RECIPROCATING & ROTARY COMPRESSORS

A lecture by
V.Manikanth
Assistant Professor
Dept. Of Mechanical Engineering
SRKR Engineering College (A)

UNIT-IV

Reciprocating and Rotary Compressors:

Reciprocating compressors-effect of clearance in compressors, volumetric efficiency single stage and multistage compressors-effect of inter cooling in multistage compressors

Vane type, blower, centrifugal compressor- Adiabatic efficiency- Diffuser- Axial flow compressors- Velocity diagrams, degree of reaction, performance characteristics.

Textbooks:

1. A Treatise on Heat Engineering by Vasandhani and Kumar.

2. Applied Thermodynamics-II by R. Yadav

Reference Books:

1. Thermal Engineering, by R. K. Rajput.

2. Gas Turbines, by V. Ganesan.

Compressors are work absorbing devices which are used for increasing pressure of fluid at the expense of work done on fluid. An ***air compressor*** is a pneumatic device that converts power (using an electric motor, diesel or gasoline engine, etc.) into potential energy stored in pressurized air (i.e., compressed air).

Compressors are like fans and blowers but differ in terms of pressure ratios. ***Fan*** is said to have pressure ratio up to 1.1 and **blowers** have pressure ratio between 1.1 and 4 while ***compressors*** have pressure ratios more than 4.

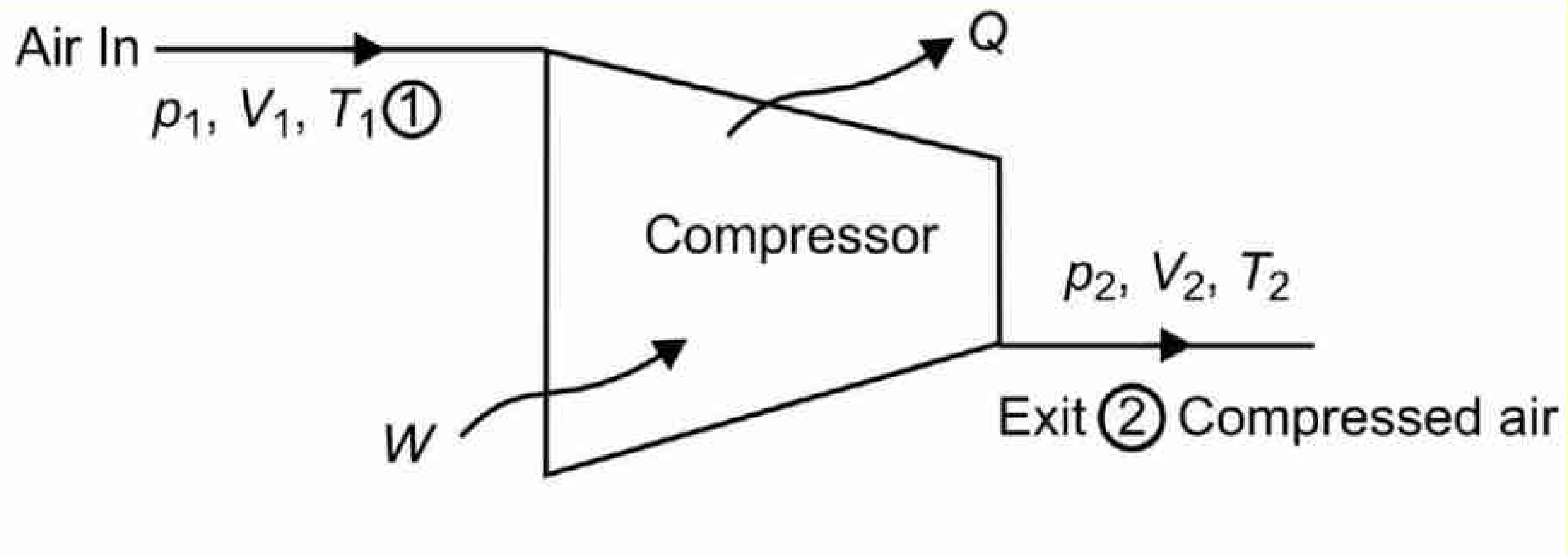
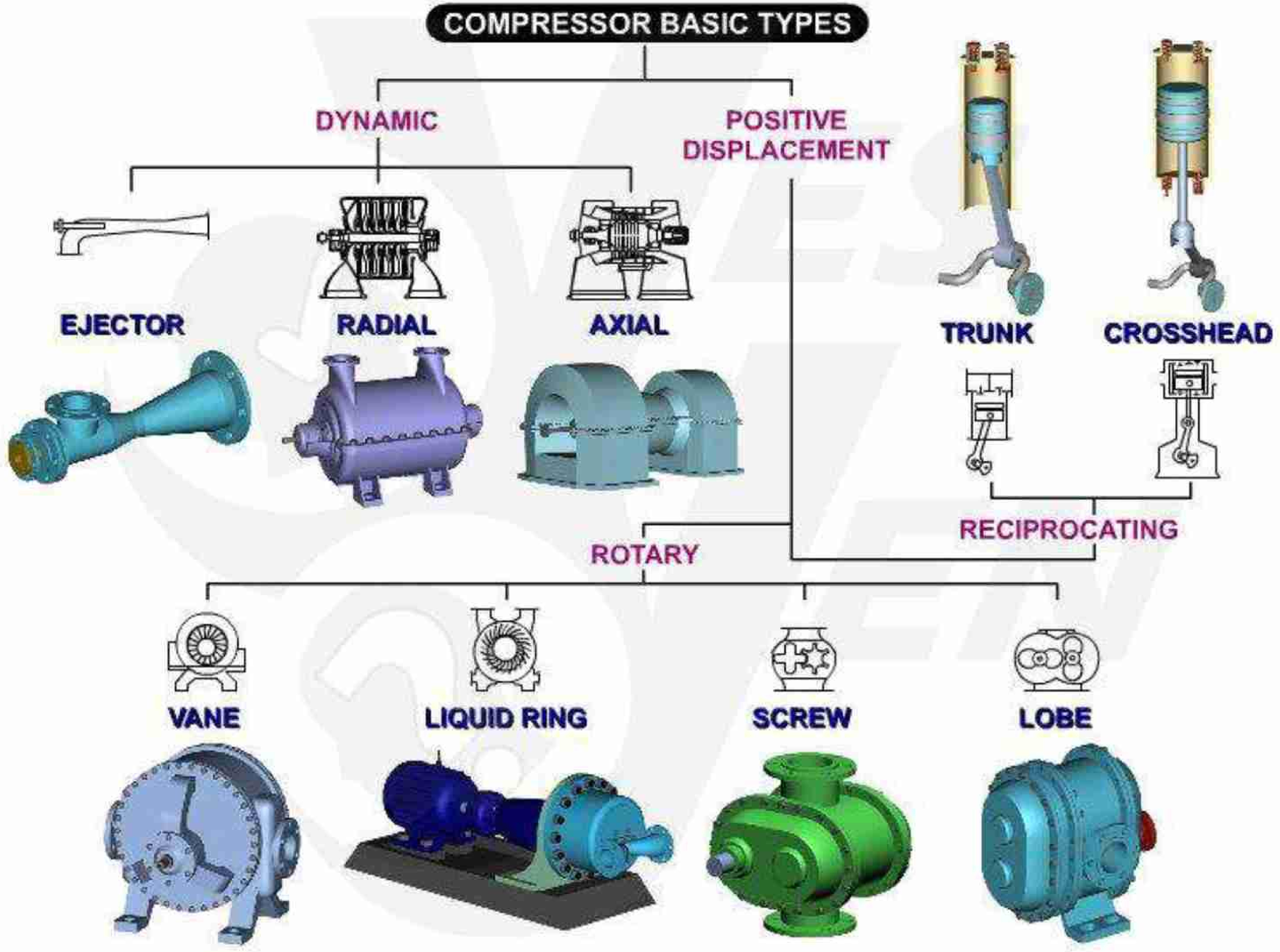


Fig. *Compressor*

By one of several methods, an air compressor forces more and more air into a storage vessel, increasing the pressure. When the tank's pressure reaches its engineered upper limit, the air compressor shuts off. An air compressor must be differentiated from a pump because it works for any gas/air, while pumps work on a liquid.

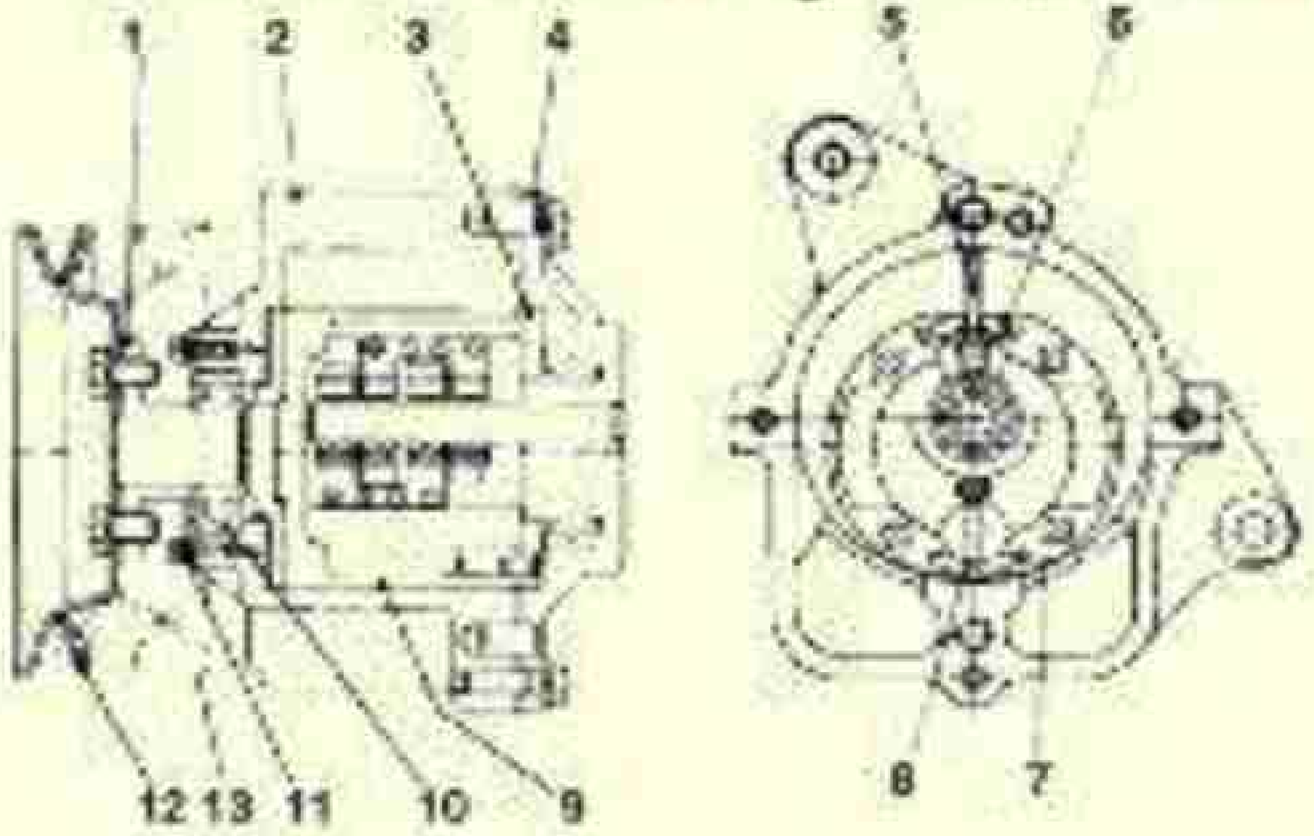

COMPRESSOR BASIC TYPES
DYNAMIC
POSITIVE DISPLACEMENT
EJECTOR
RADIAL
AXIAL
TRUNK
CROSSHEAD
RECIPROCATING
ROTARY
VANE
LIQUID RING
SCREW
LOBE

Air Compressors

Nature of Work (i.e. Use)

Air Pumps / Exhausters

To Produce Vacuum.
i.e. to remove air from a system to create low pressure zone.

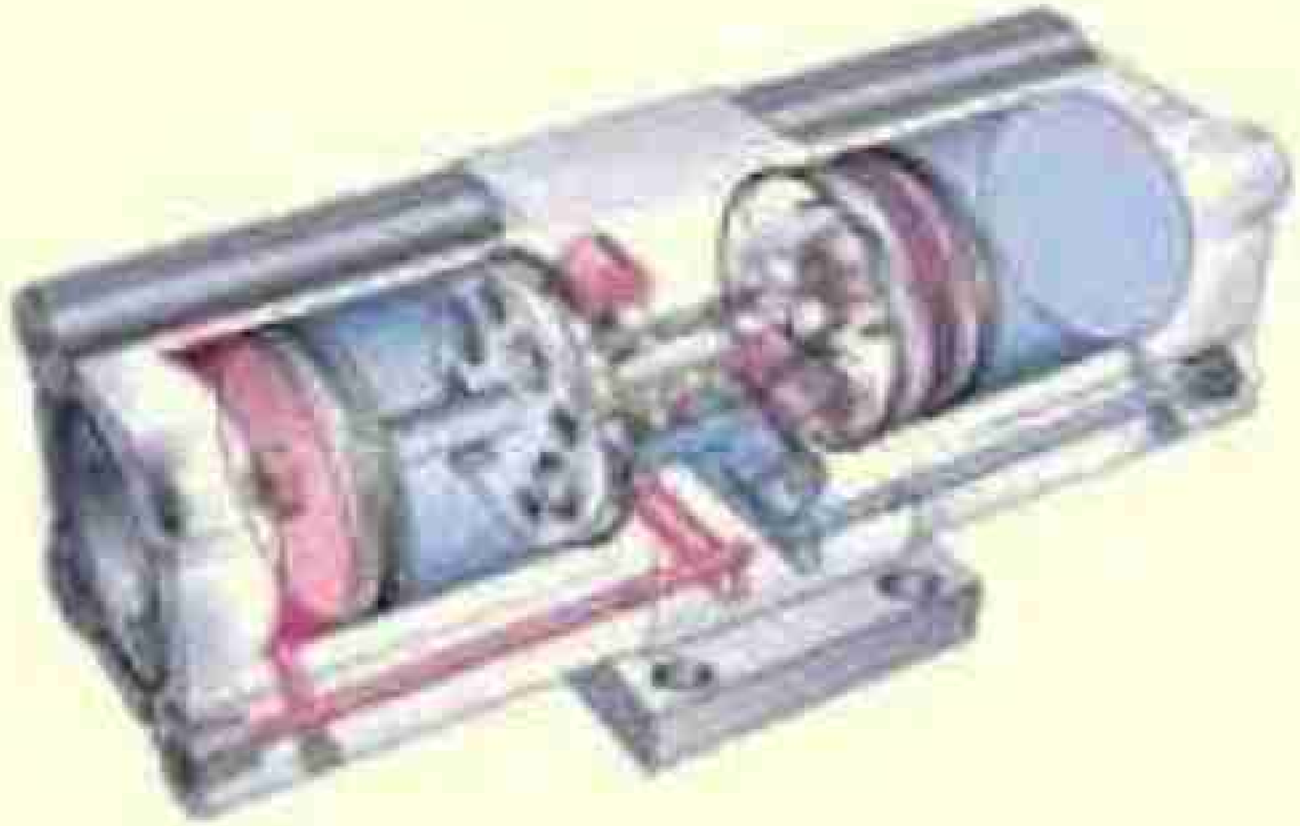


Blowers / Superchargers

Pr. rise is small.
i.e. 0.7 – 1.05 bar

Boosters

Pr. rise for HP gas
i.e. already compressed gas.



Positive displacement

In positive displacement compressors the compression is realized by displacement of solid boundary and preventing fluid by solid boundary from flowing back in the direction of pressure gradient. Due to solid wall displacement these can provide quite large pressure ratios. Positive displacement compressors can be further classified based on the type of mechanism used for compression.

(i) Reciprocating type positive displacement compressors

(ii) Rotary type positive displacement compressors

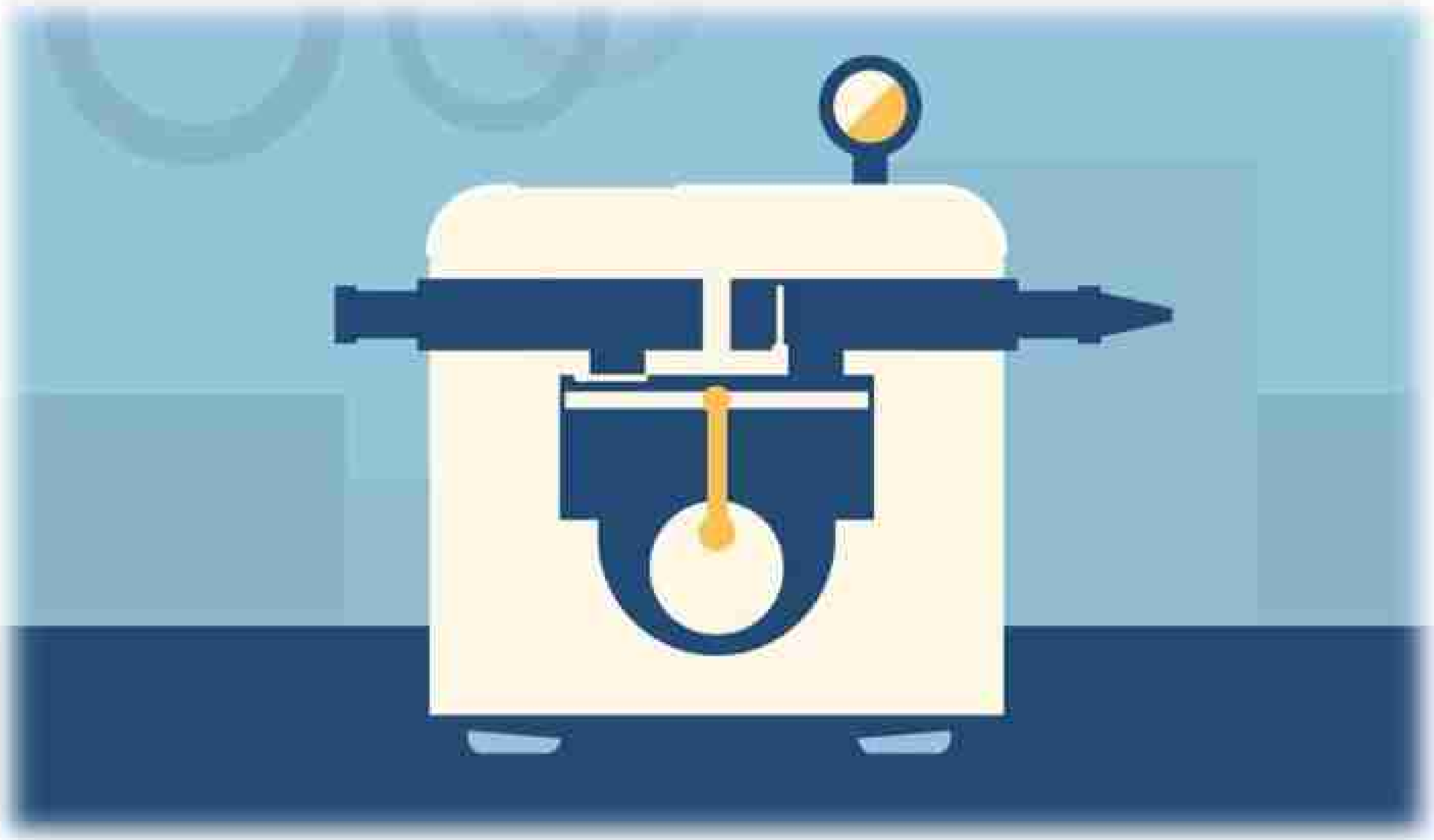


Fig: Reciprocating compressor

Fig: Rotary vane type compressor

RECIPROCATING COMPRESSORS

Construction and Working of a Reciprocating Compressor (Single-stage):

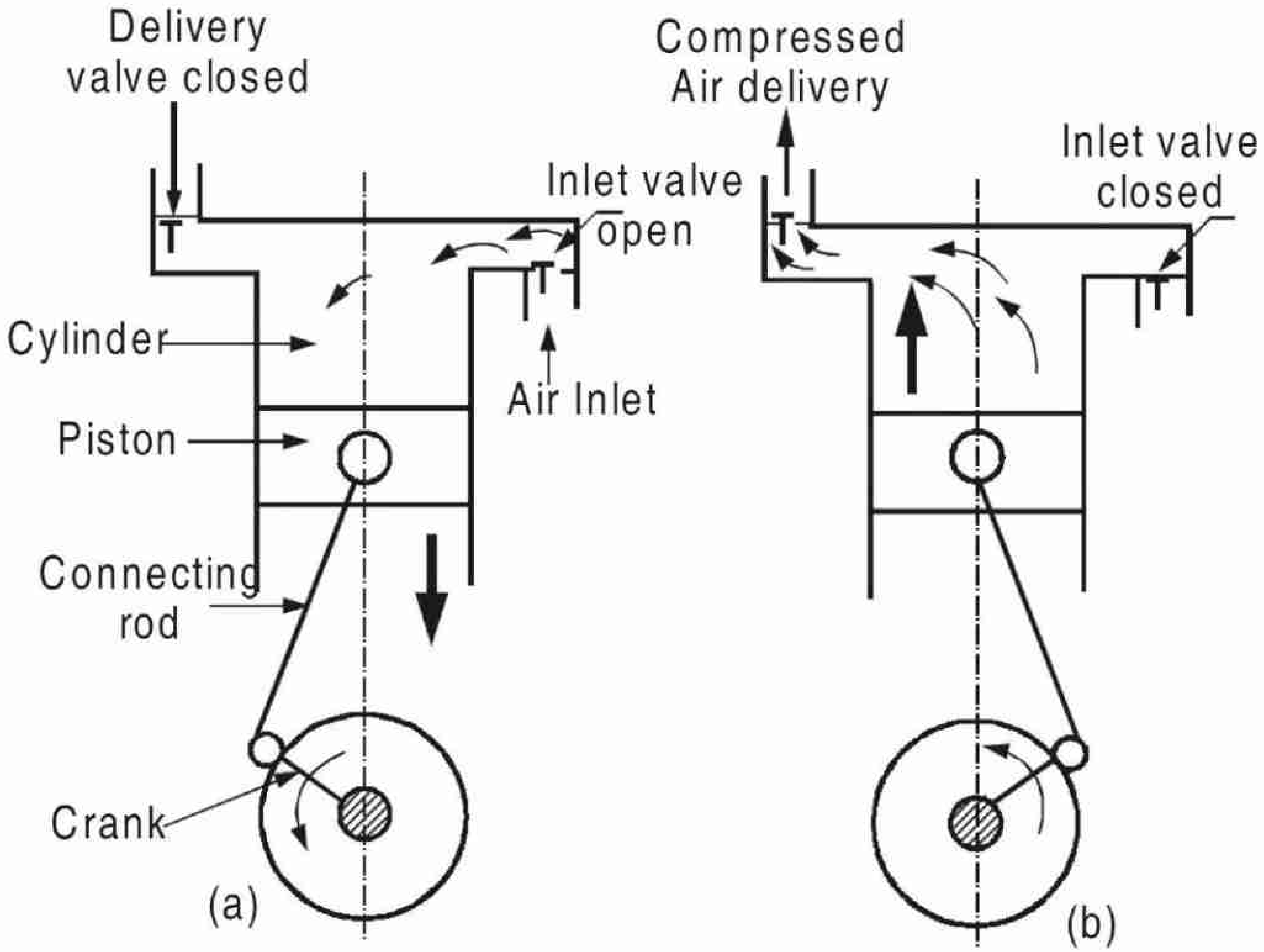
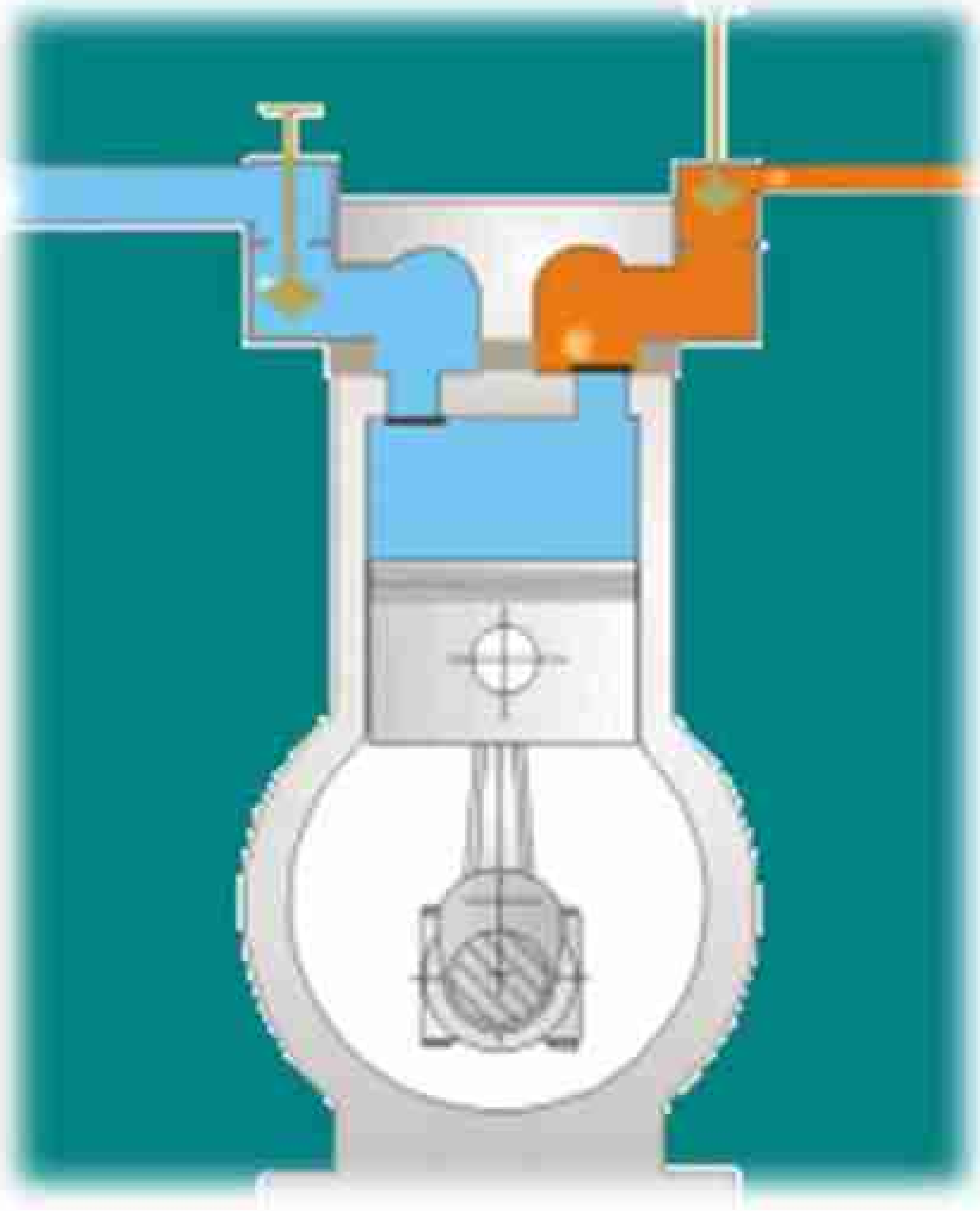


Fig. Working of Reciprocating Air Compressor



Reciprocating compressors generally, employ piston-cylinder arrangement where displacement of piston in cylinder causes rise in pressure. Reciprocating compressors can give large pressure ratios but the mass handling capacity is limited or small. Reciprocating compressor has piston, cylinder, inlet valve, exit valve, connecting rod, crank, piston pin, crank pin and crank shaft. Inlet valve and exit valves may be of spring-loaded type which get opened and closed due to pressure differential across them.

Air gets into cylinder during suction stroke and is subsequently compressed in next stroke with both inlet valve and exit valve closed. Both inlet valve and exit valves are of plate type and spring loaded to operate automatically as and when sufficient pressure difference is available to cause deflection in spring of valve plates to open them. After piston reaching BDC it reverses its motion and compresses the air inducted in previous stroke. Compression is continued till the pressure of air inside becomes sufficient to cause deflection in exit valve. Now, when exit valve plate gets lifted the exhaust of compressed air takes place. Reciprocating compressor described above has suction, compression and discharge as three prominent processes getting completed in two strokes of piston or one revolution of crank shaft.

THERMODYNAMIC ANALYSIS

Single-stage Compressor : Equation for Work (*neglecting clearance volume*)

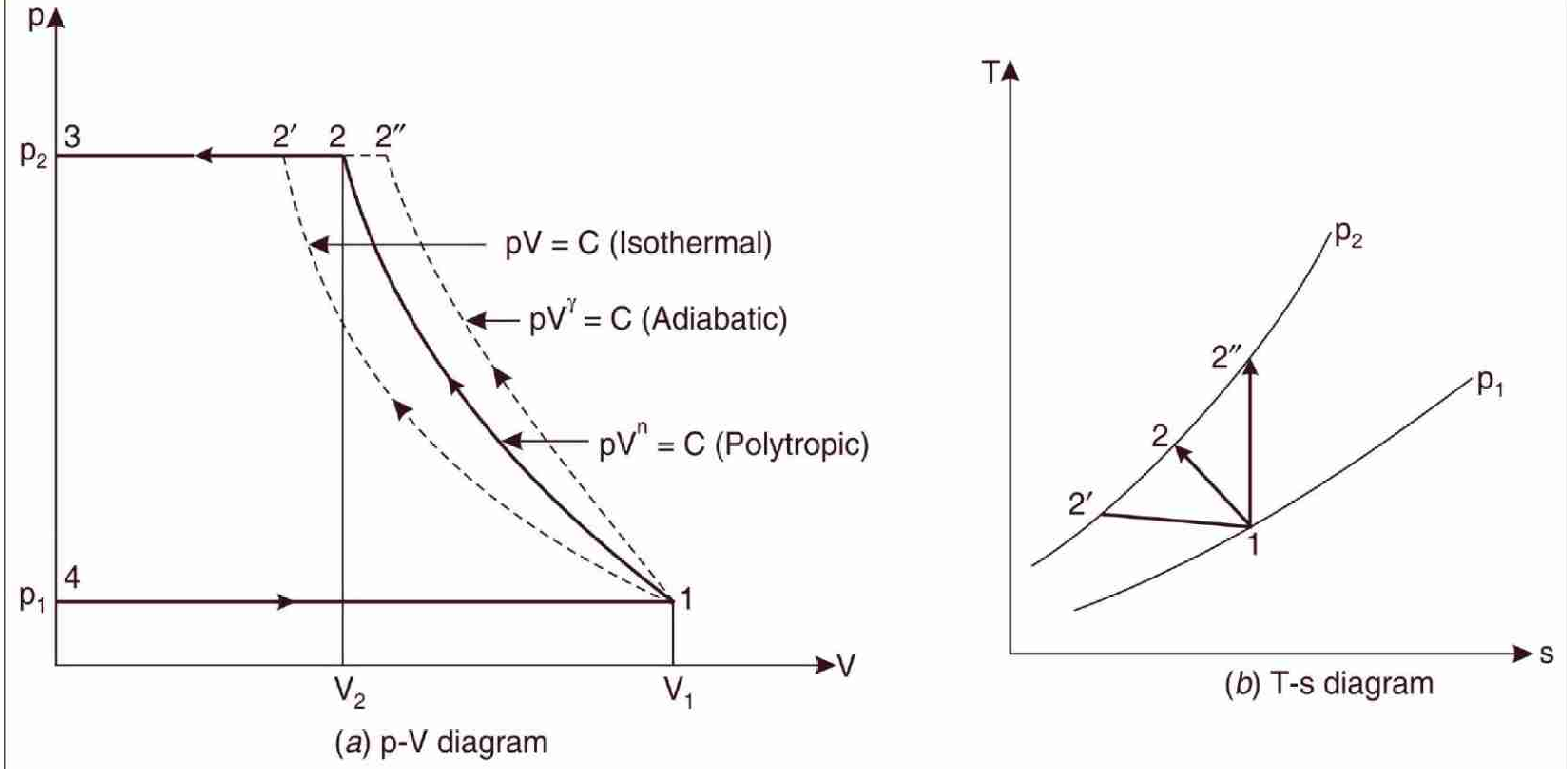


Fig. Theoretical *p-V* and *T-s* diagrams for a single-stage reciprocating air compressor.

The sequence of operations as represented on the diagram, are as follows :

(*i*) *Operation 4-1* : Volume of air V_1 aspirated into the compressor at pressure p_1 and temperature T_1.

(*ii*) *Operation 1-2* : Air compressed according to the law $pV^n = C$ from p_1 to pressure p_2. Volume decreases from V_1 to V_2. Temperature increases from T_1 to T_2.

(*iii*) *Operation 2-3* : Compressed air of volume V_2 and at pressure p_2 with temperature T_2 delivered from the compressor.

During compression, due to its excess temperature above the compressor surroundings, the air will lose some heat. Thus, neglecting the internal effect of friction which is small in the case of the reciprocating compressor, the index **n** is less than **γ**, the adiabatic index.

'Isothermal compression cannot be achieved in practice, but an attempt is made to approach the isothermal case by cooling the compressor either by addition of cooling fins or a water jacket to the compressor cylinder'.

For a reciprocating compressor, a comparison between the actual work done during compression and the ideal isothermal work done is made by means of the isothermal efficiency.

This is defined as,

$$\boxed{\text{Isothermal efficiency} = \frac{\text{Isothermal work done}}{\text{Actual work done}}}$$

Thus, the higher the isothermal efficiency, the more nearly has the actual compression approached the ideal isothermal compression.

Total shaft work done/cycle, W = Area 41234

or W = Area under 4-1 – Area under 1-2 – Area under 2-3

$$= p_1V_1 - \frac{p_2V_2 - p_1V_1}{n-1} - p_2V_2$$

$$= (p_1V_1 - p_2V_2) - \left(\frac{p_2V_2 - p_1V_1}{n-1}\right) = (p_1V_1 - p_2V_2) + \left(\frac{p_1V_1 - p_2V_2}{n-1}\right)$$

$$= \left(1 + \frac{1}{n-1}\right) (p_1V_1 - p_2V_2)$$

$$\therefore \qquad W = \left(\frac{n}{n-1}\right) (p_1V_1 - p_2V_2) \qquad ...(1)$$

This equation can be modified as follows :

$$W = \frac{n}{n-1} (p_1V_1 - p_2V_2) = \frac{n}{n-1} \cdot p_1V_1 \left(1 - \frac{p_2V_2}{p_1V_1}\right) \qquad ...(2)$$

Now $\qquad p_1V_1^n = p_2V_2^n$

$$\therefore \qquad \frac{V_2}{V_1} = \left(\frac{p_1}{p_2}\right)^{1/n}$$

and substituting this into eqn. (2), we have

$$W = \frac{n}{n-1} p_1V_1 \left\{1 - \frac{p_2}{p_1}\left(\frac{p_1}{p_2}\right)^{1/n}\right\} = \frac{n}{n-1} p_1V_1 \left\{1 - \frac{p_2}{p_1}\left(\frac{p_2}{p_1}\right)^{-\frac{1}{n}}\right\}$$

$$= \frac{n}{n-1} p_1V_1 \left\{1 - \left(\frac{p_2}{p_1}\right)^{1-\frac{1}{n}}\right\} = \frac{n}{n-1} p_1V_1 \left\{1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right\} \qquad ...(3)$$

The solution to this equation will always come out *negative* showing that work must be done *on* the compressor. Since only the *magnitude* of the work done is required from the expression then it is often written,

$$W = \frac{n}{n-1} p_1 V_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right\} \quad ...(4)$$

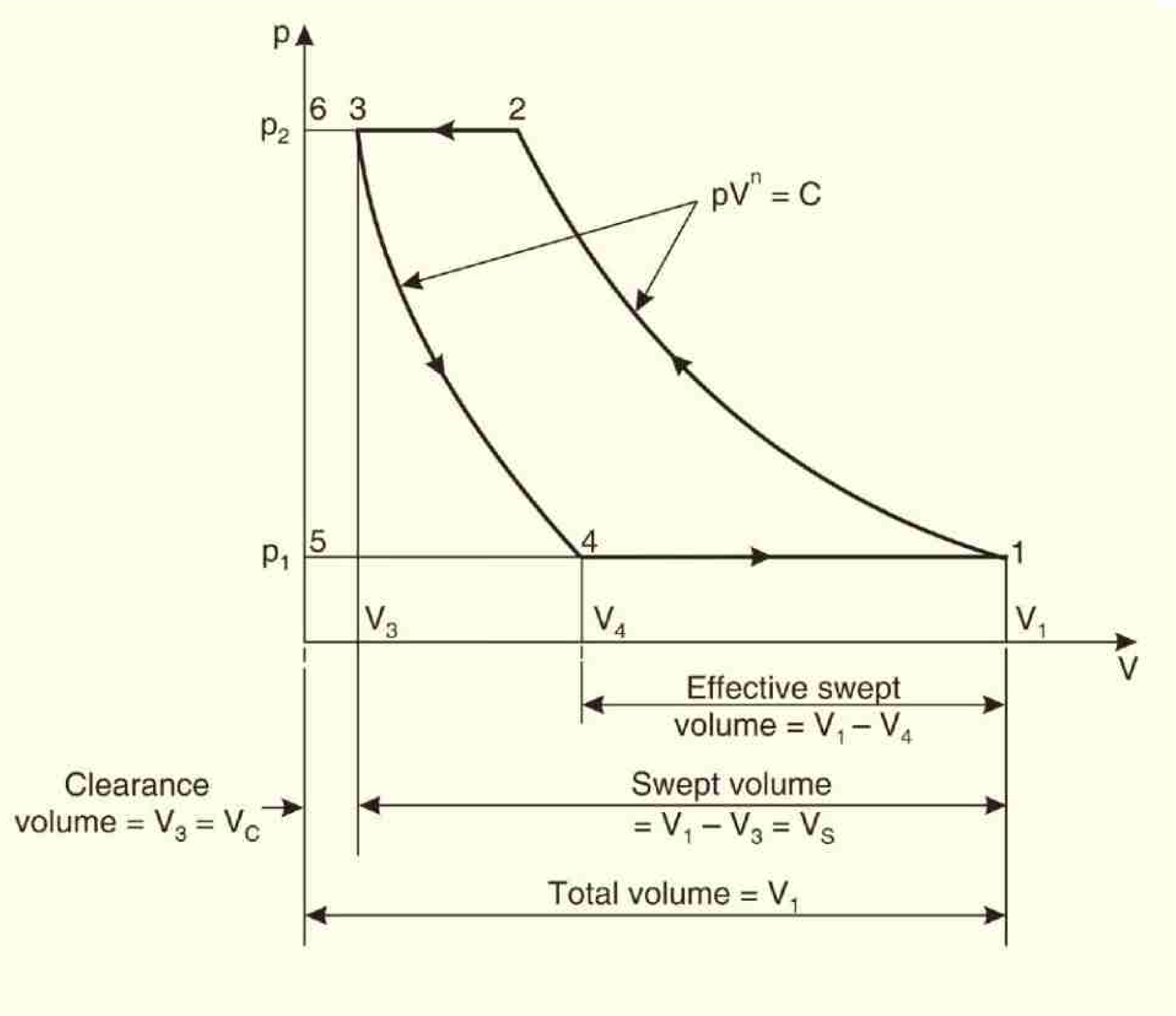
$$W = \frac{n}{n-1} mRT_1 \left\{ \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right\} \quad ...(5)$$

If the air delivery temperature T_2 is required then this can be obtained by using this equation :

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} \quad ...(6)$$

Single-stage Compressor : Equation for Work (*with clearance volume*)

In practice, all reciprocating compressors will have a clearance volume. The clearance volume is that volume which remains in the cylinder after the piston has reached the end of its inward stroke.



volume $(V_1 - V_4)$ is *effective swept volume.*

Work done/cycle, W = Net area 12341 = Area 51265 – Area 54365

Assuming the polytropic index to be same for both compression and clearance expansion, then,

$$W = \frac{n}{n-1}\, p_1 V_1 \left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right\} - \frac{n}{n-1}\, p_4 V_4 \left\{\left(\frac{p_3}{p_4}\right)^{\frac{n-1}{n}} - 1\right\} \qquad \text{...(7)}$$

But $p_4 = p_1$ and $p_3 = p_2$, then eqn. (7) becomes,

$$W = \frac{n}{n-1}\, p_1 V_1 \left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right\} - \frac{n}{n-1}\, p_1 V_4 \left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right\}$$

$$\boxed{W = \frac{n}{n-1}\, p_1 (V_1 - V_4) \left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right\}} \qquad \text{...(8)}$$

The clearance volume reduces the effective swept volume i.e., the mass of air handled but the work done per kg of air delivered remains unaffected.

