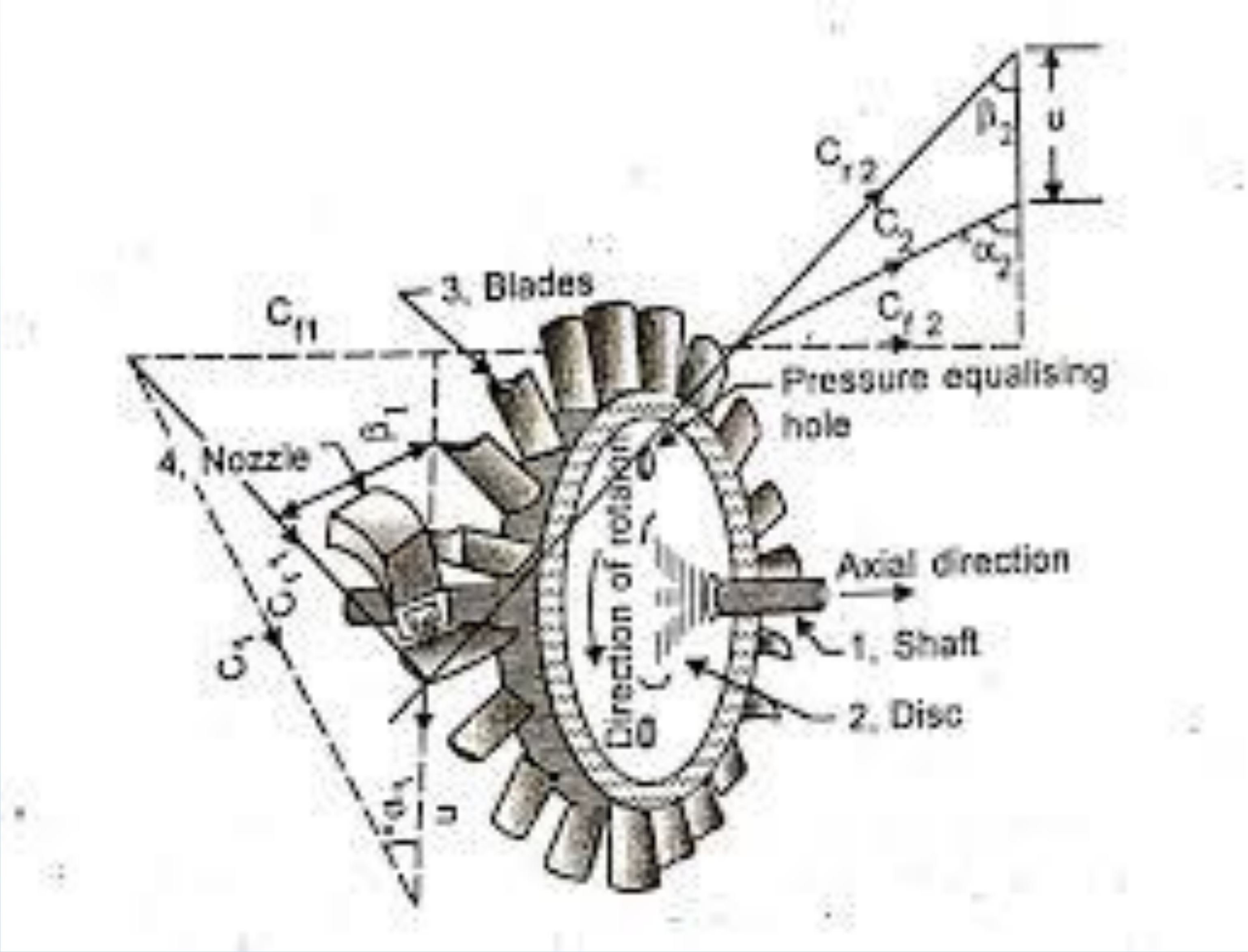
i) The fig shows rings of fixed nozzles incorporated between the rings of moving blades.
ii) The steam at boiler pressure enters the first set of nozzles and expands partially.
iii) This method of compounding is used in Rateau and Zoelly turbine.
iv) This is most efficient turbine since the speed ratio remains constant but it is expensive owing to a large number of stages.



3. Pressure velocity compounding

i. This method of compounding is the combination of two previously discussed method.
ii. The total drop in steam pressure is divided into stages and the velocity obtained in each stage is also compounded.
iii. The rings of nozzles are fixed at the beginning of each stage and pressure remains constant during each stage.
iv. This method of compounding is used in Curits and Moore turbine.




3. Blades
Pressure equalising hole
4. Nozzle
Axial direction
1. Shaft
2. Disc
Direction of rotation

1) Power developed by the turbine = $\frac{m(C_{wi} + Cwe)u}{1000}$

2) Blade efficiency = $\frac{2u(C_{wi} + C_{we})}{C_i^2}$

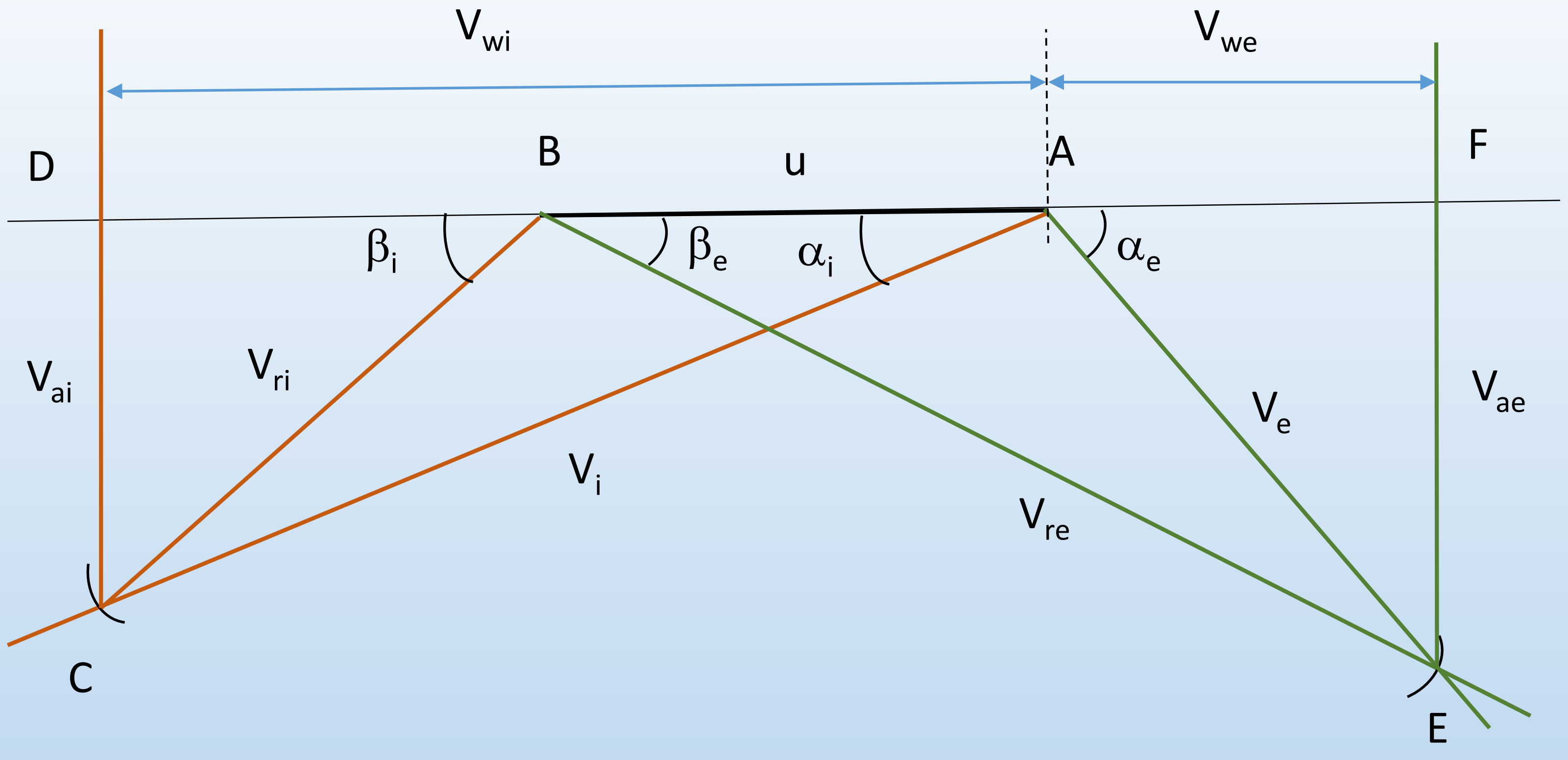
3) Stage efficiency = $\frac{(Cwi + Cwe)}{H_d}$

4) Maximum Blade efficiency = $\cos^2 \alpha_i$

5) Axial force on blades = $m(C_{ai} - C_{ae})$

6) Tangential force on blades = $m(C_{wi} \pm C_{we})$

7) Energy lost due to blade friction = $\frac{1}{2} m(C_{ri}^2 - C_{re}^2)$



u : Blade Velocity
V_i: Inlet Absolute Velocity
V_e: Exit Absolute Velocity
V_{ri}: Inlet Relative Velocity
V_{re}: Exit Relative Velocity
V_{ai}: Inlet Axial Velocity
V_{ae}:Exit Axial Velocity
V_{wi}: Inlet Tangential Velocity
V_{we}: Exit Tangential Velocity

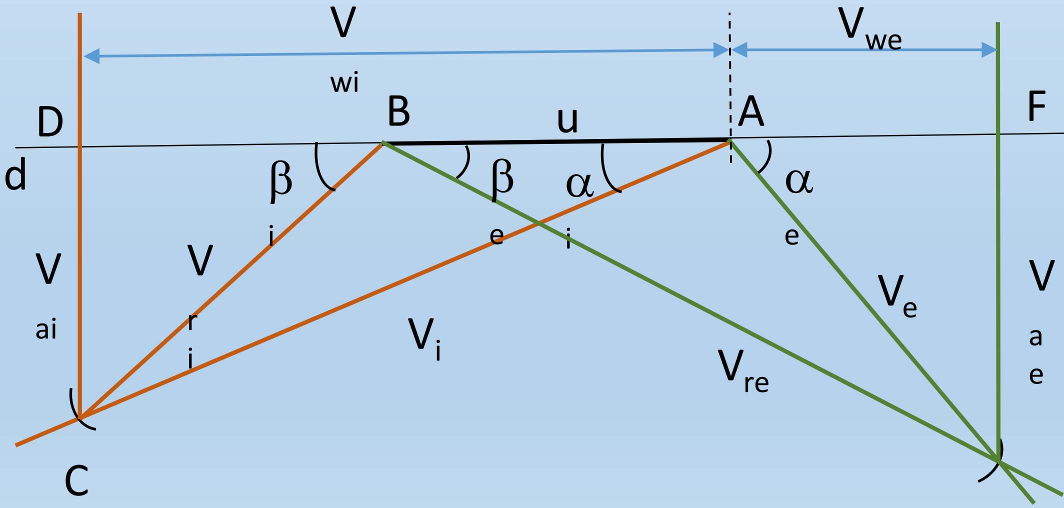
α_i: Inlet Nozzle Angle.
β_i: Inlet Blade Angle.
α_e: Exit Nozzle Angle.
β_e: Exit Blade Angle.
$K = \frac{V_{re}}{V_{ri}}$ = Blade velocity coefficient

All the Velocities are in m/s

VELOCITY DIAGRAM FOR MOVING BLADES FOR AN IMPULSE TURBINE

The procedure for drawing the combined velocity diagram is given below:

- Draw a horizontal line *AB* equal to blade velocity *u* to some suitable scale.
- Draw a line *AC* at an angle α_i with *AB.* Cut *AC* = V_i
- Join *Band* C. The line *BC* represents the relative velocity at inlet.
- The blade inlet angle β_i is measured and its value is noted down.
- From point C draw a perpendicular *CD* on *AB* produced.
- *CD* represents axial velocity at inlet and *AD* represents tangential velocity at inlet.
- From point *B* draw a line *BE* at an angle β_e (the blade outlet angle).
- Cut *BE* = $V_{re} = K V_{ri}$.
- Join *A* and *E.*
- *AE* represents the absolute velocity at outlet.
- The angle α_e is measured and noted down.
- From point *E* draw a perpendicular *EF* on *BA.*
- *Then AF* represents the tangential velocity of steam at outlet and
- *EF* represents the axial velocity outlet.
- This completes the velocity triangle.