From the cycle work estimated as above the theoretical power required for running compressor shall be as given ahead.

For single acting compressor running with N rpm, power input required, assuming clearance volume.

$$\boxed{\text{Power required} = \left[\left(\frac{n}{n-1}\right) p_1 (V_1 - V_4) \left\{\left(\frac{p_2}{p_1}\right)^{\frac{(n-1)}{n}} - 1\right\}\right] \times N}$$

$$\boxed{\text{for double acting compressor, power} = \left[\left(\frac{n}{n-1}\right) p_1 (V_1 - V_4) \left\{\left(\frac{p_2}{p_1}\right)^{\frac{(n-1)}{n}} - 1\right\}\right] \times 2N}$$

Volumetric Efficiency

Volumetric efficiency of compressor is the measure of the deviation from volume handling capacity of compressor. Mathematically, the volumetric efficiency is given by the ratio of actual volume of air sucked (effective swept vol.) and swept volume of cylinder. Ideally the volume of air sucked should be equal to the swept volume of cylinder, but it is not so in actual case because of clearance volume. Practically the volumetric efficiency lies between 60 and 90% i.e.., always less than unity.

$$i.e., \quad \text{Volumetric efficiency} = \frac{\text{Effective swept volume}}{\text{Swept volume}} = \frac{V_1 - V_4}{V_1 - V_3}$$

$$\text{The ratio,} \quad \frac{\text{Clearance volume}}{\text{Swept volume}} = \frac{V_3}{V_1 - V_3} = \frac{V_c}{V_s} = k$$ is the clearance ratio.

As a percentage, this ratio will have a value, in general, of between 4% and 10%.

The greater the pressure ratio through a reciprocating compressor, then the greater will be the effect of the clearance volume since the clearance air will now expand through a greater volume before intake conditions are reached. The cylinder size and stroke being fixed, however will mean that (V1– V4), the effective swept volume, will reduce as the pressure ratio increases and thus the volumetric efficiency reduces.

Now

$$\text{Volumetric efficiency, } \eta_{vol.} = \frac{V_1 - V_4}{V_1 - V_3}$$

$$= \frac{(V_1 - V_3) + (V_3 - V_4)}{(V_1 - V_3)} = 1 + \frac{V_3}{V_1 - V_3} - \frac{V_4}{V_1 - V_3}$$

$$= 1 + \frac{V_3}{V_1 - V_3} - \frac{V_4}{V_1 - V_3} \cdot \frac{V_3}{V_3} = 1 + \frac{V_3}{V_1 - V_3} - \frac{V_3}{V_1 - V_3} \cdot \frac{V_4}{V_3}$$

$$= 1 + k - k \cdot \frac{V_4}{V_3}$$

$$= 1 + k - k \left(\frac{p_3}{p_4}\right)^{1/n} \qquad \left| \begin{array}{l} p_3 V_3^{\,n} = p_4 V_4^{\,n} \\ \dfrac{V_4}{V_3} = \left(\dfrac{p_3}{p_4}\right)^{1/n} \end{array} \right|$$

or $$\eta_{vol.} = 1 + k - k \left(\frac{p_2}{p_1}\right)^{1/n} \qquad (\because \quad p_3 = p_2, p_4 = p_1) \qquad ...(11)$$

or $$\eta_{vol.} = 1 + k - k \left(\frac{V_1}{V_2}\right) \qquad ...(12)$$

The above equations are valid if the index of expansion and compression is same. However, it may be noted that the clearance volumetric efficiency is dependent only on the index of expansion of the clearance volume from V3 to V4.

if the index of compression = n_c and index of expansion = n_e, the volumetric efficiency is given by

$$\eta_{vol.} = 1 + k - k\left(\frac{p_3}{p_4}\right)^{1/n_e}$$

$$= 1 + k - k\left(\frac{p_2}{p_1}\right)^{1/n_e}$$

$$= 1 + k - k\left(\frac{V_4}{V_3}\right)$$

In this case volumetric efficiency = $1 + k - k\left(\dfrac{V_1}{V_2}\right)$.

In practice the air that is sucked in during the induction (suction) stroke gets heated up while passing through the hot valves and coming in contact with hot cylinder walls. There is wire drawing effect through the valves resulting in drop in pressure. Thus the ambient conditions are different from conditions obtained at state 1.

Let $p_{amb.}$ = Pressure of ambient air, and

$T_{amb.}$ = Temperature of ambient air

$$\therefore \quad \frac{p_{amb.} V_{amb.}}{T_{amb.}} = \frac{p_1 (V_1 - V_4)}{T_1}$$

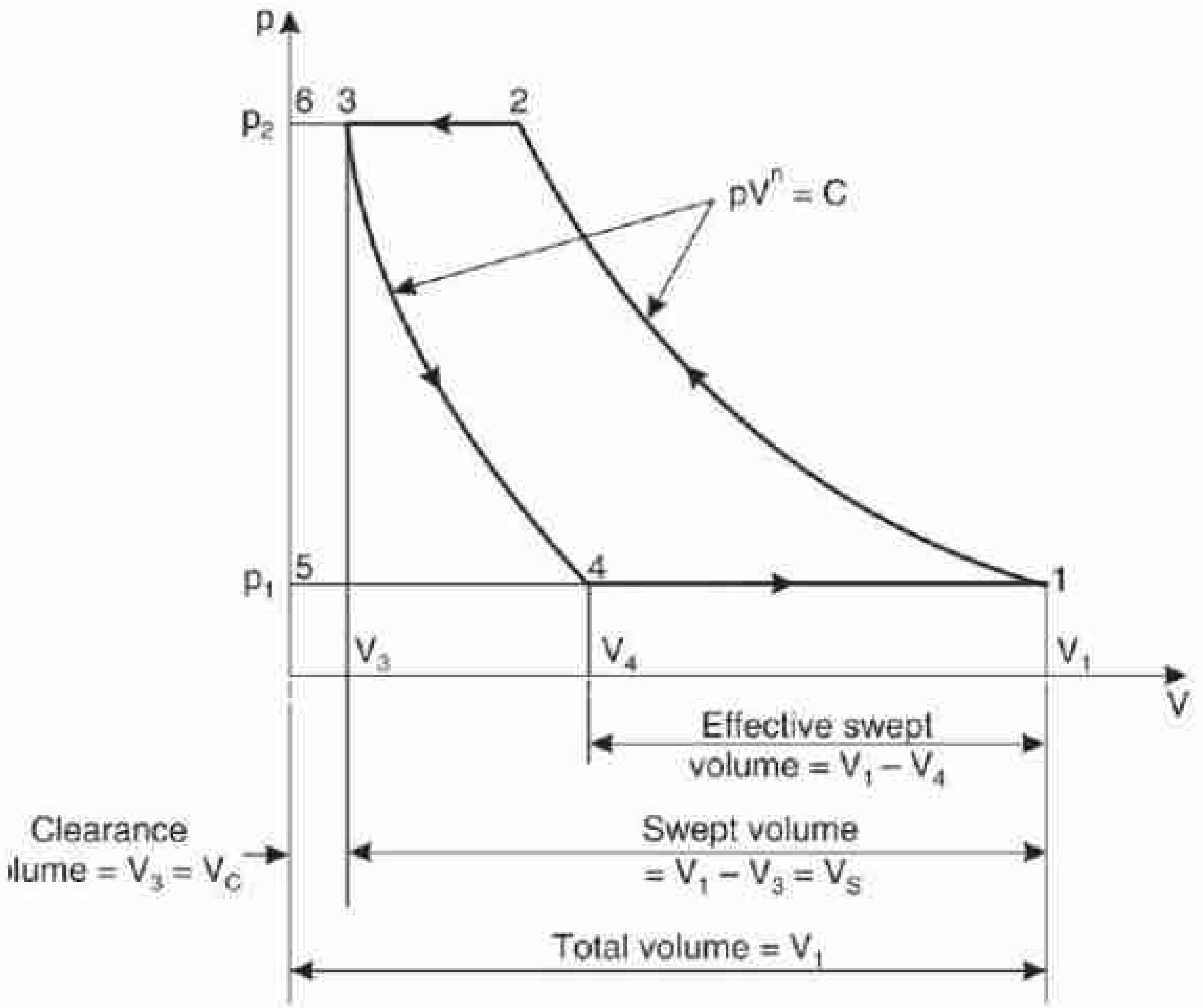
Thus,
$$V_{amb.} = \frac{p_1 \times T_{amb.}}{T_1 \times p_{amb.}} \times (V_1 - V_4)$$

Thus volumetric efficiency referred to ambient conditions may be written as

$$\eta_{vol.\ (amb.)} = \frac{V_{amb.}}{V_1 - V_3} = \frac{p_1 \times T_{amb.}}{T_1 \times p_{amb.}} \times \frac{V_1 - V_4}{V_1 - V_3}$$

But from eqn. (11)

$$\frac{V_1 - V_4}{V_1 - V_3} = 1 + k - k \left(\frac{p_2}{p_1}\right)^{1/n}$$



$$\therefore \quad \boxed{\eta_{vol.\ (amb.)} = \frac{p_1 \times T_{amb.}}{T_1 \times p_{amb.}} \left[1 + k - k\left(\frac{p_2}{p_1}\right)^{1/n}\right]} \qquad \ldots(16)$$

$$\boxed{\eta_{vol.\ (amb.)} = \frac{p_1 \times T_{amb.}}{T_1 \times p_{amb.}} \left[1 + k - k\left(\frac{V_2}{V_1}\right)\right]} \qquad \ldots(17)$$

<u>Note:-</u> This efficiency should not be used for finding out the dimensions of the cylinder. For finding out the dimensions of the cylinder, the volumetric efficiency based on suction condition only should be used.

Fig. shows the manner in which the volumetric efficiency varies with delivery pressure. Theoretically, the volumetric efficiency is 100% when the delivery pressure equals that of the surroundings, and in fact no compression takes place at all. It decreases rapidly with increase in delivery pressure at first, and then more slowly for increase in delivery pressure at higher pressure.

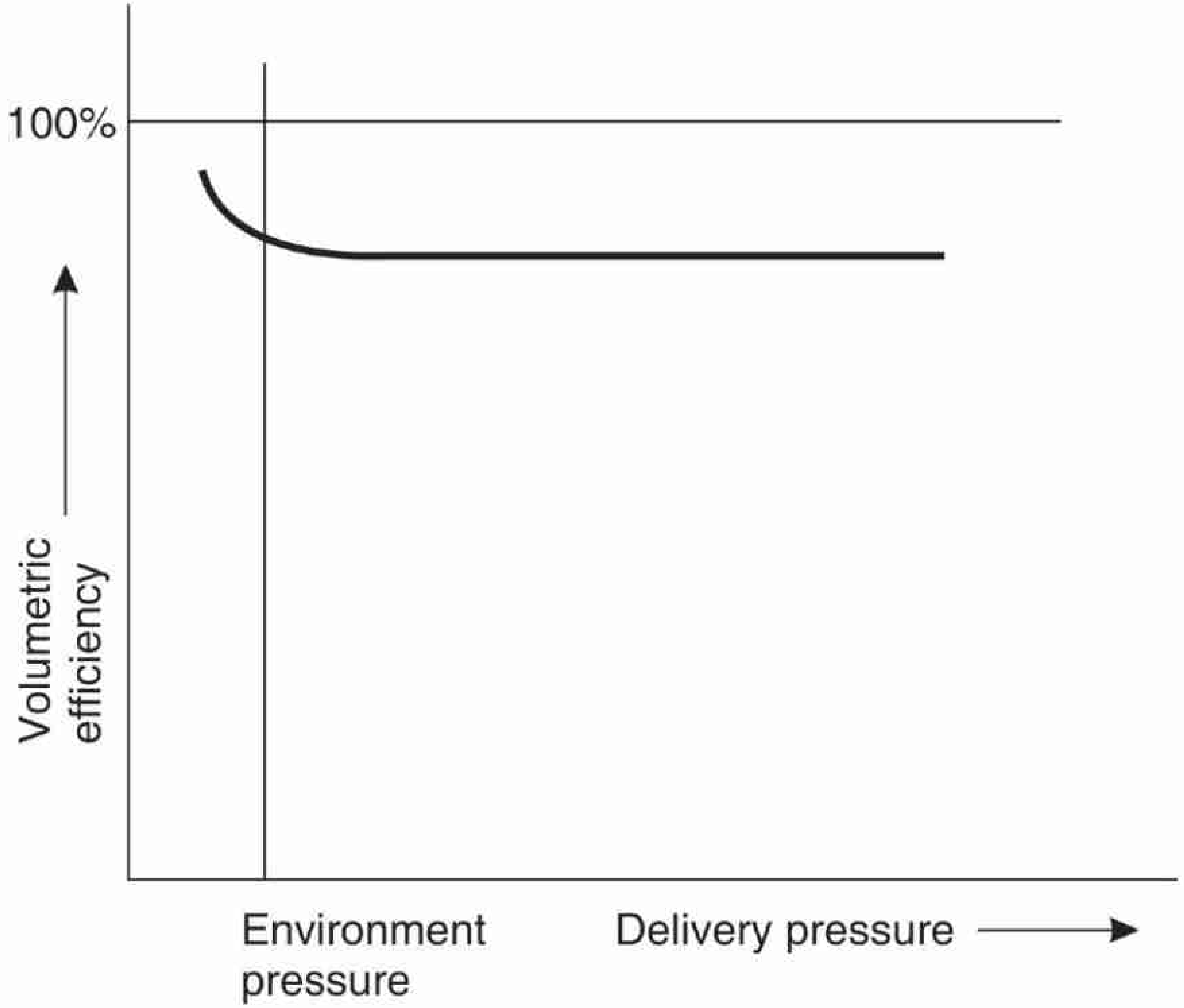


Fig. Variation of volumetric efficiency with delivery pressure.

Free Air Delivered (FAD:)

The free air condition refers to the standard conditions. Free air condition may be taken as **1 atm** or **1.01325 bar** and **15°C** or **288 K**. Consideration for free air is necessary as otherwise the different compressors can not be compared using volumetric efficiency because specific volume or density of air varies with altitude. It may be seen that a compressor at datum level (sea level) shall deliver large mass than the same compressor at high altitude. This concept is used for giving the capacity of compressor in terms of '**free air delivery' (FAD)**'. "*Free air delivery is the volume of air delivered being reduced to free air conditions*." In case of air the free air delivery can be obtained using perfect gas equation as,

$$\frac{p_a \cdot V_a}{T_a} = \frac{p_1(V_1 - V_4)}{T_1} = \frac{p_2(V_2 - V_3)}{T_2}$$

where subscript a or p_a, V_a, T_a denote properties at free air conditions

or, $$V_a = \frac{p_1 \cdot T_a \cdot (V_1 - V_4)}{p_a \cdot T_1} = \text{FAD per cycle.}$$

This volume V_a gives 'free air delivered' per cycle by the compressor.

The *volumetric efficiency is lowered* by any of the following conditions :

(*i*) Very high speed

(*ii*) Leakage past the piston

(*iii*) Too large a clearance volume

(*iv*) Obstruction at inlet valves

(*v*) Overheating of air by contact with hot cylinder walls.

(*vi*) Inertia effect of air in suction pipe.

By paying careful attention in the design of the compressor to these causes of loss, an improvement in volumetric efficiency can be obtained.

ACTUAL INDICATOR DIAGRAM

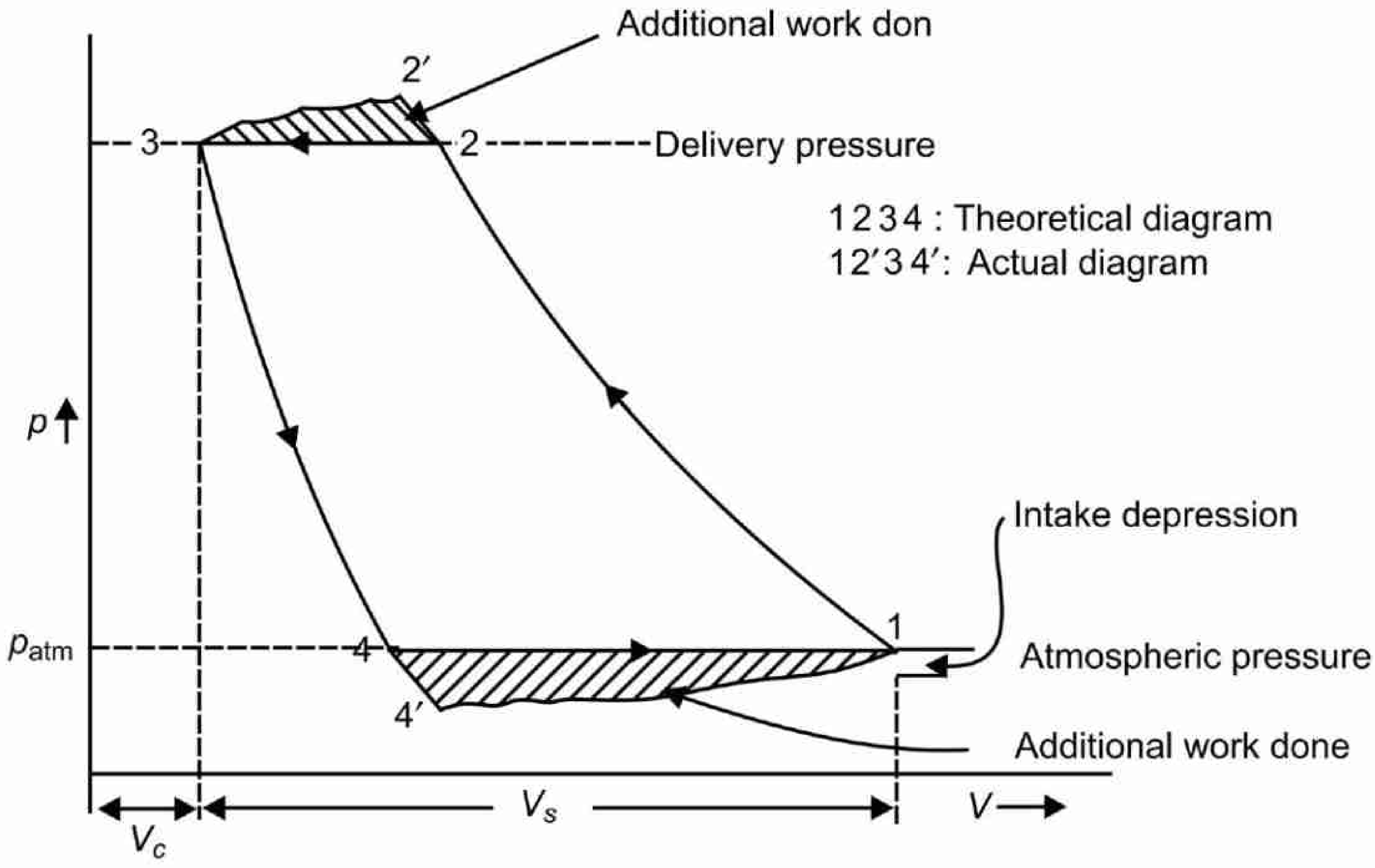


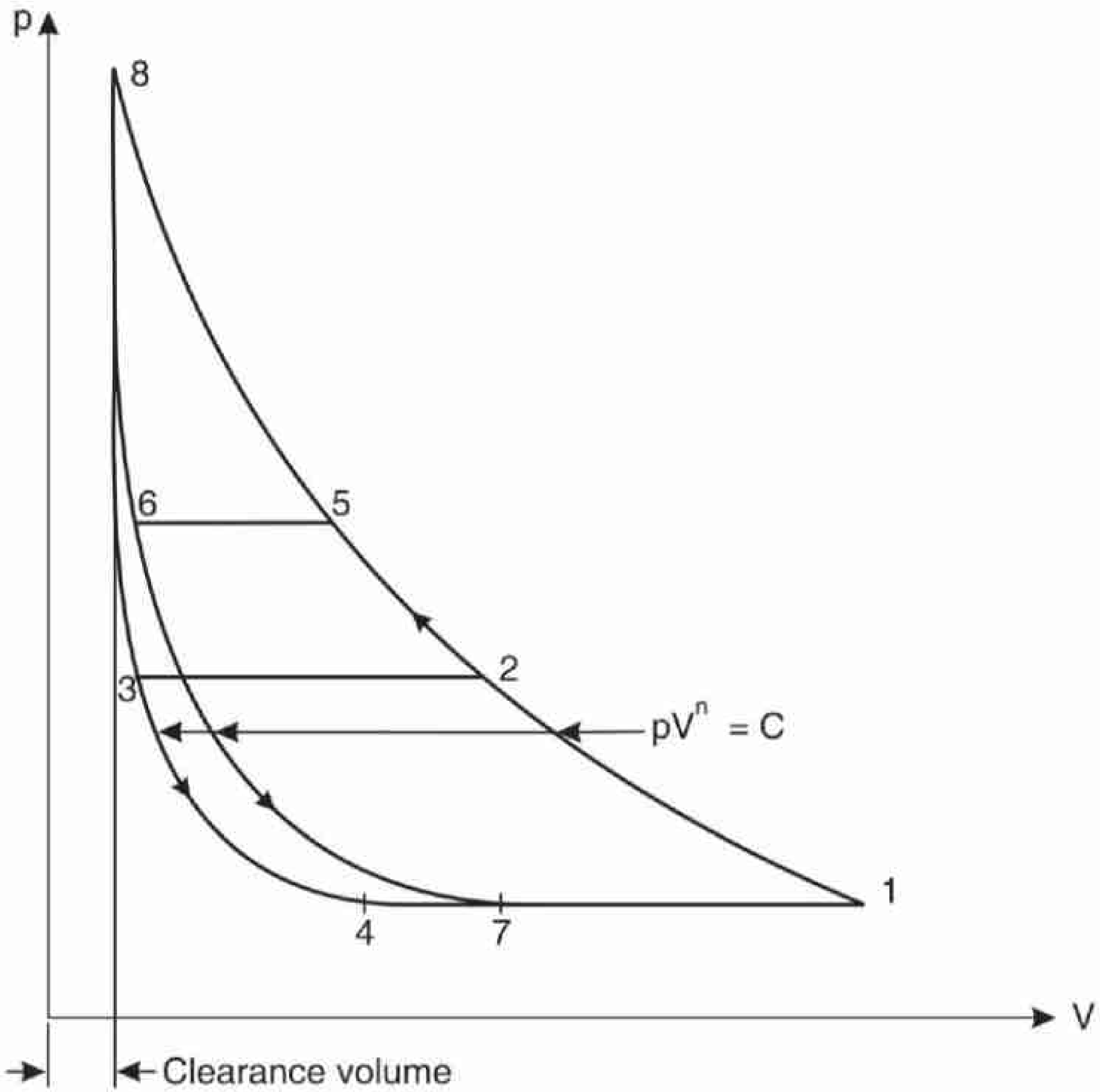
Fig. *Actual indicator diagram*

Theoretical indicator diagram of reciprocating compressor as shown in earlier discussion refers to the ideal state of operation of compressor. The practical limitations, when considered in the indicator diagram yield actual indicator diagram as shown in Fig.

Multi-stage Compression

Multi-stage compression is very efficient and is now-a-days almost universally adopted except for compressors where the overall pressure rise required is small. The method is not only advantageous from a thermodynamic point of view, but also has mechanical advantages over single-stage compression.

It is seen, therefore, that as the delivery pressure for a single-stage, reciprocating compressor is increased so the mass flow through the compressor decreases. Note, also that as the delivery pressure is increased, so also will the delivery temperature increase. Referring to Fig., T8 > T5 > T2. If high temperature air is not a requirement of the compressed air delivered, then, any increase in temperature represents an energy loss.



If high pressure is to be delivered by a single-stage machine, then it will require heavy working parts in order to accommodate the high-pressure ratio through the machine. This will increase the balancing problem and the high torque fluctuation will require a heavier flywheel installation. Such disadvantages can be overcome by multi-stage compression.

Advantages :

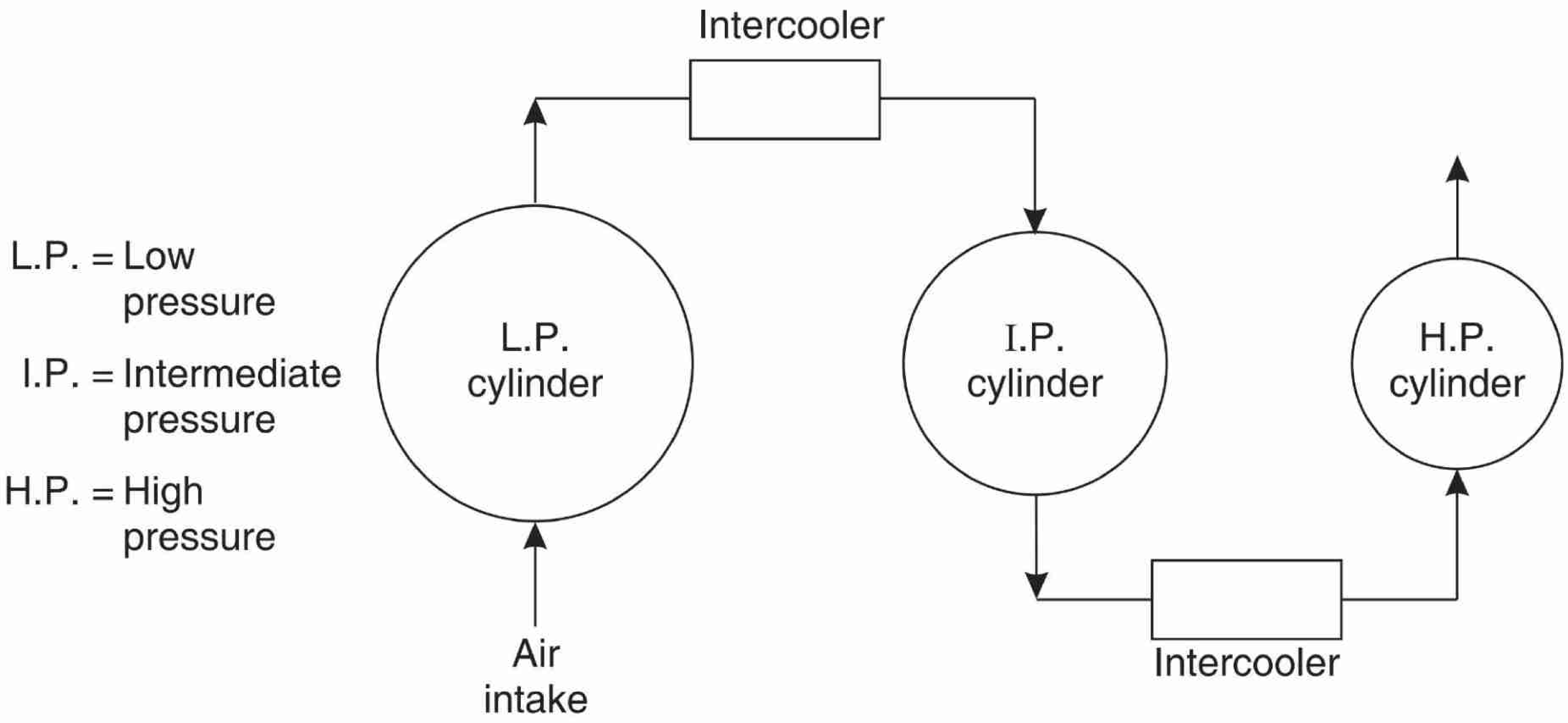
The important advantages of multi-stage compression can be summed up as follows :

1. The air can be cooled at pressures intermediate between intake and delivery pressures.
2. The power required to drive a multi-stage machine is less than would be required by a single-stage machine delivering the same quantity of air at the same delivery pressure.
3. Multi-stage machines have better mechanical balance.
4. The pressure range (and hence also the temperature range) may be kept within desirable limits. This results in (i) reduced losses due to air leakage (ii) improved lubrication, due to lower temperatures and (iii) improved volumetric efficiency.

5. The cylinder, in a single-stage machine, must be robust enough to withstand the delivery pressure. The down pressure cylinders of a multi-stage machine may be lighter in construction since the maximum pressure therein is low.

Disadvantages :

Despite all these advantages, a multi-stage compressor with intercoolers is likely to be more expensive in initial cost than a single-stage compressor of the same capacity.



3-Stage compressor

Intercooler:

The cooler which is placed in between stages is called Intercooler. With the object of removing moisture, coolers are sometimes fitted after the last stage, and for this reason are called '***Aftercoolers***', but it should be understood that aftercoolers cannot influence the work done in compression.

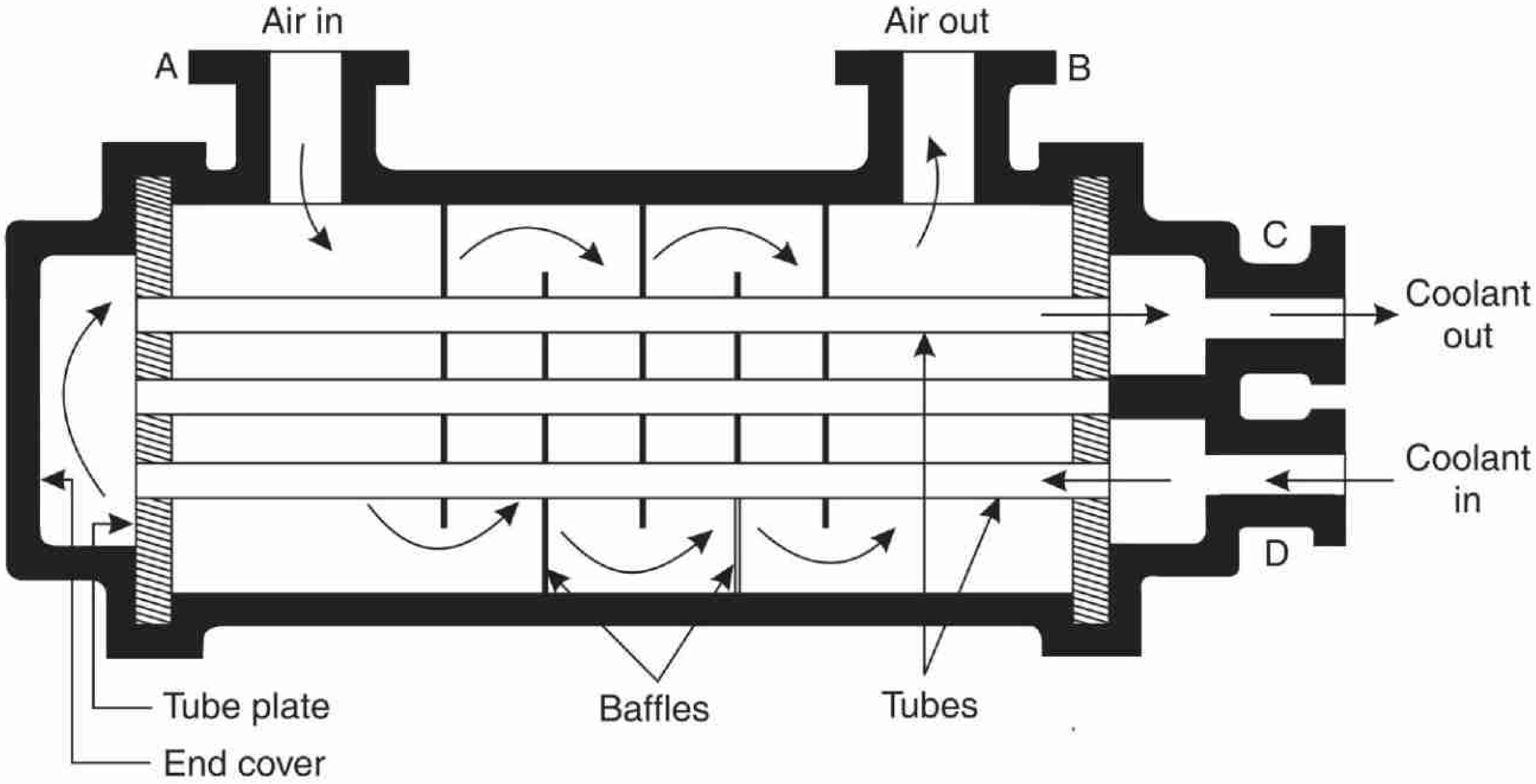
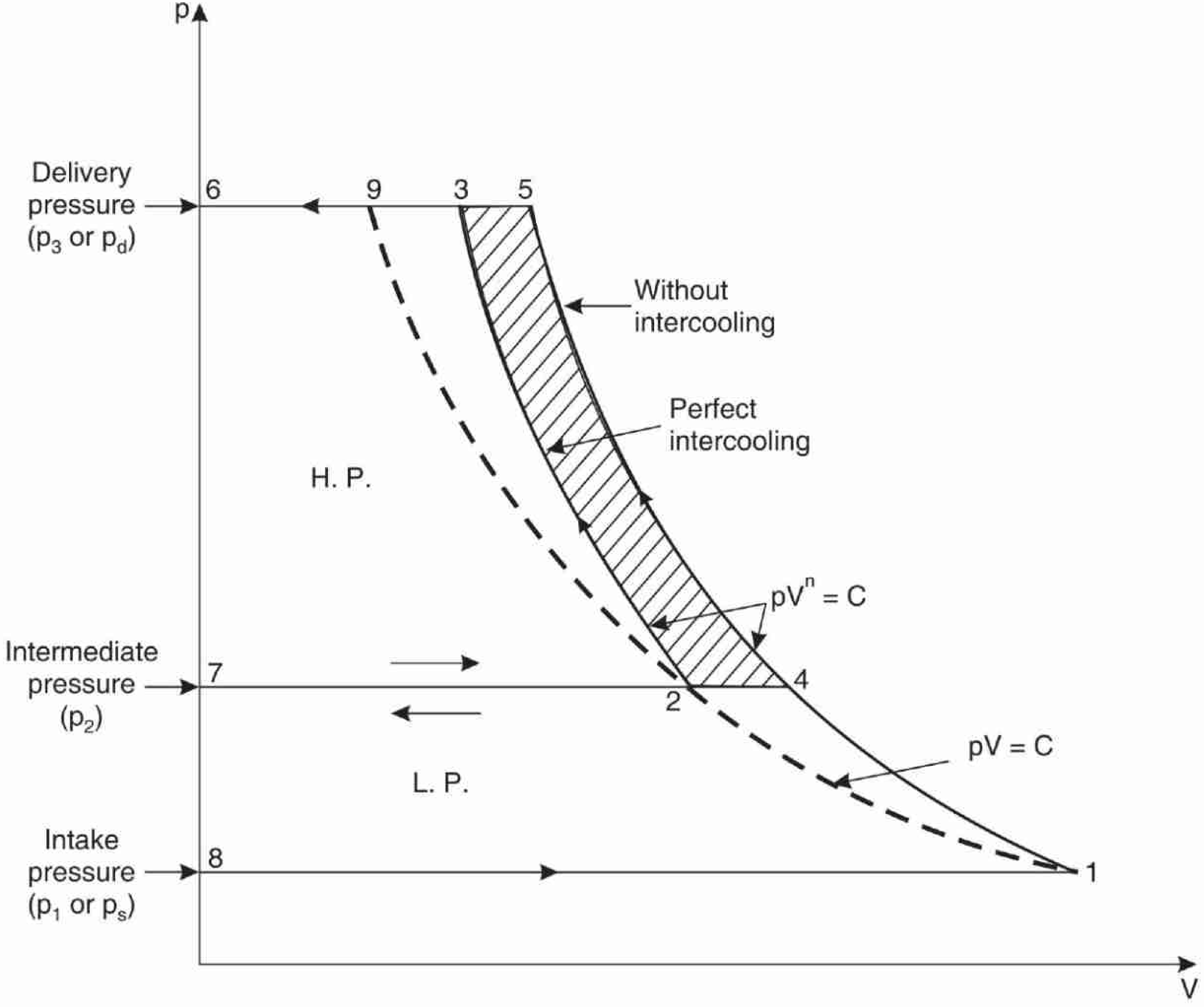


Fig. Intercooler.


p
V
Delivery pressure (p_3 or p_d)
Intermediate pressure (p_2)
Intake pressure (p_1 or p_s)
6
9
3
5
7
2
4
8
1
Without intercooling
Perfect intercooling
H. P.
L. P.
$pV^n = C$
$pV = C$

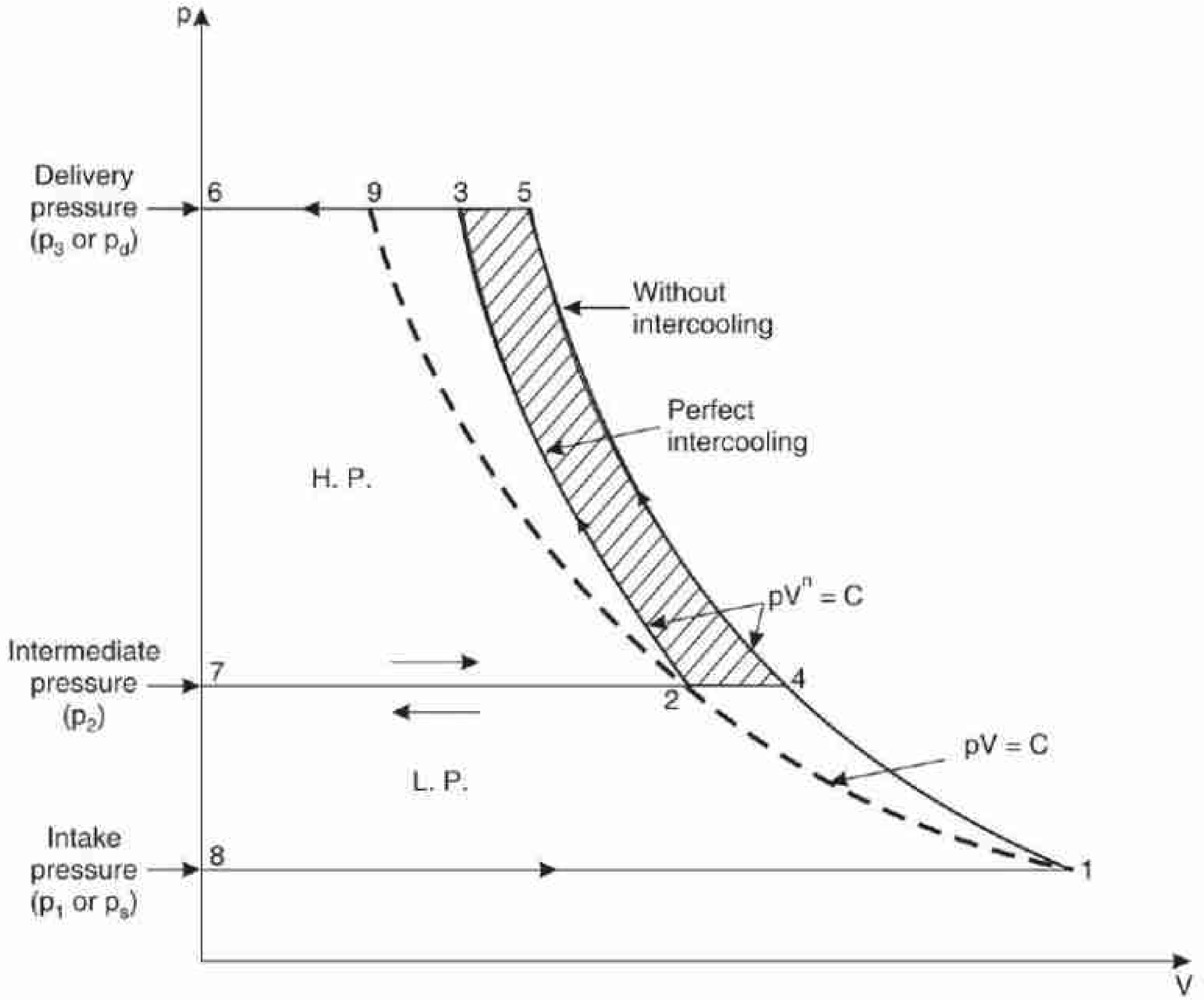
Ideal conditions for multi-stage compressors :

Case 1. Single-stage compressor :

As earlier stated cycle 8156 is that of a single-stage compressor, neglecting clearance. For this cycle,

$$W = \frac{n}{n-1}\, p_1 V_1 \left[\left(\frac{p_5}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

Delivery temperature, $\quad T_5 = T_1 \left(\frac{p_5}{p_1} \right)^{\frac{n-1}{n}}$



Case 2. Two-stage compressor :

(*i*) ***Without intercooling***

8147 ...... Low pressure cycle

7456 ...... High pressure cycle.