Steam with absolute velocity 360 m/s enters the stage of an impulse turbine provided with a single row wheel. The nozzles are inclined at 20° to the plane of the wheel. The blade rotor with diameter 95.5 cm rotates with a speed of 3000 r.p.m. Find

(a) suitable inlet and outlet angle for the moving blade so that there is no axial thrust on the blade.

It may be assumed that friction In blade passages is 19% of the kinetic energy corresponding to relative velocity at inlet to blades,

(b) Power developed in blading for a steam flow of 1 kg/s, and

(c) Kinetic energy of steam finally leaving the stage.

Given : $V_i = 360$ m/s *N = 3000 rpm* $\alpha_i = 20°$ *D = 95.5 cm* $m_s = 1$ kg/s $V_{ai} = V_{ae}$

$$u = \frac{\pi DN}{60} = \frac{\pi \times 0.955 \times 3000}{60} = 150$$

For Drawing

Scale 1:30

$$u = \frac{150}{30} = 5$$

$$V_i = \frac{360}{30} = 12$$

$$V_{ai} = V_{ae}$$

$$\frac{V_{r2}^2}{2} = (1\text{-}0.19)\frac{V_{r1}^2}{2}$$

$$= (1\text{-}0.19)\frac{7.5^2}{2}$$

$$= 6.8$$

From Diagram

$V_i = 12$	$V_e = 5.6$
$V_{ri} = 7.5$	$V_{re} = 6.8$
$V_{ai} = 4$	$V_{ae} = 4$
$V_{wi} = 11.3$	$V_{we} = 0.35$

For Calculation

Multiply all the values taken from diagram by 30

$V_i = 12 \times 30 = 360$ m/s

$V_{ri} = 7.5 \times 30 = 225$ m/s

$V_{ai} = 4 \times 30 = 120$ m/s

$V_{wi} = 11.3 \times 30 = 339$ m/s

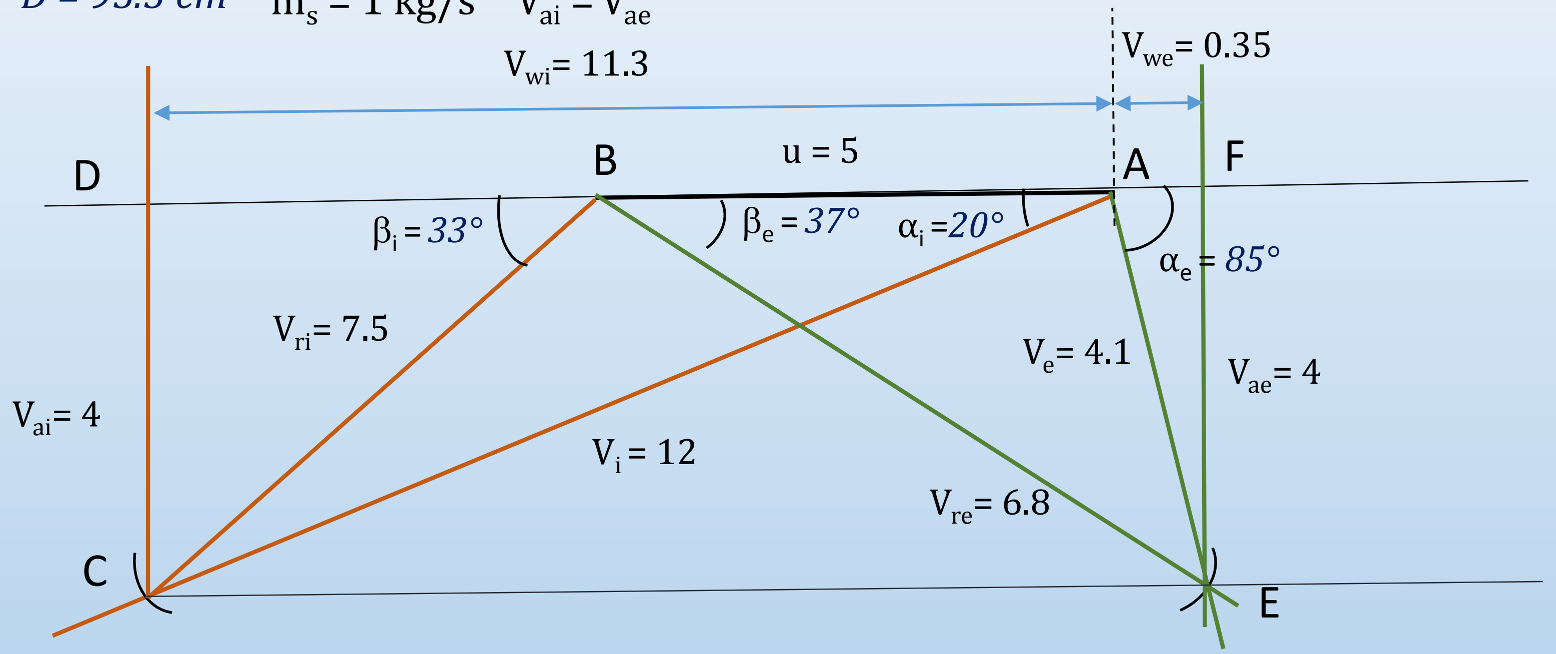
$V_e = 4.1 \times 30 = 123$ m/s

$V_{re} = 6.8 \times 30 = 204$ m/s

$V_{ae} = 4 \times 30 = 120$ m/s

$V_{we} = 0.35 \times 30 = 10.5$ m/s

$u = 5 \times 30 = 150$ m/s



a) Inlet angle of blade $\beta_i = 33°$ *Outlet angle of blade* $\beta_e = 37°$

b) Power developed by the turbine $= \dfrac{m(Vwi \pm Vwe)u}{1000} = \dfrac{1\,(339 + 10.5)\,150}{1000} = 52$ *kW*

c) Kinetic energy of steam finally leaving the stage $= \dfrac{V_e^2}{2} = \dfrac{123^2}{2} = 7564.5$ *Nm/kg*

The blade speed of a single ring of an impulse turbine is 300 m/s and the nozzle angle is 20°. The isentropic heat drop is 473 kJ/kg and the nozzle efficiency is 0.85. Given that the blade velocity coefficient is 0.7 and the blades are symmetrical, draw the vector diagrams and calculate for a mass flow of 1 kg/s: (a) axial thrust on the blading. (b) steam consumption per B.P. hour if the mechanical efficiency is 90 per cent. (c) blade efficiency, stage efficiency and maximum blade efficiency. (d) heat equivalent of the friction of blading.

Given : $u = 300\ m/s$ $\alpha_i = 20°$ $m_s = 1\ kg/s$ $(h_1 - h_2) = 473\ kJ/kg$ $\eta_n = 0.85$ $k = \frac{V_{re}}{V_{ri}} = 0.7$ $\beta_i = \beta_e$

For Drawing

Scale 1:75

$V_1 = 44.72\sqrt{\eta_n(h_1 - h_2)}$

$= 44.72\sqrt{0.85(473)}$

$= 896\ m/s \approx 900\ m/s$

$\beta_i = \beta_e$

$u = \frac{300}{75} = 4$

$V_i = \frac{900}{75} = 12$

$k = \frac{V_{re}}{V_{ri}} = 0.7$

$V_{re} = k * V_{ri} = 5.88 \approx 5.9$

From Diagram

$V_i = 12$	$V_e = 3.1$
$V_{ri} = 8.2$	$V_{re} = 5.8$
$V_{ai} = 4.2$	$V_{ae} = 2.9$
$V_{wi} = 12$	$V_{we} = 0.16$

For Calculation

Multiply all the values taken from diagram by 75

$V_i = 12 \times 75 = 900\ m/s$

$V_{ri} = 8.4 \times 75 = 630\ m/s$

$V_{ai} = 4.1 \times 75 = 307.5\ m/s$

$V_{wi} = 11.3 \times 75 = 847.5\ m/s$

$V_e = 3.1 \times 75 = 232.5\ m/s$

$V_{re} = 5.9 \times 75 = 442.5\ m/s$

$V_{ae} = 2.9 \times 75 = 217.5\ m/s$

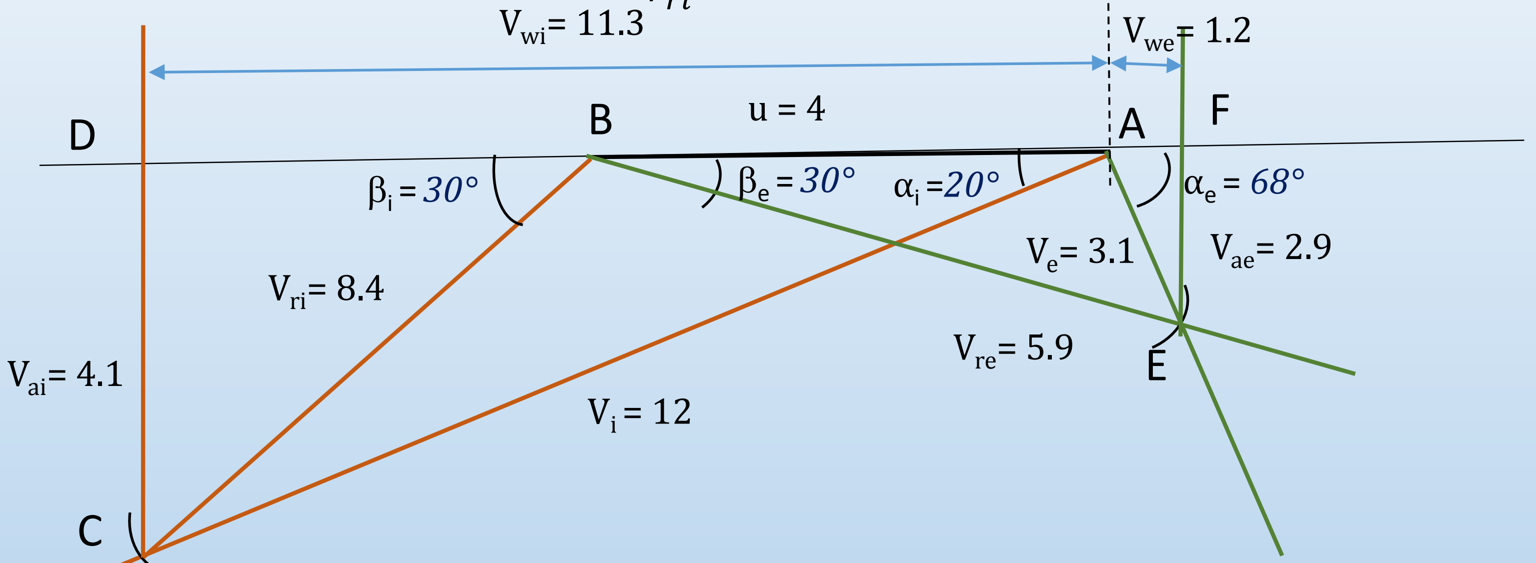
$V_{we} = 1.2 \times 75 = 90\ m/s$

a) Axial force on blades $= m(V_{ai} - V_{ae})$

$= 1\ (307.5 - 217.5) = 90\ N$

Maximum Blade efficiency $= \cos^2 \alpha_i$

$= \cos^2 20 = 0.88 = 88\ \%$



b) Power developed by the turbine $= \frac{m(Vwi \pm Vwe)u}{1000} = \frac{1\ (900 + 90)\ 300}{1000} = 297\ kW$

B.P. $= 0.9 * 297 = 267.3\ kW$ *Steam consumption per B.P. hour* $= \frac{3600}{267.3} = 13.4\ kg$

c) Blade efficiency $= \frac{2u(Vwi \pm Vwe)}{V_i^2} = \frac{2 \times 300 \times (900+90)}{900^2} = 0.755 = 75\ \%$

Stage efficiency $= \frac{(Vwi \pm Vwe)}{H_d} = \frac{300(900 \pm 90)}{473 * 1000} = 0.627 = 62.7\ \%$

d) Energy lost due to blade friction $= \frac{1}{2}m(V_{ri}^2 - V_{re}^2) = \frac{1}{2}(630^2 - 442.5^2)$

$= 100504\ J$

Steam issues from the nozzle of a simple impulse turbine with a velocity of 900 m/sec. The nozzle angle is 20°, the mean diameter of the blades is 25 cm and the speed of rotation is 20,000 r.p.m. m The mass flow through the turbine nozzles and blading is 0.18 kg of steam per sec. Draw the velocity diagram and derive or calculate the following : (a) Tangential force on blades, (b) Axial force on blades, (c) Power developed by the turbine wheel, (d) Efficiency of the blading, and (e) Inlet angles of blades for shock less inflow of steam. Assume that the outlet angle of the blades is equal to the inlet angle and frictional losses are negligible.