For this arrangement work done,

$$W = \frac{n}{n-1}\ p_1V_1\left[\left(\frac{p_4}{p_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1}\ \cdot p_4V_4\left[\left(\frac{p_5}{p_4}\right)^{\frac{n-1}{n}} - 1\right] \qquad ...(20)$$

This will give the same result as that of eqn. (18). The final delivery temperature will also be given by eqn. (19), because there is no intercooling.

(*ii*) ***With perfect intercooling***

8147 ...... Low pressure cycle

7236 ...... High pressure cycle.

For this arrangement,

$$W = \frac{n}{n-1}\ p_1V_1\left[\left(\frac{p_4}{p_1}\right)^{\frac{n-1}{n}} - 1\right] + \frac{n}{n-1}\ p_2V_2\left[\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 1\right] \qquad ...(21)$$

Delivery temperature is given by

$$T_3 = T_2\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} = T_1\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}},\ \text{Since } T_2 = T_1 \qquad ...(22)$$

Now, since $T_2 = T_1$, then

$$p_2V_2 = p_1V_1 \quad ...(23)$$

Also $$p_4 = p_2 \quad ...(24)$$

Inserting eqns. (23) and (24) in eqn. (21)

$$W = \frac{n}{n-1} p_1V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}} - 2\right] \quad ...(25)$$

Now, inspection of Fig. 9 *shows the shaded area 2453 which is the work saving which occurs as a result of using an intercooler.*

Conditions for minimum work

It will be observed from the Fig. 9 that as intermediate pressure $p_2 \to p_1$, then area 2453 $\to$ 0. Also as $p_2 \to p_3$, then area 2453 $\to$ 0. This means, therefore, that an intermediate pressure p_2 exists which makes area 2453 a maximum. This is the condition when W is a minimum.

Inspection of eqn. 25 shows that for minimum W, $[(p_2/p_1)^{n-1/n} + (p_3/p_2)^{n-1/n}]$ must be minimum, all other parts of the equation being constant in this consideration and p_2 is the variable.

Hence, for *minimum*, W, $\dfrac{dW}{dp_2} = \dfrac{d\left[\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}} + \left(\dfrac{p_3}{p_2}\right)^{\frac{n-1}{n}}\right]}{dp_2} = 0$

pressure ratio per stage is equal.

$$\therefore \qquad p_2^{\,2} = p_1 p_3$$

$$p_2 = \sqrt{p_1 p_3}$$

$$\frac{p_2}{p_1} = \frac{p_3}{p_2}$$

With these ideal conditions, inserting equations (23), (24) and (28) into eqn. (21) shows that *there is equal work per cylinder.*

$$\therefore \qquad W = \frac{2n}{n-1}\, p_1 V_1 \left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1\right] \qquad \text{...(30)}$$

Note that p_3 / p_1 is the pressure ratio through the compressors.

Case 3. Multi-stage compressor

From the analysis of compressor so far, for a *single-stage* compressor,

$$W = \frac{n}{n-1} p_1V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

For a *two-stage* compressor,

$$W = \frac{2n}{n-1} p_1V_1 \left[\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}} - 1\right]$$

It seems reasonable to assume, therefore, that for a *three-stage* machine,

$$W = \frac{3n}{n-1} p_1V_1 \left[\left(\frac{p_4}{p_1}\right)^{\frac{n-1}{3n}} - 1\right]$$

and for *x-stage* compressor,

$$W = \frac{xn}{n-1} p_1V_1 \left[\left(\frac{p_{(x+1)}}{p_1}\right)^{\frac{n-1}{xn}} - 1\right] \quad ...(31)$$

This equation is very important, since it *applies to any type of compressor or motor, and even to vapour engines, provided* $n = \text{or} < \gamma$.

Note that $\dfrac{p_{(x+1)}}{p_1}$ is the pressure ratio through the compressor, in each case.

Note, also, that since for an ideal compressor there is equal work per cylinder, for an *x*-stage compressor

$$W = \frac{xn}{n-1}\, p_1 V_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \qquad \text{...(32)}$$

To determine the intermediate pressures for an *x-stage* machine running under ideal conditions, use is made of eqn. (28).

Here it is shown that the pressure ratio per stage is equal.

Hence, for an *x-stage* machine,

$$\frac{p_2}{p_1} = \frac{p_3}{p_2} = \frac{p_4}{p_3} \; \ldots\ldots \; \frac{p_{(x+1)}}{p_x} = Z, \; say$$

$$\therefore \qquad Z = \sqrt[x]{\frac{p_{(x+1)}}{p_1}} = x\ \sqrt{(\text{Pressure ratio through compressor})} \qquad \text{...(34)}$$

Inserting the value of Z in eqn. (33) will determine the intermediate pressures.

In the event of *intercooling being imperfect we must treat each stage as a separate compressor*, in which case 'x' in eqn. (31) will be unity. With this special value of 'x' the power per stage can be calculated, and finally the total power is the sum of the powers per stage :

$$W = \frac{n_1}{n_1 - 1} \, p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n_1 - 1}{n_1}} - 1 \right] + \frac{n_2}{n_2 - 1} \, p_2 V_2 \left[\left(\frac{p_3}{p_2} \right)^{\frac{n_2 - 1}{n_2}} - 1 \right] + \ldots\ldots$$

Heat rejection per stage per kg of air :

If the air is cooled to its initial temperature the whole of the work done in compression must be rejected to the cooling medium.

$$W = \frac{n}{n-1} \, (c_p - c_v)(T_2 - T_1) \qquad \ldots(36)$$

$\therefore$ ***Heat rejected with perfect intercooling***

$$= \left[c_p + c_v\left(\frac{\gamma - n}{n - 1}\right)\right] (T_2 - T_1) \text{ per kg of air} \qquad \ldots(37)$$

$$\left[\textbf{Note. } \frac{n}{n-1}(c_p - c_v) = c_p + \frac{c_v(\gamma - n)}{n-1}\right]$$

The first term in eqn. (37) represents the *heat rejected at constant pressure in the intercooler* ; whilst the second term represents the *heat rejected during compression alone* ; and writing $c_v = R/J\,(\gamma - 1)$ it may be reduced to the form

$$\frac{\gamma - n}{\gamma - 1} \times \text{work done in heat units}$$

To find the change in entropy (s) during the first stage of compression, we have, from the definition of entropy,

$$ds = \frac{dW}{T} = \frac{dQ}{T} \qquad (\because \quad \text{Work done} = \text{Heat rejected})$$

$$(s_2 - s_1) = \frac{n}{n-1}\,(c_p - c_v)\log_e\left(\frac{p_3}{p_1}\right)^{\frac{n-1}{2n}}$$

$$= \left(\frac{c_p - c_v}{2}\right)\log_e p_3/p_1$$

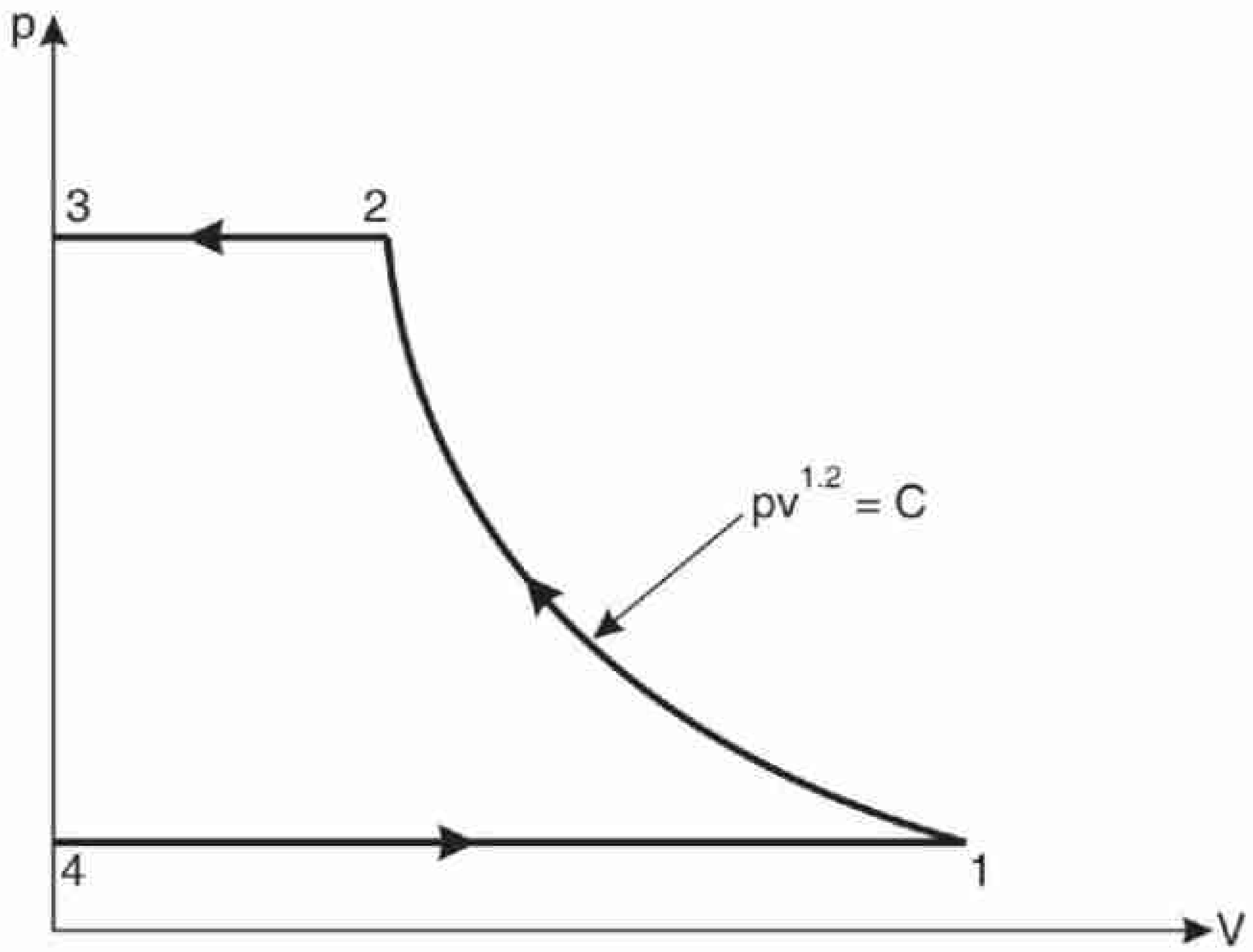
Example 3. *An air compressor takes in air at 1 bar and 20°C and compresses it according to law $pv^{1.2}$ = constant. It is then delivered to a receiver at a constant pressure of 10 bar. $R = 0.287$ kJ/kg K. Determine :*

(*i*) *Temperature at the end of compression ;*

(*ii*) *Work done and heat transferred during compression per kg of air.*

Solution. Refer Fig.

$$T_1 = 20 + 273 = 293 \text{ K} ;\ p_1 = 1 \text{ bar} ;\ p_2 = 10 \text{ bar}$$



Law of compression : $pv^{1.2} = C$; $R = 0.287$ J kJ/kg K

(*i*) **Temperature at the end of compression, T_2 :**

For *compression process 1–2*, we have

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = \left(\frac{10}{1}\right)^{\frac{1.2-1}{1.2}} = 1.468$$

or $$T_2 = T_1 \times 1.468 = 293 \times 1.468 = 430 \text{ K or } \mathbf{157°C.} \ \textbf{(Ans.)}$$

(*ii*) **Work done and heat transferred during compression per kg of air :**

$$\textit{Work done, } W = mRT_1 \, \frac{n}{n-1}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right] \qquad \text{...[Eqn. (5)]}$$

$$= 1 \times 0.287 \times 293 \times \left(\frac{1.2}{1.2-1}\right)\left[\left(\frac{10}{1}\right)^{\frac{1.2-1}{1.2}} - 1\right] = \textbf{236.13 kJ/kg of air. (Ans.)}$$

Heat transferred during compression,

$$Q = W + \Delta U$$

$$= \frac{p_1 v_1 - p_2 v_2}{n-1} + c_v (T_2 - T_1)$$

$$= \frac{R(T_1 - T_2)}{n-1} + c_v (T_2 - T_1) = (T_2 - T_1)\left[c_v - \frac{R}{n-1}\right]$$

$$= (430 - 293)\left[0.718 - \frac{0.287}{1.2-1}\right] = \textbf{-- 98.23 kJ/kg. (Ans.)}$$

Example 6. *A single-stage double-acting air compressor is required to deliver 14 m^3 of air per minute measured at 1.013 bar and 15°C. The delivery pressure is 7 bar and the speed 300 r.p.m. Take the clearance volume as 5% of the swept volume with the compression and expansion index of n = 1.3. Calculate :*

(i) Swept volume of the cylinder ; *(ii) The delivery temperature ;*

(iii) Indicated power.

Solution. Quantity of air to be delivered = 14 m^3/min

Intake pressure and temperature $p_1 = 1.013$ bar,

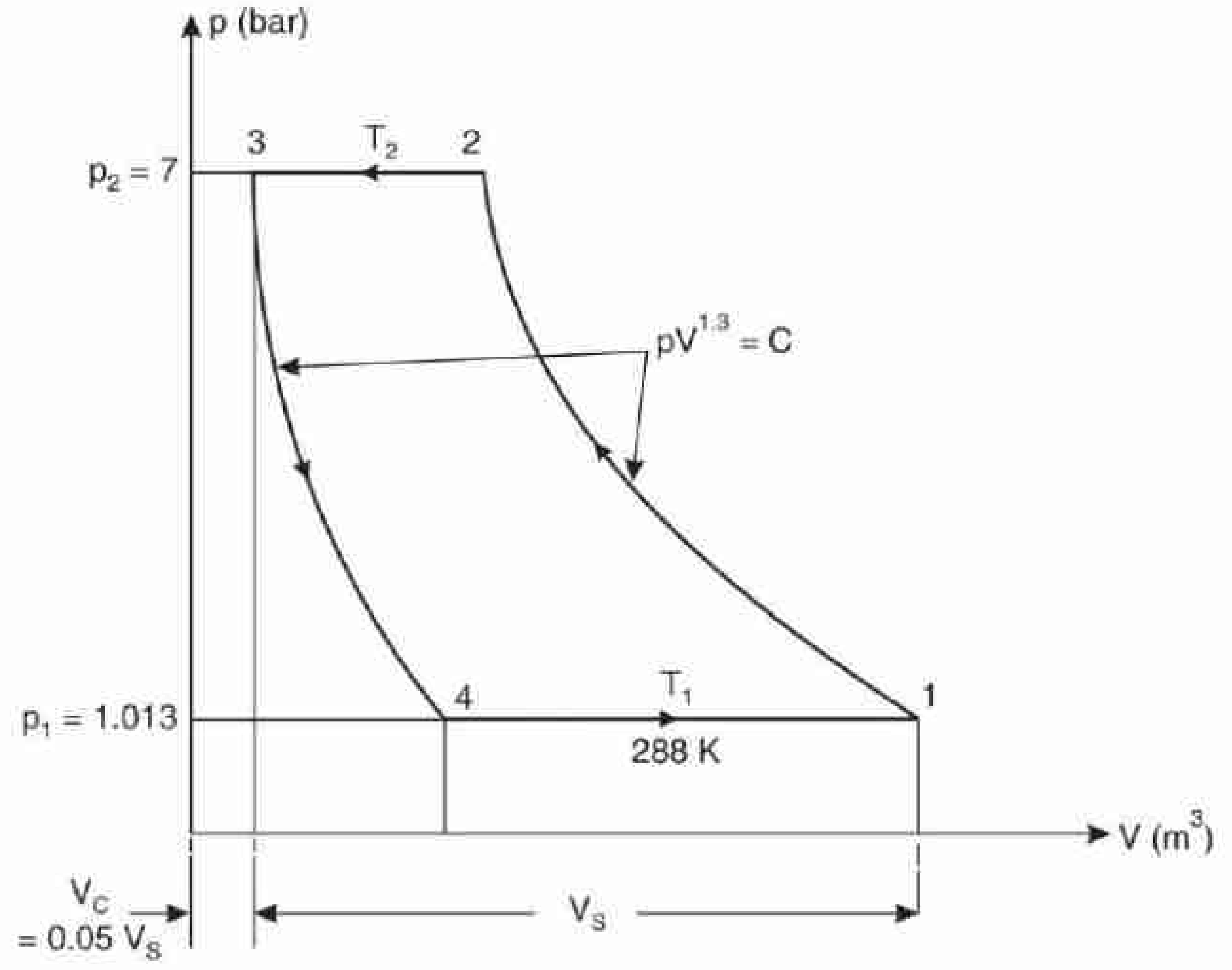
$T_1 = 15 + 273 = 288$ K

Delivery pressure, $p_2 = 7$ bar

Compressor speed, $N = 300$ r.p.m.

Clearance volume, $V_c = 0.05\ V_s$

Compression and expansion index, $n = 1.3$



(*i*) **Swept volume of the cylinder, V_s :**

Swept volume, $V_s = V_1 - V_3 = V_1 - V_c = V_1 - 0.05\ V_s$

$\therefore$ $V_1 = 1.05\ V_s$

Volume induced per cycle $= (V_1 - V_4)$

$$V_1 - V_4 = \frac{14}{300 \times 2} = 0.0233\ \text{m}^3$$

Now, $$V_1 = 1.05\ V_s \text{ and } \frac{V_4}{V_3} = \left(\frac{p_2}{p_1}\right)^{1/n} = \left(\frac{7}{1.013}\right)^{1/1.3} = 4.423$$

i.e., $$V_4 = 4.423\ V_3 = 4.423 \times 0.05\ V_s = 0.221\ V_s$$

$$\therefore \quad (V_1 - V_4) = 1.05\ V_s - 0.221\ V_s = 0.0233$$

$$\therefore \quad V_s = \frac{0.0233}{(1.05 - 0.221)} = 0.0281\ \text{m}^3$$

i.e., *Swept volume of the cylinder* $=$ **0.0281 m³. (Ans.)**

(*ii*) **The delivery temperature, T_2 :**

Using the relation, $$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$$

$$\therefore \qquad T_2 = T_1 \times \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = 288 \times \left(\frac{7}{1.013}\right)^{\frac{1.3-1}{1.3}} = 450 \text{ K}$$

$\therefore$ *Delivery temperature* $= 450 - 273 =$ **177°C. (Ans.)**

(*iii*) **Indicated power :**

Indicated power $$= \frac{n}{n-1}\, p_1(V_1 - V_4) \left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right\}$$

$$= \frac{1.3-1}{1.3} \times \frac{1.013 \times 10^5 \times 14}{10^3 \times 60} \left\{\left(\frac{7}{1.013}\right)^{\frac{1.3-1}{1.3}} - 1\right\} \text{ kW}$$

$= 57.56$ kW.

i.e., *Indicated power* = **57.56 kW. (Ans.)**

Example 16. *Air at 103 kPa and 27°C is drawn in L.P. cylinder of a two-stage air compressor and is isentropically compressed to 700 kPa. The air is then cooled at constant pressure to 37°C in an intercooler and is then again compressed isentropically to 4 MPa in the H.P. cylinder, and is delivered at this pressure. Determine the power required to run the compressor if it has to deliver 30 m^3 of air per hour measured at inlet conditions.* **(M.U.)**

Pressure of intake air (L.P. cylinder), $p_1 = 103$ kPa

Temperature of intake air, $T_1 = 27 + 273 = 300$ K

Pressure of air entering H.P. cylinder, $p_2 = 700$ kPa

Temperature of air entering H.P. cylinder, $T_2 = 37 + 273 = 310$ K

Pressure of air after compression in H.P. cylinder, $p_3 = 4$ MPa or 4000 kPa

Volume of air delivered $= 30\ m^3/h$

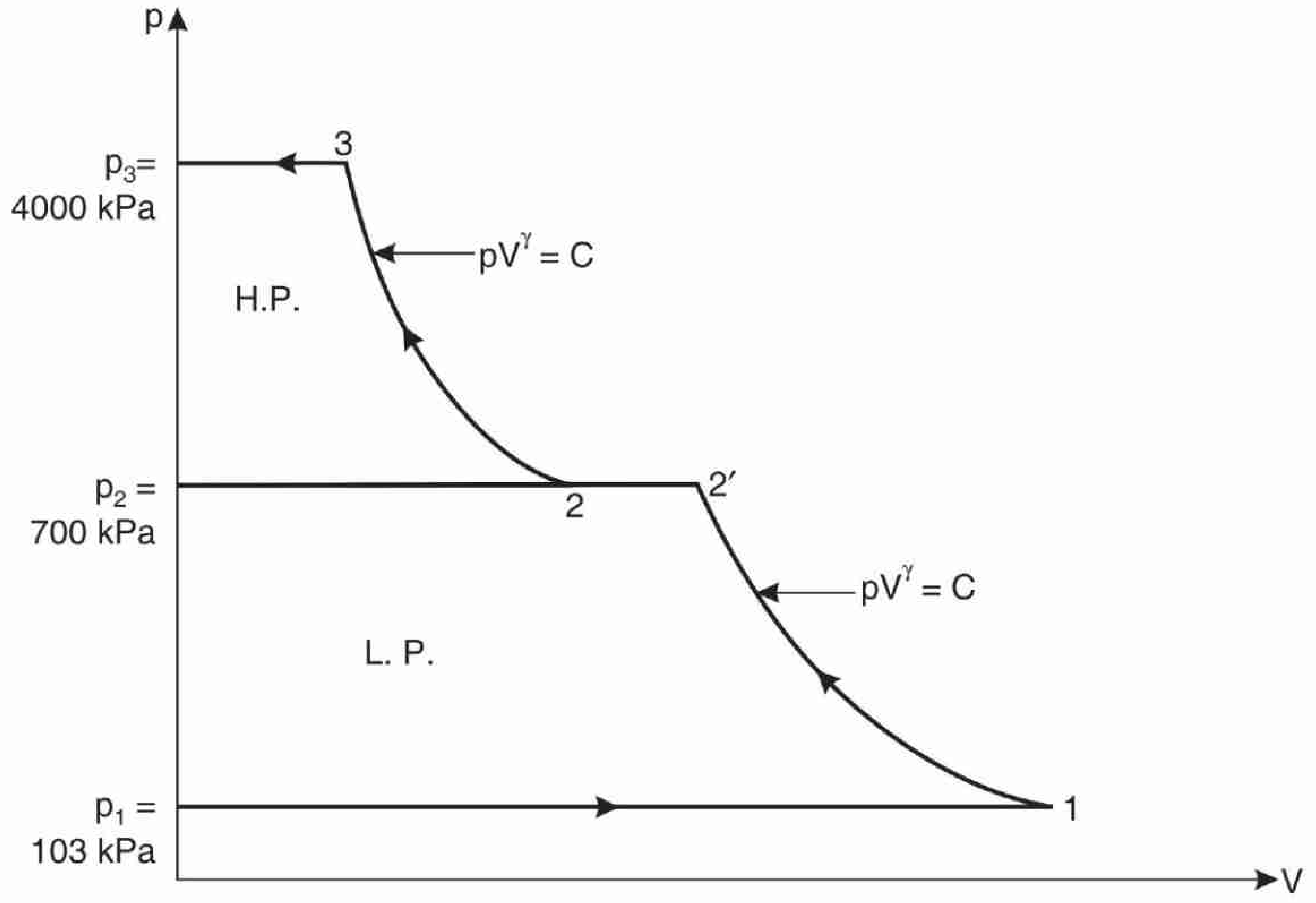

p
3
p_3 =
4000 kPa
$pV^\gamma = C$
H.P.
p_2 =
700 kPa
2
2′
$pV^\gamma = C$
L. P.
p_1 =
103 kPa
1
V

Fig. 26

Power required to run the compressor, P :

Mass of air compressed, $m = \dfrac{(103 \times 10^3) \times 30}{(0.287 \times 1000) \times 300} = 35.89$ kg/h

For the *compression process 1-2′*, we have

$$\frac{T_2'}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{700}{103}\right)^{\frac{1.4-1}{1.4}} = 1.7289 \text{ or } T_2' = 300 \times 1.7289 = 518.7 \text{ K}$$

Similarly for the *compression process 2–3*, we have

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{4000}{700}\right)^{\frac{1.4-1}{1.4}} = 1.6454 \text{ or } T_3 = 310 \times 1.645 = 510.1 \text{ K}$$

∴ Work required to run the compressor,

$$W = \frac{\gamma}{\gamma - 1}\,[mR\,(T_2' - T_1) + mR\,(T_3 - T_2)]$$

$$= \frac{\gamma}{\gamma - 1} \times mR\,[(T_2' - T_1) + (T_3 - T_2)]$$

$$= \frac{1.4}{1.4 - 1} \times \frac{35.89}{3600} \times 0.287\,[(518.7 - 300) + (510.1 - 310)] = 4.194 \text{ kN m/s}$$

Hence power required to run the compressor = **4.194 kW. (Ans.)**

ROTARY COMPRESSORS

Rotary compressors are those compressors in which rotating action is used for compression of fluid. Rotary air compressors have capability of running at high speeds up to 40,000 rpm and can be directly coupled to any prime mover such as electric motor, turbine etc. due to compact design, no balancing problem and less no. of sliding parts. Comparative study of rotary compressor with reciprocating compressor shows that rotary compressors can be used for delivering large quantity of air but the maximum pressure at delivery is less compared to reciprocating compressors. Generally, rotary compressors can yield delivery pressure up to 10 bar and free air delivery of 3000 m3/min. Rotary compressors are less bulky, and offer uniform discharge compared to reciprocating compressor even in the absence of big size receiver. Lubrication requirement and wear and tear is less due to rotary motion of parts in rotary compressors compared to reciprocating compressors.

Rotary compressors may work on the principle of positive displacement and dynamic action both. Rotary compressors having positive displacement may be of following types:

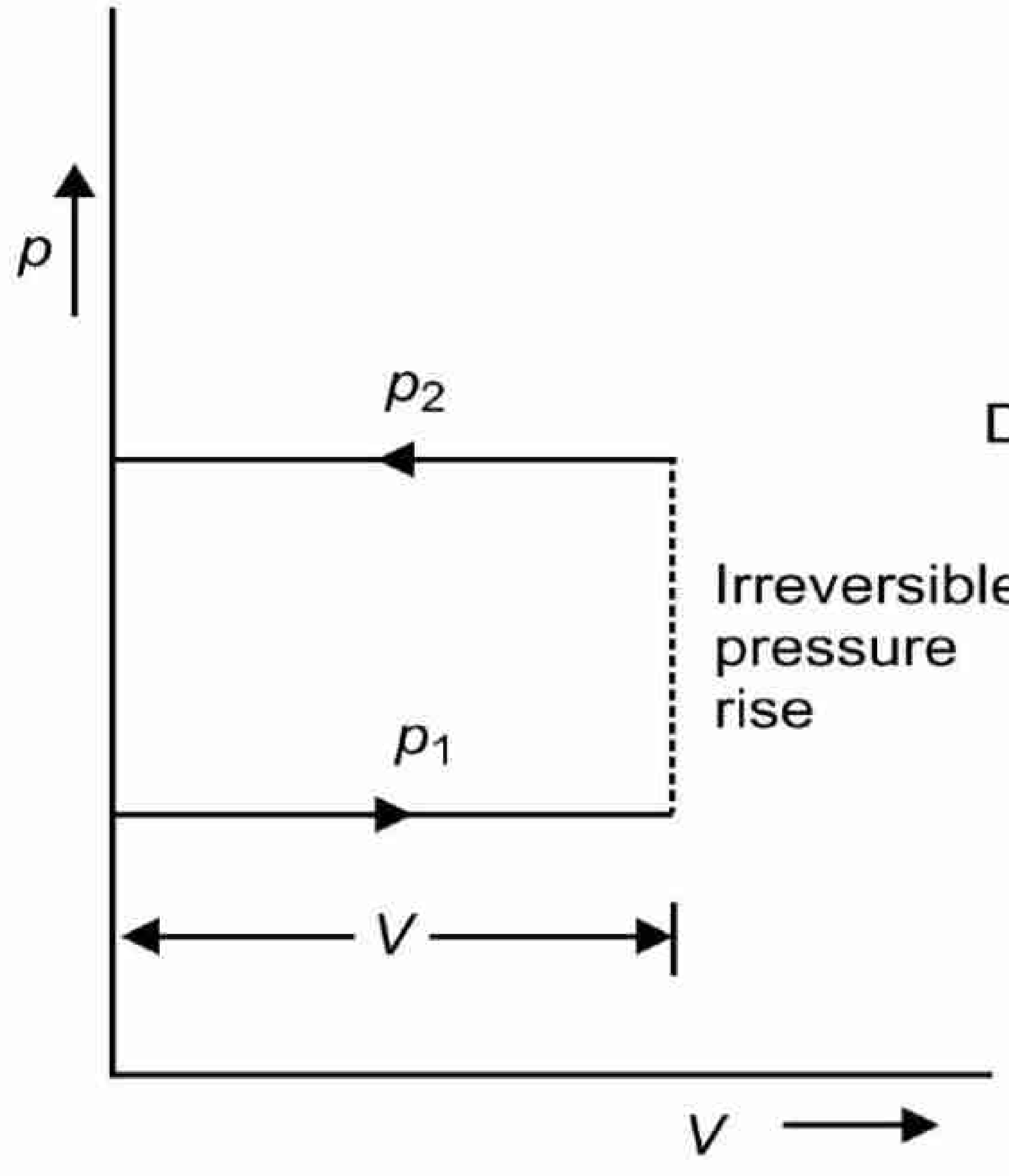
(i) Roots blower

(ii) Screw type or Helical type compressor

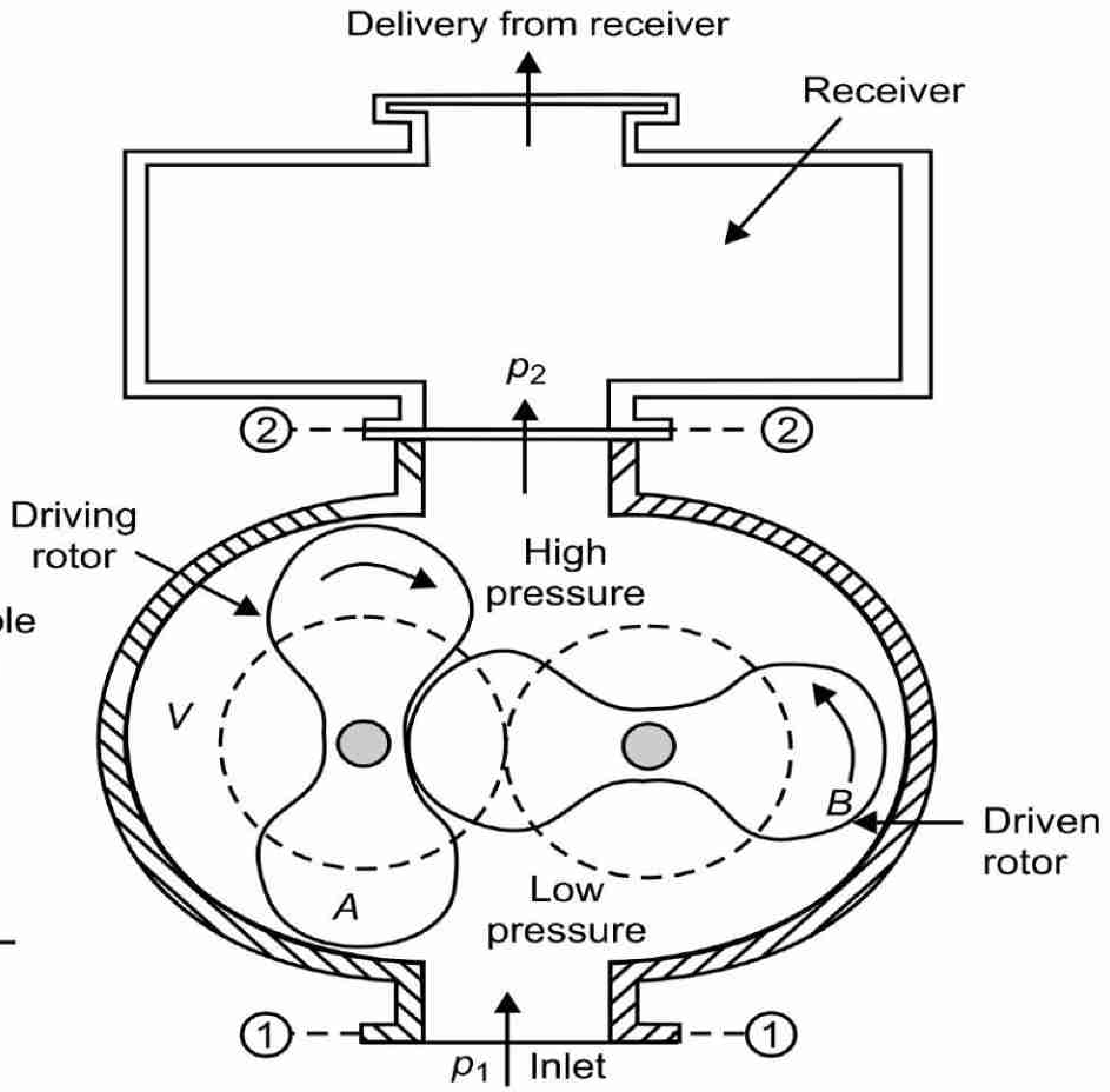
(iii) Vane type compressor

Rotary compressors employing dynamic action may be of centrifugal type or axial type depending upon the direction of flow. These centrifugal type or axial compressors may also be termed as nonpositive displacement type steady flow compressors.

(i) Roots blower



(a) p-V representation



(b) Schematic of roots blower

Fig. *Roots blower*

Roots blower is a positive displacement type rotary compressor. It has two rotors having two or three lobes having epicycloid and hypocycloid or involute profiles such that they remain in proper contact.

Applications include oil refineries, gas pipelines, chemical plants, natural gas processing plants, air conditioning and refrigeration plants. One specialty application is the blowing of plastic bottles made of polyethylene terephthalate (PET).

For this machine the p-V diagram is shown in Fig.   in which the pressure rise from p_1 to p_2 is shown as an irreversible process at constant volume.

Work done per cycle $= (p_2 - p_1)V$

$\therefore$ Work done per revolution $= 4(p_2 - p_1)V$

If V_s is the volume dealt with per minute at p_1 and T_1 then

Work done/min. $= (p_2 - p_1)V_s$

The ideal compression process from p_1 to p_2 is a reversible adiabatic (*i.e.*, isentropic) process. The work done per minute ideally is given by,

$$\text{Work done/min.} = \frac{\gamma}{\gamma - 1}\, p_1 V_s \left\{\left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}} - 1\right\}$$

Then a comparison may be made on the basis of a roots efficiency,

i.e., Roots efficiency $= \dfrac{\text{Work done isentropically}}{\text{Actual work done}}$

$$= \frac{\dfrac{\gamma}{\gamma - 1} p_1 V_s \left\{ \left(\dfrac{p_2}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right\}}{V_s (p_2 - p_1)}$$

$$= \frac{\dfrac{\gamma}{\gamma - 1} p_1 V_s \left\{ (r)^{\frac{\gamma - 1}{\gamma}} - 1 \right\}}{p_1 V_s (r - 1)} \qquad \left(\text{where } r = \frac{p_2}{p_1} = \text{pressure ratio} \right)$$

Also, $$\frac{\gamma}{\gamma - 1} = \frac{c_p}{R}$$

$\therefore$ Roots efficiency $$= \frac{c_p}{R} \left[\frac{(r)^{\frac{\gamma - 1}{\gamma}} - 1}{(r - 1)} \right]$$

In case of a Roots air blower values of pressure ratio, r of 1.2, 1.6, and 2 give values for the Roots efficiency of 0.945, 0.84 and 0.765 respectively. These values show that the *efficiency decreases as the pressure ratio increases.*

This machine has a number of *imperfections but is well suited to such tasks as the scavenging and supercharging of I.C. engines.*

Example *A roots blower compresses 0.08 m^3 of air from 1.0 bar to 1.5 bar per revolution. Calculate the compressor efficiency.*

Solution. Volume of air to be compressed, $V = 0.08 \text{ m}^3$

Intake pressure, $p_1 = 1.0$ bar

Pressure after compression, $p_2 = 1.5$ bar

Actual work done, $W_{actual} = (p_2 - p_1)\, V = 10^5(1.5 - 1.0) \times 0.08 = 4000$ Nm.

Also ideal work done per revolution is given by,

$$W_{ideal} = \frac{\gamma}{\gamma - 1}\, p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{(\gamma - 1)/\gamma} - 1 \right]$$

$$= \frac{1.4}{1.4 - 1} \times 1.0 \times 10^5 \times 0.08 \left[\left(\frac{1.5}{1.0} \right)^{\frac{1.4 - 1}{1.4}} - 1 \right] = 3438.89 \text{ Nm.}$$

$$\therefore \qquad \eta_{compressor} = \frac{W_{ideal}}{W_{actual}} = \frac{3438.89}{4000} = \mathbf{0.8597 \text{ or } 85.97\%.} \quad \textbf{(Ans.)}$$

Vane Type Blower

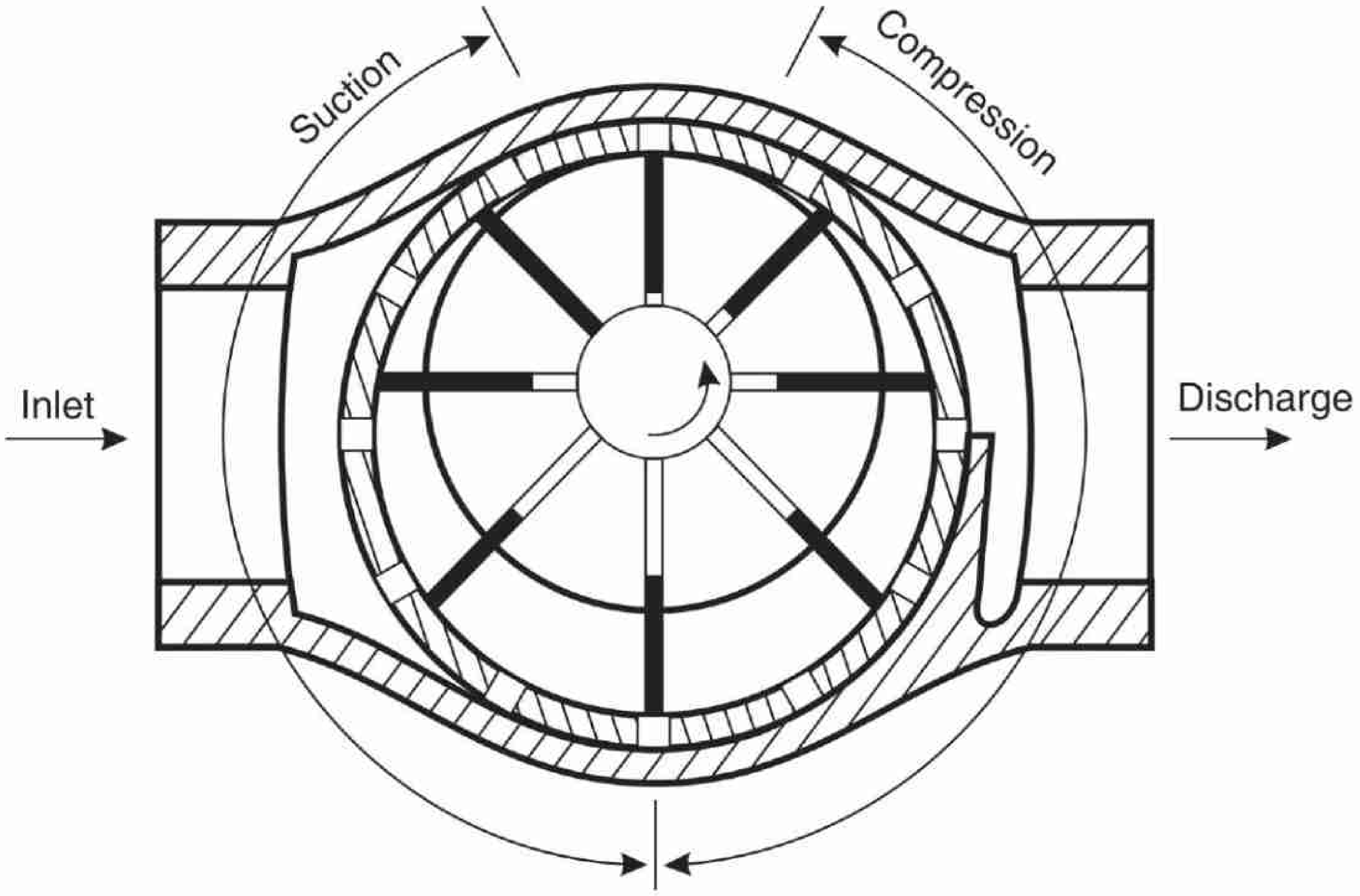


Fig. Vane type blower.

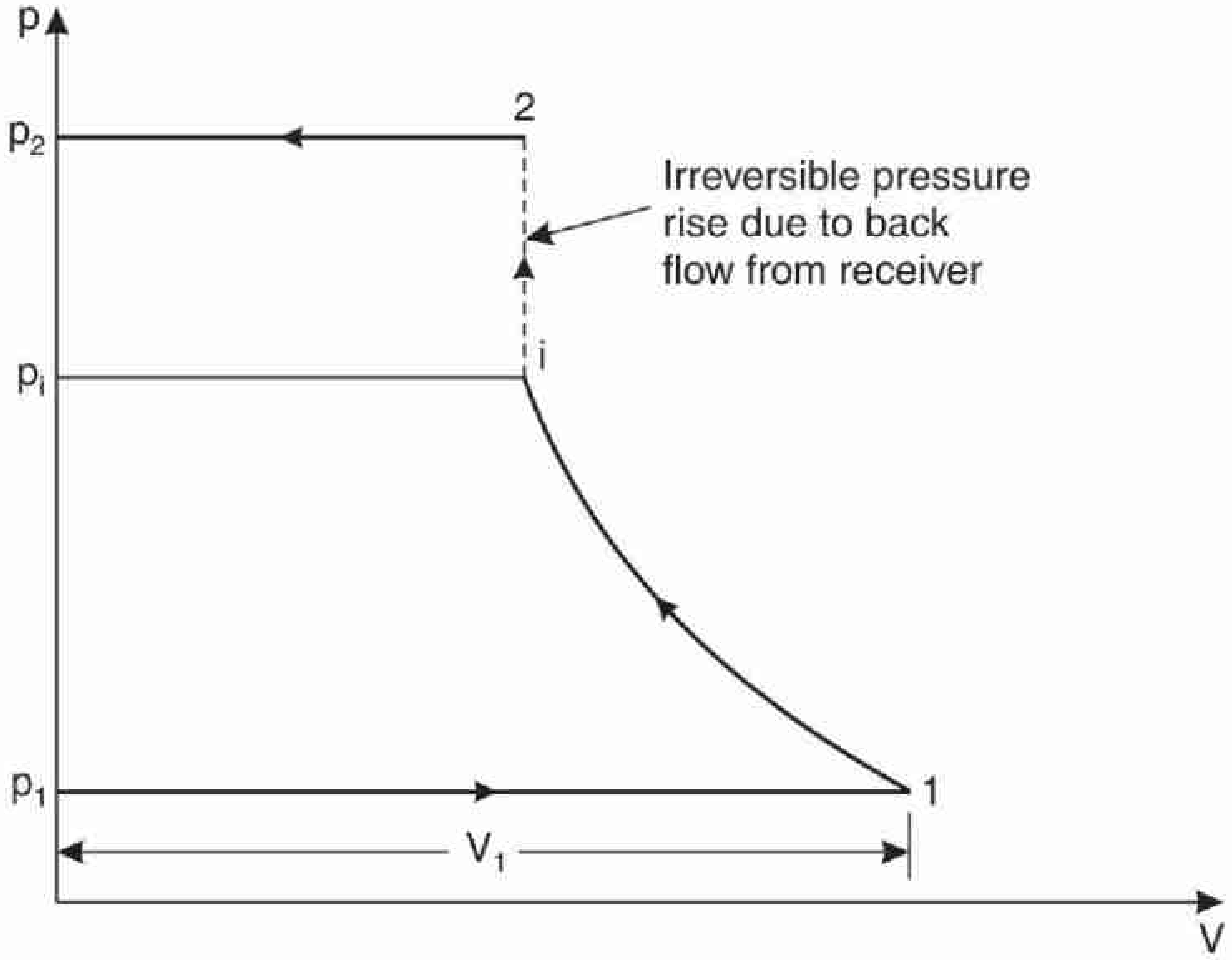


Fig. . *p-V* diagram for vane blower.

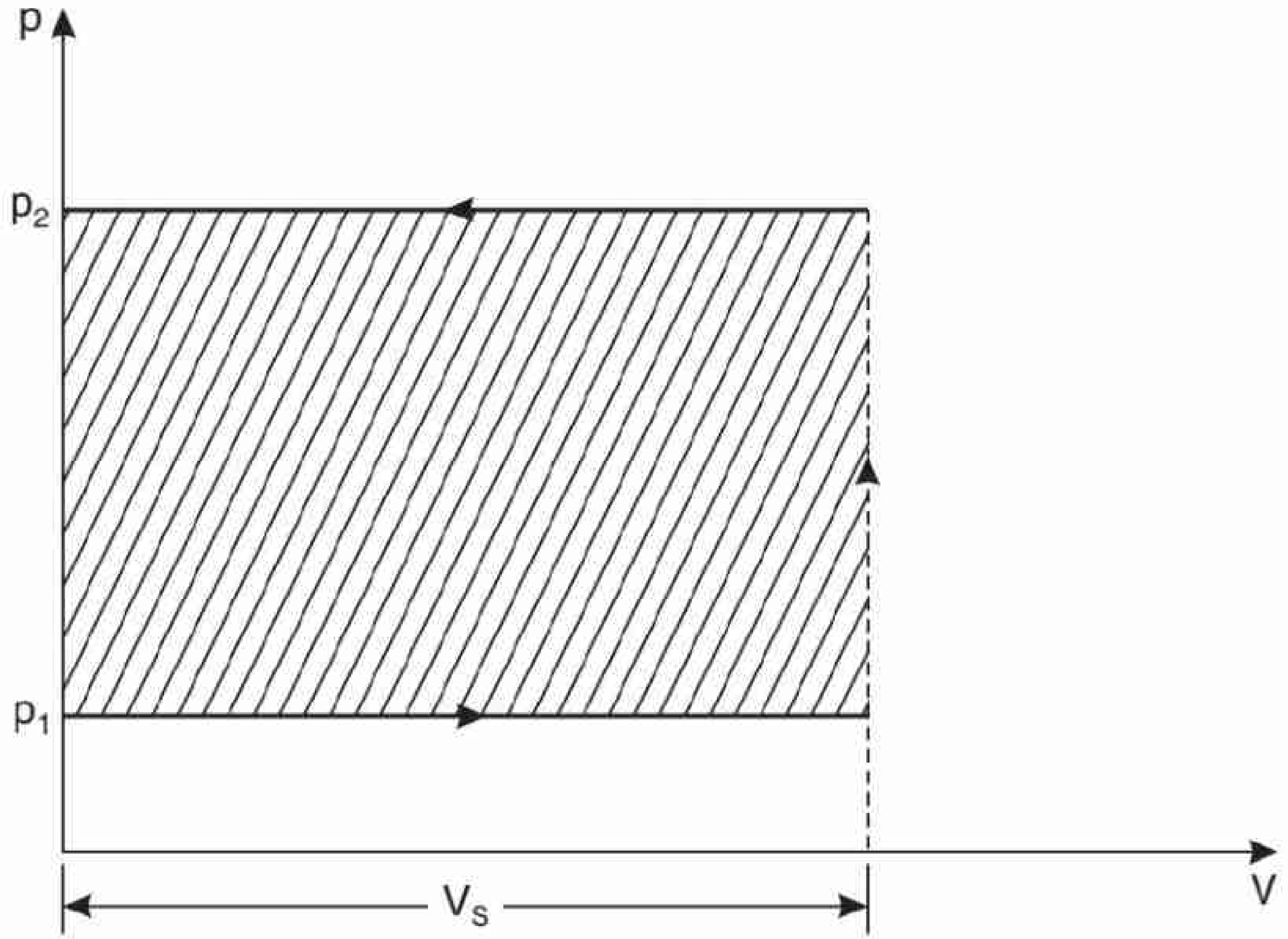
Compression occurs to the pressure p_i, the ideal form for an uncooled machine being isentropic. At this pressure the displaced gas is opened to the receiver and gas flowing back from the receiver raises the pressure irreversibly to p_2. The work done per revolution with N vanes is given by the following expression :

$$W = N \frac{\gamma}{\gamma - 1} p_1 V_1 \left[\left(\frac{p_i}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} - 1 \right] + N(p_2 - p_i)V_i$$

- The vane blowers *require less work* compared to roots blower for the same capacity and pressure rise.
- They are commonly used to deliver upto 150 m^3 of air per minute at pressure ratio upto 8.5.
- The speed limit of a vane blower is 3000 r.p.m.

Example *Compare the work inputs required for a Roots blower and a Vane type compressor having the same induced volume of 0.03 m^3/rev., the inlet pressure being 1.013 bar and the pressure ratio 1.5 to 1. For the Vane type assume that internal compression takes place through half the pressure range.*

Solution. Inlet pressure, p_1 = 1.013 bar



Pressure ratio, $\frac{p_2}{p_1} = 1.5$

$\therefore$ $p_2 = 1.5\ p_1 = 1.5 \times 1.013 = 1.52$ bar.

Work done/rev. $= (p_2 - p_1)V_s$

$= (1.52 - 1.013) \times \frac{10^5 \times 0.03}{10^3}$ = **1.52 kJ.** **(Ans.)**

For the **Vane type,** $p_i = \dfrac{1.52 + 1.013}{2} = 1.266$ bar

Refer Fig. 43.

Work required = (Area A + Area B)

Now,

$$\text{Area } A = \frac{\gamma}{\gamma - 1} \, p_1 V_s \left[\left(\frac{p_i}{p_1} \right)^{(\gamma - 1)/\gamma} - 1 \right]$$

$$= \frac{1.4}{1.4 - 1} \times \frac{1.013 \times 10^5 \times 0.03}{10^3} \left[\left(\frac{1.266}{1.013} \right)^{\frac{1.4 - 1}{1.4}} - 1 \right] \text{ kJ/rev.}$$

$$= 3.5 \times 1.013 \times 100 \times 0.03 \times 0.066 = 0.702 \text{ kJ/rev.}$$

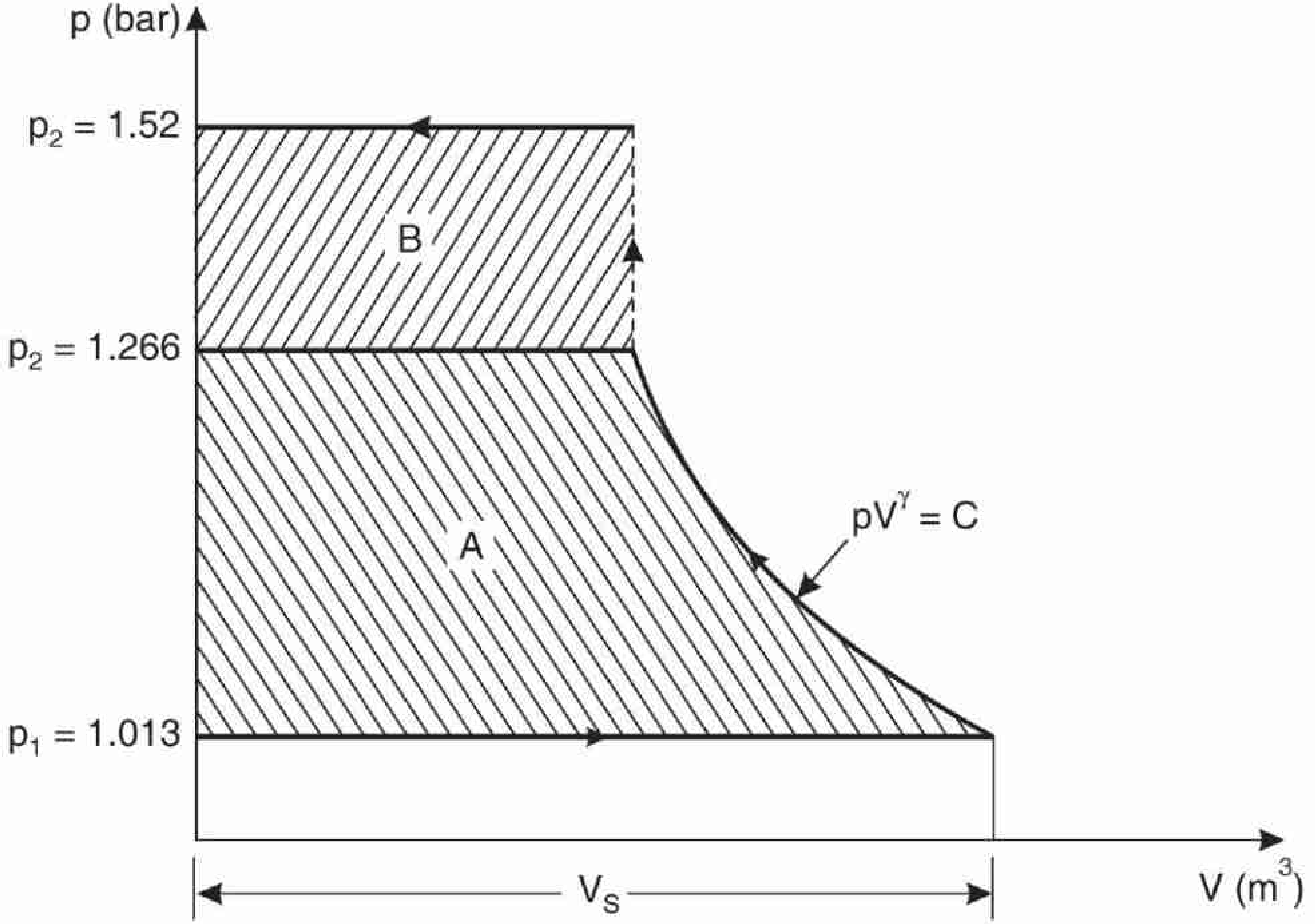


Fig. 43

Area $B = (p_2 - p_1)V_b$

Now, $$V_b = V_s \left(\frac{p_1}{p_2}\right)^{1/\gamma} = 0.03 \left(\frac{1.013}{1.266}\right)^{1/1.4} = 0.0256 \text{ m}^3$$

Area $$B = \frac{(1.52 - 1.266) \times 10^5 \times 0.0256}{10^3} = 0.65 \text{ kJ/rev.}$$

$\therefore$ Work required = 0.702 + 0.65 = **1.352 kJ/rev. (Ans.)**

Screw type or Helical type compressor :

Screw type compressor is very much similar to roots blower. These may have two spiral lobed rotors, out of which one may be called *male rotor* having 3–4 lobes and other *female rotor* having 4–6 lobes which intermesh with small clearance. Meshing is such that lobes jutting out of male rotor get placed in matching hollow portion in female rotors. Initially, before this intermeshing the hollows remain filled with gaseous fluid at inlet port. As rotation begins the surface in contact move parallel to the axis of rotors toward the outlet end gradually compressing the fluid till the trapped volume reaches up to outlet port for getting discharged out at designed pressure. Since the number of lobes are different so the rotors operate at different speed.

These compressors are capable of handling gas flows ranging from 200 to 20000 m3/h under discharge pressures of 3 bar gauge in single stage and up to 13 bar gauge in two stages. Even with increase in number of stages pressures up to 100 bar absolute have been obtained with stage pressure ratio of 2.

Mechanical efficiency of these compressors is quite high, and their isothermal efficiencies are even more than vane blowers and may be compared with centrifugal and axial compressors. But these are very noisy, sensitive to dust and fragile due to small clearances.

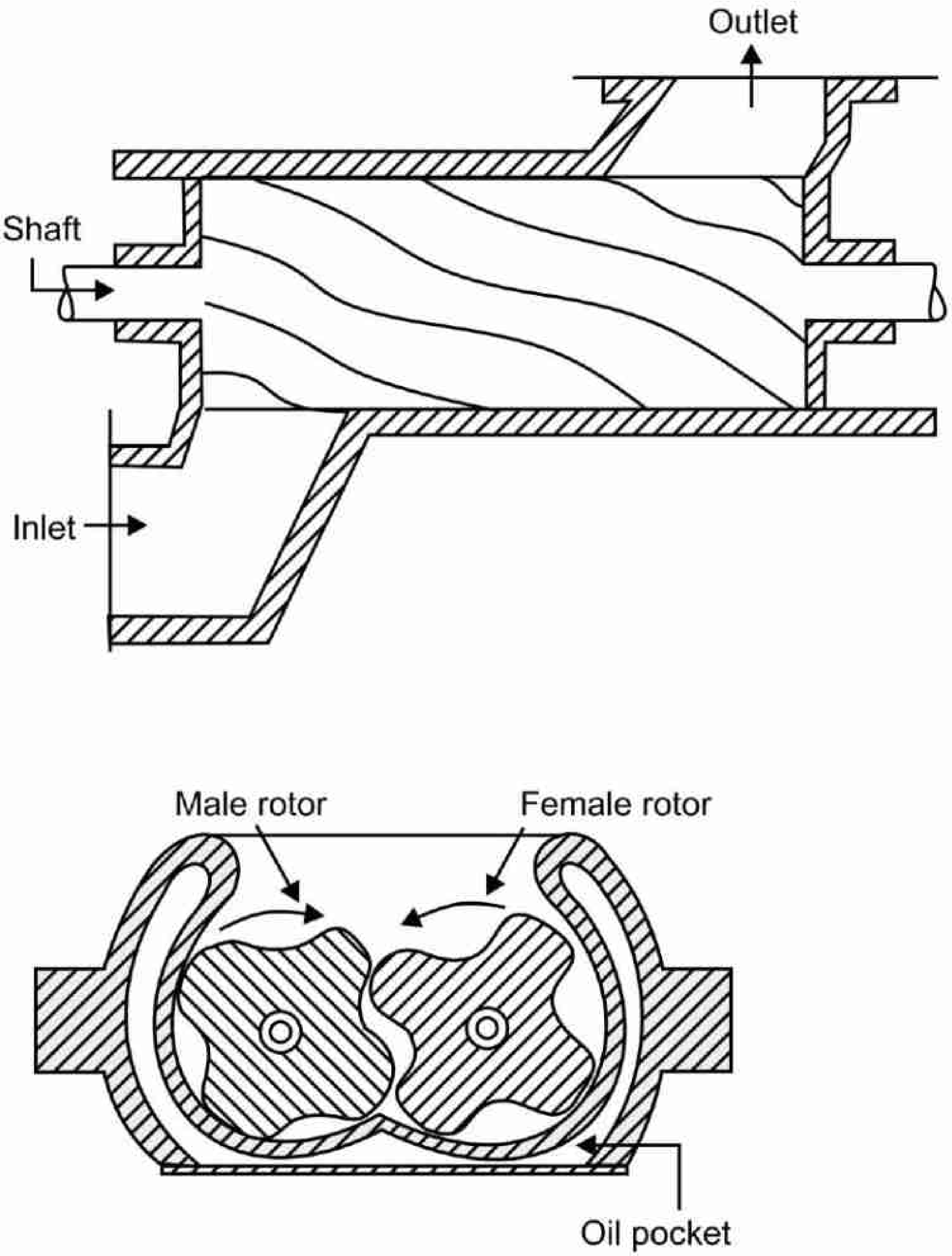
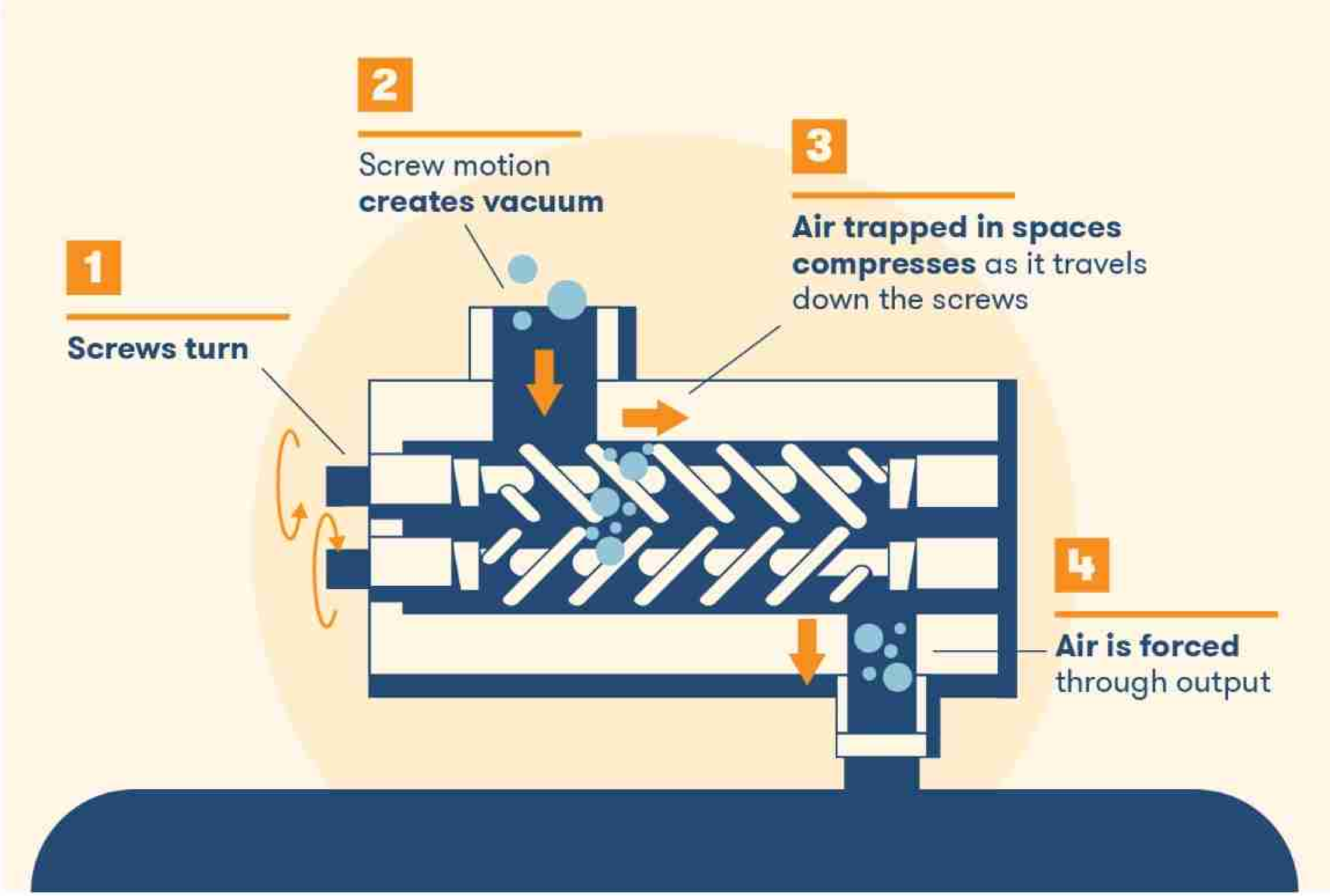


Fig. Line sketch of screw compressor



Centrifugal Compressor

Centrifugal compressor is a ***radial flow*** machine compressing the fluid due to the dynamic action of impeller. Centrifugal compressors have impeller mounted on driving shaft, diffuser and volute casing as shown in Fig. Centrifugal compressors have air inlet at the center of impeller. The portion of impeller in front of inlet passage is called impeller eye.

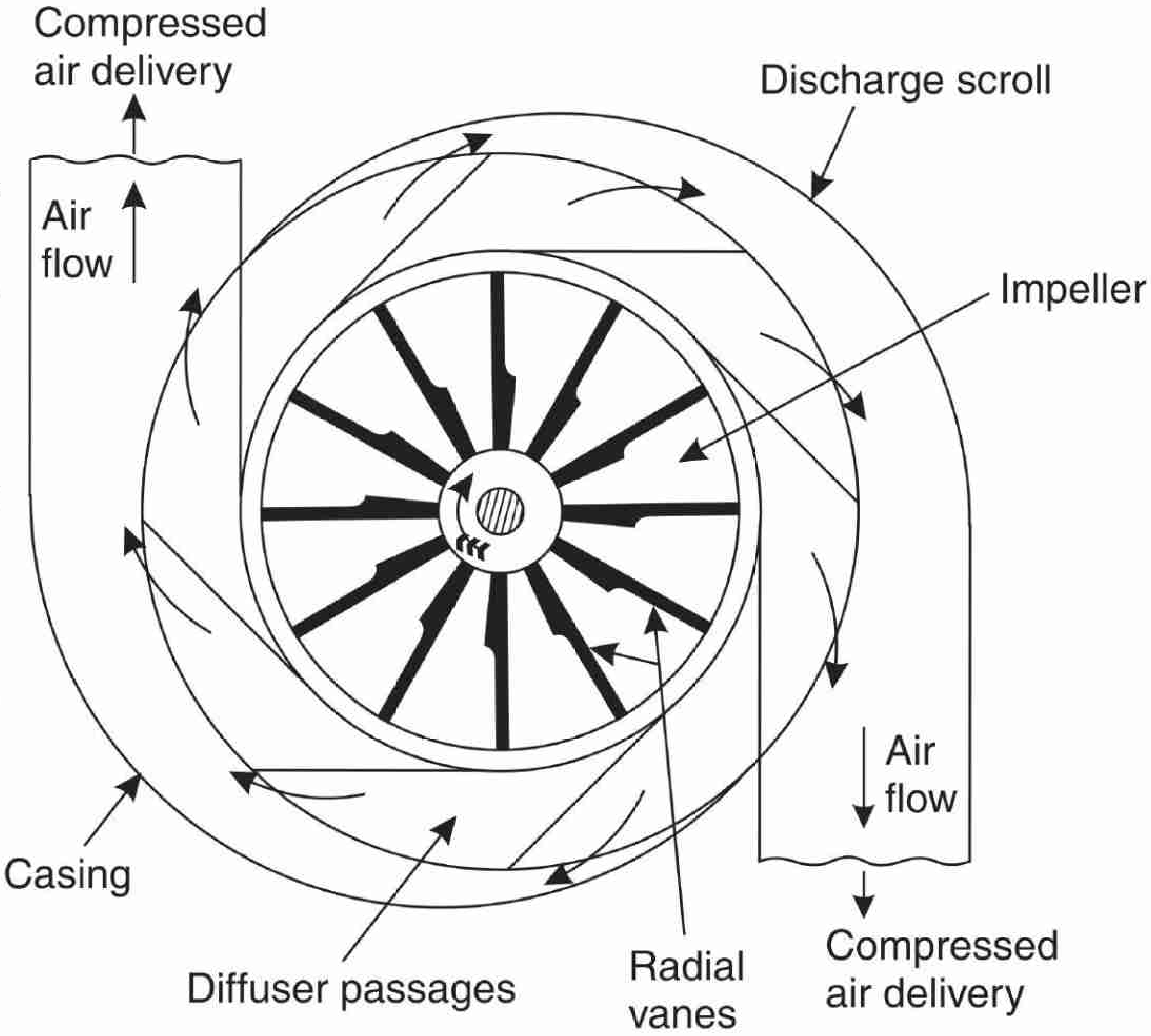


Fig. Centrifugal compressor.

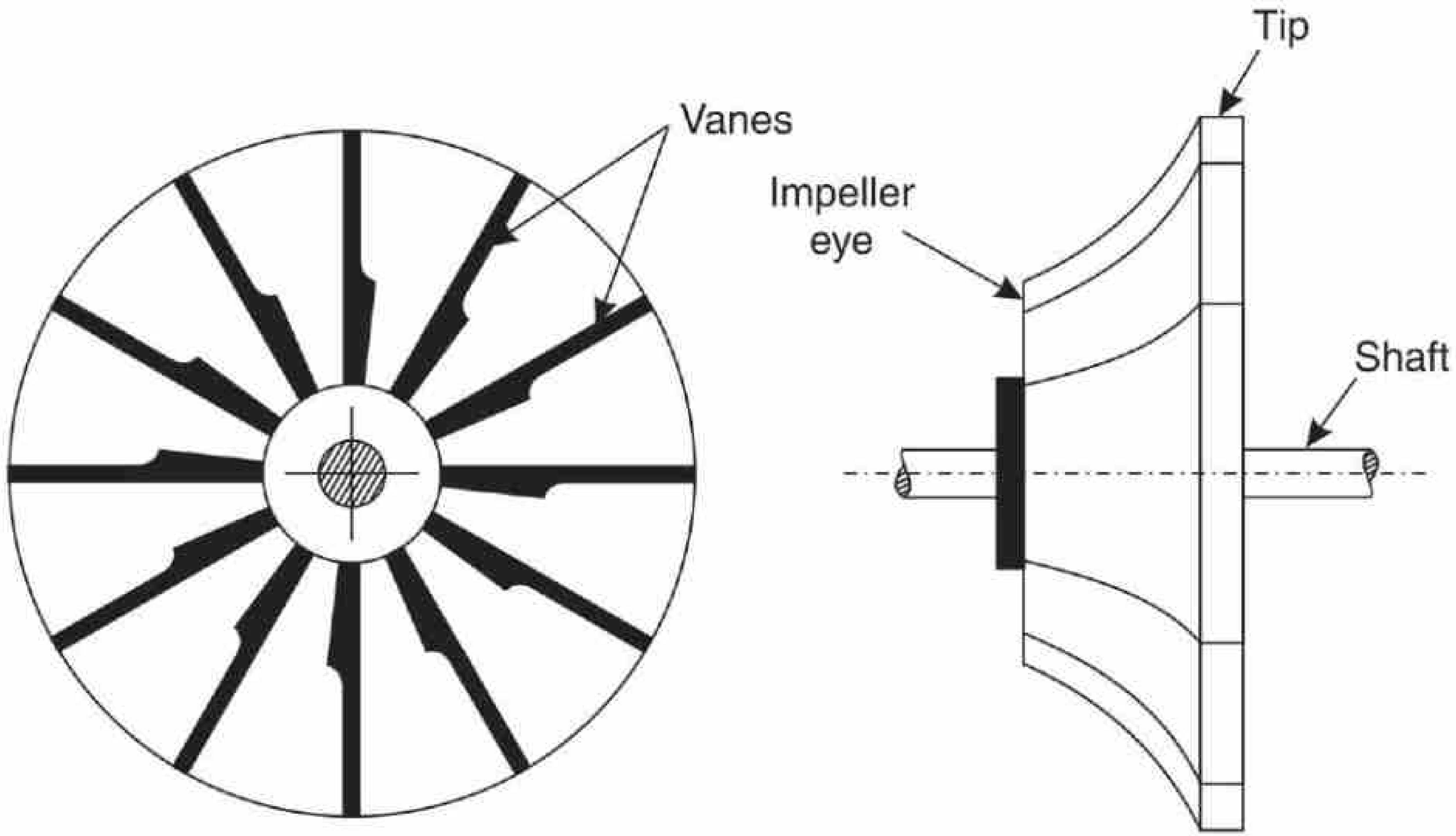


Fig. Impeller (single-eyed) and radial vanes of centrifugal compressor.

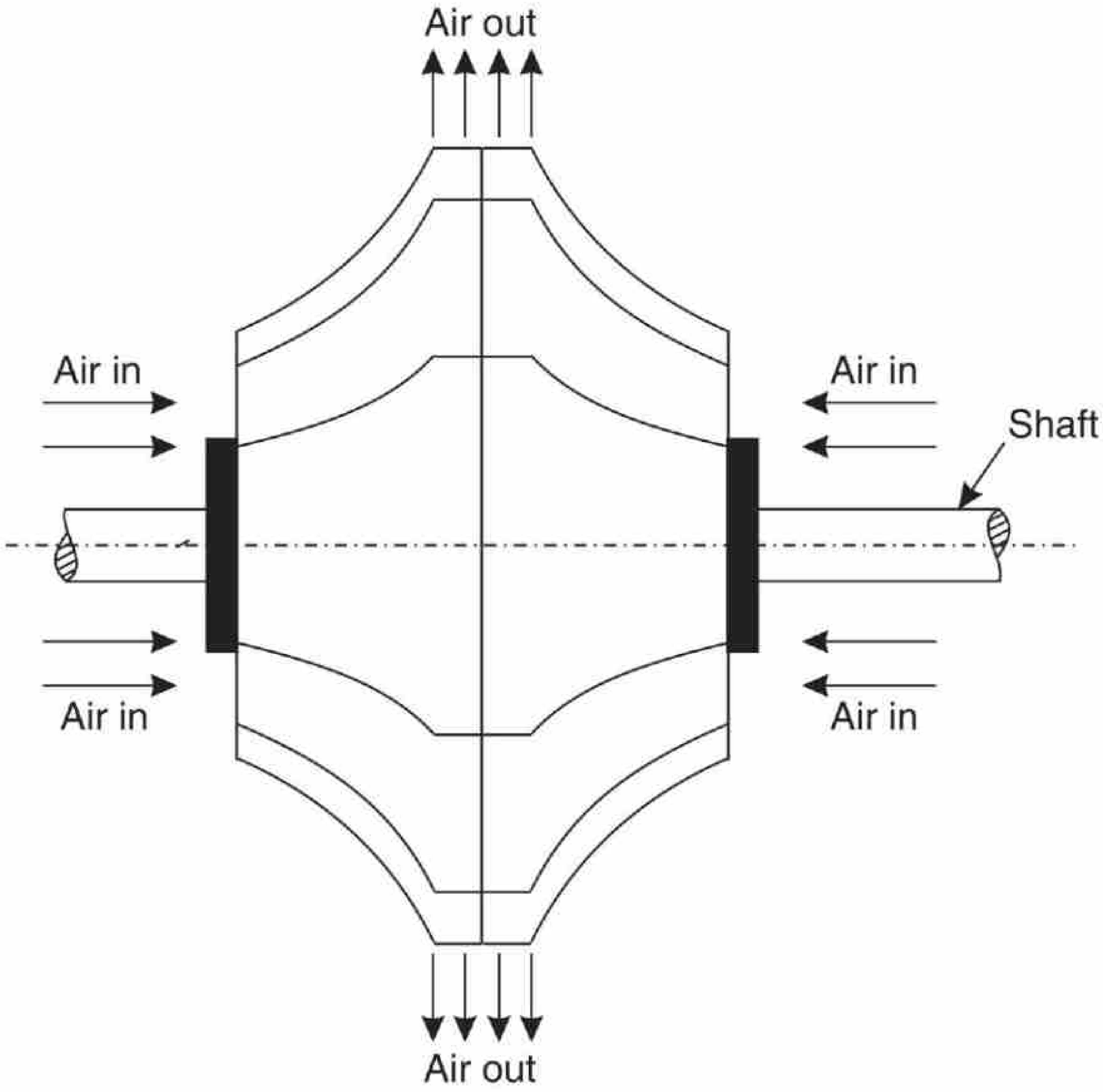


Fig. Double eyed impeller.