Given : $V_i = 900$ m/s $N = 20{,}000$ rpm $\alpha_i = 20°$ $D = 25$ cm $m_s = 0.18$ kg/s $\beta_i = \beta_e$ $V_{ri} = V_{re}$

$$u = \frac{\pi DN}{60} = \frac{\pi \times 0.25 \times 20{,}000}{60} = 262$$

For Drawing

Scale 1:75

$$u = \frac{262}{75} = 3.5$$

$$V_i = \frac{900}{75} = 12$$

$V_{ri} = V_{re}$

$\beta_i = \beta_e$

From Diagram

$V_i = 12$	$V_e = 5.6$
$V_{ri} = 8.8$	$V_{re} = 8.8$
$V_{ai} = 4.1$	$V_{ae} = 4.1$
$V_{wi} = 11.3$	$V_{we} = 4.3$

$u = 3.5$

For Calculation

Multiply all the values taken from diagram by 75

$V_i = 12 \times 75 = 900$ m/s

$V_{ri} = 8.8 \times 75 = 660$ m/s

$V_{ai} = 4.1 \times 75 = 307.5$ m/s

$V_{wi} = 11.3 \times 75 = 847.5$ m/s

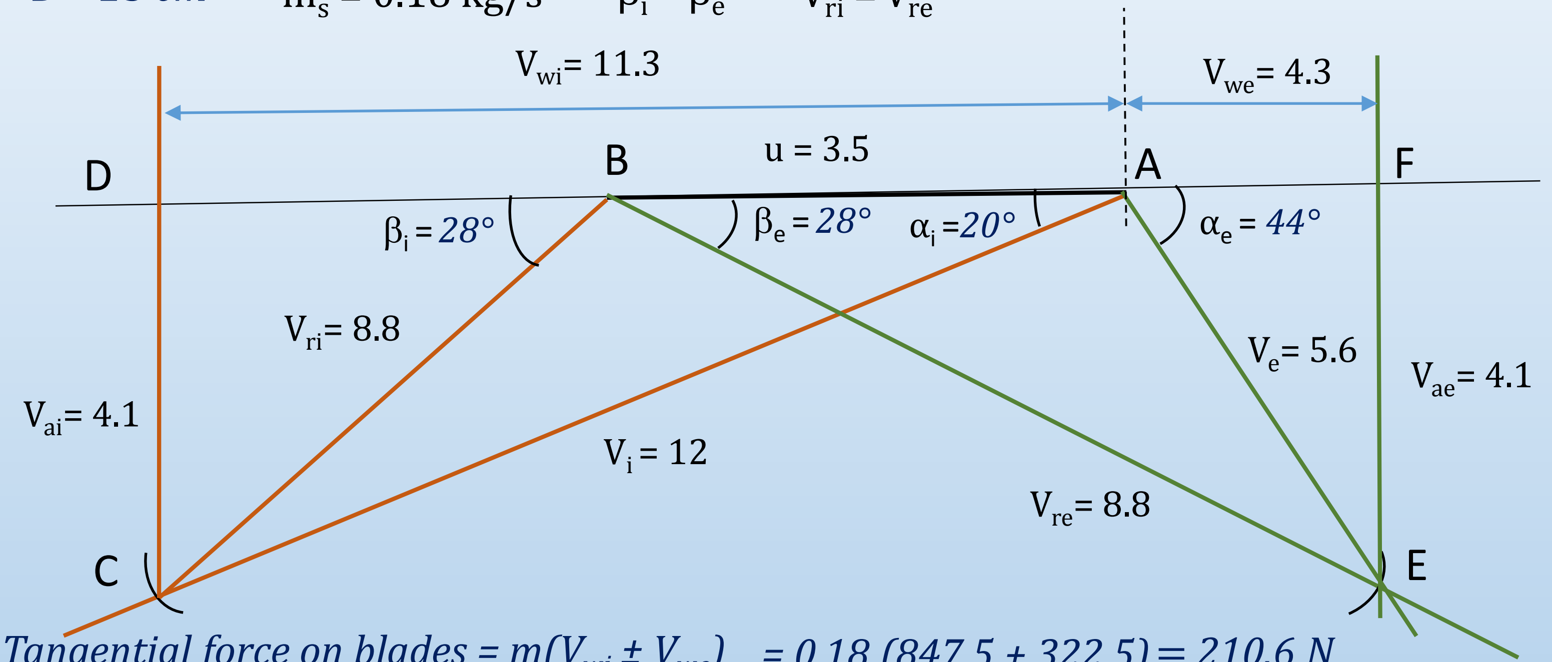
$V_e = 5.6 \times 75 = 900$ m/s

$V_{re} = 8.8 \times 75 = 660$ m/s

$V_{ae} = 4.1 \times 75 = 307.5$ m/s

$V_{we} = 4.3 \times 75 = 322.5$ m/s

$u = 3.5 \times 75 = 900$ m/s



a) *Tangential force on blades* = $m(V_{wi} \pm V_{we})$ = 0.18 (847.5 + 322.5) = 210.6 N

b) *Axial force on blades* = $m(V_{ai} - V_{ae})$ = 0.18 (307.5 - 307.5) = 0 N

c) *Power developed by the turbine* $= \frac{m(Vwi \pm Vwe)u}{1000} = \frac{0.18\,(847.5 + 322.5)\,262}{1000} = 55$ kW

d) *Blade efficiency* $= \frac{2u(Vwi \pm Vwe)}{V_i^2} = \frac{2 \times 262 \times (847.5+322.5)}{900^2} = 0.75$

e) *Inlet angles of blade* $\beta_i = 28°$

A single row impulse turbine develops 132.4 kW at a blade speed of 175 m/s using 2 kg of steam per second. Steam leaves the nozzle at 400 m/s. Velocity co-efficient of the blades are 0.9. Steam leaves the turbine blades axially. Determine (a) nozzle angle and (b) blade angle at entry and exit, assuming no shock.

Given : $u = 175$ m/s *P = 132.4 kW* $V_i = 400$ m/s $k = \frac{V_{re}}{V_{ri}} = 0.9$ $\alpha_e = 90°$ $V_e = V_{ae}$

For Drawing

Scale 1:50

$u = \frac{175}{50} = 3.5$

$V_i = \frac{400}{50} = 8$

$\alpha_e = 90°$

$V_e = V_{ae}$

$P = \frac{m(Vwi \pm Vwe)u}{1000}$

$132.4 = \frac{2(Vwi \pm Vwe)175}{1000}$

$(Vwi \pm Vwe) = 378.28\ m/s$

$= \frac{378}{50}$

$= 7.6$

$k = \frac{V_{re}}{V_{ri}} = 0.9$

$V_{re} = k * V_{ri}$

$V_{re} = 0.9 * 4.8 = 4.3$

From Diagram

$V_i = 8$ | $V_e = 2.5$

$V_{ri} = 4.8$ | $V_{re} = 4.3$

$V_{ai} = 2.5$ | $V_{ae} = 2.5$

$V_{wi} = 7.6$ | $V_{we} = 0$

For Calculation

Multiply all the values taken from diagram by 50

$V_i = 8 \times 50 = 400$ m/s

$V_{ri} = 4.8 \times 50 = 240$ m/s

$V_{ai} = 2.5 \times 50 = 125$ m/s

$V_{wi} = 7.6 \times 50 = 380$ m/s

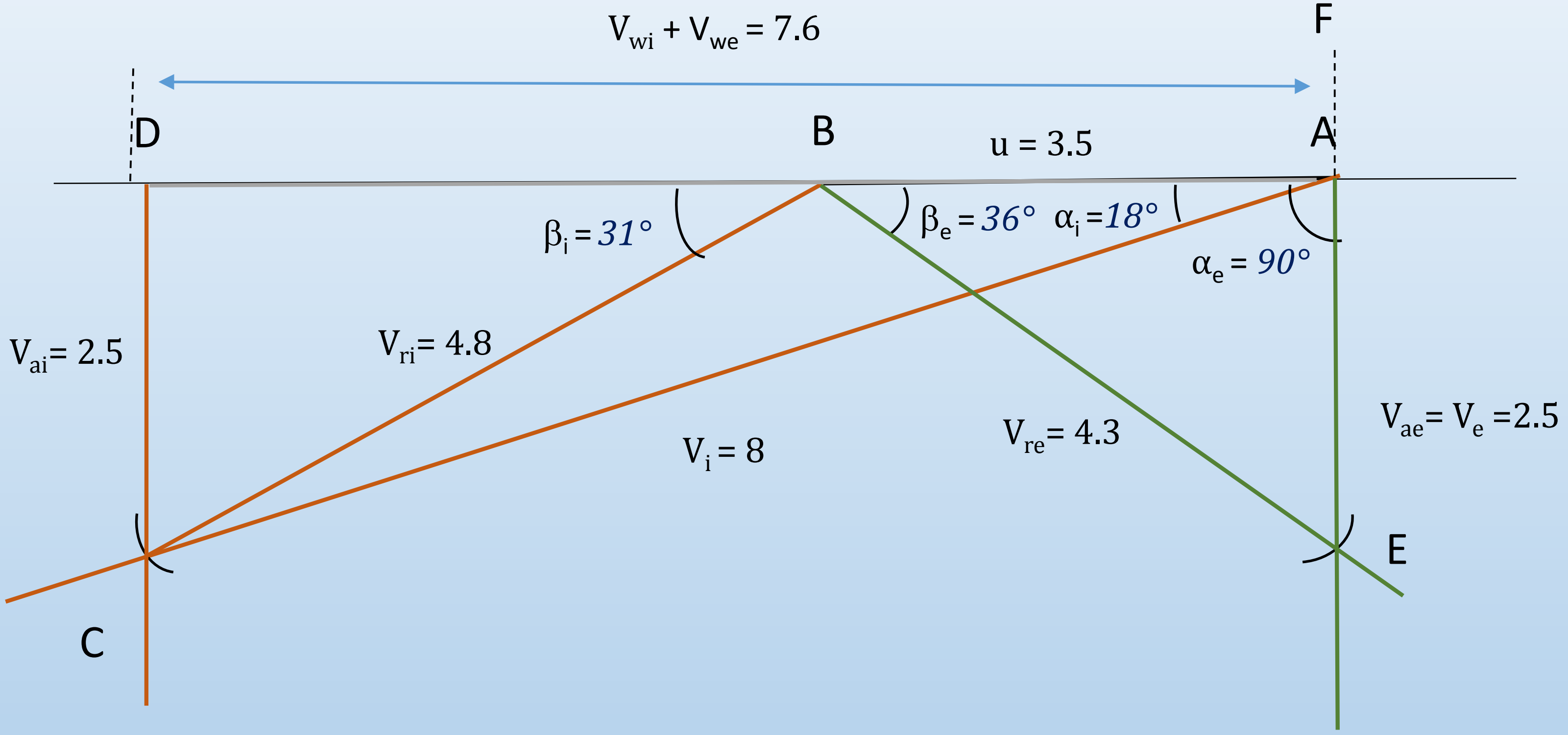
$V_e = 2.5 \times 50 = 125$ m/s

$V_{re} = 4.3 \times 50 = 215$ m/s

$V_{ae} = 2.5 \times 50 = 125$ m/s

$V_{we} = 0 \times 50 = 0$ m/s

$u = 3.5 \times 50 = 175$ m/s



1) *Inlet angle of blade* $\beta_i = 31°$

2) *Exit angle of blade* $\beta_i = 36°$

3) *Inlet angle of Nozzle* $\alpha_i = 18°$

The mean diameter of the blades of an impulse turbine with a single row wheel is 105 cm and the speed is 3000 r.p.m. The nozzle angle is 18°, the ratio of blade speed to steam speed is 0.42 and the ratio of the relative velocity at outlet from the blades to that at inlet is 0.84. The outlet angle of the blade is to be made 3° less than the inlet angle. The steam flow is 8 kg per sec. Draw the velocity diagram for the blades and derive the (a) resultant thrust on the blades, (b) tangential thrust on the blades, (c) axial thrust on the blades, (d) power developed in blades, and (e) blading efficiency