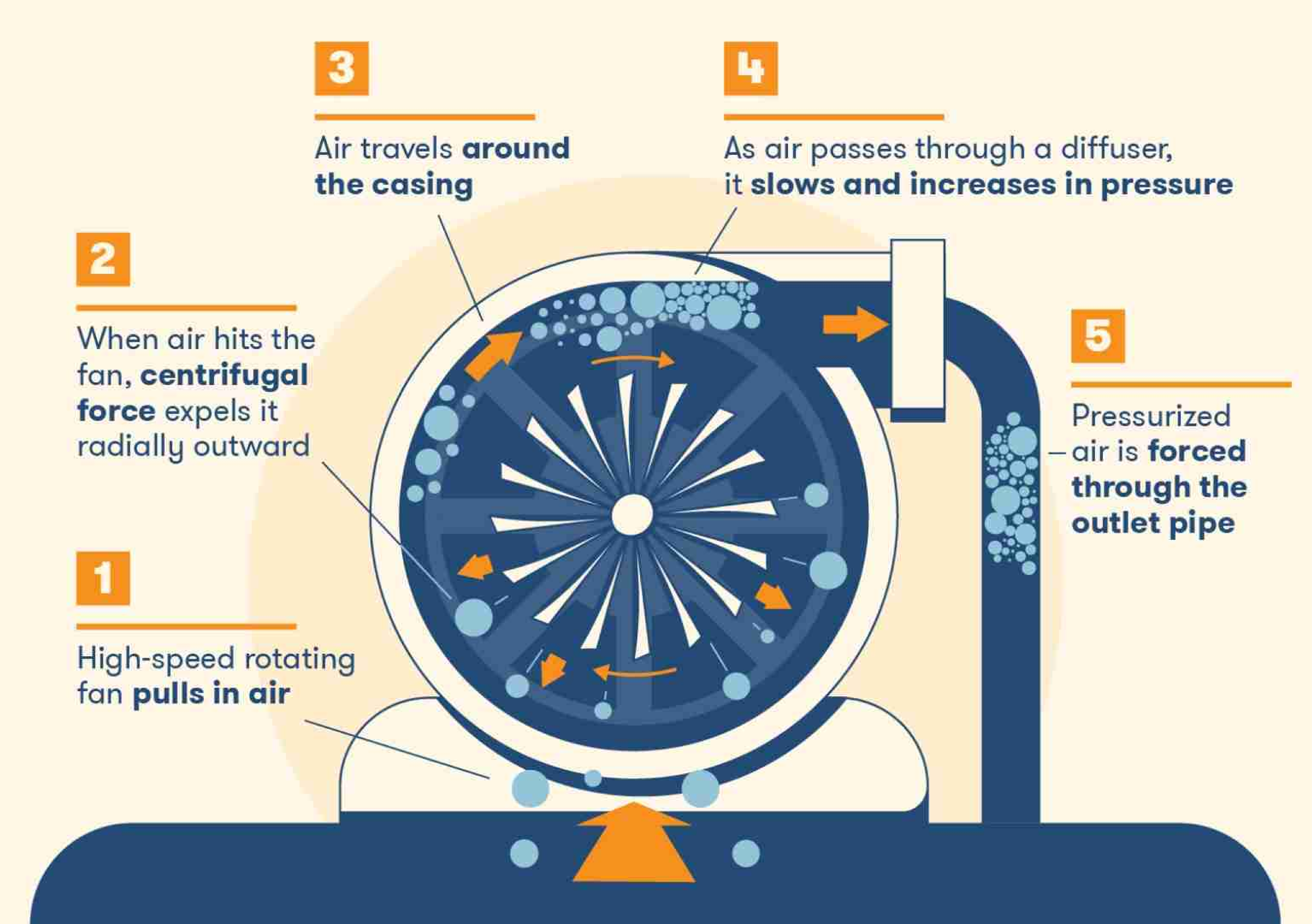
Centrifugal compressors are used in aircrafts, blowers, superchargers, etc. where large quantity of air is to be supplied at smaller pressure ratios. Generally, pressure ratio up to 4 is achieved in single stage centrifugal compressors while in multistage compressors the pressure ratio up to 12 can be achieved. These compressors run at speed of 20,000–30,000 rpm.

1. Curved radial vanes: A series of curved radial vanes are attached to and rotate with the shaft.

2. Impeller: The impeller is a disc fitted with radial vanes. The impeller is generally *forged or die-casted of low silicon aluminium alloy.*

3. Casing: The casing surrounds the rotating impeller.

4. Diffuser: The diffuser is housed in a radial portion of the casing.


3
Air travels **around the casing**
4
As air passes through a diffuser, it **slows and increases in pressure**
2
When air hits the fan, **centrifugal force** expels it radially outward
5
Pressurized air is **forced through the outlet pipe**
1
High-speed rotating fan **pulls in air**

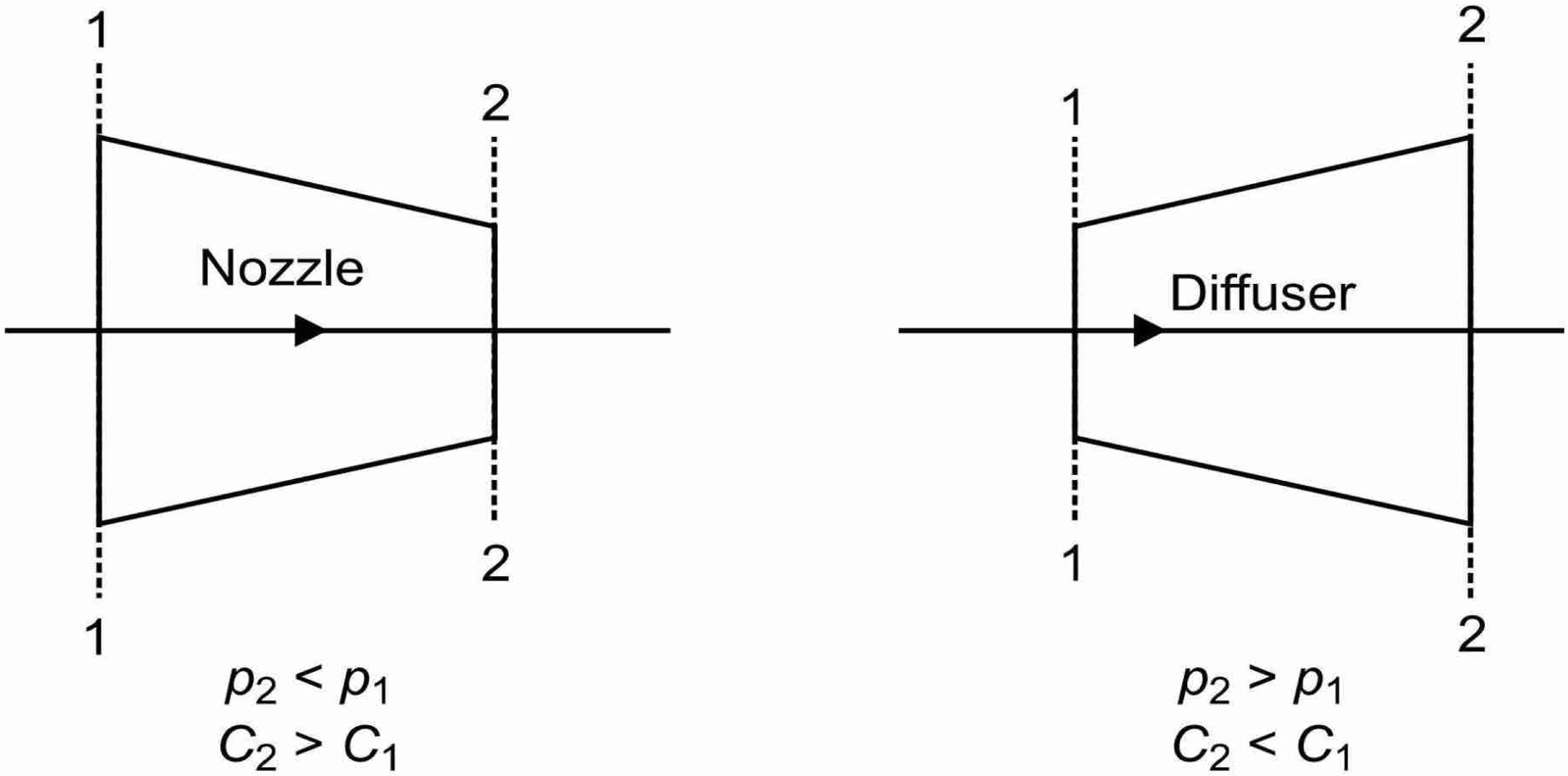
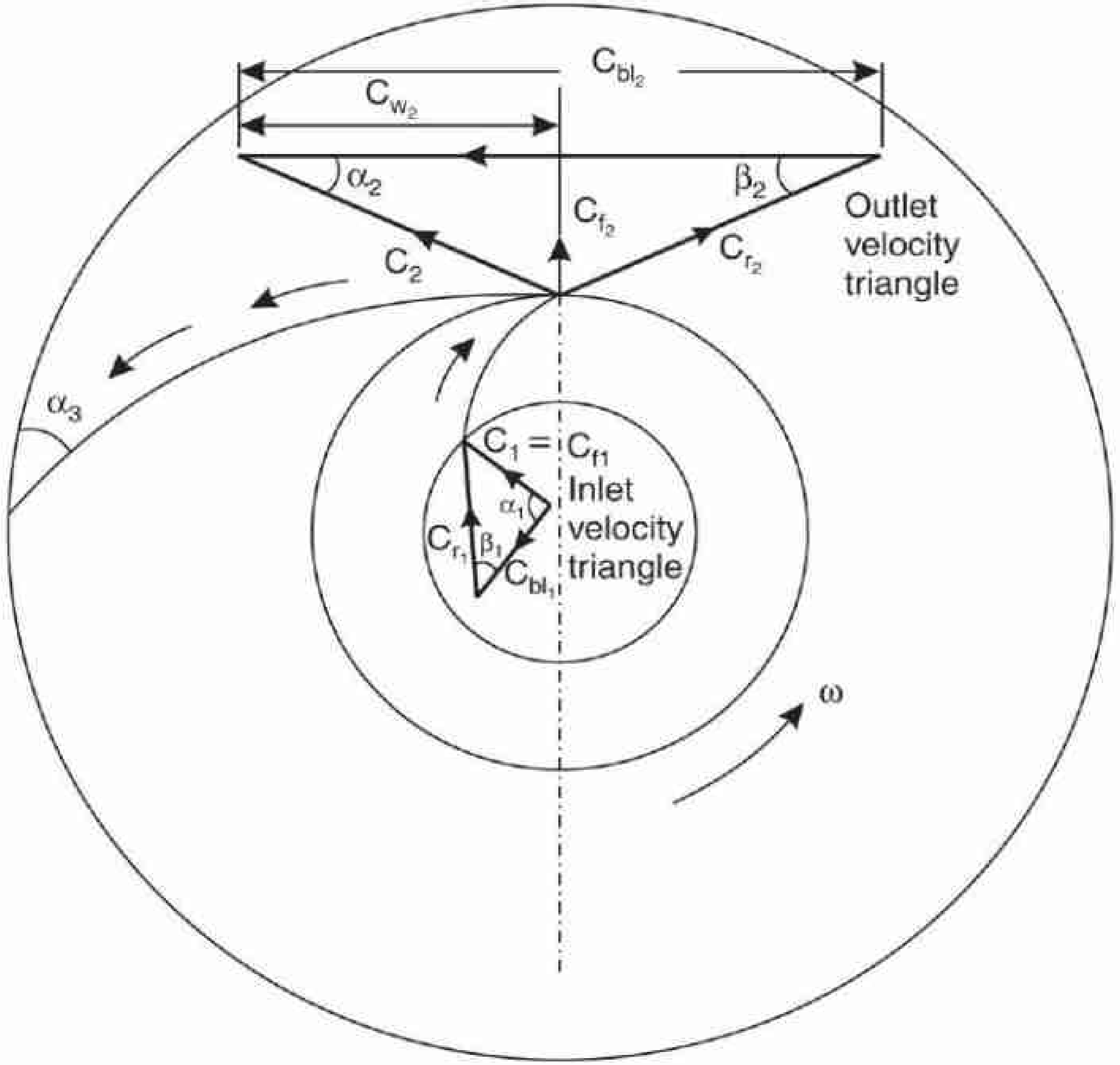


Fig. *General arrangement in nozzle and diffuser*

FS ELLIOTT

Velocity Diagrams and Theory of Operation of Centrifugal Compressors:



Let, C_{bl_1} = Mean blade velocity at entrance,

C_{bl_2} = Mean blade velocity at exit,

C_1 = Absolute velocity at inlet to the rotor,

C_2 = Absolute velocity at outlet to the rotor,

C_{r_1} = Relative velocity of air at entry of rotor,

C_{r_2} = Relative velocity of air at exit of rotor,

C_{w_1} = Velocity of whirl at inlet,

C_{w_2} = Velocity of whirl at outlet,

C_{f_1} = Velocity of flow at inlet,

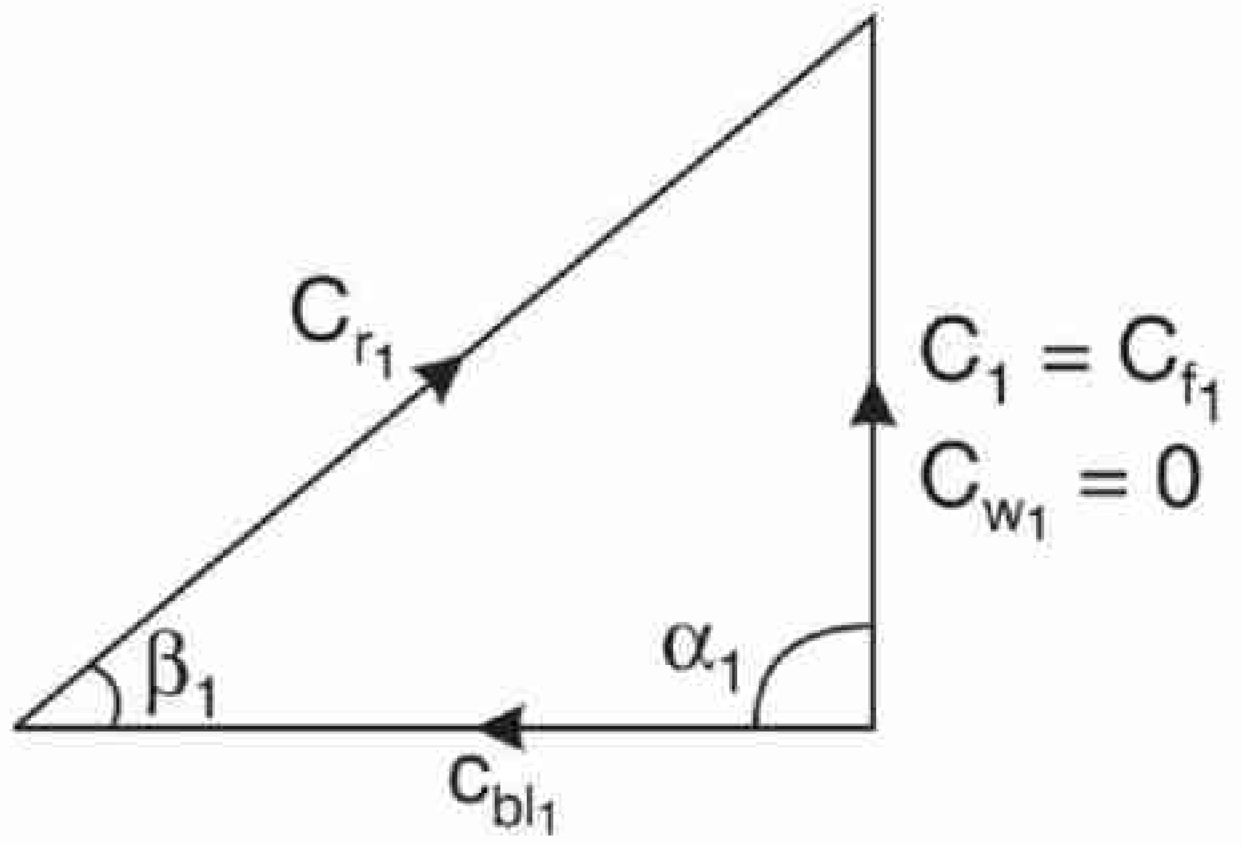
C_{f_2} = Velocity of flow at outlet,

α_1 = Exit angle from the guide vane or inlet angle of the guide vane,

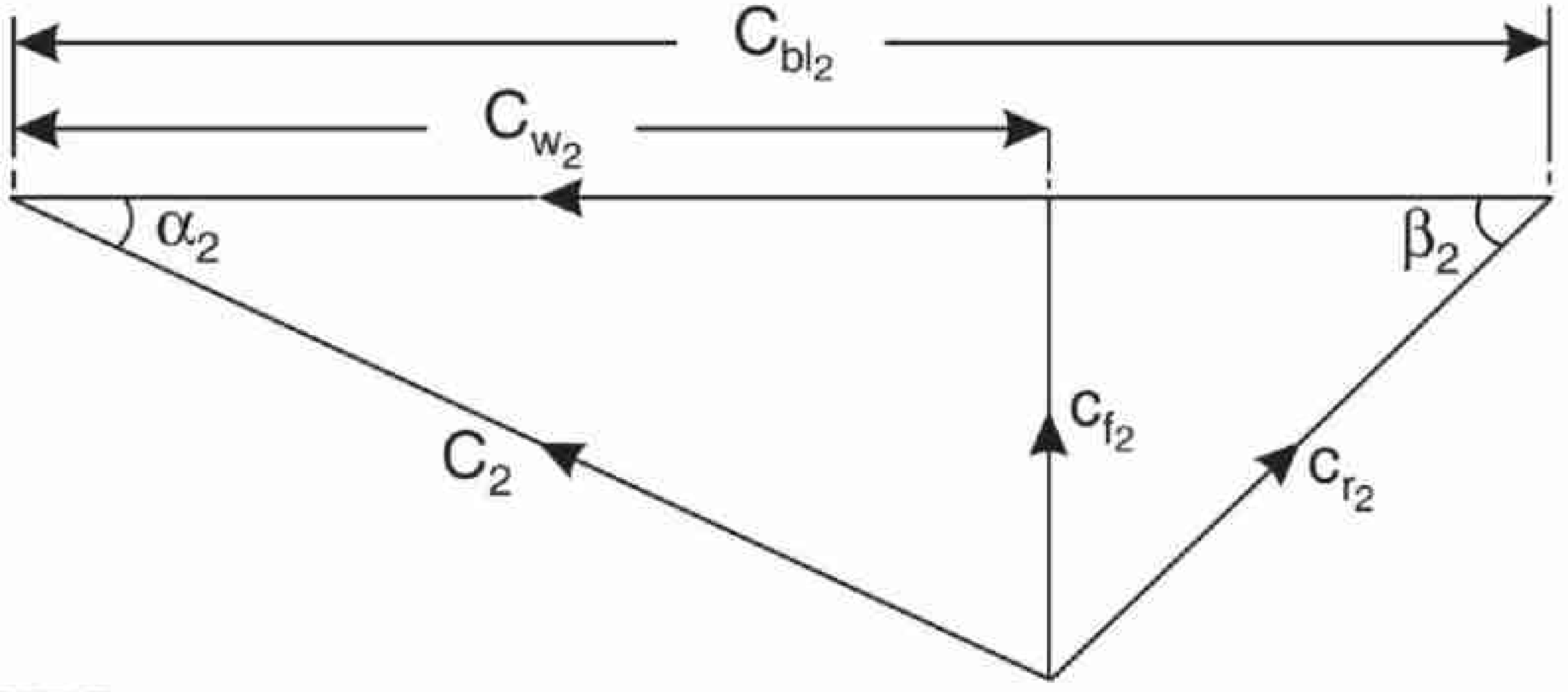
β_1 = Inlet angle to the rotor or impeller,

β_2 = Outlet angle from the rotor or impeller, and

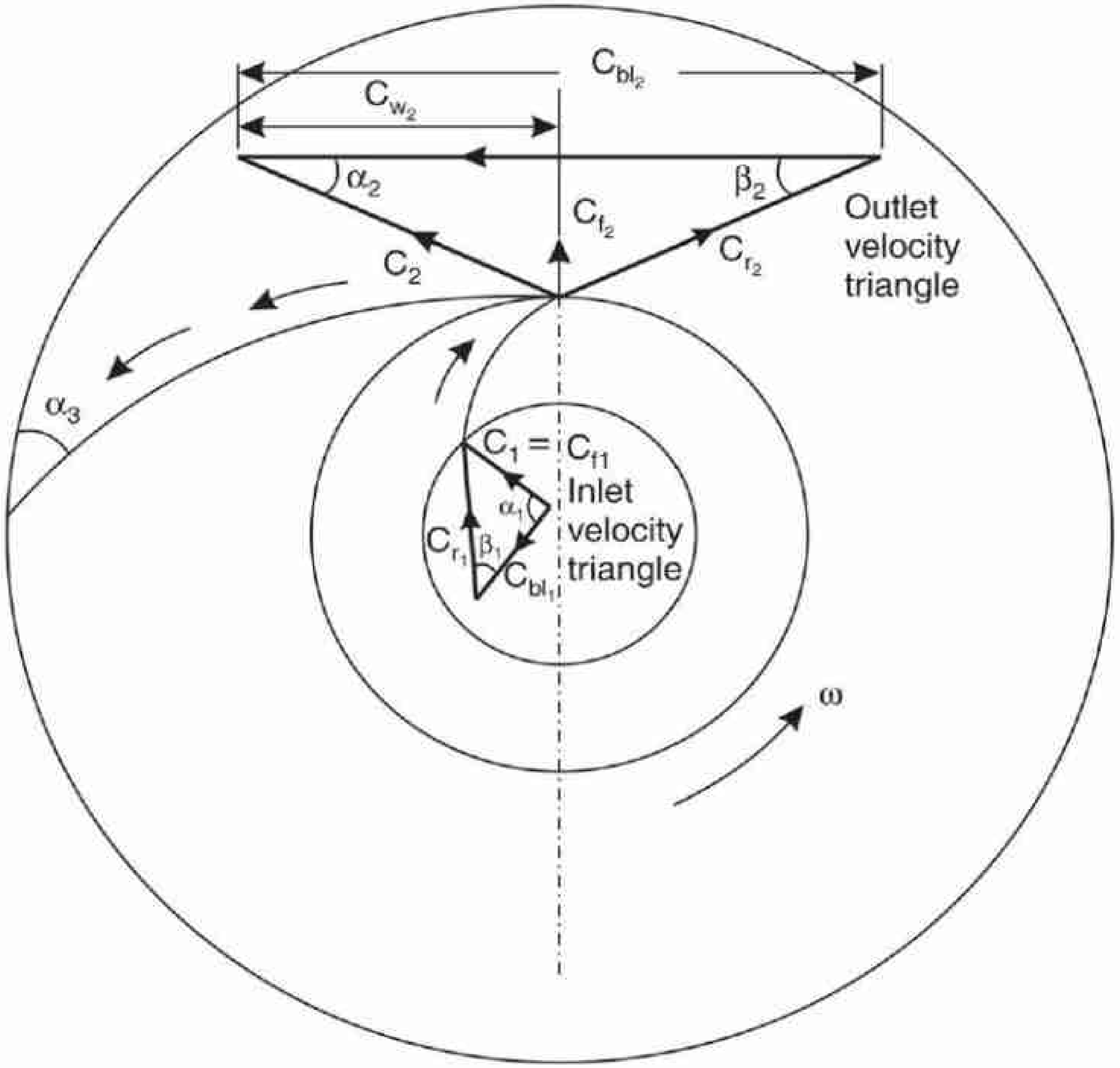
α_2 = Inlet angle to the diffuser.



Inlet velocity diagram
(*a*)



Outlet velocity diagram
(*b*)



It is assumed that the entry of the air is 'axial', therefore the whirl component at the inlet (C_{w1}) is zero and therefore $C_1 = C_{f1}$. The enlarged views of inlet and outlet velocity diagrams are shown in Fig. (a) and (b). To avoid shock at entry and exit the blade must be parallel to the relative velocity of air at inlet or outlet and therefore β_1 and β_2 are the impeller blade angles at the inlet and outlet. The diffuser blade angle must be parallel to the absolute velocity of air from the impeller (C_2), therefore α_2 is the diffuser blade angle at the inlet and α_3 is the diffuser blade angle at the outlet. If the discharge from the diffuser is circumferential, then its blade angle at outlet (α_3) should be as small as possible.

Work done by impeller (Euler's work) :

$$\boxed{W = \underbrace{\frac{C_2{}^2 - C_1{}^2}{2}}_{\text{First term}} + \underbrace{\frac{C_{r_1}^2 - C_{r_2}^2}{2}}_{\text{Second term}} + \underbrace{\frac{C_{bl_2}^2 - C_{bl_1}^2}{2}}_{\text{Third term}}}$$

— The first term shows the increase in K.E. of 1 kg of working fluid in the impeller that has to converted into the pressure energy in the 'diffuser'.

— The second term shows the pressure rise in the impeller due to 'diffusion action' (as the relative velocity decreases from inlet to outlet).

— The third term shows the pressure rise in the impeller due to 'centrifugal action' (as the working fluid enters at a lower diameter and comes out at a higher diameter).

$$\frac{C_2^{\,2} - C_4^{\,2}}{2} = \int_2^4 \frac{dp}{\rho} + \Delta h_{\text{dif. loss}}$$

i.e., the difference of K.E. at the impeller outlet and diffuser outlet introduces the work partly utilised for pressure increase, and partly irreversibly inverted into heat due to losses in diffuser.

Power required per impeller for *m* kg of air flow in one second,

$$P = \frac{\dot{m}\, C_{w_2}\, C_{bl_2}}{1000}\ \text{kW},$$

If the blade is radial (ideal case), then the velocity diagram at the outlet of the impeller is as shown in Fig.

As Cw2 = Cbl2 , the work done per kg of air flow per second is given by

$$W = C_2^{\,2}$$

The power input to the compressor depends upon the following factors :

(i) Mass flow of air through the compressor.

(ii) Total temperature at the inlet of the compressor.

(iii) Total pressure ratio of the compressor which depends upon the square of the impeller tip velocity.

Width of Blades of Impeller and Diffuser

If the mass of the air flowing per second is constant and is known, then the width of blades of impeller and diffuser can be calculated as follows :

Let, $\dot{m}$ = Mass of air flowing per second,

b_1 = Width (or height) of impeller at inlet,

C_{f_1} = Velocity of flow at inlet of the impeller,

v_1 = Volume of 1 kg of air at the inlet,

r_1 = Radius of impeller at the inlet, Then, $\dot{m} = \frac{\text{Volume of air flowing per second}}{\text{Volume of 1 kg of air}} = \frac{2\pi r_1 b_1 \times C_{f_1}}{v_1}$

But as the air is trapped radially,

$$C_{f_1} = C_1$$

$$\therefore \quad \dot{m} = \frac{2\pi r_1 b_1 \times C_1}{v_1}$$

$$b_1 = \frac{\dot{m}\, v_1}{2\pi_1 r_1 C_1}$$

Similarly the width of impeller blade at the outlet can be found by using suffix 2

$$\dot{m} = \frac{2\pi r_2 b_2 \times C_{f_2}}{v_2}$$

The width or height of the impeller blades at the outlet and height of diffuser blade at the inlet should be same theoretically.

The width or height of the diffuser blades at the outlet, is given by

$$\dot{m} = \frac{2\pi r_d\, b_d \times C_{fd}}{v_d}$$

where suffix '*d*' represents the quantities at the *outlet of the diffuser*.

If, n = Number of blades on the impeller, and

t = Thickness of the blade,

then eqns. are expressed as follows :

$$\dot{m} = \frac{(2\pi r_1 - nt)b_1 C_{f_1}}{v_1}$$

$$\dot{m} = \frac{(2\pi r_2 - nt)b_2 C_{f_2}}{v_2}$$

$$\dot{m} = \frac{(2\pi r_d - nt)b_d C_{f_d}}{v_d}$$

Slip Factor and Pressure Co-efficient

In the earlier analysis it was assumed that Cw2 = Cbl 2 but this condition is not satisfied in actual practice due to secondary flow effects and therefore in actual compressors

$$C_{w_2} < C_{bl_2}$$

The difference between $(C_{bl_2} - C_{w_2})$ is known as **slip.**

Slip factor (ϕ_s). It is defined as the *ratio of actual whirl component* (C_{w_2}) *and the ideal whirl component* (C_{bl_2})

$$\phi_s = \frac{C_{w_2}}{C_{bl_2}} = 1 \text{ if } C_{w_2} = C_{bl_2}$$

The actual work done per kg of air by the compressor is always greater than $C_{bl_2} C_{w_2}$ due to fluid friction and windage losses, therefore the actual work is obtained by multiplying $C_{bl_2} C_{w_2}$ by a factor ϕ_w known as **work factor** or **power input factor.**

$$\therefore \qquad c_p(T_{02} - T_{01}) = \phi_w C_{bl_2} C_{w_2}$$

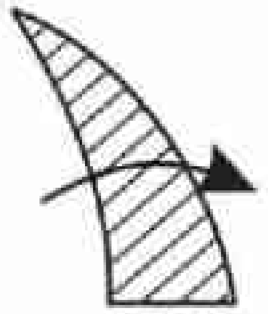
Pressure Co-efficient (ϕ_p). It is defined as the *ratio of isentropic work to Euler work.*

$$\therefore \qquad \phi_p = \frac{\text{Isentropic work}}{\text{Euler work}} = \frac{c_p (T_{02}' - T_{01})}{C_{bl_2} C_{w_2}}$$

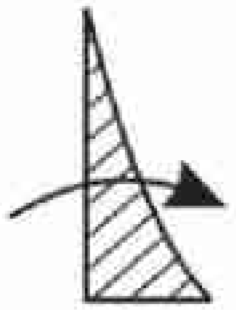
The Effect of Impeller Blade Shape on Performance

The following shapes of blades are utilized in the impellers of centrifugal compressors :

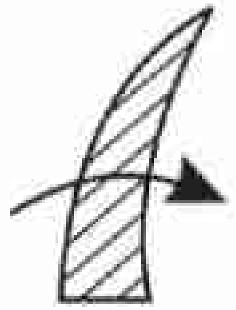
1. *Backward-curved blades* ($\beta_2 < 90°$)



2. *Radial-curved blades* ($\beta_2 = 90°$)



3. *Forward-curved blades* ($\beta_2 > 90°$)



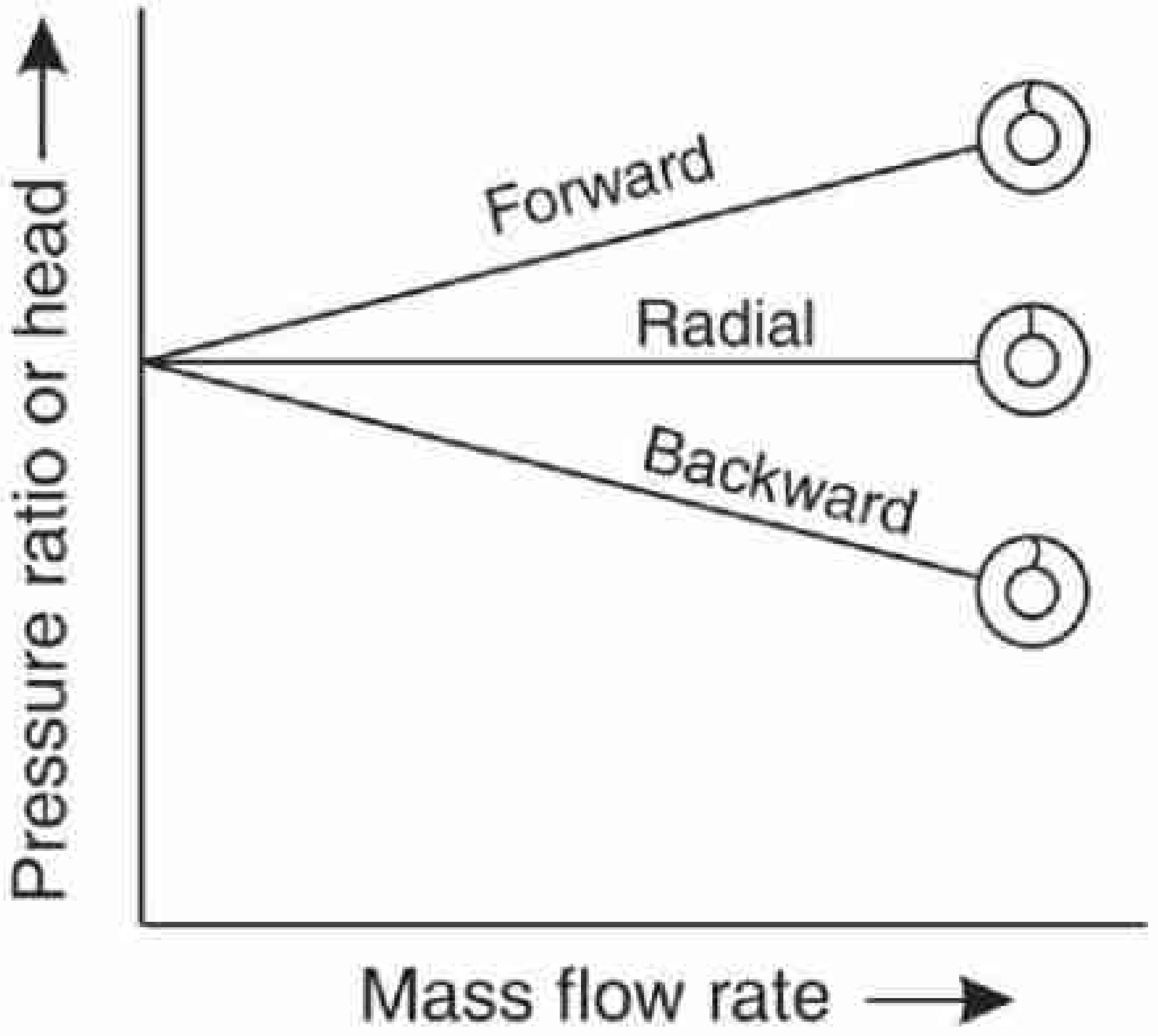


Fig. Characteristics of backward-curved, radial-curved, and forward-curved vanes.

Advantages of radial-blade impellers :

1. Can be manufactured *easily*.

2. Lowest unit blade stress for a given diameter and rotational speed, hence *highest weight*.

3. Free from complex bending stresses.

4. Equal energy conversion in impeller and diffuser, *giving high pressure ratios with good efficiency*.

In view of the above reasons, the *impeller with radial blades has been the logic choice of the designers of aircraft centrifugal compressors.*

A centrifugal compressor used as a supercharger for aero-engines handles 150 kg/min. of air. The suction pressure and temperature are 1 bar and 290 K. The suction velocity is 80 m/s. After compression in the impeller the conditions are 1.5 bar 345 K and 220 m/s. Calculate :

(*i*) *Isentropic efficiency.*

(*ii*) *Power required to drive the compressor.*

(*iii*) *The overall efficiency of the unit.*

It may be assumed that K.E. of air gained in the impeller is entirely converted into pressure in the diffuser.

Solution. *Given :* $\dot{m} = \frac{150}{60} = 2.5$ kg/s ; $p_1 = 1$ bar ; $T_1 = 290$ K ; $C_1 = 80$ m/s ;

$p_2 = 1.5$ bar ; $T_2 = 345$ K ; $C_2 = 220$ m/s.

(*i*) **Isentropic efficiency,** $\eta_{\text{isen.}}$ **:**

$$\frac{T_2'}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.5}{1}\right)^{\frac{1.4-1}{1.4}} = 1.1228$$

or $$T_2' = 290 \times 1.1228 = 325.6 \text{ K}$$

$\therefore$ Isentropic work done $= c_p(T_2' - T_1) + \dfrac{C_2^2 - C_1^2}{2 \times 1000}$

$$= 1.005(325.6 - 290) + \frac{(220)^2 - (80)^2}{2 \times 1000}$$

$$= 35.778 + 21 = 56.78 \text{ kJ/kg.}$$

Work done in the impeller $= c_p(T_2 - T_1) + \dfrac{(220)^2 - (80)^2}{2 \times 1000}$

$$= 1.005(345 - 290) + \frac{(220)^2 - (80)^2}{2 \times 1000}$$

$$= 55.275 + 21 = 76.27 \text{ kJ/kg}$$

$\therefore$ $$\eta_{isen} = \frac{\text{Isentropic work}}{\text{Actual work}} = \frac{56.78}{76.27} = \mathbf{0.7445} \text{ or } \mathbf{74.45\%}. \quad \textbf{(Ans.)}$$

(*ii*) **Power required to drive the compressor, P :**

$$P = \dot{m} \times \text{Work done in the impeller (kJ/kg)}$$

$$= 2.5 \times 76.27 = \mathbf{190.67 \ kW.} \quad \textbf{(Ans.)}$$

(*iii*) **The overall efficiency of the unit, $\eta_{overall}$:**

As K.E. gained in the impeller is converted into pressure, hence

$$c_p(T_3 - T_2) = \frac{C_2^{\,2} - C_1^{\,2}}{2 \times 1000}$$

$$1.005(T_3 - 345) = \frac{(220)^2 - (80)^2}{2 \times 1000}$$

$$T_3 = 365.9 \text{ K.}$$

The *pressure of air after leaving the diffuser, p_3* :

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{p_3}{p_2} = \left(\frac{T_3}{T_2}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{365.9}{345}\right)^{\frac{1.4}{1.4-1}} = 1.2286$$

$\therefore$ $$p_3 = 1.5 \times 1.2286 = 1.843 \text{ bar.}$$

After isentropic compression, the *delivery temperature from diffuser*, T_3' :

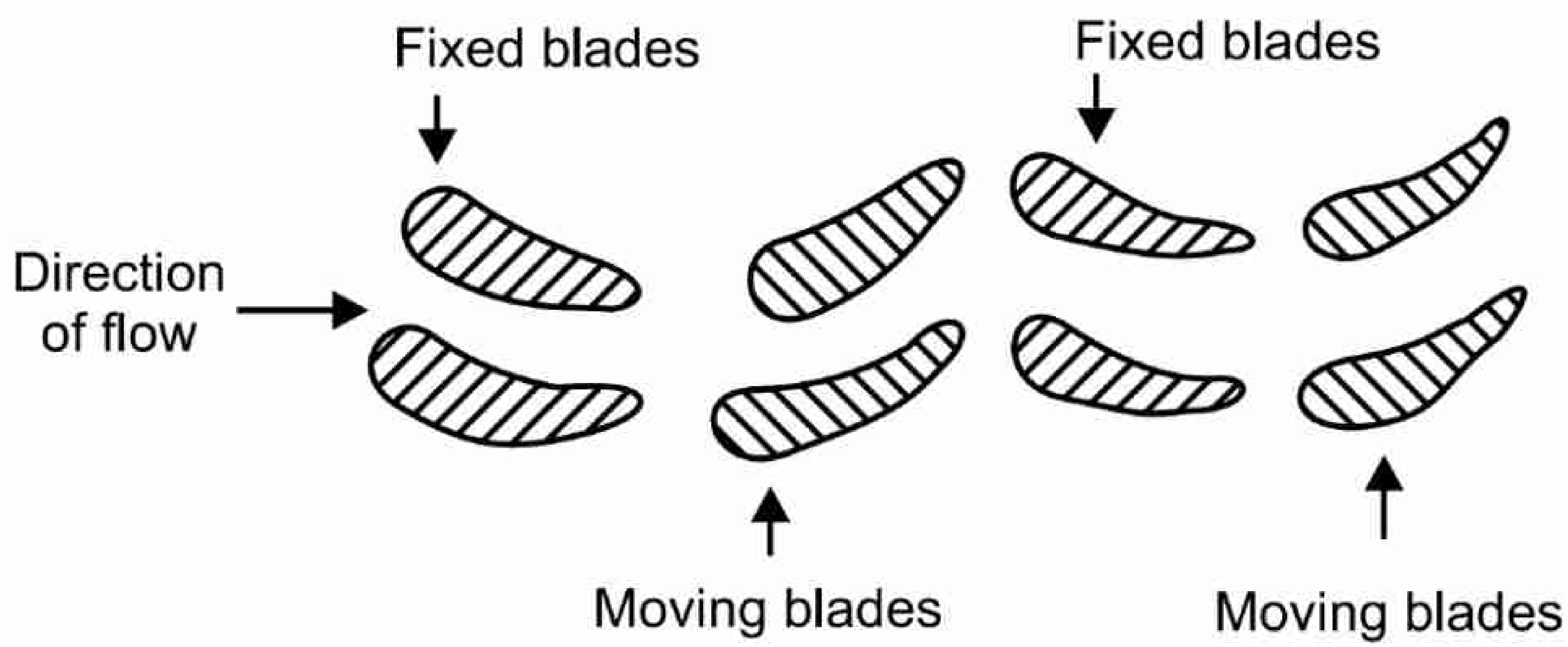
$$\frac{T_3'}{T_1} = \left(\frac{p_3}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.843}{1}\right)^{\frac{1.4-1}{1.4}} = 1.191$$

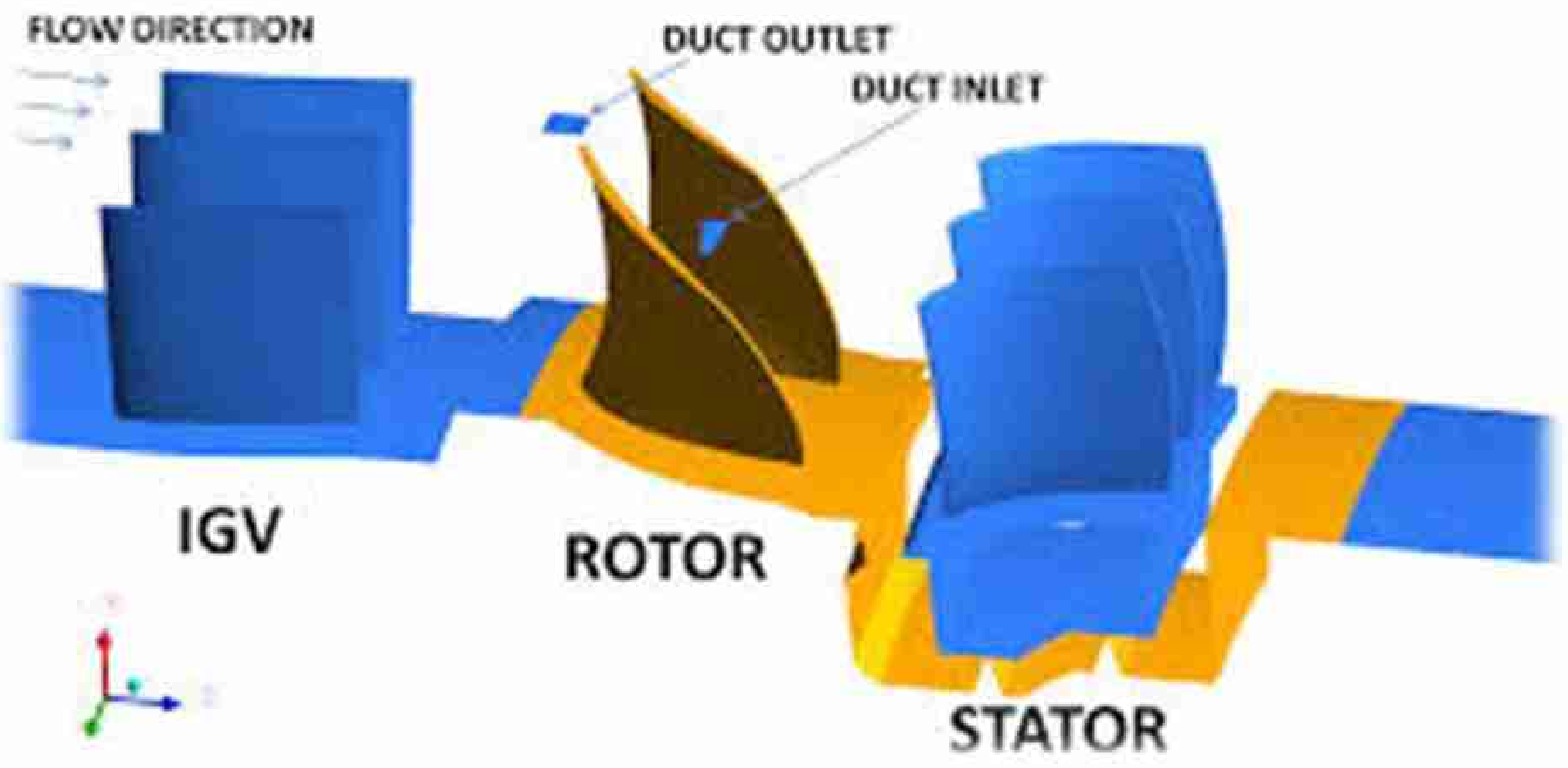
$$T_3' = 290 \times 1.191 = 345.39 \text{ K}$$

$$\therefore \qquad \eta_{overall} = \frac{T_3' - T_1}{T_3 - T_1} = \frac{345.39 - 290}{365.9 - 290} = \mathbf{0.7298} \text{ or } \mathbf{72.98\%}. \quad \textbf{(Ans.)}$$

AXIAL FLOW COMPRESSORS

Axial flow compressors have the fixed blades and moving blades mounted along the axis of compressor. Air enters axially and leaves axially. It has primarily two components i.e. rotor and casing. The rotor has blades mounted on it constituting moving blade ring. Blades are also mounted on the inner side of casing thereby constituting stages as fixed blade ring followed by moving blade ring followed by fixed blade ring, moving blade ring and so on.





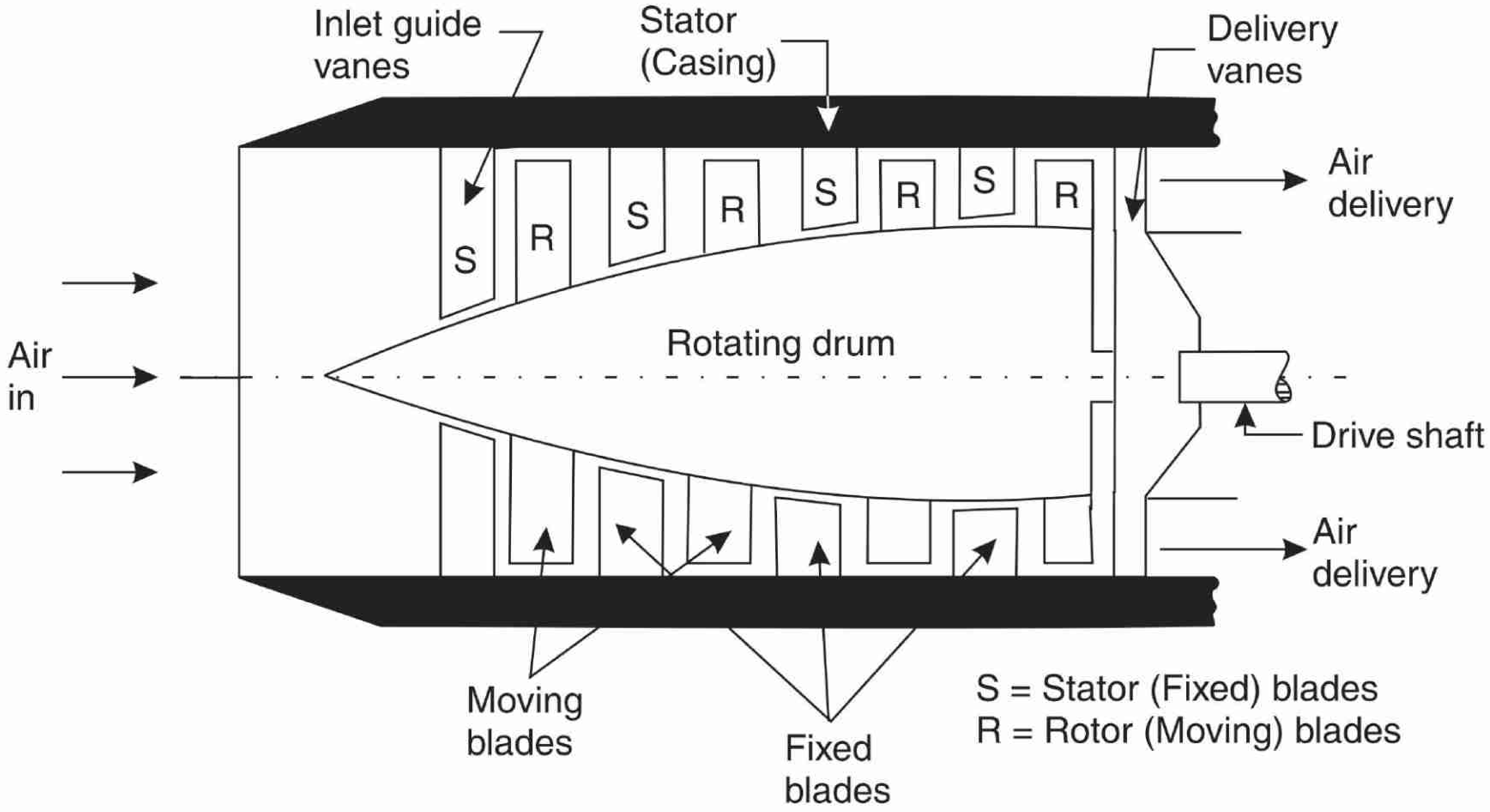
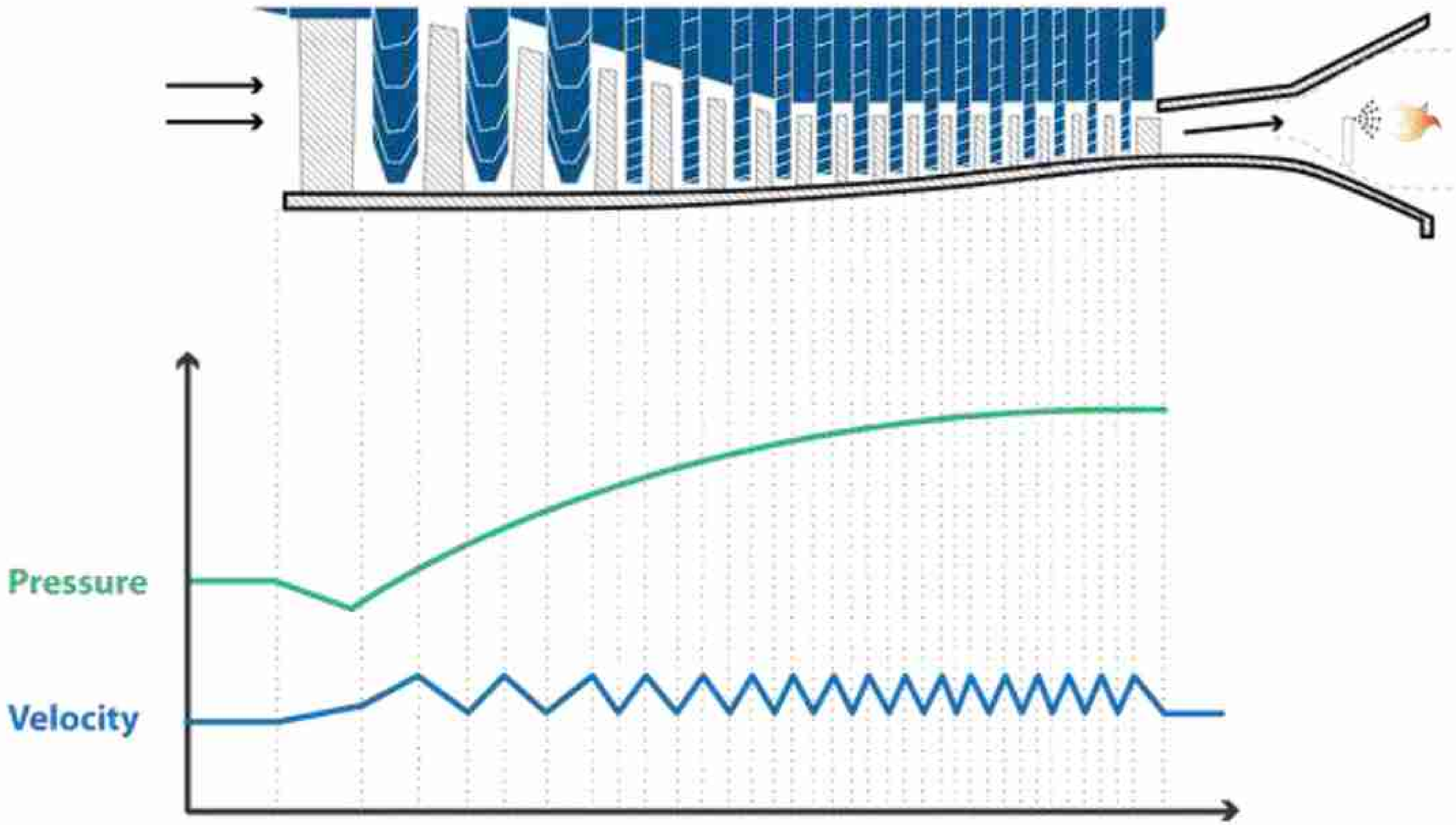
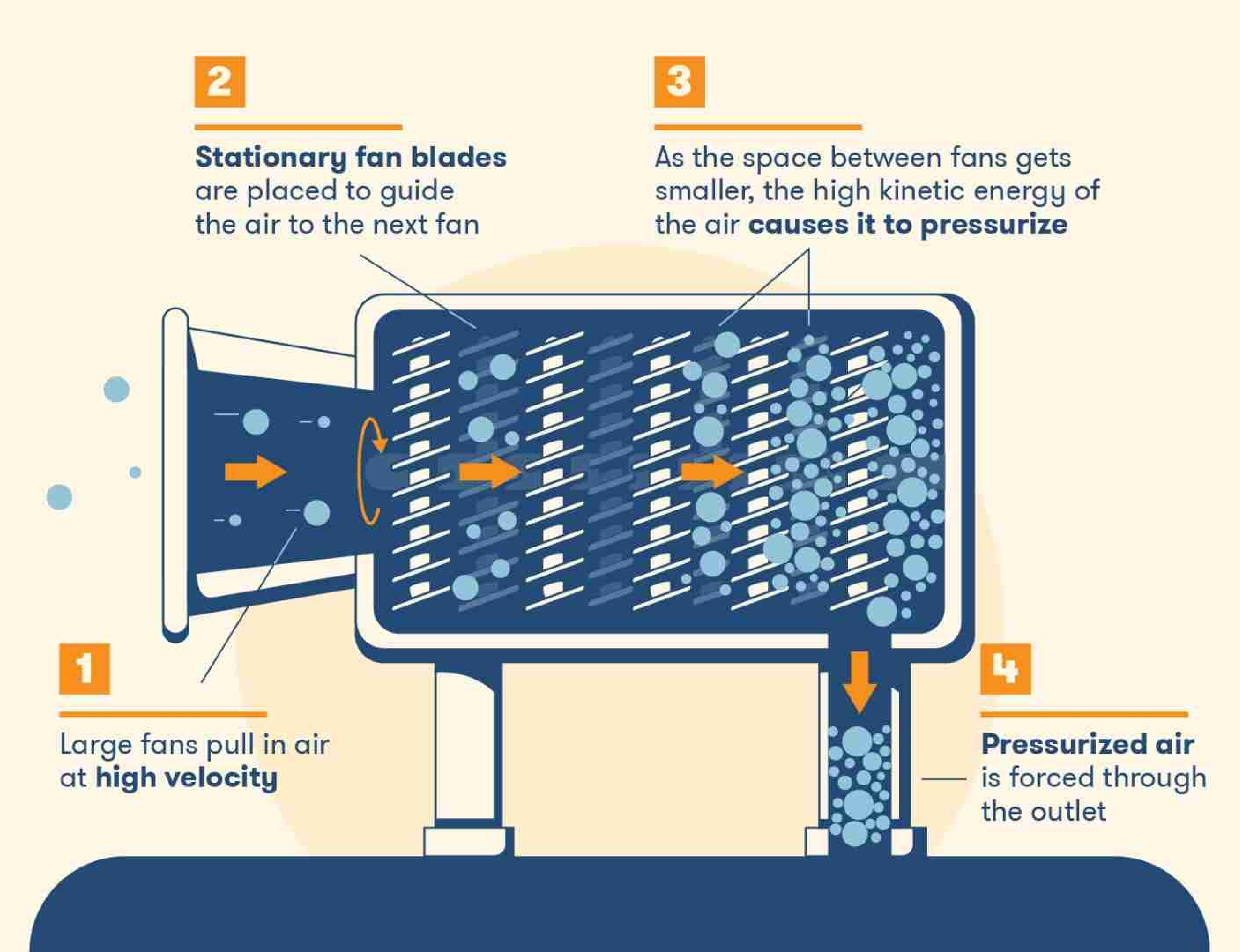


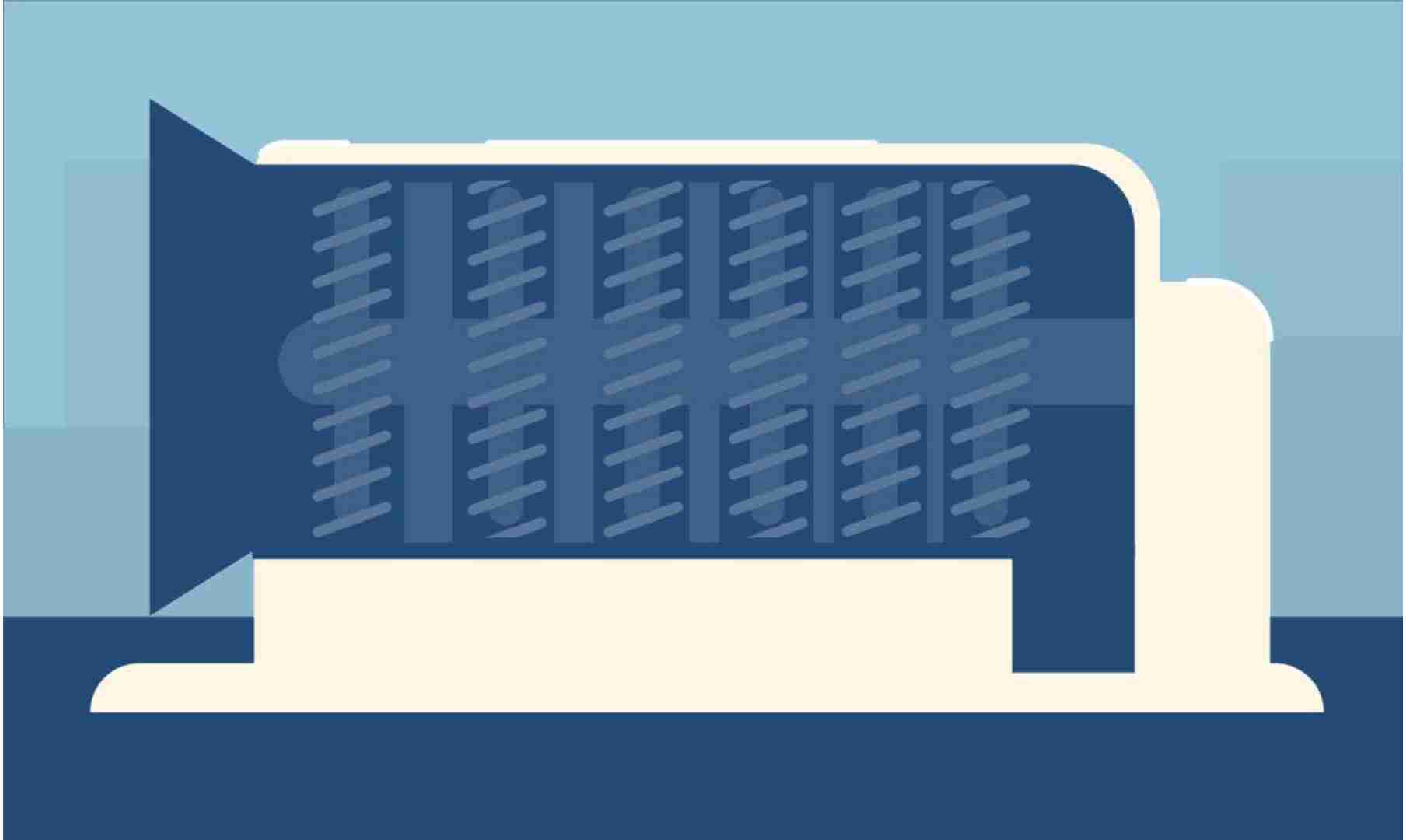
Fig. Axial flow compressor.

PV Changes in an Axial Compressor





2
Stationary fan blades are placed to guide the air to the next fan
3
As the space between fans gets smaller, the high kinetic energy of the air **causes it to pressurize**
1
Large fans pull in air at **high velocity**
4
Pressurized air is forced through the outlet



The *"fixed blades"* serve the following two purposes :

(i) Convert a part of the K.E. of the fluid into pressure energy. This conversion is achieved by diffusion process carried out in the diverge blade passages.

(ii) Guide and redirect the fluid flow so that entry to the next stage is without shock.

The blades are so arranged that the spaces between the blades form diffuser passages, and hence the velocity of the air relative to the blades is decreased as the air passes through them, and there is a rise in pressure. The air is then further diffused in the stator blades, which are also arranged to form diffuser passages. In the fixed stator blades the air is turned through an angle so that its direction is such that it can be allowed to pass to a second row of moving rotor blades. It is usual to have a relatively large number of stages and to maintain a constant work input per stage.

→ The necessary reduction in volume may be allowed by flaring the stator or by flaring the rotor. It is more common to use a flared rotor.

It is usually arranged to have an equal temperature rise in the moving and the fixed blades, and to keep the axial velocity of air constant throughout the compressor. Thus, each stage of the compression is exactly similar with regard to air velocity and blade inlet and outlet angles.

A diffusing flow is less stable than a converging flow, and for this reason the blade shape and profile is much more important for a compressor than for a reaction turbine. The design of compressor blades is based on aerodynamic theory and an aerofoil shape is used.

Note: Two forms of rotors have been used namely the ***drum*** and ***disc*** types. The disc type is used where consideration of low weight is more important than cost as in aircraft applications. The drum type is more suitable for static industrial applications. In some applications, combination of both types has been used.

Materials. The following materials are used for the various components of an axial flow compressor :

1. *Rotor bladings.* The materials listed below are in the *increasing* order of weight and their ability to withstand high temperature :

(*i*) Fibrous composites
(*ii*) Aluminium
(*iii*) Titanium
(*iv*) Steel
(*v*) Nickel alloy.

2. *Rotor :*

- For *rotor shafts* and disc .... "*steel.*"
- *Aircraft engines may use titanium at the front stages and "nickel alloy" in the rest.*

3. *Stator bladings :*

- Same materials as that of rotor but *steel is the most common.*

4. *Castings.* These may be of *cast magnesium*, *aluminium, steel or iron or fabricated from titanium or steel.*

— NC (Numerically controlled) machines make dies and the blades are manufactured by precision forging. Blades are also machined by CNC copying machines.

Velocity Diagrams and Work Done of a stage of Axial Flow Compressors :

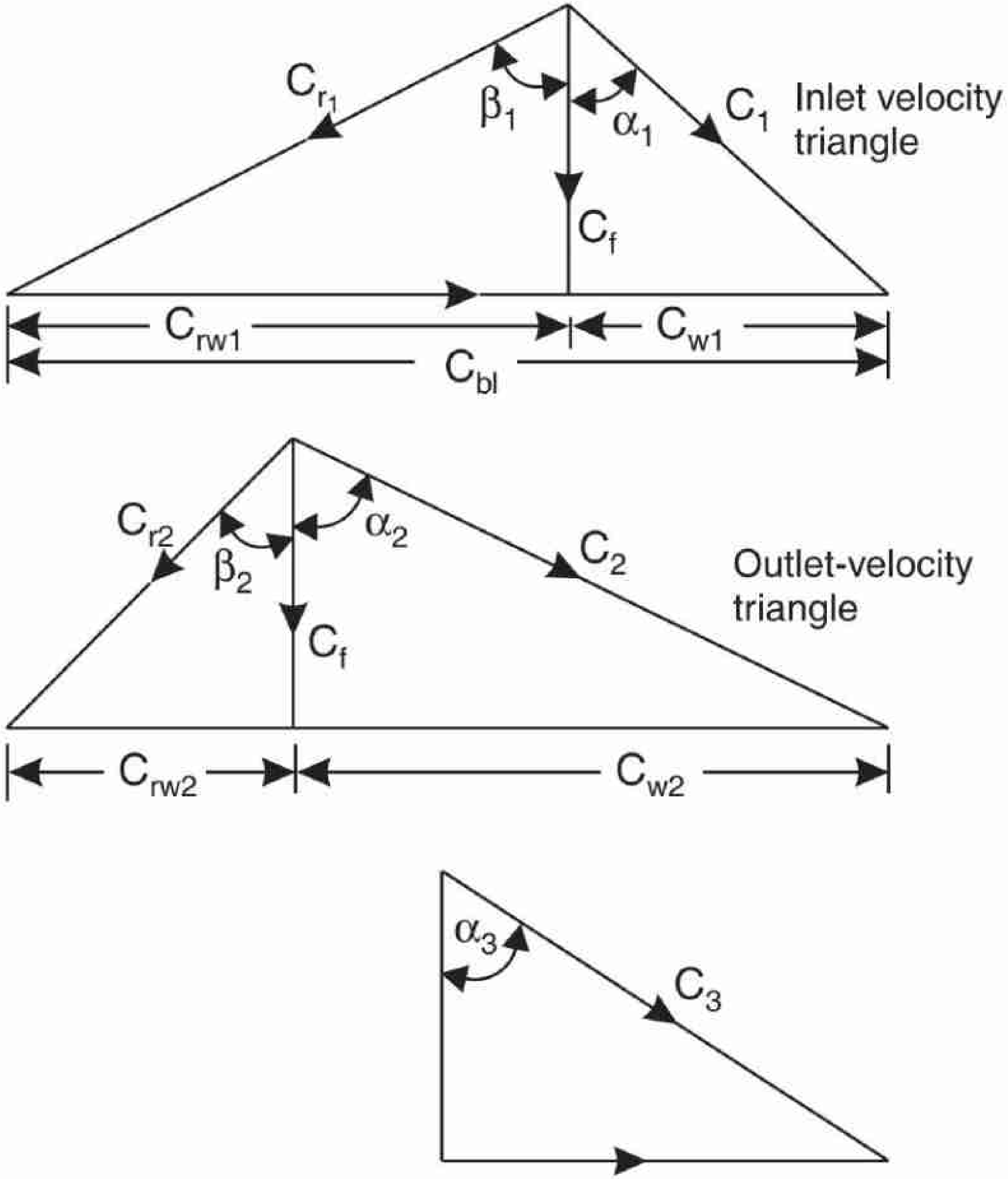
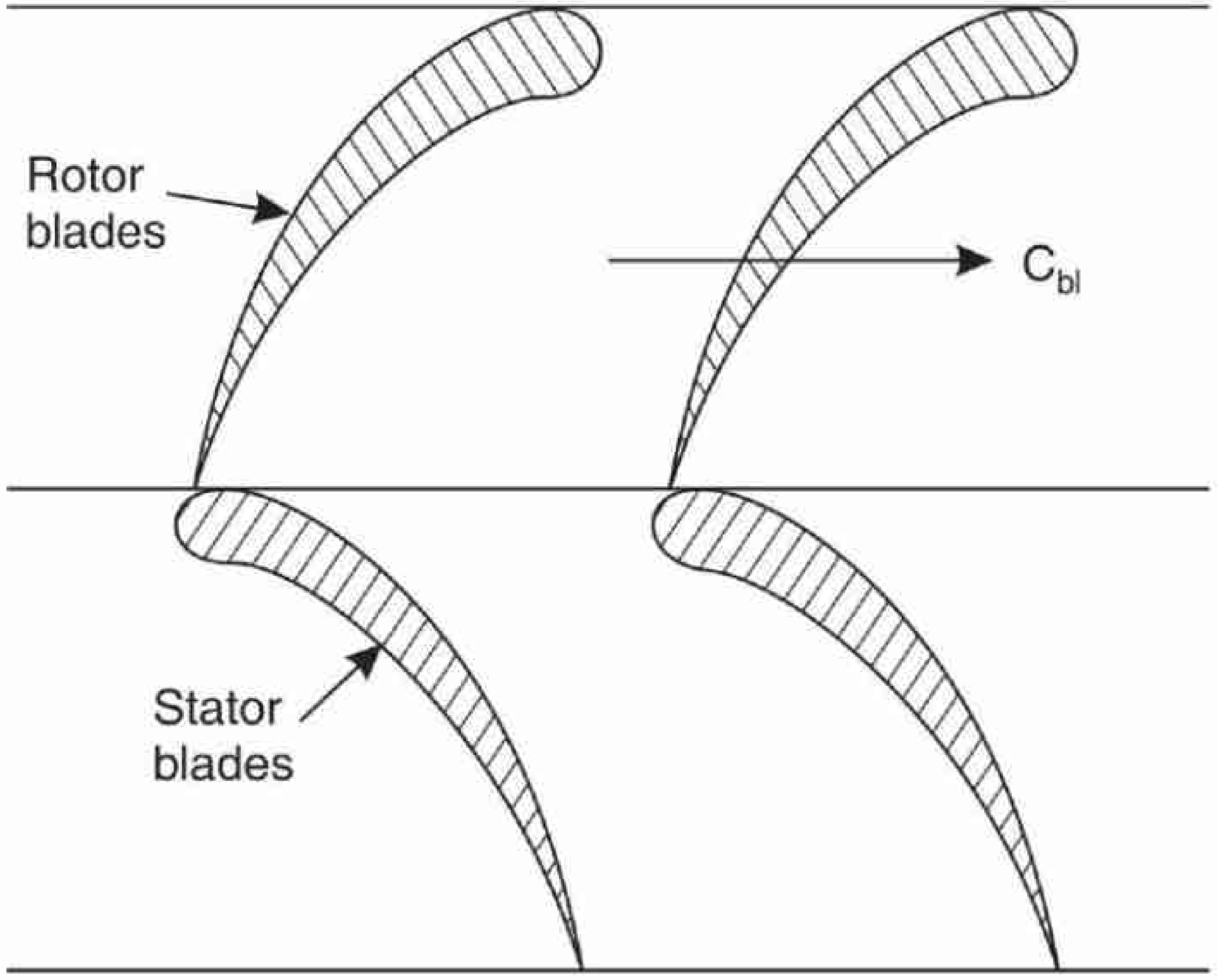


Fig. shows the velocity triangles for one stage of an axial flow compressor. All angles are measured from the axial direction and the blade velocity C_{bl} is taken to be same at blade entry and exist. This is because the air enters and leaves the blades at almost equal radii.



- Air approaches the rotor blade with absolute velocity C_1 and at an angle α_1. The relative velocity C_{r1}, obtained by the vectorial addition of absolute velocity C_1 and blade velocity C_{bl}, has the inclination β_1 with the axial direction.

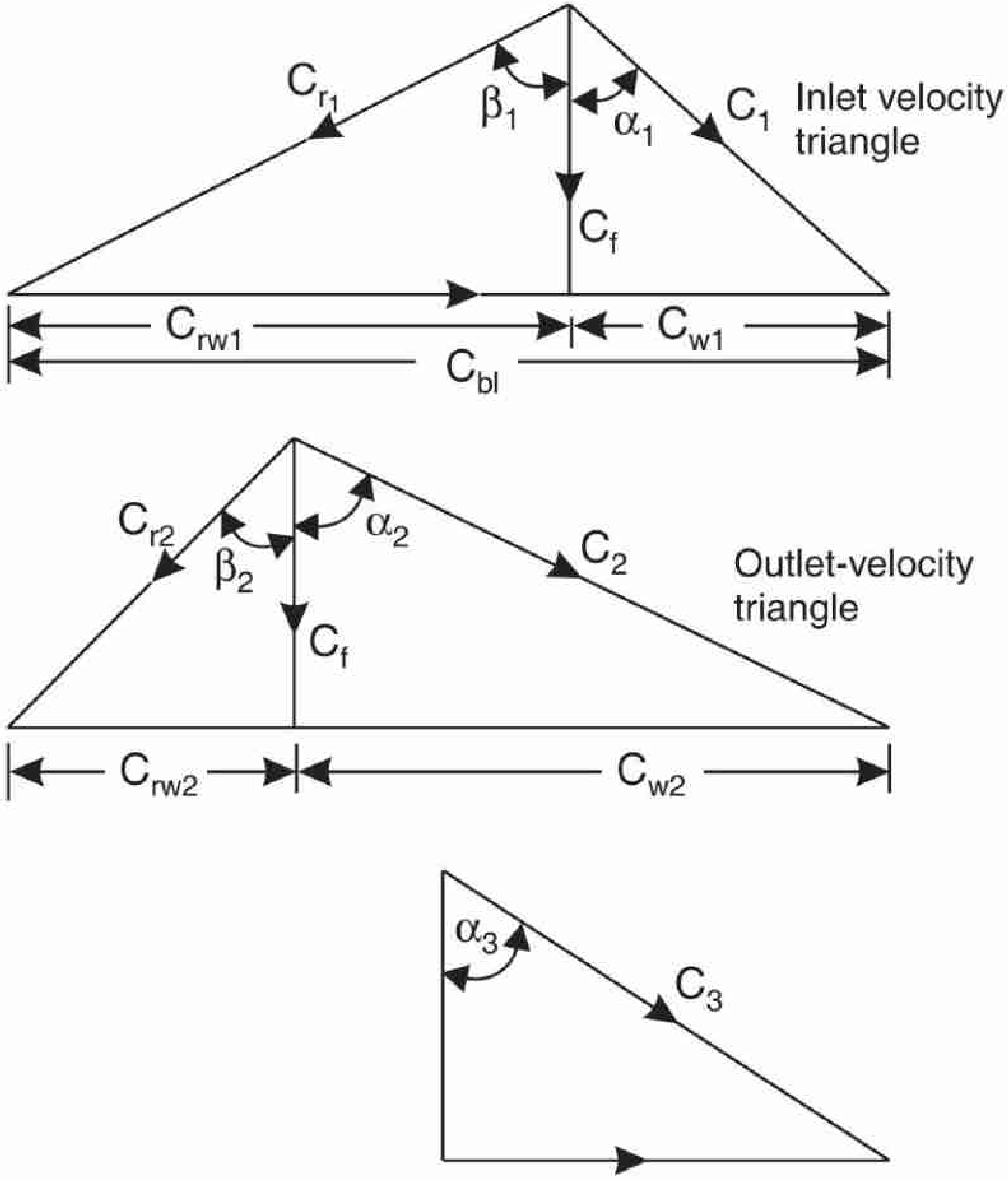
- Due to diffusion in the diverging passages formed by rotor blades, there is some pressure rise. This is at the expense of relative velocity and so the relative velocity decreases from C_{r1} to C_{r2}. Since work is being done on the air by rotor blades, the air would ultimately leave the rotor with increased absolute velocity C_2.

- The air then enters the stator blades, and the diffusion and deceleration takes place in the diverging passage of stator blades. Finally, the air leaves the stator blades with velocity C_3 at an angle α_3 and is redirected to the next stage. Generally, it is assumed that absolute velocity C_3 leaving the compressor stage equals the approach velocity C_1.

From the velocity triangles, we have

$$\frac{C_{bl}}{C_f} = \tan \alpha_1 + \tan \beta_1$$

$$\frac{C_{bl}}{C_f} = \tan \beta_2 + \tan \alpha_2$$



Work absorbed by the stage per kg of air,

$$W_{st} = C_{bl}(C_{w2} - C_{w1}) = c_p(T_{02} - T_{01})_{act}$$

By use of velocity triangle and cosine theorem,

$$W_{st} = \underbrace{\frac{C_{r1}^{\;2} - C_{r2}^{\;2}}{2}}_{\text{(First term)}} + \underbrace{\frac{C_2^{\;2} - C_3^{\;2}}{2}}_{\text{(Second term)}}$$

Here $\quad C_3 = C_1$

By use of velocity triangle and cosine theorem,

$$W_{st} = \underset{\text{(First term)}}{\frac{C_{r1}^{\ 2} - C_{r2}^{\ 2}}{2}} + \underset{\text{(Second term)}}{\frac{C_2^{\ 2} - C_3^{\ 2}}{2}}$$

Here $$C_3 = C_1$$

- The first term on the right side of the above equation introduces the part of the work supplied by a rotating cascade, which is converted into pressure due to diffusion action in rotating cascade itself.

- The second term represents the increment of K.E. in rotating cascade that has to be converted into pressure energy in stationary cascade. Comparing this equation to the work input to centrifugal compressor, we find that the term centrifugal action

$\left[\frac{C_{bl_2}^{\ 2} - C_{bl_1}^{\ 2}}{2}\right]$ is missing in axial flow compressors. Due to this reason the *pressure ratio per stage in axial flow compressor is much less than that of centrifugal compressor.*

The stage temperature rise, regardless of efficiency of compression, will be given by the equation

$$(\Delta T)_{act} = \frac{C_{bl} C_f}{c_p} (\tan \beta_1 - \tan \beta_2)$$

Pressure rise in isentropic flow through a cascade :

$$\therefore \quad (\Delta p)_{isen} = \frac{\rho}{2} (C_{w1}{}^2 - C_{w2}{}^2) = \frac{\rho}{2} C_f^2 (\tan^2 \alpha_1 - \tan^2 \alpha_2)$$

$$[\because \quad C_{w1} = C_f \tan \alpha_1, \text{ and } C_{w2} = C_f \tan \alpha_2]$$

Degree of Reaction:

Degree of reaction (R_d) is defined as the ratio of pressure rise in the compressor stage.

$$R_d = \frac{\text{Pressure rise in the rotor blades}}{\text{Pressure rise in the stage}}$$

$$R_d = \frac{1}{2} \frac{C_f}{C_{bl}} (\tan \beta_1 + \tan \beta_2)$$

Degree of reaction is usually kept as 0.5,

$$\frac{C_{bl}}{C_f} = \tan \beta_1 + \tan \beta_2 = \tan \alpha_1 + \tan \beta_1 = \tan \alpha_2 + \tan \beta_2$$

From this $\alpha_1 = \beta_2 \; ; \alpha_2 = \beta_1$

So with 50% reaction blading, the compressors have symmetrical blades and with this type of set-up losses in flow path are amply reduced.

In symmetrical blades, the tip clearance and fluid friction losses are minimum.

Surging, Choking and Stalling—Compressor Characteristics

Surging: In axial flow and centrifugal compressors ***"surging"*** is an unstable limit of operation. *Surging is caused due to unsteady, periodic and reversal of flow through the compressor when the compressor must operate at less mass flow rate than a predetermined value* (a value corresponding to maximum pressure). As the flow is drastically reduced than this predetermined value, this surge can reach such a magnitude as to endanger the compressor and, in many cases, mechanical failures may result. The alternating stresses to which the rotor of the machine is subjected during this irregular working condition, may damage compressor bearings, rotor blading and scales. Severe surge have been known to bend the rotor shaft.

Choking: When the pressure ratio is unity (i.e., there is no compression), theoretically mass flow rate becomes maximum. This generally occurs when the Mach number corresponding to relative velocity at inlet becomes sonic. The maximum mass flow rate possible in compressor is known as ***choking flow***. "Choking" means fixed mass flow rate regardless of pressure ratio (i.e., characteristic becomes vertical).

"Stalling" of a stage of axial flow compressor is defined as the aerodynamic stall or the breakway of the flow from suction side of the blade aerofoil. It may be due to lesser flow rate than designed value or due to non-uniformity in the blade profile. Thus stalling is ahead phenomenon of surging.

A multi-stage compressor may operate stable in the unsurged region with one or more of the stages stalled and rest of the stages unstalled. In other words, stalling is a local phenomenon whereas surging is a complete system phenomena

Example *An axial flow compressor having eight stages and with 50% reaction design compresses air in the pressure ratio of 4 : 1. The air enters the compressor at 20°C and flows through it with a constant speed of 90 m/s. The rotating blades of compressor rotate with a mean speed of 180 m/s. Isentropic efficiency of the compressor may be taken as 82%. Calculate :*

(i) Work done by the machine *(ii) Blades angles.*

Assume γ = 1.4 and c_p = 1.005 kJ/kg K.