Given : V_i = 900 m/s *N = 3000 rpm* α_i = *18°* *D = 105 cm* m_s = 8 kg/s $\beta_e = \beta_i - 3$ $k = \frac{V_{re}}{V_{ri}} = 0.84$ $\frac{u}{V_i} = 0.42$

$$u = \frac{\pi DN}{60} = \frac{\pi \times 1.05 \times 3{,}000}{60} = 165$$

For Drawing

Scale 1:50

$$u = \frac{165}{50} = 3.3$$

$$\frac{u}{V_i} = 0.42$$

$$V_i = \frac{165}{0.42} = 393$$

$$V_i = \frac{393}{50} = 7.9$$

$$k = \frac{V_{re}}{V_{ri}} = 0.84$$

$$V_{re} = k * V_{ri}$$

$$V_{re} = 0.84 * 4.9 = 4.1$$

From Diagram

V_i = 7.9 V_e = 2

V_{ri} = 4.9 V_{re} = 4.1

V_{ai} = 2.5 V_{ae} = 1.9

V_{wi} = 7.5 V_{we} = 0.3

For Calculation

Multiply all the values taken from diagram by 50

V_i = 7.9 × 50 = 395 m/s

V_{ri} = 4.9 × 50 = 245 m/s

V_{ai} = 2.5 × 50 = 125 m/s

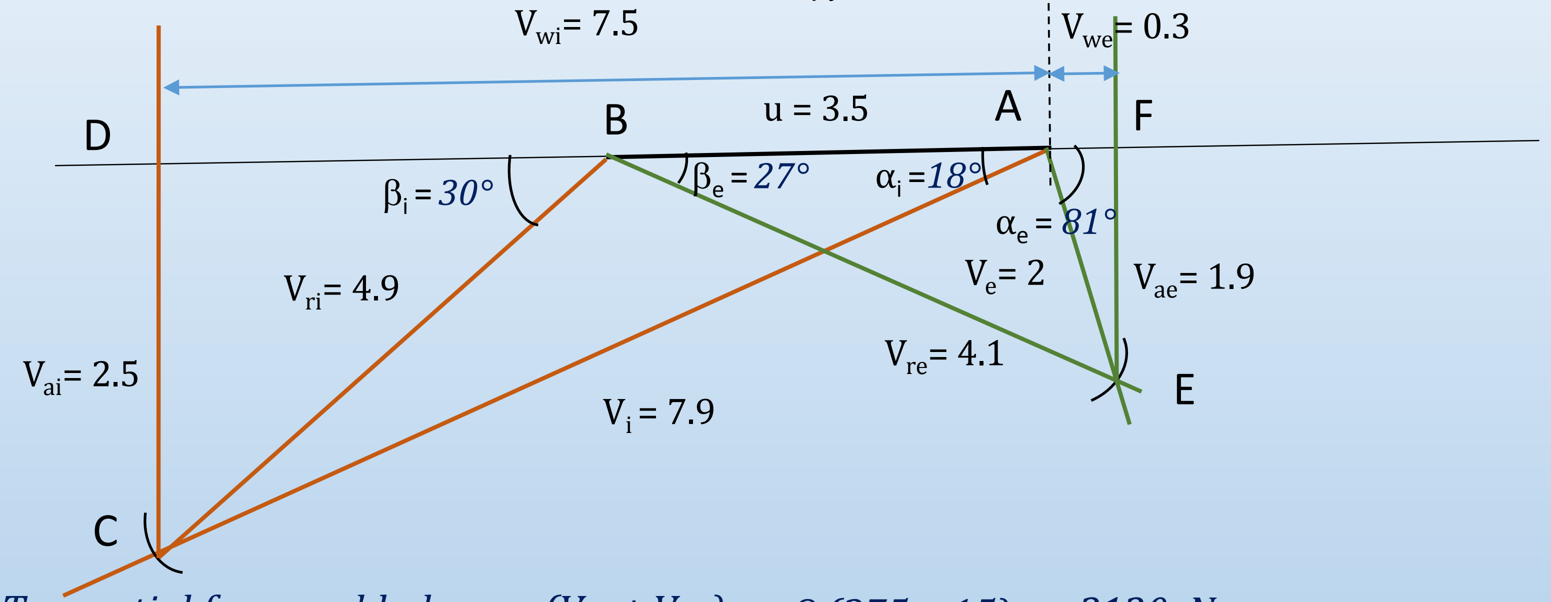
V_{wi} = 7.5 × 50 = 375 m/s

V_e = 2 × 50 = 100 m/s

V_{re} = 4.1 × 50 = 205 m/s

V_{ae} = 1.9 × 50 = 95 m/s

V_{we} = 0.3 × 50 = 15 m/s



b) Tangential force on blades = $m(V_{wi} \pm V_{we})$ *= 8 (375 + 15) = 3120 N*

c) Axial force on blades = $m(V_{ai} - V_{ae})$ *= 8 (125 - 95) = 240 N*

d) Power developed by the turbine = $\frac{m(Vwi \pm Vwe)u}{1000} = \frac{8\,(375 + 15)\,165}{1000}$ *= 515 kW*

e) Blade efficiency = $\frac{2u(Vwi \pm Vwe)}{V_i^2} = \frac{2 \times 165 \times (375+15)}{395^2}$ *= 0.82 = 82 %*

a) Resultant thrust = $\sqrt{3120^2 - 240^2}$ *= 3110 N*

In a certain stage of an impulse turbine the nozzle angle is 20° with the plane of the wheel. Four nozzles each of 1 cm diameter expand steam isentropically from 15.2 bar and 250°C to 0.5 bar. The mean diameter of the blade ring is 2.8 m.1t develops 55.2 kW at 2400 r.p.m. The axial thrust is 3.45 N. Calculate blade angles at entrance and exit

Given : $\alpha_i = 18°$ $D_n = 1$ cm $P_1 = 15.2$ bar $T_1 = 250°C$ $P_2 = 0.5$ bar $D_b = 2.8$ cm $P = 55.2$ kW $N = 2400$ rpm Axial thrust = 3.45 N

$V_i = 44.72\sqrt{(h_1 - h_2)}$

Using mollier chart @ inlet

$P_1 = 10$ bar & $T_1 = 250°C$

$h_1 = 2910$ kJ/kg

@ Exit

$P_2 = 0.5$ bar & $S_1 = S_2$

$h_2 = 2340$ kJ/kg

$v_2 = 3\ m^3/kg$

$x = 0.88$

$\therefore (h_1 - h_2)$

$2910 - 2340 = 570$ kJ/kg

$\therefore V_i = 44.72\sqrt{(h_1 - h_2)}$

$= 44.72\sqrt{570} = 1067$ m/s

$u = \frac{\pi DN}{60} = \frac{\pi \times 0.28 \times 2400}{60}$

$= 352$ m/s

$P = \frac{m(Vwi \pm Vwe)u}{1000}$

$m = \frac{nAV_i}{x\ v}$

$A = \frac{\pi}{4} D_n^2 = \frac{\pi}{4} 0.1^2$

$= 0.785 \times 10^{-4}\ m^2$

$\therefore m = \frac{4 \times 1067 \times 0.785 \times 10{-}4}{0.88 \times 3}$

$= 0.12$ kg/s

$V_{wi} \pm V_{we} = \frac{P \times 1000}{m \times u}$

$= \frac{55.2 \times 1000}{0.12 \times 352} = 1306$ m/s

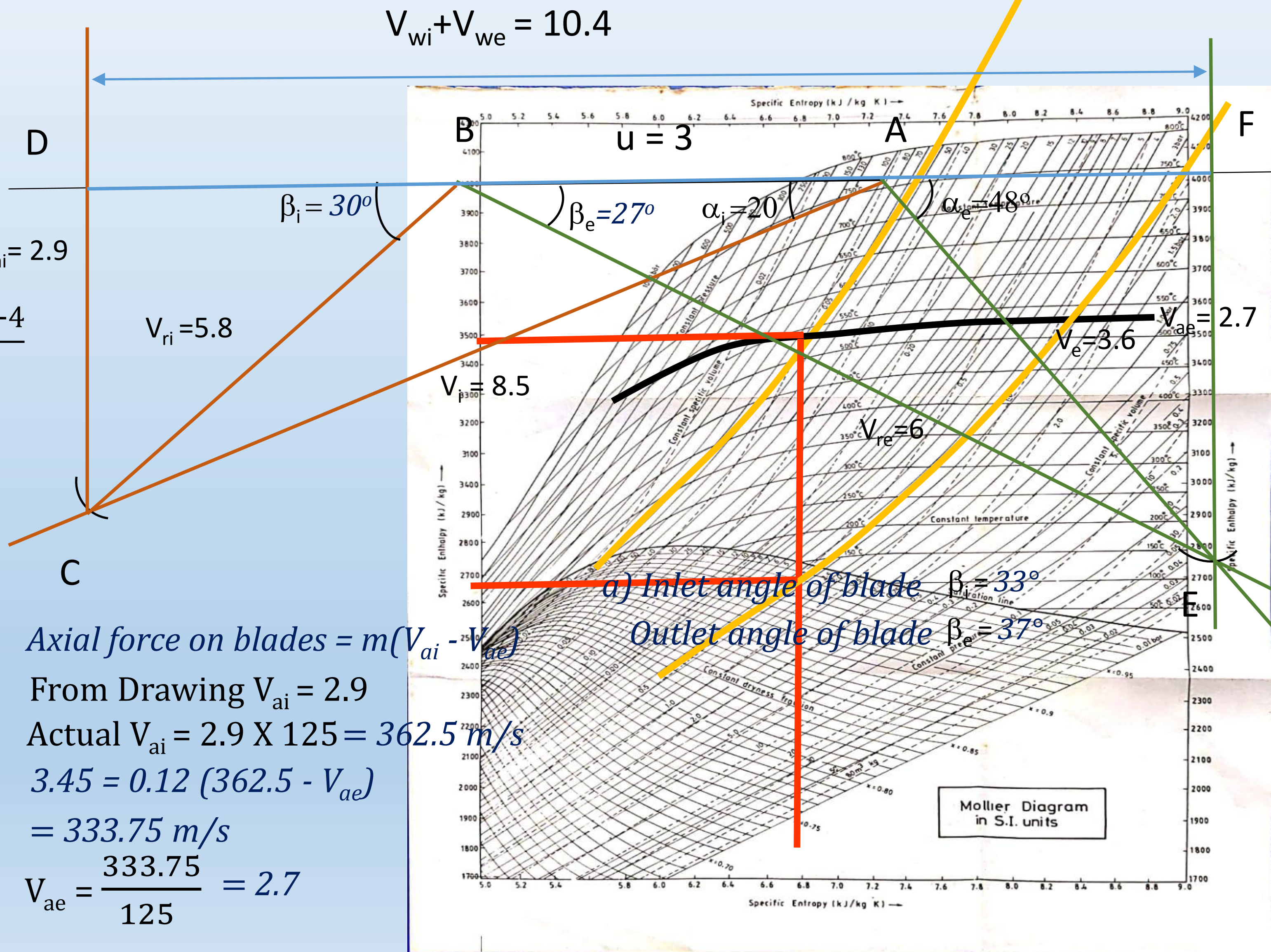
For Drawing

Scale 1:125

$u = \frac{352}{125} = 2.8 \approx 3$

$V_i = \frac{1067}{125} = 8.5$

$V_{wi} \pm Vwe = \frac{1306}{125} = 10.4$



a) *Inlet angle of blade* $\beta_i = 33°$

Outlet angle of blade $\beta_e = 37°$

Axial force on blades $= m(V_{ai} - V_{ae})$

From Drawing $V_{ai} = 2.9$

Actual $V_{ai} = 2.9 \times 125 = 362.5$ m/s

$3.45 = 0.12\ (362.5 - V_{ae})$

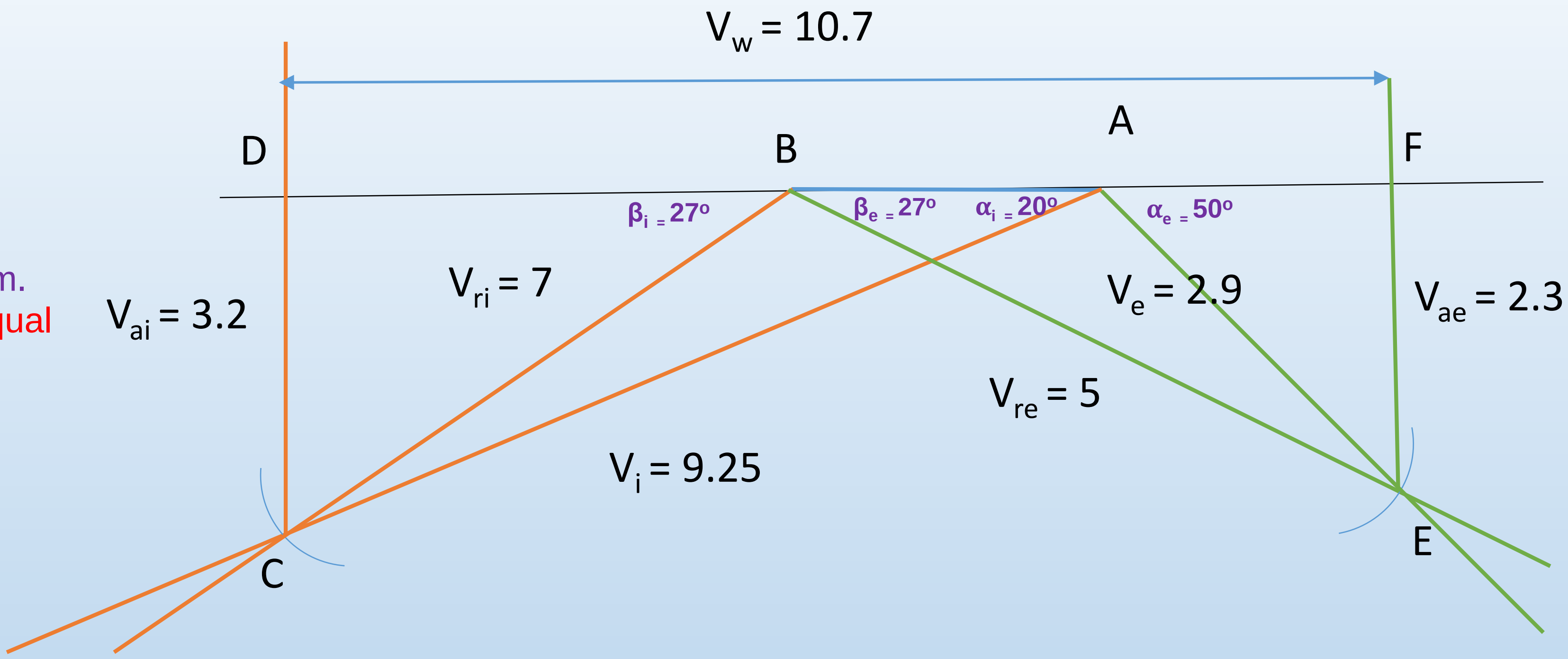
$= 333.75$ m/s

$V_{ae} = \frac{333.75}{125} = 2.7$

The velocity of steam leaving a nozzle is 925 m/s and the nozzle angle is 20°. The blade speed is 250 m/s. The mass flow through the turbine nozzles and blading is 0.182 kg/s and the blade velocity coefficient is 0.7. Calculate the following:

1. Velocity of whirl.
2. Tangential force on blades.
3. Axial force on blades.
4. Work done on blades.
5. Efficiency of blading.
6. Inlet angle of blades for shock less inflow of steam.

Assume that the inlet and outlet blade angles are equal



$u = 250$	After scaling
$V_i = 925$	1 :100
$\alpha_i = 20^o$	$u = 250 = 2.5$
$K = 0.7$	$V_i = 925 = 9.25$
$m = 0.18$	$\alpha_i = 20^o$
	$K = \frac{Vre}{Vri} = 0.7$
$\beta_i = \beta_e$	$m = 0.18$

$\beta_i = \beta_e$

$Vre = 0.7\ Vri$

$V_w = 1070$ m/s

TF = 191.63 N.

AF = 17.26 N

$\beta_i = 27^o$

WD = 47.91 kW.

$\eta = 61.53\%$

The steam velocity leaving the nozzle is 590 m/s and the nozzle angle is 20°. The blade is running at 2800 rpm and blade diameter is 1050 mm. The axial velocity at rotor outlet is155 m/s, and the blades are symmetrical. Calculate the work done, the diagram efficiency and the blade velocity coefficient.

Given Data

V_i = 590 m/s

α_i = 20°

N = 2800 rpm

D = 1050 mm = 1.05 m

V_{ae} = 155 m/s

$\beta_i = \beta_e$

$$u = \frac{\pi DN}{60} = 154$$

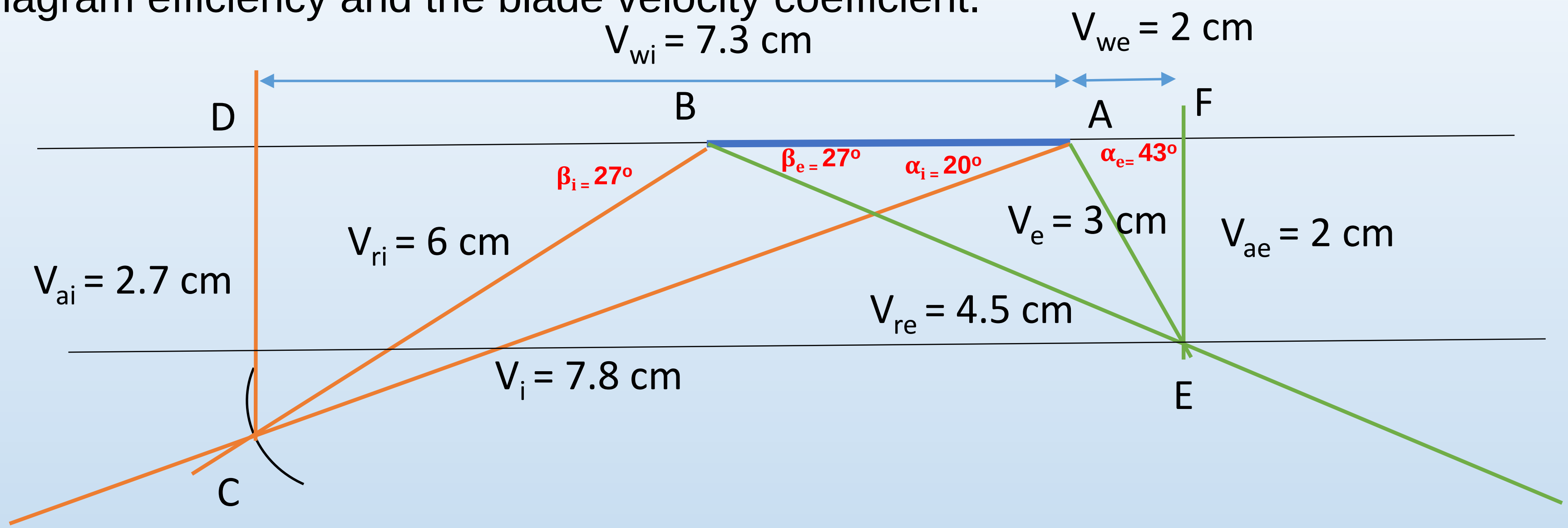
Scale

1:75

u = 2 cm

V_i = 7.8 cm

V_{ae} = 2 cm



WD = 109 kW.

$$\eta_d = \frac{2u(Vw)}{V_i^2} = 62.63\%$$

K = 0.77

Gas turbines	*Steam turbines*
The important components are compressor and combustion chamber.	The important components are steam boiler and accessories.
The mass of gas turbine per kW developed is less.	The mass of steam turbine per kW developed is more.
It requires less space for installation.	It requires more space for installation.
The installation and running cost is less.	The installation and running cost is more.
The starting of gas turbine is very easy and quick.	The starting of steam turbine is difficult and takes long time.
Its control, with the changing load conditions, is easy.	Its control, with the changing load conditions, is difficult.
A gas turbine does not depend on water supply.	A steam turbine depends on water supply.
Its efficiency is less.	Its efficiency is higher.

Gas Turbines

- A gas turbine is a machine delivering mechanical power or thrust. It does this using a gaseous working fluid. The mechanical power generated can be used by, for example, an industrial device.
- The outgoing gaseous fluid can be used to generate thrust. In the gas turbine, there is a continuous flow of the working fluid.

Efficiency is 20 to 30% whereas that of steam power plant is 38 To 48%

Major Applications of Gas Turbine

1. Aviation(self contained, light weight don’t require cooling
2. Power Generation
3. Oil and Gas industry(cheaper supply of fuel and low installation cost)
4. Marine propulsion

Gas Turbine

Hot gases move through a multistage gas turbine.

Like in steam turbine, the gas turbine also has stationary and moving blades.

The stationary blades

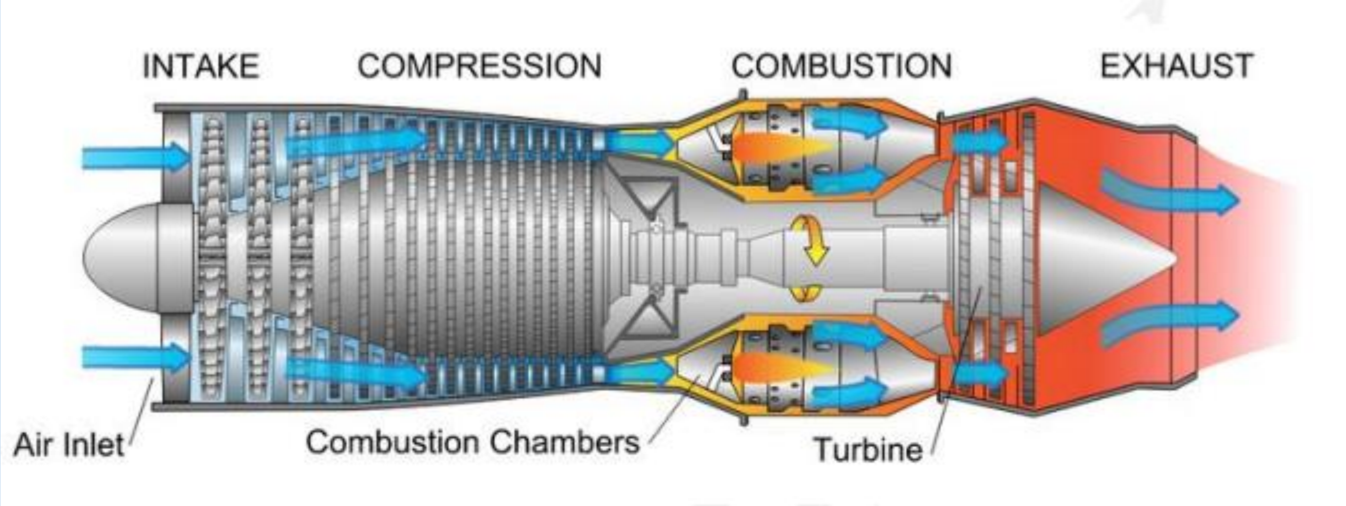
- ✓ guide the moving gases to the rotor blades
- ✓ adjust its velocity.

The shaft of the turbine is coupled to a generator.

Working principle :

- Air is compressed(squeezed) to high pressure by a compressor.
- Then fuel and compressed air are mixed in a combustion chamber and ignited.
- Hot gases are given off, which spin the turbine wheels.
- Gas turbines burn fuels such as oil, natural gas and pulverized(powdered) coal.
- Gas turbines have three main parts:

  i) Air compressor
  ii) Combustion chamber
  iii) Turbine



Advantages of Gas turbine power plants.

- Storage of fuel requires less area and handling is easy.
- The cost of maintenance is less.
- It is simple in construction. There is no need for boiler, condenser and other accessories as in the case of steam power plants.
- Cheaper fuel such as kerosene , paraffin, benzene and powdered coal can be used which are cheaper than petrol and diesel.
- Gas turbine plants can be used in water scarcity areas.
- Less pollution and less water is required.

Disadvantages of gas turbine power plant

1. 66% of the power developed is used to drive the compressor. Therefore the gas turbine unit has a low thermal efficiency.
2. The running speed of gas turbine is in the range of (40,000 to 100,000 rpm) and the operating temperature is as high as 1100 – 1260^{0}C. For this reason special metals and alloys have to be used for the various parts of the turbine.
3. High frequency noise from the compressor is objectionable.