Solution.
$$\eta_{isen} = \frac{T_2' - T_1}{T_2 - T_1}$$

Also
$$\frac{T_2'}{T_1'} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (4)^{\frac{1.4-1}{1.4}} = (4)^{\frac{0.4}{1.4}} = 1.486$$

$\therefore$
$$T_2' = (20 + 273) \times 1.486 = 435.4 \text{ K}$$

$\therefore$
$$\eta_{isen} = \frac{435.4 - 293}{T_2 - 293} \quad \text{or} \quad 0.82 = \frac{142.4}{T_2 - 293}$$

$\therefore$
$$T_2 = \frac{142.4}{0.82} + 293 = 466.6 \text{ K}$$

Work required/kg $= c_p(T_2 - T_1) = 1.005(466.6 - 293) =$ **174.47 kJ/kg. (Ans.)**

Now, work done/kg $= \text{Number of stages} \times C_{bl}(C_{w_2} - C_{w_1})$

$$174.47 = 8 \times C_{bl}\, C_f\, (\tan \alpha_2 - \tan \alpha_1)$$

$$\therefore \quad \tan \alpha_2 - \tan \alpha_1 = \frac{174.47 \times 1000}{8 \times 180 \times 90} = 1.346$$

For 50% reaction blading, $\alpha_2 = \beta_1$ and $\alpha_1 = \beta_2$

$$\therefore \quad 1.346 = \tan \beta_1 - \tan \alpha_1$$

Now, $$\tan \alpha_1 + \tan \beta_1 = \frac{C_{bl}}{C_f} = \frac{180}{90} = 2$$

i.e., $$\tan \beta_1 - \tan \alpha_1 = 1.346$$

$$\tan \beta_1 + \tan \alpha_1 = 2$$

From (*i*) and (*ii*), we get

$$2 \tan \beta_1 = 3.346$$

$\therefore$ $\beta_1 = \mathbf{59.1°} = \alpha_2$. **(Ans.)**

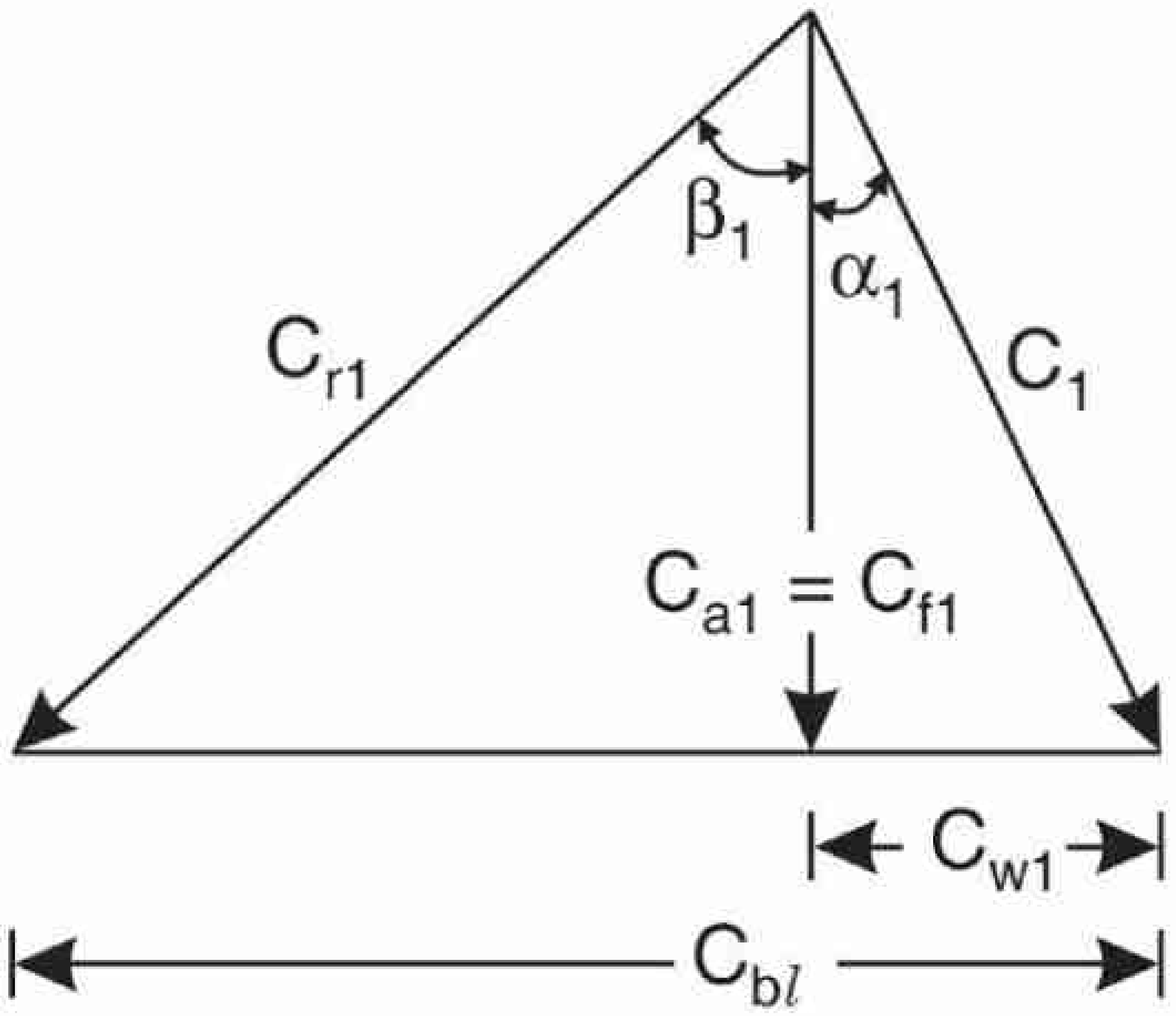
and $\alpha_1 = \mathbf{18.1°} = \beta_2$. **(Ans.)**

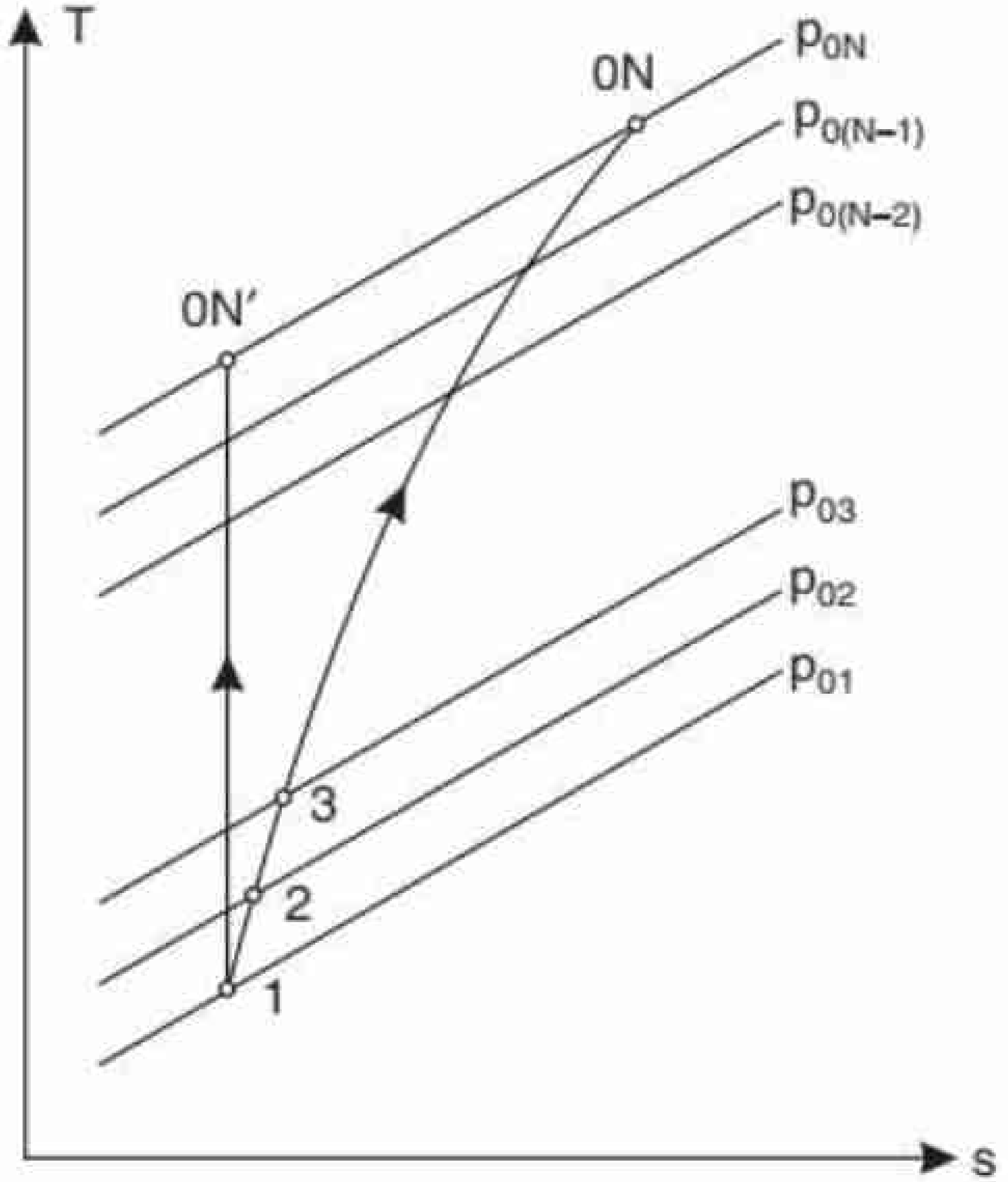
Example *In an eight stage axial flow compressor, the overall stagnation pressure ratio achieved is 5 : 1 with an overall isentropic efficiency of 92 per cent. The inlet stagnation temperature and pressure at inlet are 290 K and 1 bar. The work is divided equally between the stages. The mean blade speed is 160 m/s and 50% reaction design is used. The axial velocity through the compressor is constant and is equal to 90 m/s. Calculate :*

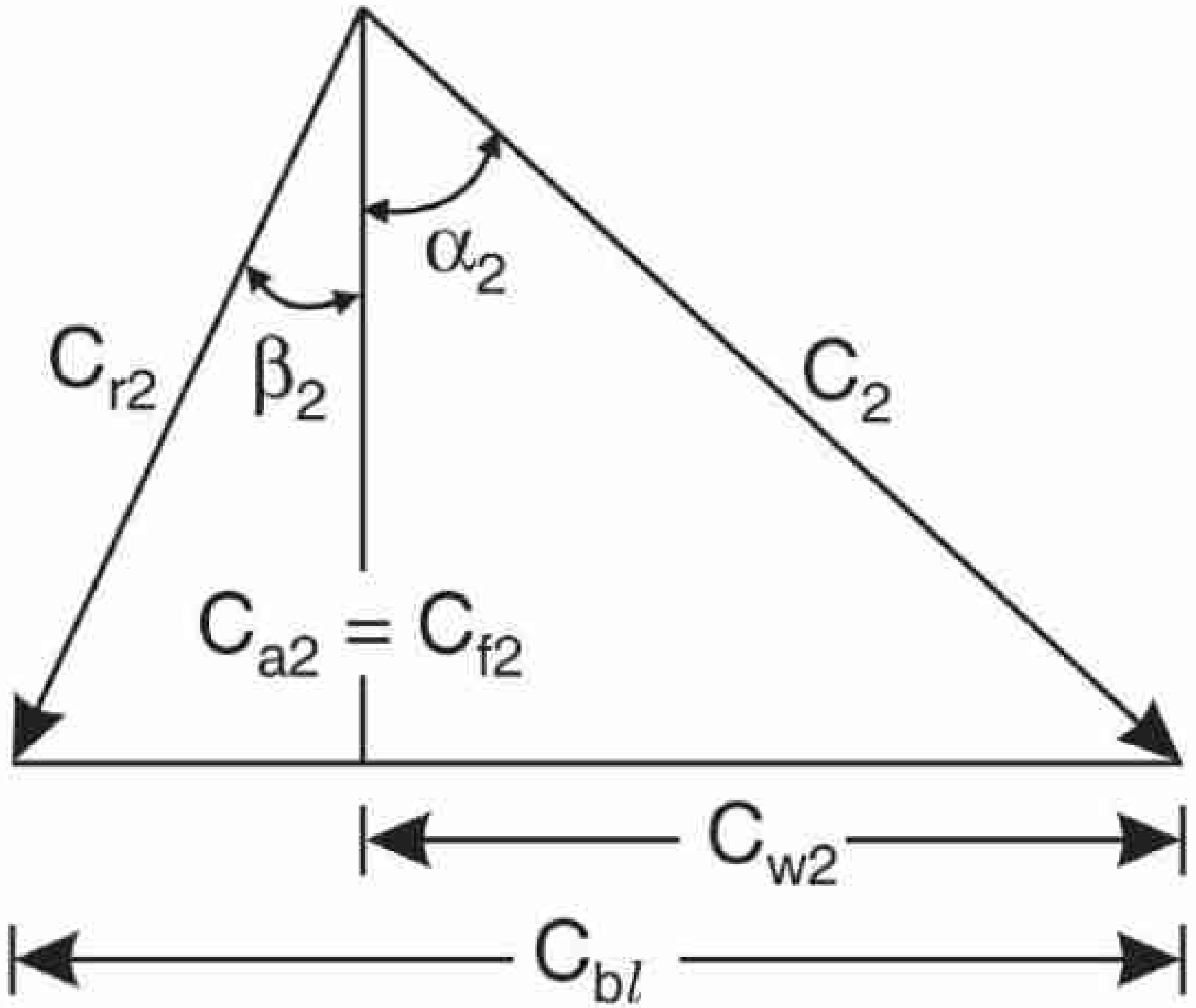
(*i*) *The blade angles.* (*ii*) *The power required.*

Solution. *Given* : $N = 8$; $r_p = 5 : 1$; $\eta_{isen} = 92\%$; $T_{01} = 290$ K ; $p_{01} = 1$ bar ; $C_{bl} = 160$ m/s ; Degree of reaction = 50% ; $C_f = 90$ m/s.

(i) The blade angles, α_1, β_1, α_2, β_2







Refer to Fig. 69 for velocity diagrams. Since the degree of reaction is 50% reaction, the *blades are symmetrical* and hence the velocity diagrams are *identical*. Thus

$$\alpha_1 = \beta_2 \text{ and } \alpha_2 = \beta_1$$

Let suffix N denotes the *number of stages*.

With isentropic compression the temperature of air leaving the compressor stage is

$$T_{0N}' = T_{01}\left(\frac{p_{0N}}{p_{01}}\right)^{(\gamma-1/\gamma)} = 290 \times (5)^{\frac{1.4-1}{1.4}} = 459.3 \text{ K}$$

But
$$\eta_{isen} = \frac{T_{0N}' - T_{01}}{T_{0N} - T_{01}}$$

$$0.92 = \frac{459.3 - 290}{T_{oN} - 290}$$

$$\therefore \quad T_{0N} = \frac{459.3 - 290}{0.92} + 290 = 474 \text{ K}$$

The work consumed by the compressor

$$= c_p(T_{0N} - T_{01}) = (C_{w2} - C_{w1})\, C_{bl} \times \text{N}$$

or
$$c_p(T_{0N} - T_{01}) = C_f\,(\tan \alpha_2 - \tan \alpha_1)\, C_{bl}\, .\, N$$

$$\therefore \qquad \tan \alpha_2 - \tan \alpha_1 = \frac{c_p(T_{0N} - T_{01})}{C_f . C_{bl} . N} = \frac{1.005\,(474 - 290) \times 10^3}{90 \times 160 \times 8} = 1.605 \qquad \text{...}(i)$$

From velocity triangles, we have

$$\frac{C_{bl}}{C_f} = \tan \alpha_1 + \tan \beta_1 = \frac{160}{90} = 1.778 \qquad \text{...}(ii)$$

Adding (*i*) and (*ii*), we get

$$\tan \beta_1 = \frac{1.605 + 1.778}{2} = 1.6915 \qquad (\because \quad \alpha_1 = \beta_1)$$

or
$$\beta_1 = \tan^{-1}(1.6915) = 59.4°$$

$$\therefore \qquad \boldsymbol{\beta_1 = \alpha_2 = 59.4°.} \quad \textbf{(Ans.)}$$

Putting the value of tan β_1 in (*ii*), we have

$$\tan \alpha_1 + 1.6915 = 1.778$$

or $$\tan \alpha_1 = 0.0865 \quad \text{or} \quad \alpha_1 = \tan^{-1} (0.0865) = 4.94°$$

$\therefore$ $$\alpha_1 = \beta_2 = \mathbf{4.94°.} \quad \textbf{(Ans.)}$$

(*ii*) **The power required by compressor P :**

$$P = \dot{m}\ c_p(T_{0N} - T_{01})$$

$$= 1 \times 1.005(474 - 290) = \mathbf{184.9\ kW.} \quad \textbf{(Ans.)}$$

GAS TURBINES

APPLIED THERMODYNAMICS
B20ME2202
R20
2/4 SEM-II

A lecture by
V.Manikanth M.Tech.,(Ph.D.)
Assistant Professor
Dept. Of Mechanical Engineering
SRKR Engineering College (A)

Estd. in 1980
SAGI RAMA KRISHNAM RAJU ENGINEERING COLLEGE
AUTONOMOUS INSTITUTION | APPROVED BY AICTE, NEW DELHI | AFFILIATED TO JNTUK, KAKINADA
6 PROGRAMMES ACCREDITED BY NBA | ACCREDITED BY NAAC | RECOGNIZED AS SIRO | ARIIA & NIRF RANKED INSTITUTE

UNIT-V

Gas Turbines: *Simple gas turbine plant- Ideal cycle, closed cycle and open cycle for gas turbines Efficiency, work ratio and optimum pressure ratio for simple gas turbine cycle Parameters of performance- Actual cycle, regeneration, Inter-cooling and reheating, semi-closed cycle*

Jet propulsion and Rockets.

Reference Books:

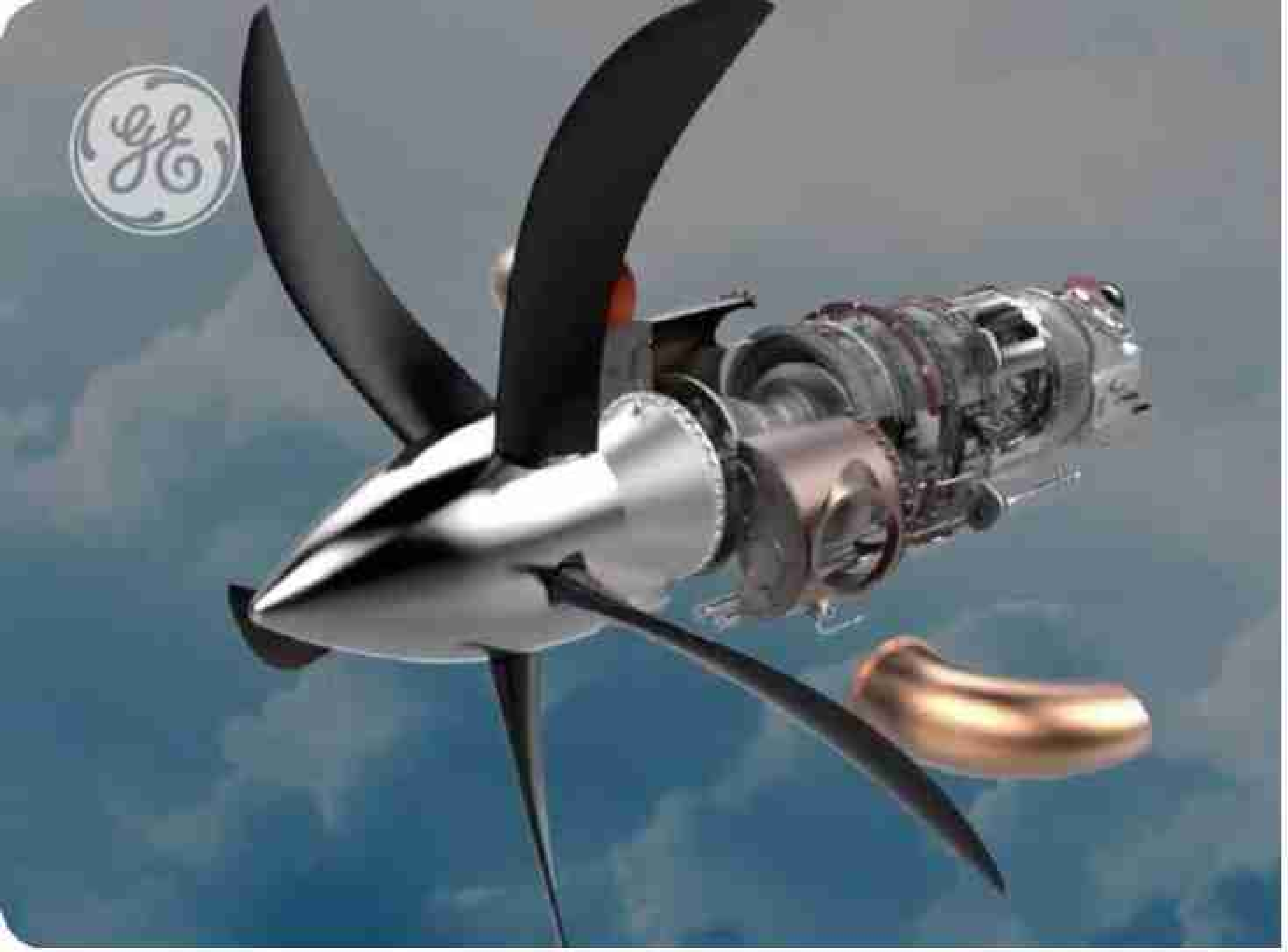
1. Gas Turbines by V. Ganesan.
2. Thermal Engineering, by R.K.Rajput
3. Gas Turbines, by Cohen and Rogers.

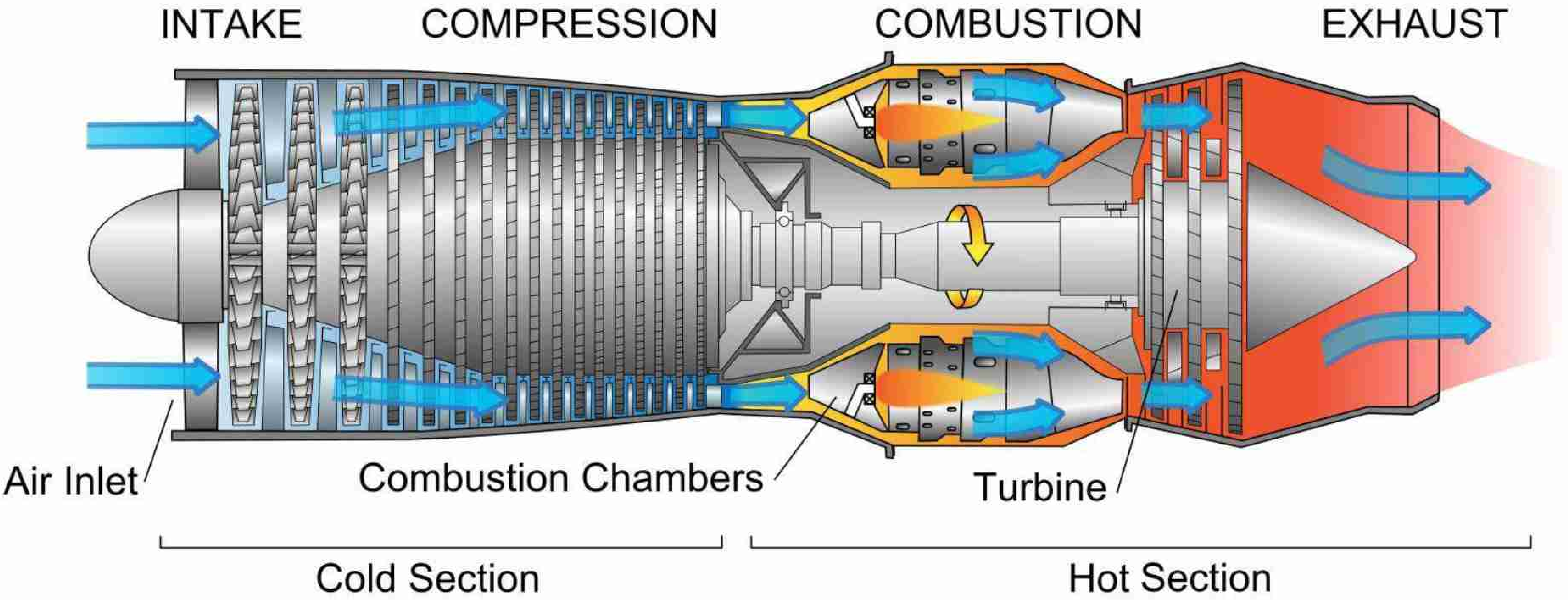
GAS TURBINE:

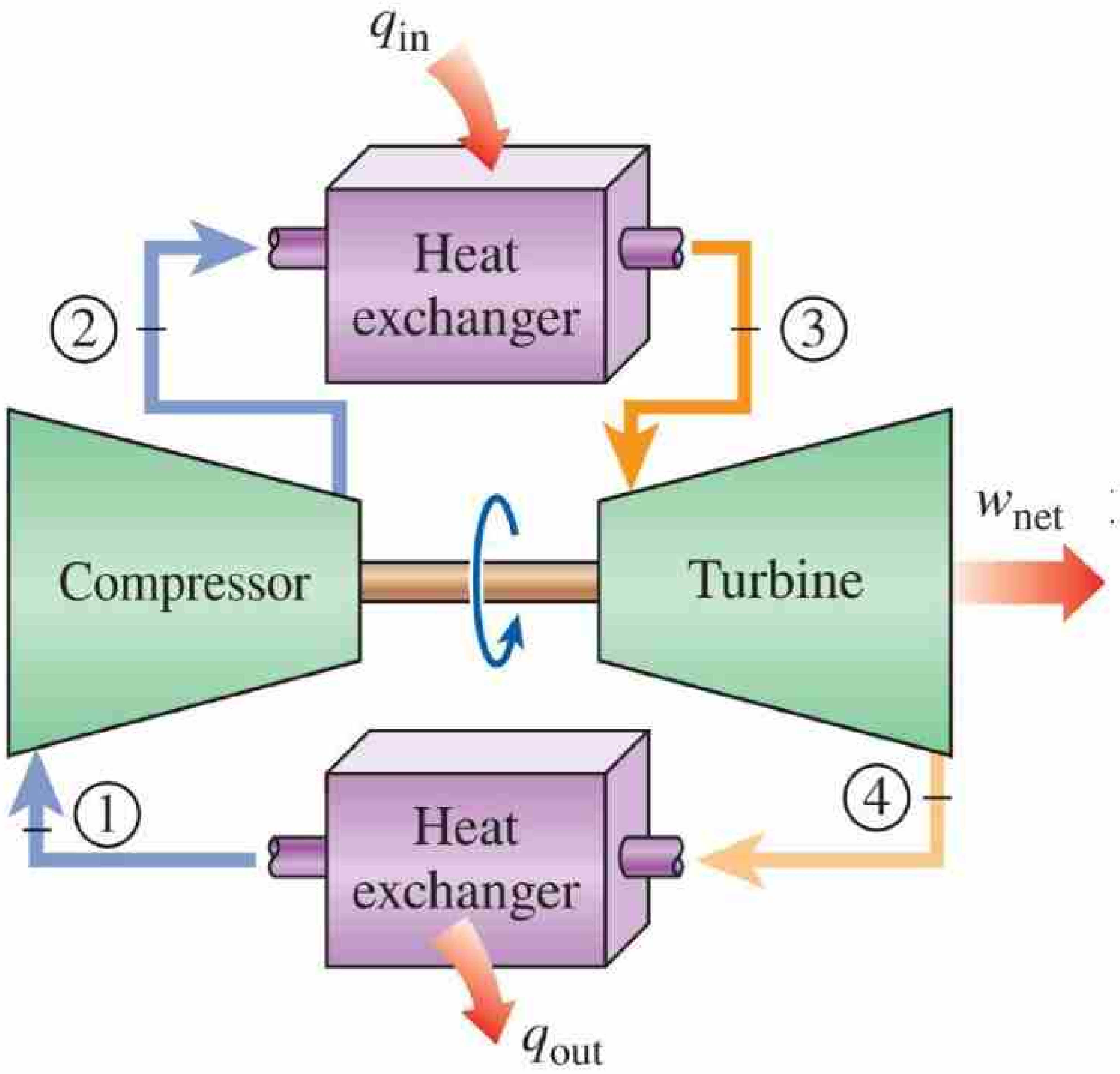
The gas turbine is an air breathing heat engine, said to be the heart of the power plant produces electric power, by burning of gas (or) liquid fuels along with fresh air. The fresh air performs two main functions in gas turbine. The fresh air acts as a cooling agent for various parts of the power plants and gives required amount of oxygen for combustion of fuel.

The gas turbine engines have application for aircraft propulsion and electric power generation. In the case of aircraft propulsion, the gas turbine supplies power to run the compressor and the generator to power the auxiliary equipment's. The gas turbine cycle can be applied as closed cycle in nuclear power plants.

@V.Manikanth, Asst. Prof., Dept. of Mechanical, SRKREC(A).




INTAKE
COMPRESSION
COMBUSTION
EXHAUST
Air Inlet
Combustion Chambers
Turbine
Cold Section
Hot Section



A closed-cycle gas-turbine engine.

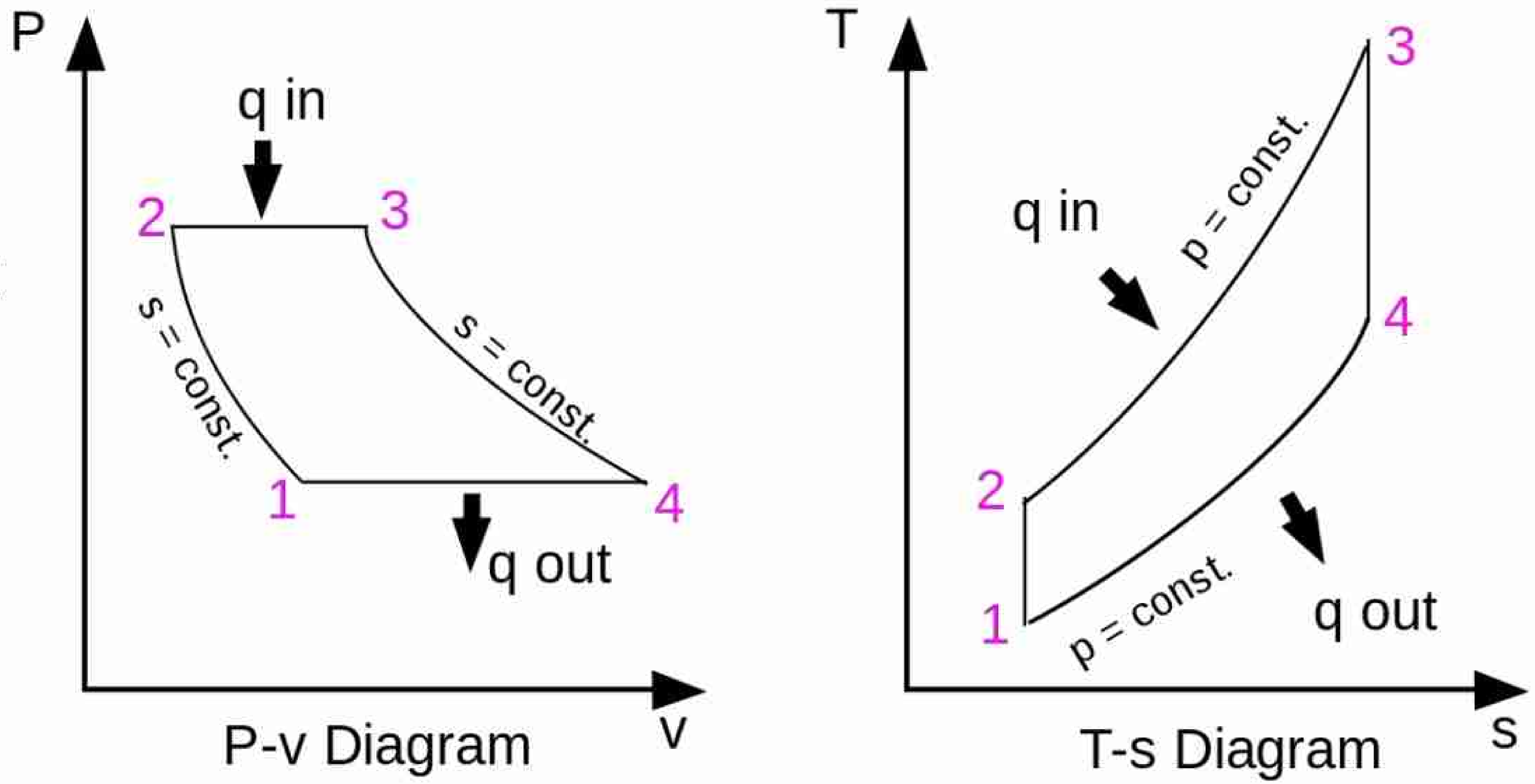
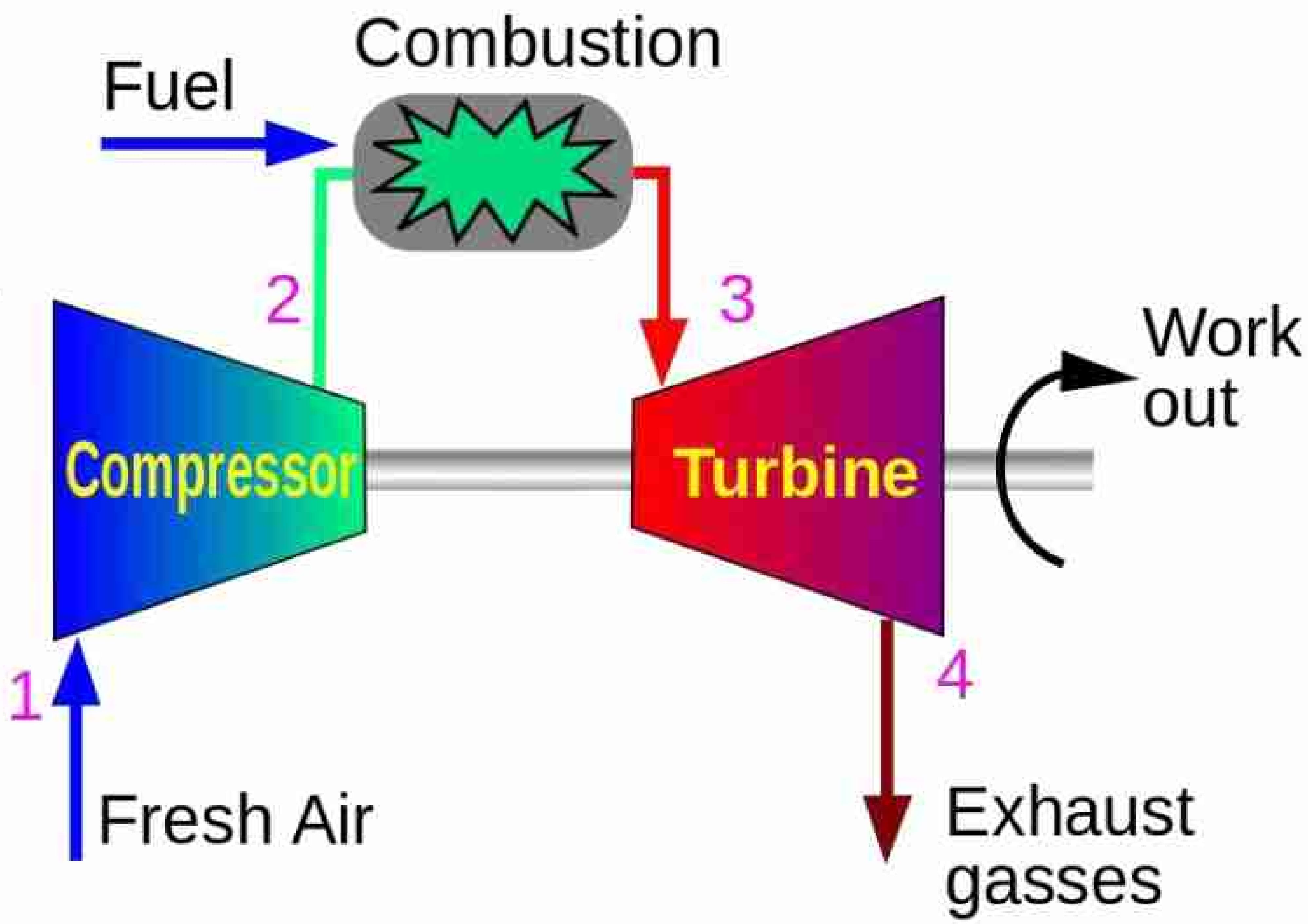


Fig: Brayton cycle

COMPONENTS OF GAS TURBINE PLANT



The gas turbine plant comprises of three important components.

(i) A compressor

(ii) Combustion chamber

(iii) Turbine

The fresh air is compressed isentropically in the compressor, then mixed with fuel and burned by combustor under constant pressure condition in the combustion chamber. This process is said to be constant pressure heat addition. The obtained hot flue gas expands through the turbine isentropically. Maximum amount of power obtained from the turbine is used to run the compressor and the remaining power is used for useful work.

CLASSIFICATION OF GAS TURBINE PLANTS:

Gas turbine plants are classified as

Based on operation

(a) Open cycle

(b) Closed cycle

(c) Semi closed cycle

Based on Arrangement

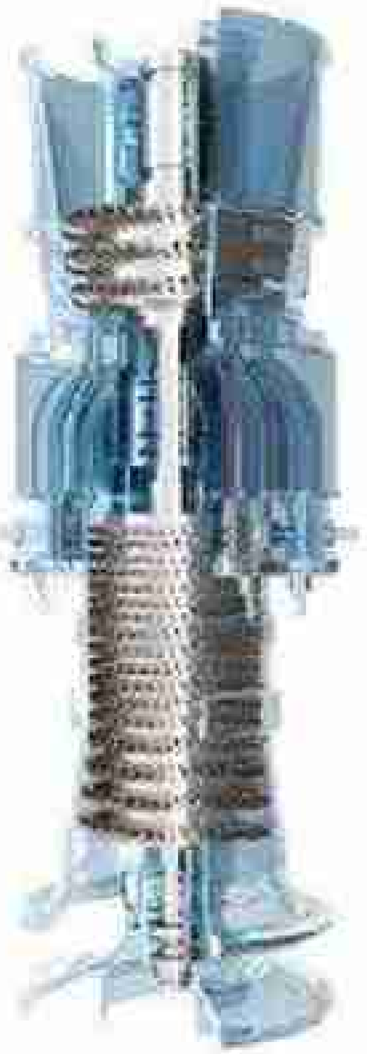
(a) Simple

(b) Reheat

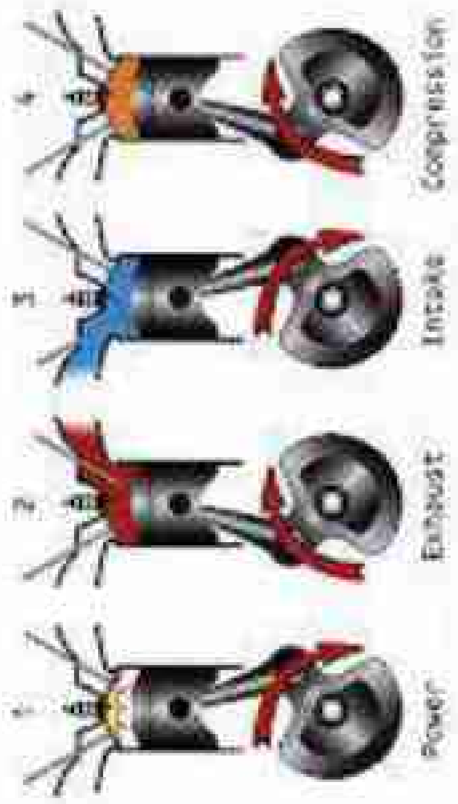
(c) Regenerative

(d) Intercooled

GAS TURBINE



VS



I.C ENGINE

Comparison between I.C. Engines and Gas Turbines

	I.C. Engines	Gas Turbines
1.	Ignition and lubrication systems is complicated as compared to gas turbines.	Ignition and lubrication systems is simpler as compared to IC. engines.
2.	Weight per HP developed is greater.	Weight per HP developed is lesser.
3.	Exhaust gas pollution is more.	Exhaust gas pollution is comparatively less polluting since excess air is used for combustion.
4.	Thermal efficiency is higher compared to simple gas turbine.	Thermal efficiency is lesser than I.C. engine for simple turbine cycle.
5.	Fuel cost comparatively costlier fuel required.	Fuel cost comparatively cheaper fuel can be used.
6.	Work developed per kg of air is lesser.	Work developed per kg of air are more.
7.	Mechanical efficiency is lower than gas turbine.	Mechanical efficiency is higher than I.C. engine.
8.	It produces lesser exhaust gases.	It produces exhaust gases five times greater than IC. engines
9.	Requirement of flywheel is a must.	Requirement of flywheel is not required.
10.	IC. engine can't be driven at higher speeds.	Gas turbine can be driven at higher speeds (40.000 rpm).
11.	Starting is easier.	Starting is difficult.
12.	Fuel control is comparatively easier.	Fuel control is comparatively difficult due to wide operating speeds.