Performance Terms

- **Pressure Ratio-** Ratio of the cycle's highest pressure to its lowest pressure.
- **Work Ratio**: Ratio of network output to the total work developed in the turbine.
- **Air Ratio**: kg of air entering the compressor inlet per unit of cycle net output, Kg/kWh
- **Compression efficiency**: Ratio of work needed for ideal air compressor through a given pressure range to work actually used by the compressor.
- **Engine Efficiency**: It is the ratio of the work actually developed by the turbine expanding hot power gas through a given pressure range to that would be yeilded for ideal expansion conditions
- **Machine Efficiency**: Collective term of engine efficiency and compressor efficiency of turbine and compressor.
- **Combustion Efficiency**: It is the ratio of heat actually released by 1 g of the fuel to heat that would be released by complete perfect combustion.
- **Thermal Efficiency**: It is the percentage of total energy input appearing as net work output of the cycle.

TYPES OF GAS TURBINE POWER PLANTS

The gas turbine power plants can be classified mainly into two categories. These are :open cycle gas turbine power plant and closed cycle gas turbine power plant.

Open Cycle Gas Turbine Power Plant In this type of plant the atmospheric air is charged into the combustor through a compressor and the exhaust of the turbine also discharge to the atmosphere.

Closed Cycle Gas Turbine Power Plant In this type of power plant, the mass of air is constant or another suitable gas used as working medium, circulates through the cycle over and over again.

OPEN CYCLE GAS TURBINE POWER PLANTAND ITS CHARACTERISTICS

Gas turbines usually operate on an open cycle

Air at ambient conditions is drawn into the compressor, where its temperature and pressure are raised. The high pressure air proceeds into the combustion chamber, where the fuel is burned at constant pressure. The high-temperature gases then enter the turbine where they expand to atmospheric pressure while producing power output.

Some of the output power is used to drive the compressor.

The exhaust gases leaving the turbine are thrown out (not re-circulated), causing the cycle to be classified as an **open cycle**

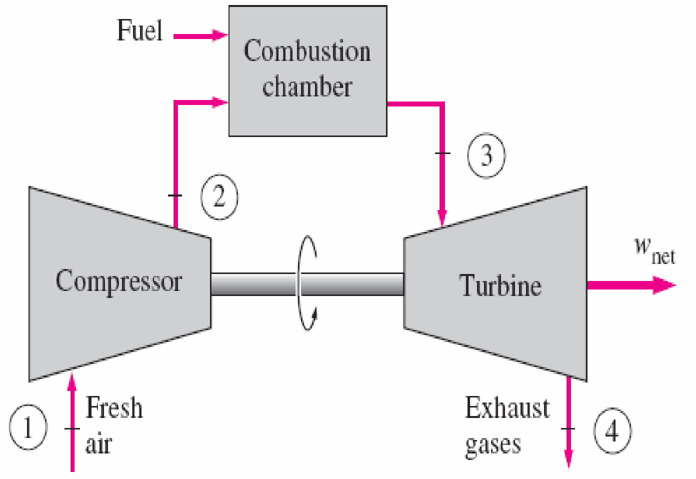


FIGURE 9–29

An open-cycle gas-turbine engine.

- The ideal cycle that the working fluid undergoes in the closed loop is the **Brayton cycle.** It is made up of four internally reversible processes:

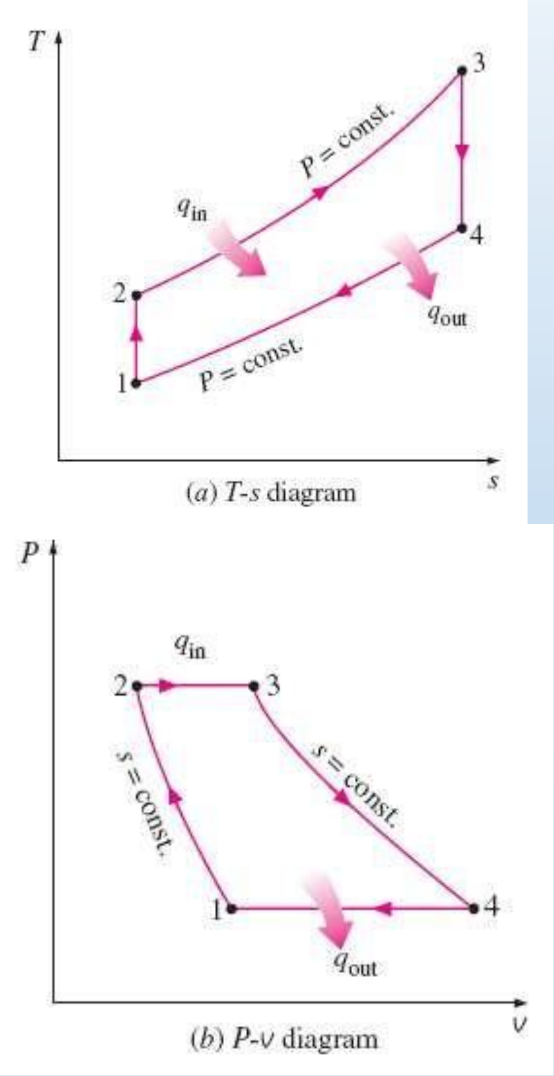
1-2 Isentropic compression;(No change in entropy)

2-3 Constant-pressure heat addition;

3-4 Isentropic expansion;

4-1 Constant-pressure heat rejection.

The *T-s* diagrams of an ideal Brayton cycle.



(a) *T-s* diagram

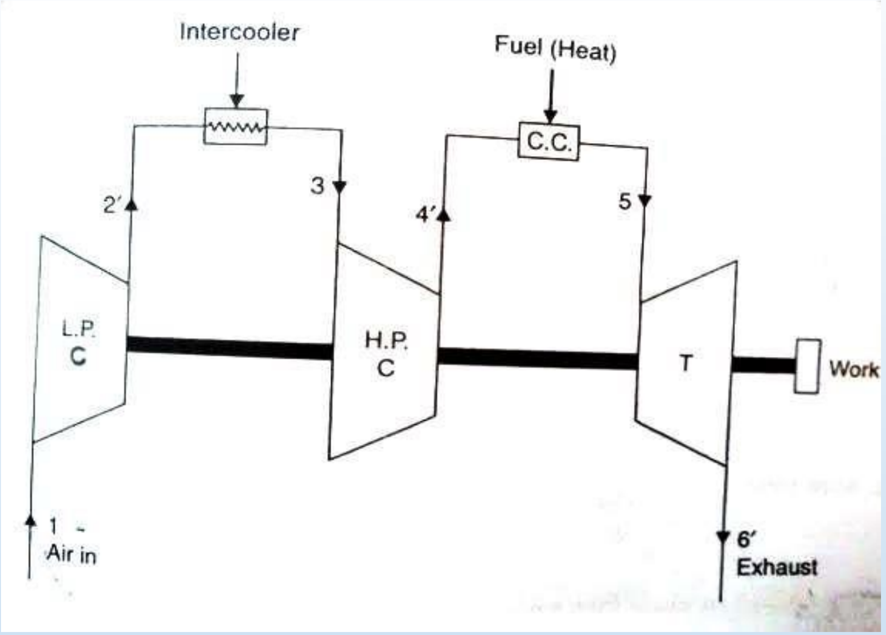
(b) *P-v* diagram

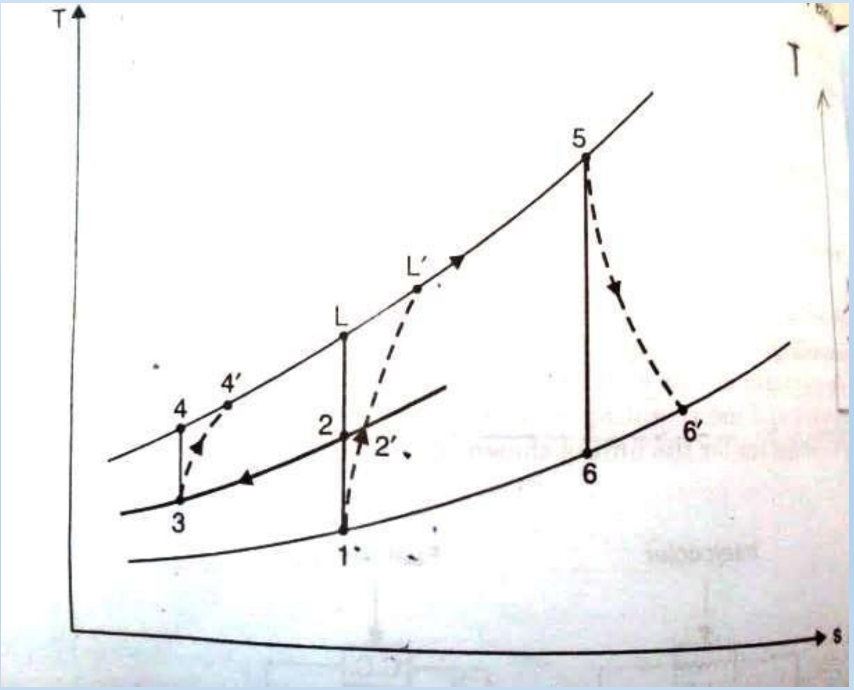
Methods of Improvement of Thermal Efficiency of Open Cycle Gas Turbine Plant

1. Intercooling
2. Reheating
3. Regeneration

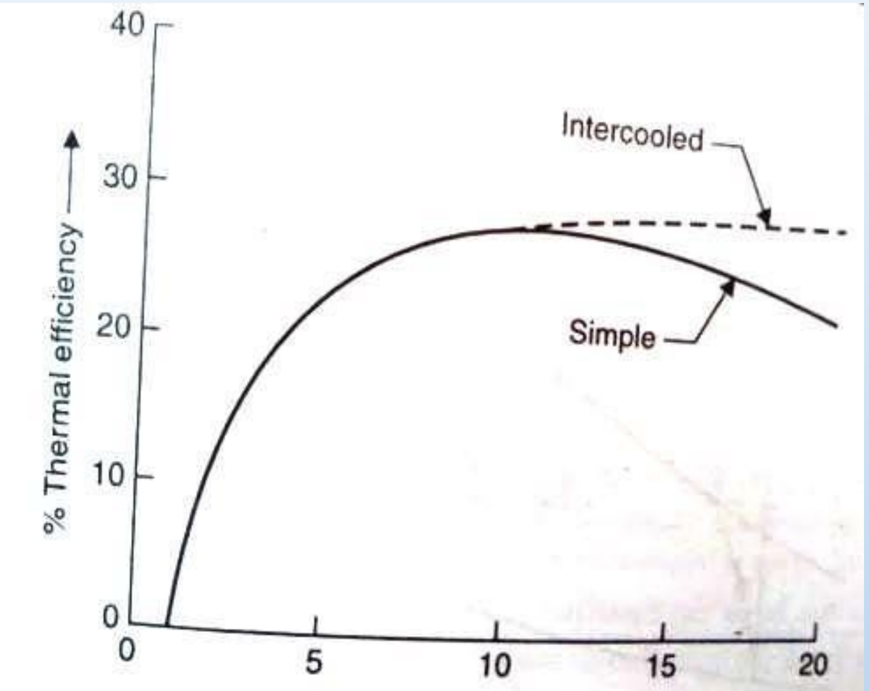
Intercooling

- A compressor utilizes the major percentage of power developed by the gas turbine. The work required by the compressor can be reduced by compressing the air in two stages and incorporation a intercooler between the two.
  - 1-2’: LP compression
  - 2’-3: Intercooling
  - 3-4’: H.P. compression
  - 4’-5: C.C. Combustion chamber(heating)
  - 5-6’: T(Turbine) -Expansion





- Work Ratio is increased
- Thermal efficiency decreases but it increases at high pressure ratio.



Reheating

- The output of gas turbine can be improved by expanding the gasses in two stages with a reheater between the two.
- The H.P. turbine drives the compressor and the LP turbine provides useful power output.

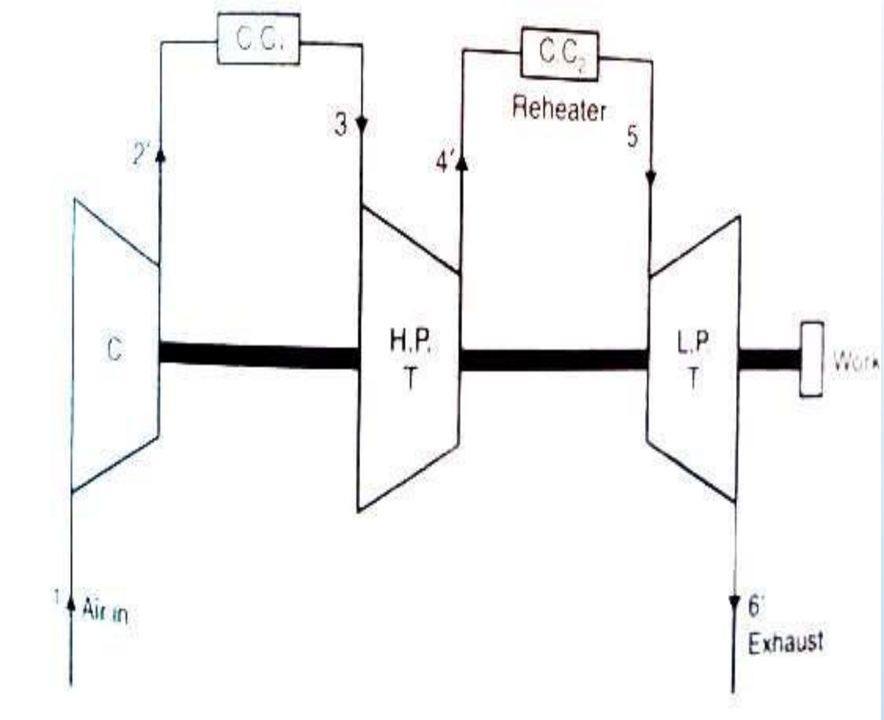
1-2’: Compression

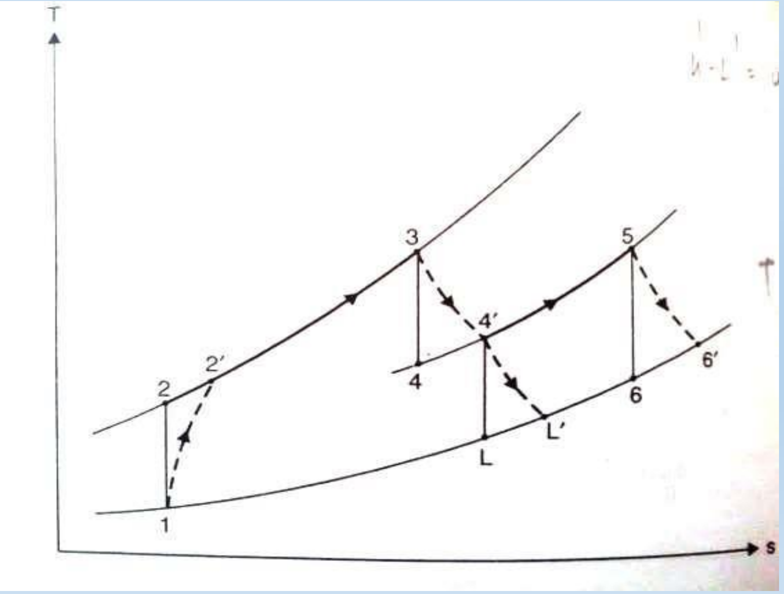
2’-3: C.C (heating)

3’-4’: Turbine(Expansion)

4’-5: Reheater(heating)

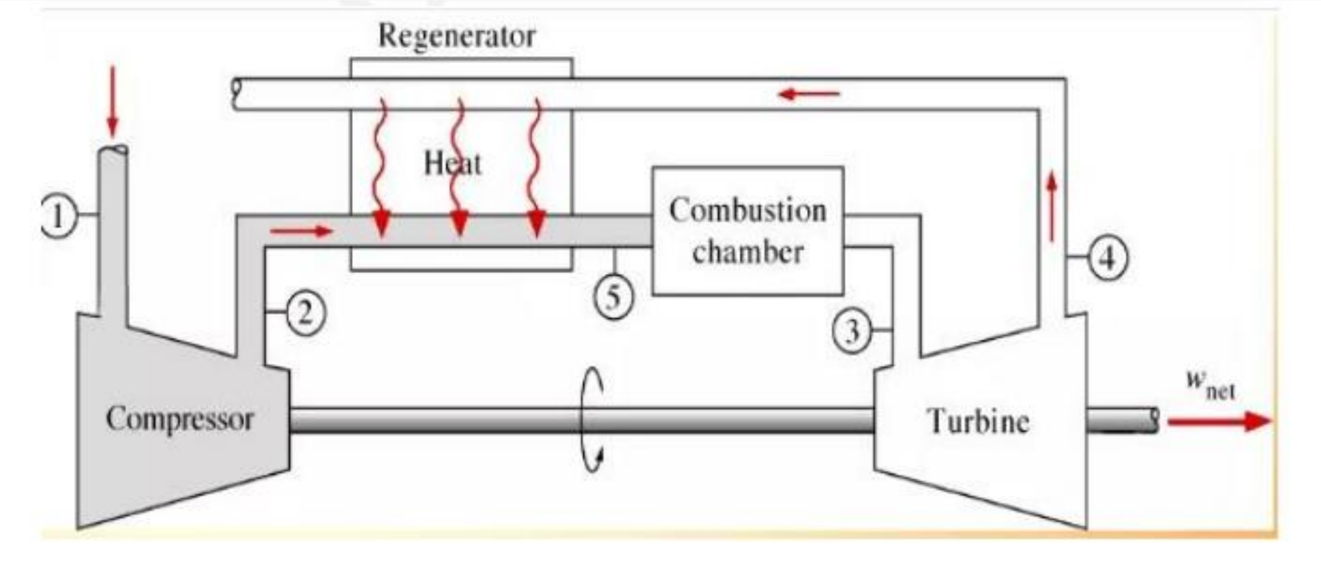
5-6’: Turbine(Expansion)

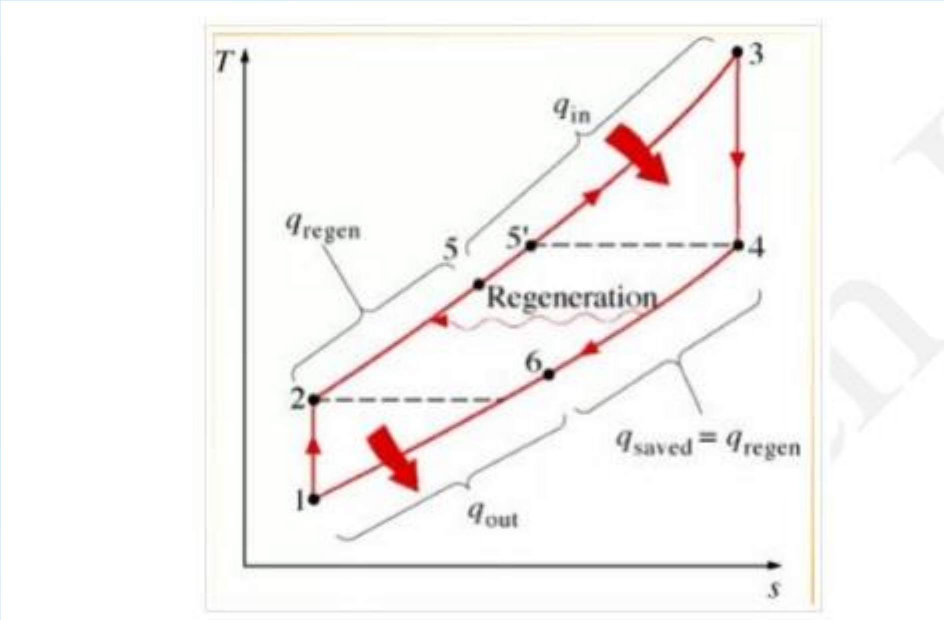




Regeneration

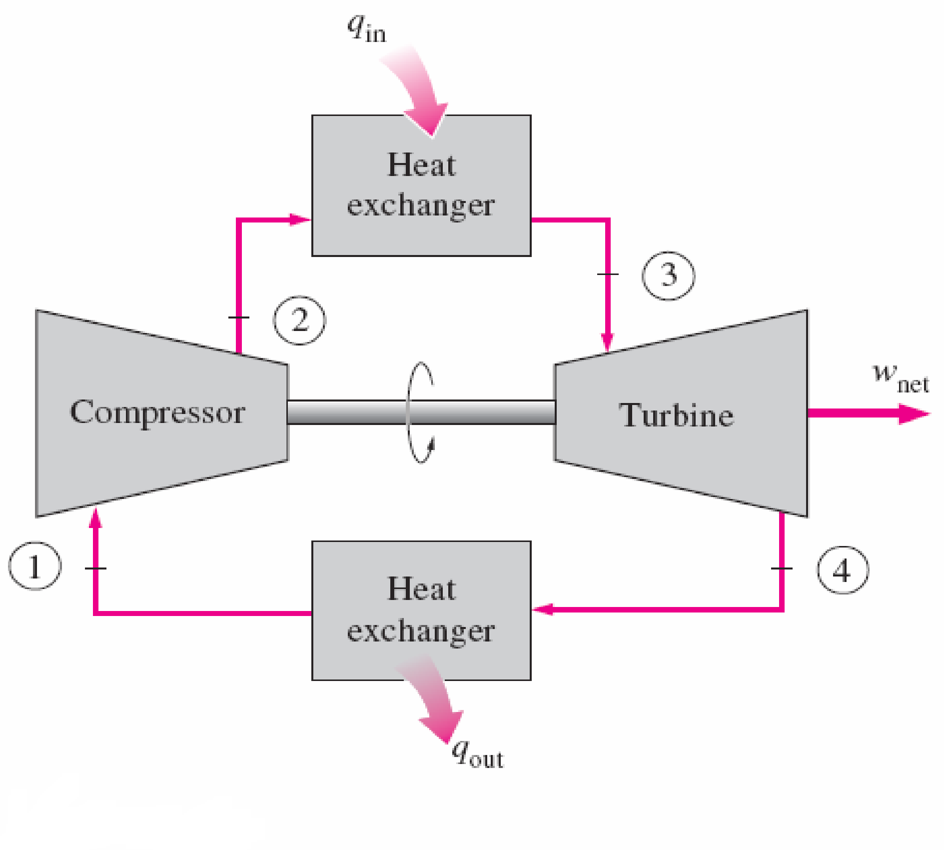
- The exhaust gasses from the turbine carry a large quantity of heat with them since their temperature is far above the ambient temperature.
- They can be used to heat air coming from the compressor there by reducing the mass of fuel supplied in the combustion chamber.





CLOSED CYCLE GAS TURBINE POWER PLANT AND ITS CHARACTERISTICS

- The compression and expansion processes remain the same, but the combustion process is replaced by a constant-pressure heat addition process from an external source.
- The exhaust process is replaced by a constant-pressure heat rejection process to the ambient air.

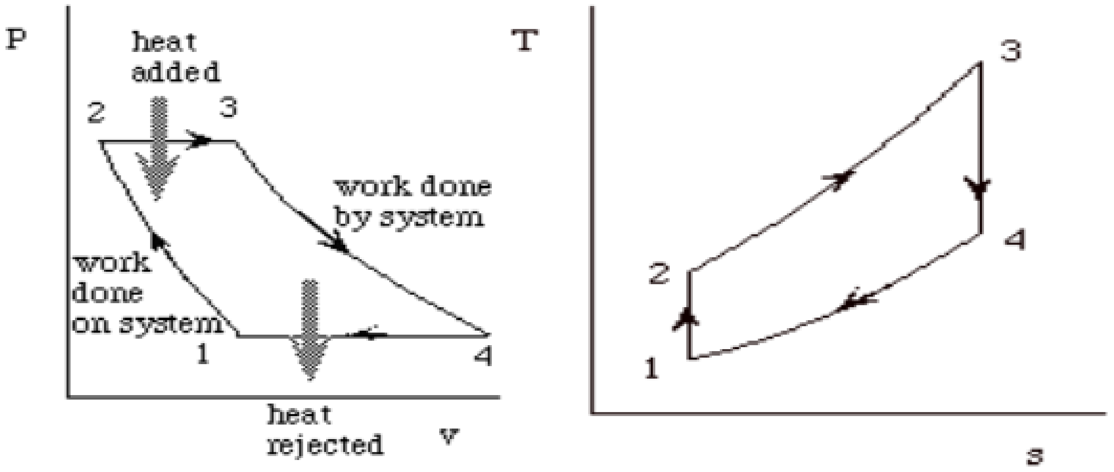


A closed-cycle gas-turbine engine.

DIFFERENCE BETWEEN OPEN AND CLOSED GAS TURBINE:

S. No	Closed cycle gas turbine	Open cycle gas turbine
1.	Combustion of fuel is external	Combustion of fuel is internal.
2.	Gas from turbine is passed into cooling chamber.	Gas from turbine is exhausted to atmosphere.
3.	Any type of fluid is used.	Only air can be used.
4.	Turbine blades cannot be contaminated.	Turbine blades get contaminated.
5.	Working fluid circulated continuously.	Working fluid replaced continuously.
6.	Mass of installation per KW is more.	Mass of installation per KW is less.
7.	Heat exchanger is used.	Heat exchanger is not used.
8.	This system required more space.	This system required less space.
9.	Since exhaust is cooled by circulating water, it is best suited fo stationary installation, marine use.	Since turbine exhaust is discharged into atmosphere, it is best suited for moving Vehicle like Aircraft.
10.	Maintenance cost is high.	Maintenance cost is low.

PERFORMANCE CALCULATION OF OPEN AND CLOSED GAS TURBINE CYCLE:



1-2 Process: Adiabatic compression process

Compressor Work $(W_C) = mc_p(T_2 - T_1)$ **kJ**

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}}$$

2-3 Process: Constant pressure heat addition

$$Q_S = mc_p(T_3 - T_2)\ \mathbf{kJ}$$

3-4 Process: Adiabatic Expansion process

Turbine Work $(W_T) = mc_p(T_3 - T_4)$ **kJ**

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}}$$

4-1 Process: Constant pressure Heat Rejection

$$Q_R = mc_p(T_4 - T_1)\ \mathbf{kJ}$$

Net Work done

$$W_{net} = Q_s - Q_R\ \mathbf{kJ}$$

Thermal Efficiency

$$\eta = \frac{\text{Work done}}{\text{Heat Supplied}} = \frac{W_{net}}{Q_S} = \frac{mc_p(T_3 - T_2) - mc_p(T_3 - T_4)}{mc_p(T_3 - T_2)} = 1 - \frac{T_3 - T_4}{T_3 - T_2}$$

Efficiency

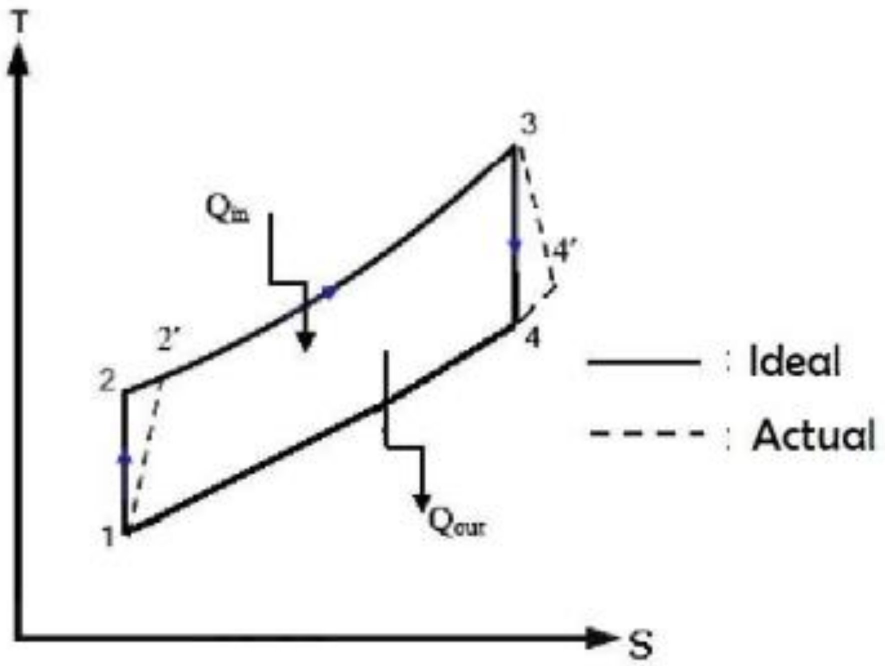
$$\eta = \mathbf{1} - \frac{\mathbf{1}}{(\mathbf{r_p})^{\frac{\gamma-1}{\gamma}}}$$

Turbine Work Ratio

$$W_R = \frac{W_T}{W_{net}} = 1 - \frac{T_1}{T_3}(r_c^{\gamma-1})$$

Causes of Departure of Actual Cycle from Ideal Cycle

In practice, losses due to friction, heat transfer, shock, etc. occur in both the compressor and turbine components (i.e. the compression in the compressor and expansion in the turbine are not isentropic) so that actual power absorbed by the compressor increases and actual output of the turbine decreases compared with isentropic operation. Thus, in practice the compression is polytropic (1-2) and not isentropic (1-2'). Similarly, expansion is polytropic (3-4) and not isentropic (3-4'). If η_c and η_T are isentropic efficiencies of compressor and turbine respectively,



Ideal Gas turbine cycle: 1-2-3-4

Actual Gas turbine cycle: $1' - 2' - 3' - 4'$

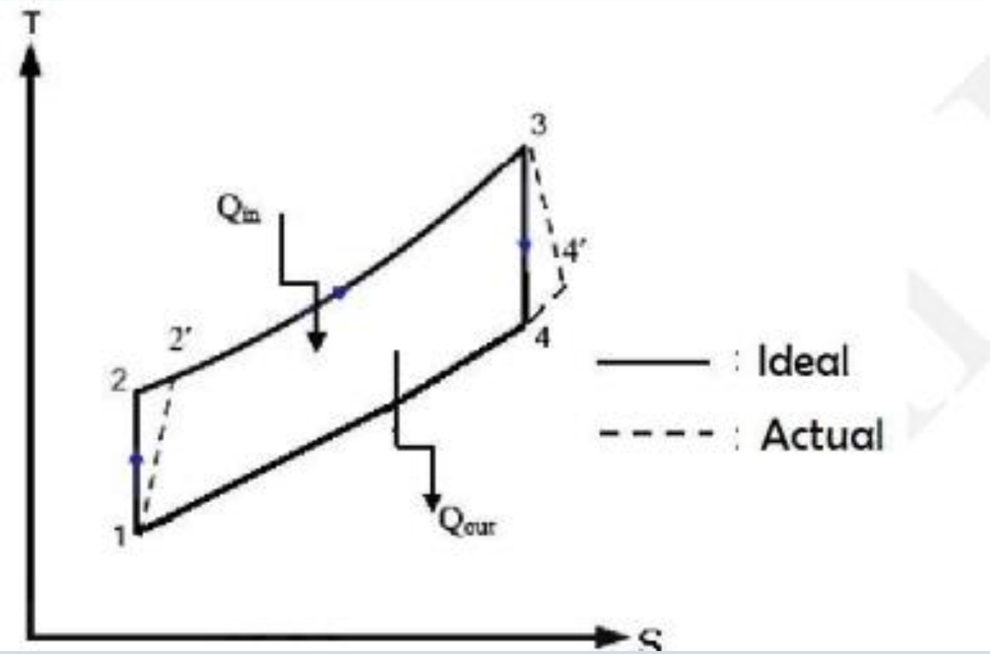
Actual compression work: $pV^n = C$

Actual Compressor Work $(W_C) = c_p(T_{2'} - T_1)$

Actual Compressor Work $(W_C) = \frac{\text{Ideal compressor work}}{\text{Compressor efficiency}}$ ⇨ $W_C = \frac{c_p(T_2 - T_1)}{\eta_c}$ ⇨ $W_C = \frac{c_p T_1\left(\frac{T_2}{T_1} - 1\right)}{\eta_c}$

Where,

$$\textbf{Actual Compressor Work } (W_C) = \frac{c_p T_1\left((r_c)^{\frac{\gamma-1}{\gamma}} - 1\right)}{\eta_c} \qquad \therefore \frac{T_2}{T_1} = (r_c)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}}$$



Actual Turbine Work: $pV^n = C$

Actual Turbine Work $(W_T) = c_p(T_3 - T_{4'})$

Actual Turbine Work (W_T) = Ideal Turbine Work × Turbine Efficiency ⇨ $W_T = c_p(T_3 - T_4) \times \eta_T$

⇨ $W_T = c_p \eta_T \left(1 - \frac{T_4}{T_3}\right)$ ⇨ $\mathbf{W_T = c_p \eta_T \left(1 - \frac{1}{(r_c)^{\frac{\gamma-1}{\gamma}}}\right)}$ $\qquad \therefore \frac{T_4}{T_3} = (r_c)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}}$

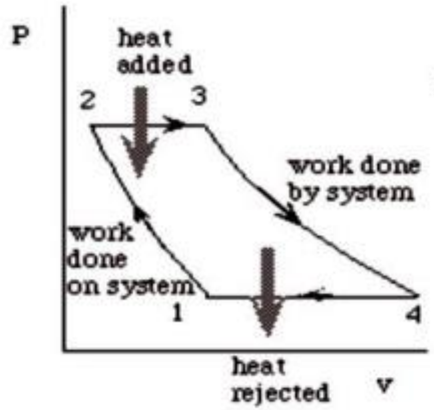
Consider an air standard cycle in which the air enters the compressor at 1.0 bar and 20°C. The pressure of air leaving the compressor is 3.5 bar and the temperature at turbine inlet is 600°C. Determine per kg of air, (i)Efficiency of the cycle, (ii) Heat supplied to air, (iii) Work available at the shaft,(iv) Heat rejected in the cooler and (v) Temperature of air leaving the turbine. For air γ = 1.4 and Cp = 1.005 kJ/kg K.

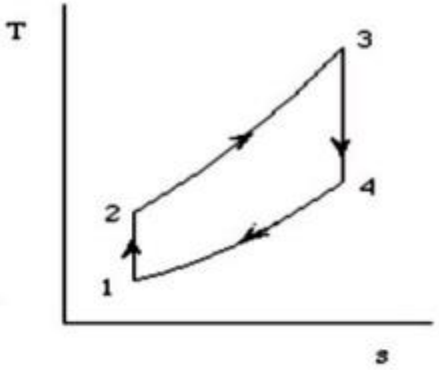
GIVEN:

$p_1 = 1\text{bar},\ T_1 = 20^\circ C + 273$

$p_3 = 3.5\text{bar},$

$T_3 = 600^\circ C + 273$





1-2 Process: Adiabatic compression process

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_2 = r_p^{\frac{\gamma-1}{\gamma}} \times T_1 = 3.5^{\frac{1.4-1}{1.4}} \times 293 \Rightarrow \mathbf{T_2 = 419.09\ K}$$

$$\text{Compressor Work } (W_C) = c_p(T_2 - T_1) = 1.005(419.09 - 293) \Rightarrow \mathbf{W_C = 126.72\ kJ/kg}$$

2-3 Process: Constant pressure heat addition

$$Q_S = c_p(T_3 - T_2) = 1.005(873 - 419.09) \Rightarrow \mathbf{Q_S = 456.18\ kJ/kg}$$

3-4 Process: Adiabatic Expansion process

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = r_p^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4 = \frac{T_3}{r_p^{\frac{\gamma-1}{\gamma}}} = \frac{873}{3.5^{\frac{1.4-1}{1.4}}} \Rightarrow \mathbf{T_4 = 610.33\ K}$$

$$\text{Turbine Work } (W_T) = c_p(T_3 - T_4) = 1.005(873 - 610.33) \Rightarrow \mathbf{W_T = 263.98\ kJ/kg}$$

4-1 Process: Constant pressure Rejection addition

$$Q_R = c_p(T_4 - T_1) = 1.005(610.33 - 293) \Rightarrow \mathbf{Q_R = 318.92\ kJ/kg}$$

Work done

$$W = Q_s - Q_R = 456.18 - 318.92 \Rightarrow \mathbf{W_{net} = 137.26\ kJ/kg}$$

Thermal Efficiency

$$\eta = \frac{\textit{Work done}}{\textit{Heat Supplied}} = \frac{W_{net}}{Q_S} = \frac{137.26}{456.18} \Rightarrow \mathbf{\eta = 30.08\%}$$

Thank you