@V.Manikanth, Asst. Prof, Dept. of Mechanical, SRKREC(A).

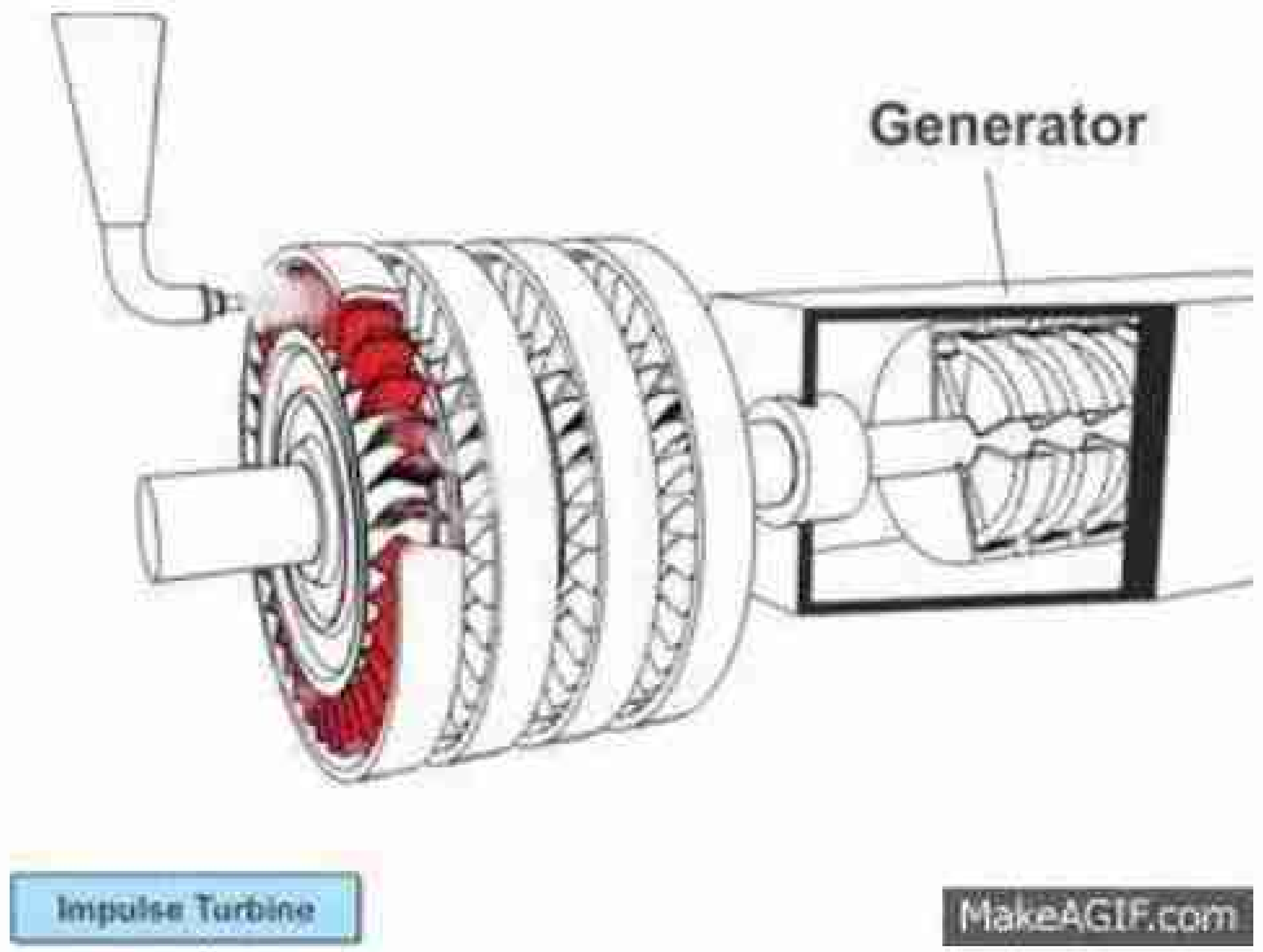
	Reciprocating engines	Gas Turbines
1	Larger size & weight	Much smaller in size & weight
2	Higher efficiency for medium range Only (1-10 MW)	Lower efficiency in medium range but higher efficiency for range of power >10 MW
3	The engine put in operation very fast	Startup time is longer
4	Not affected by weather condition	Extremely affected by weather condition
5	Cheap in comparison	Very expensive
6	Required reinforced foundation to counter unbalance force produced from reciprocating motion	Lower vibration energy because of its direct rotary motion
7	Produce higher vibration level & noise	More smooth (lower vibration level & noise)
8	Practical for low speed - high torque applications (cannot used in high speed)	preferred for high speed - low torque applications
9	Lower fuel consumption	Higher fuel consumption
10	Useful in application which need higher power to heat ratio	Useful in application which need lower power to heat ratio
11	Supplied fuel pressure is lower	Supplied fuel pressure is higher
12	Ability to burn even heavy fuels	Gas turbines firing heavy fuels is uncommon
13	Fuel consumption is proportional to the power output	Consume more fuel when they are idling (compressor work)
14	Can be operated with variable load but with good fuel consumption	Better fuel consumption in a constant load rather than a fluctuating load
15	Required higher maintenance cost (more moving parts)	Required lower maintenance cost
16	Limited power output (up to 15 MW)	Much higher power output (up to 258 MW as land base)

@V.Manikanth, Asst. Prof., Dept. of Mechanical, SRKREC(A).

Merits over steam turbines :

The gas turbine entails the following *advantages over steam turbines* :

1. Capital and running cost less.
2. For the same output the space required is far less.
3. Starting is more easy and quick.
4. Weight per H.P. is far less.
5. Can be installed anywhere.
6. Control of gas turbine is much easier.
7. Boiler along with accessories not required.



PARAMETER OF PERFORMANCE

The important parameters required to analyze the performance of a gas turbine are

(i) Pressure ratio: Ratio of maximum cycle pressure to minimum cycle pressure. (P_{max}/P_{min})

(ii) Work ratio: Ratio of Network output to the total work obtained in the turbine.

(iii) Air ratio: Mass of air entering the compressor per unit net output of cycle.

(iv) Compression efficiency: Ratio of work required for ideal compression of air for a known pressure range to work actually used by the compressor.

(v) Engine efficiency: Ratio of actual work obtained by turbine during expansion of working substance for a known pressure range to that of work obtained for ideal expansion conditions.

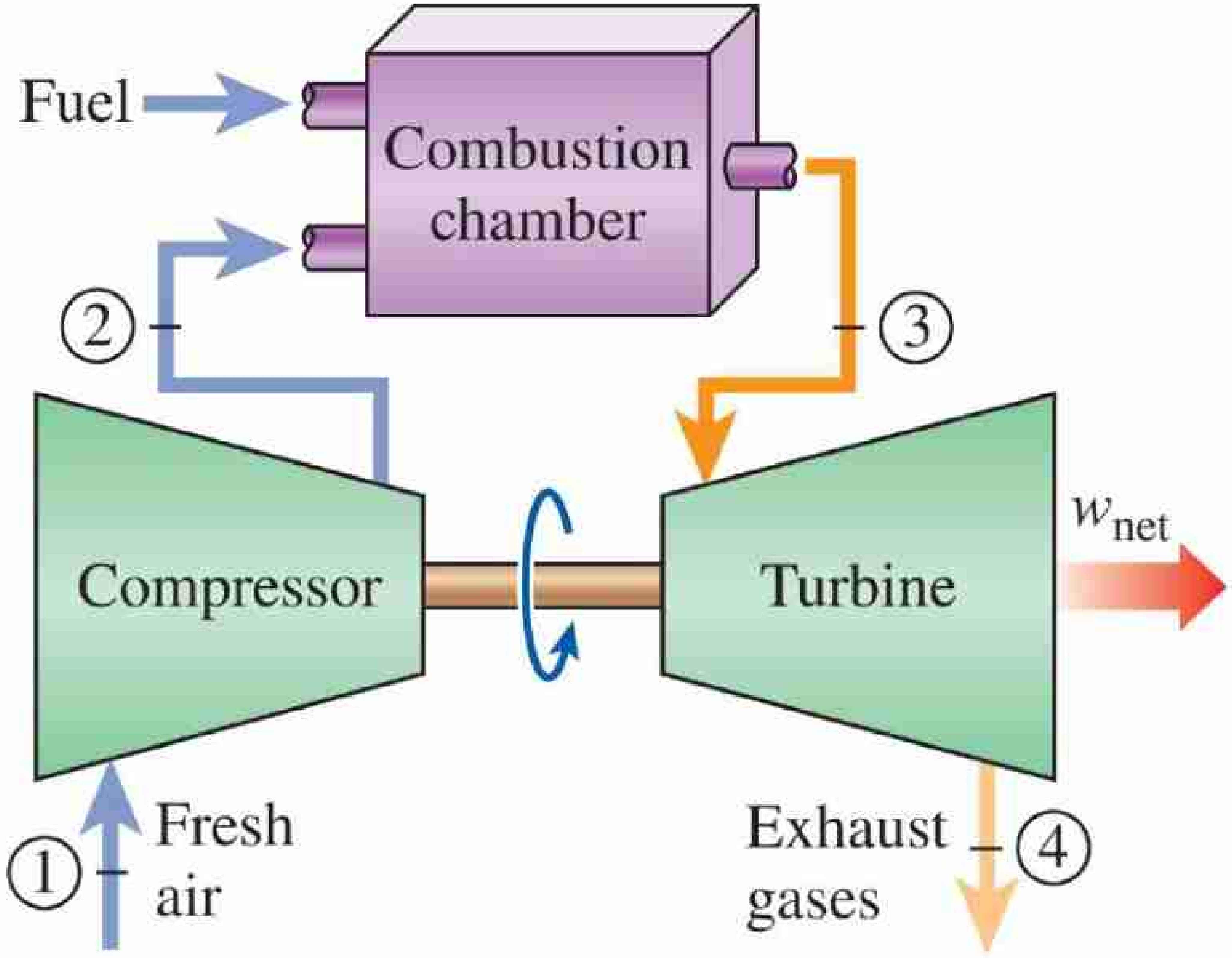
(vi) Combustion efficiency: Ratio of actual heat release by 1 kg of fuel to heat released by complete combustion.

(vii) Thermal efficiency: Percentage of total input energy converting as net work output of the cycle.

@V.Manikanth, Asst. Prof., Dept. of Mechanical, SRKREC(A).

IDEAL GAS TURBINE CYCLE (Brayton Cycle) :

I. Open cycle



An open-cycle gas-turbine engine.

The various processes in this open cycle include

(a) Isentropic compression (b) Constant pressure heat addition (c) Isentropic expansion

II) Closed cycle

Brayton cycle is called as External Combustion Engines Cycle, because the burning takes place outside the cylinder (or turbine).

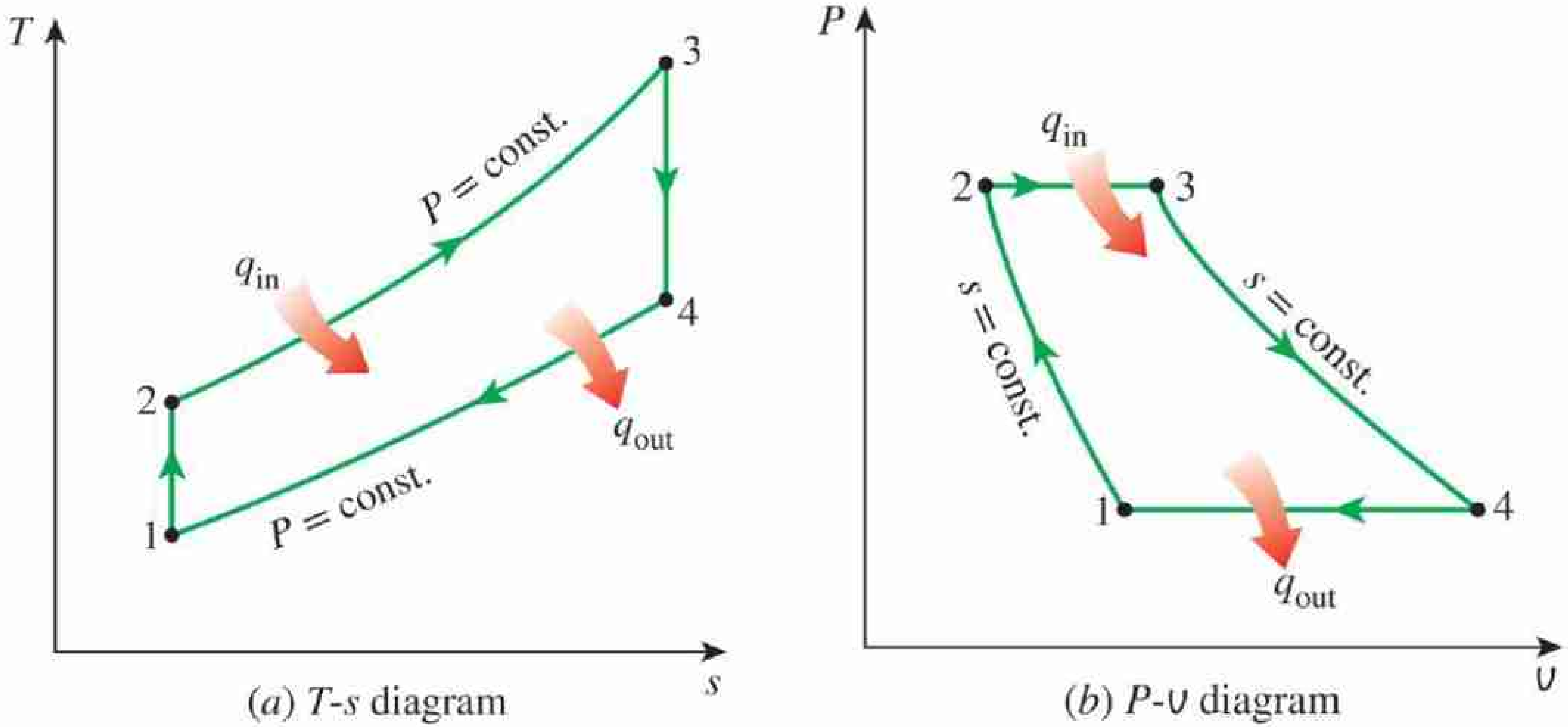
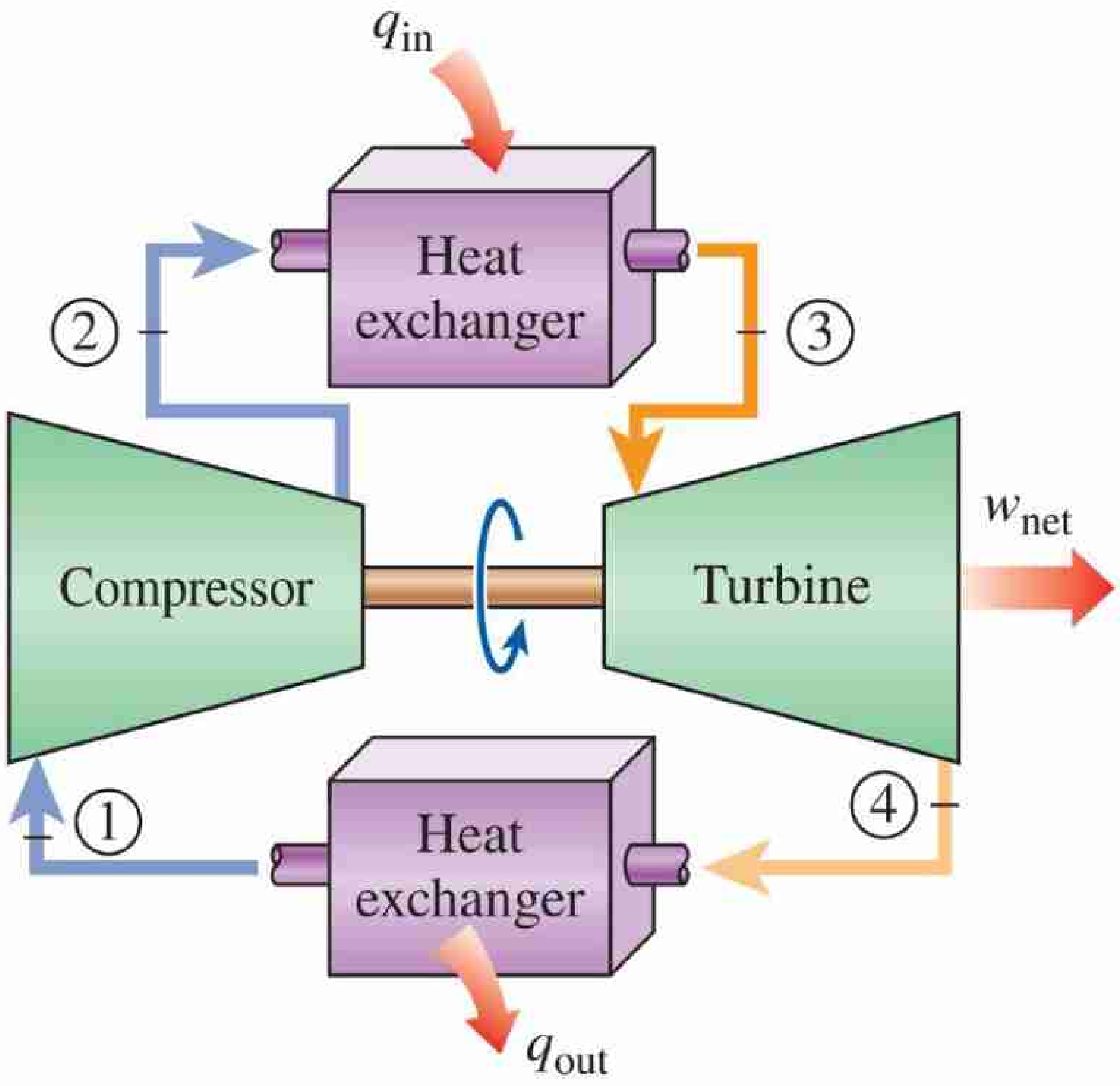


FIGURE
T-s and *P-υ* diagrams for the ideal Brayton cycle.



A closed-cycle gas-turbine engine.

Merits and Demerits of Closed Cycle Gas Turbine Over Open Cycle Gas Turbine

Merits of closed cycle :

1. Higher thermal efficiency
2. Reduced size
3. No contamination
4. Improved heat transmission
5. Improved part load efficiency
6. Lesser fluid friction
7. No loss of working medium
8. Greater output
9. Inexpensive fuel.

Demerits of closed cycle :

1. Complexity
2. Large amount of cooling water is required. This limits its use to stationary installation or marine use where water is available in abundance.
3. Dependent system.
4. The weight of the system per H.P. developed is high comparatively, therefore not economical for moving vehicles.
5. Requires the use of a very large air heater.

Brayton cycle consists of following processes.

Process (1) - (2) isentropic compression in compressor ($Q = 0$)

Process (2) - (3) constant pressure heating in combustion chamber $[Q_s = C_p (T_3 - T_2)]$

Process (3) - (4) isentropic expansion in turbine ($Q = 0$)

Process (4) - (1) constant pressure heat rejection exhaust. $[Q_r = C_p (T_4 - T_1)]$

Heat supplied $Q_s = C_p (T_3 - T_2)$ J/kg

Heat rejected $Q_r = C_p (T_4 - T_1)$ J/kg

Network done $= W_{\text{net}} = Q_s - Q_r$

(i) Air standard efficiency

$$\eta_{thermal} = \left[\frac{\text{Network done}}{\text{Heat supplied}}\right] = \frac{W_{net}}{Q_s} = \frac{Q_s - Q_r}{Q_s}$$

$$= 1 - \frac{Q_r}{Q_s} = 1 - \frac{C_p\,(T_4 - T_1)}{C_p\,(T_3 - T_2)}$$

$$\eta_{air\ standard}\ \text{(or)}\ \eta_{thermal} = 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)} \qquad ...(1)$$

$$r_p = \text{ pressure ratio during compression } = \frac{P_2}{P_1} = \frac{P_3}{P_4}$$

Process (1) - (2) isentropic proces

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

(3) - (4) isentropic process

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

$$\frac{T_2}{T_1} = (r_p)^{\frac{\gamma - 1}{\gamma}} \qquad \frac{T_3}{T_4} = (r_p)^{\frac{\gamma - 1}{\gamma}}$$

$$T_2 = T_1 \cancel{=} (r_p)^{\left[\frac{\gamma - 1}{\gamma}\right]} \qquad T_3 = T_4 \cancel{=} (r_p)^{\left[\frac{\gamma - 1}{\gamma}\right]}$$

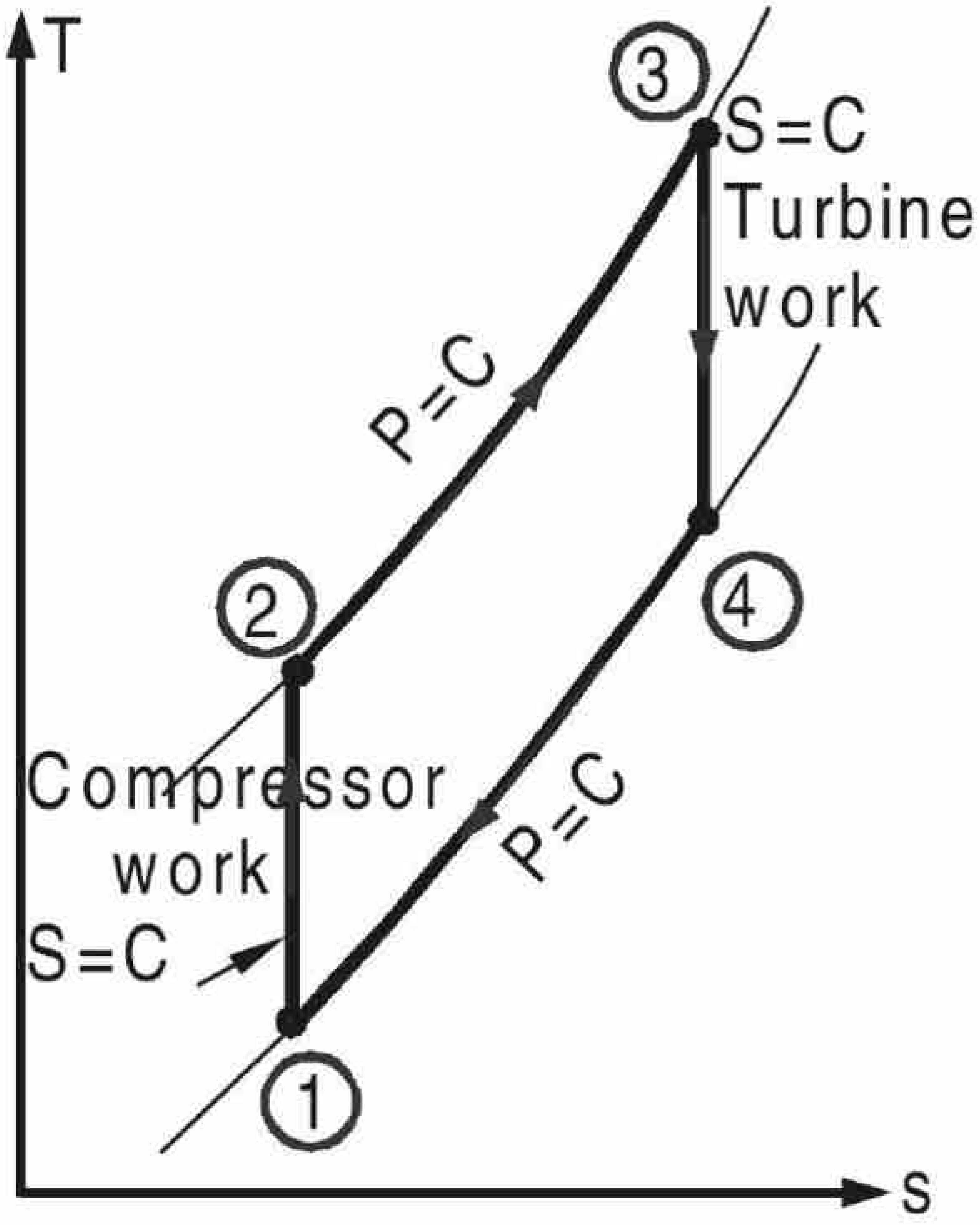
Substitute T_2 and T_3 in eqn (1), we get

$\eta_{\text{Air standard}}$ (or) η_{thermal}

$$= 1 - \frac{(T_4 - T_1)}{\left[T_4 \cdot r_p^{\left(\frac{\gamma - 1}{\gamma}\right)} - T_1 \cdot r_p^{\left(\frac{\gamma - 1}{\gamma}\right)}\right]}$$

$$= 1 - \frac{(T_4 - T_1)}{(r_p)^{\left(\frac{\gamma - 1}{\gamma}\right)} [T_4 - T_1]}$$

$$\eta_{\text{standard}} \text{ (or) } \eta_{\text{thermal}} = 1 - \frac{1}{r_p^{\left(\frac{\gamma - 1}{\gamma}\right)}}$$



@V.Manikanth, Asst. Prof., Dept. of Mechanical, SRKREC(A).

$W_T =$ turbine work $= C_p\,(T_3 - T_4)$ J/kg ; $W_C =$ compressor work $= C_p\,(T_2 - T_1)$

$$W_{\text{net}} = \text{ Net work } = (W_T - W_C)$$

$$W_{\text{net}} = Q_s - Q_r = C_p\,(T_3 - T_2) - C_p\,(T_4 - T_1)$$

$$W = C_p\,[(T_3 - T_4) - (T_2 - T_1)]$$

[Steady flow turbine work $W = \dfrac{R \cdot \gamma}{\gamma - 1}(T_1 - T_2) = C_p\,(T_1 - T_2)$]

$\dot{m}_a -$ mass rate flow or air, kg/sec; $m_f -$ mass rate of flow of fuel, kg/sec.

$$\frac{\dot{m}_a}{\dot{m}_f} = \text{Air fuel ratio}$$

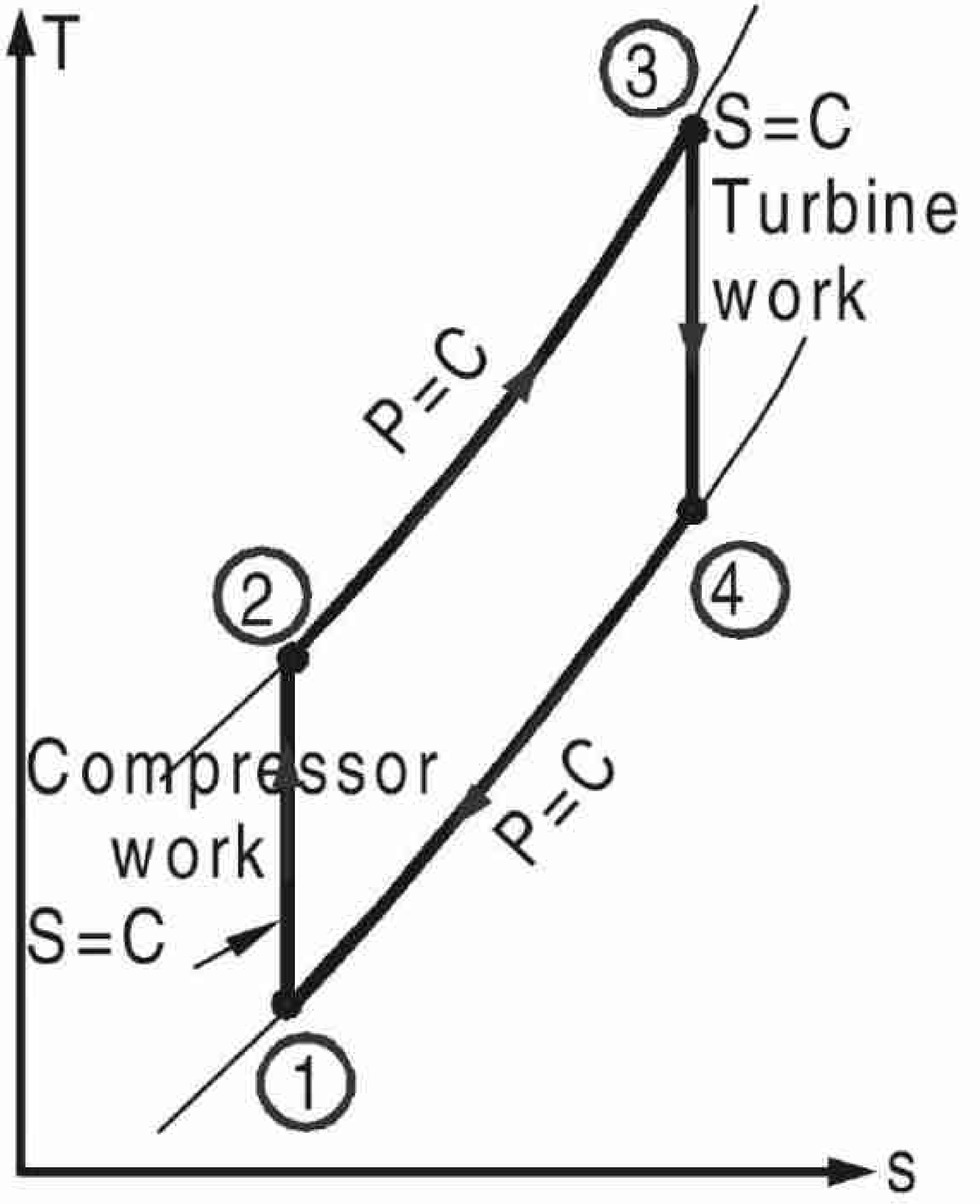
$$W_T = \text{Turbine power} = \dot{m}_a C_p (T_3 - T_4) \text{ in Watts.}$$

$$W_C = \text{compression power} = \dot{m}_a C_p (T_2 - T_1) \text{ in Watts.}$$

$$W_{\text{Net}} = \text{net power} = W_T - W_C \text{ in Watts}$$

$$Q_s = \text{Heat supplied} = \dot{m}_a C_p (T_3 - T_2) \text{ in Watts}$$

$$= \dot{m}_f (CV) \qquad \text{(where } CV \text{ – Calorific value)}$$



Expression for pressure ratio of Brayton cycle for maximum work

Work output during the cycle

$W =$ Heat received/cycle − Heat rejected/cycle

$$W = W_T - W_C = \dot{m}C_p\,[(T_3 - T_4) - (T_2 - T_1)]$$

$$W = \dot{m}C_p\left[T_3\left(1 - \frac{T_4}{T_3}\right) - T_1\left(\frac{T_2}{T_1} - 1\right)\right]$$

$$\therefore\ W = \dot{m}C_p\left[T_3\left(1 - \frac{1}{r_p^z}\right) - T_1(r_p^z - 1)\right]$$

$$\boxed{\therefore\ W = R\left[T_3\left(1 - \frac{1}{r_p^z}\right) - T_1(r_p^z - 1)\right]}$$

where $\dfrac{T_3}{T_4} = \dfrac{T_2}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}} = r_p^z$ and $\dfrac{\gamma - 1}{\gamma} = z$

where $R = \dot{m}C_p =$ constant

@V.Manikanth, Asst. Prof., Dept. of Mechanical, SRKREC(A).

Differentiating above equation and equating to zero.

$$\frac{dW}{dr_p} = R\left[T_3 \times \frac{z}{r_p\,(z+1)} - T_1\, z r_p^{(z-1)} \right] = 0$$

$$\frac{zT_3}{r_p^{(z+1)}} = T_1 z\,(r_p)^{z-1}$$

$$\text{(or)}\quad r_p^{2z} = \frac{T_3}{T_1}$$

$$\text{(or)}\quad \boxed{r_p = \left(\frac{T_3}{T_1}\right)^{1/2z} = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}}$$

Thus, the pressure ratio for maximum work is a function of the limiting temperature ratio.

is the expression for maximum pressure ratio.

Work Ratio: Work ratio is defined as ratio of net work output to the work done by the turbine.

$$\text{Work ratio} = \frac{W_T - W_C}{W_T}$$

$$\therefore \text{ Work ratio} = \frac{\dot{m} \cdot C_p\,[(T_3 - T_4) - (T_2 - T_1)]}{\dot{m} C_p\,(T_3 - T_4)}$$

$$\text{Work ratio} = 1 - \frac{T_2 - T_1}{T_3 - T_4}$$

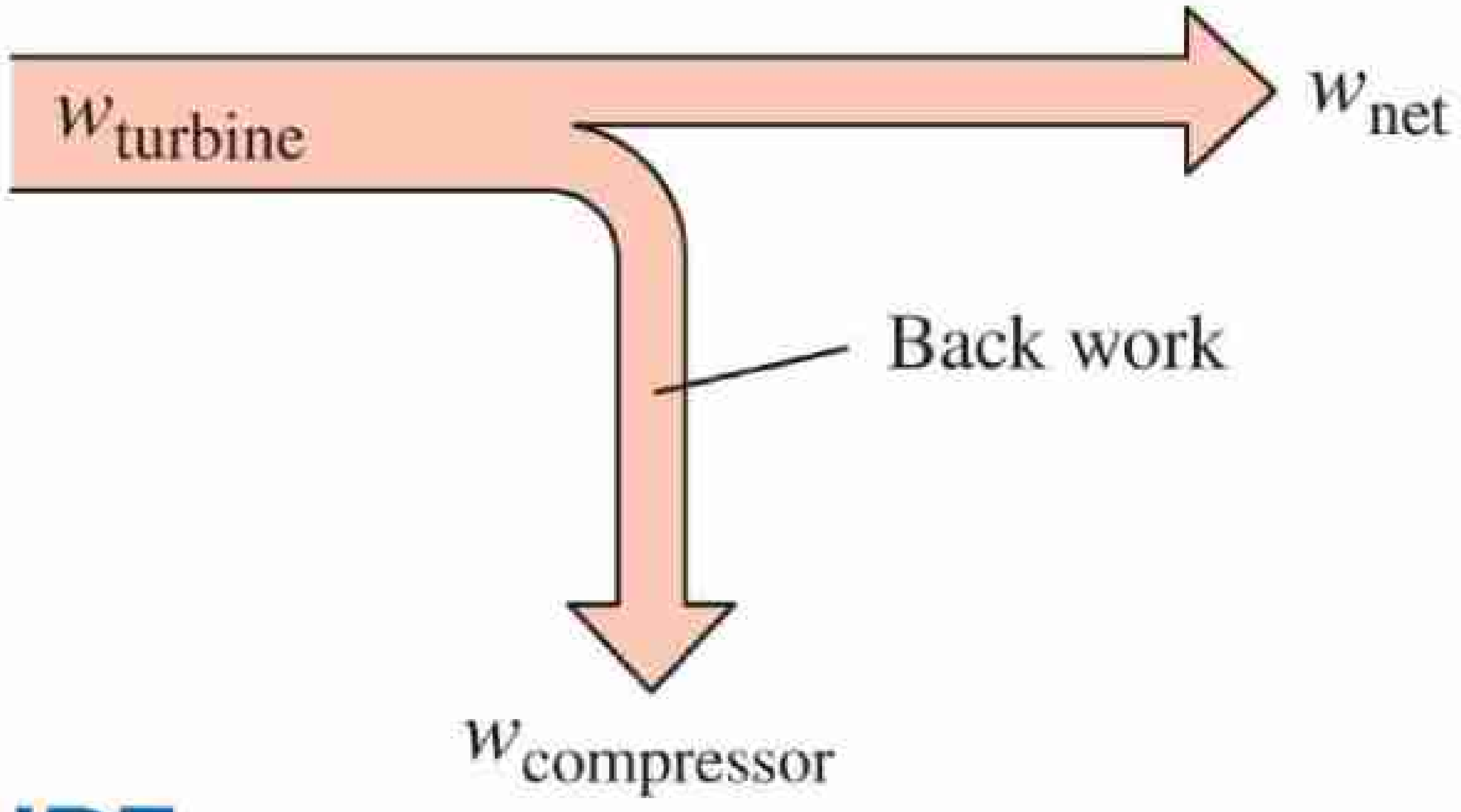


FIGURE
The fraction of the turbine work used to drive the compressor is called the back work ratio.

$$= 1 - \frac{T_1}{T_3}\left[\frac{\dfrac{T_2}{T_1} - 1}{1 - \dfrac{T_4}{T_3}}\right]$$

$$\text{Work ratio} = 1 - \frac{\pi}{T_3}\left[\frac{r_p^{\frac{\gamma-1}{r}} - 1}{1 - \dfrac{1}{(r_p)^{\frac{\gamma-1}{r}}}}\right] = 1 - \frac{T_1}{T_3}(r_p)^{\frac{\gamma-1}{r}}$$

Effect of pressure ratio on thermal efficiency:

$$\eta_{\text{thermal}} = 1 - \frac{1}{r_p^{\left(\frac{\gamma - 1}{\gamma}\right)}}$$

Where r_p = Pressure ratio (P_{max}/P_{min})

The Equation shows that under the cold-air-standard assumptions, the thermal efficiency of an ideal Brayton cycle depends on the pressure ratio of the gas turbine and the specific heat ratio of the working fluid. The thermal efficiency increases with both parameters, which is also the case for actual gas turbines.

As the pressure ratio increases, the thermal efficiency also increases until it becomes maximum and then it drops off with a further increase in the pressure ratio. Further, as the turbine inlet temperature increases, the peaks of the curves flatten out, giving a greater range of ratios of pressure optimum efficiency. Consequently, there exists an optimum pressure ratio to produce maximum thermal efficiency for a given turbine inlet temperature.

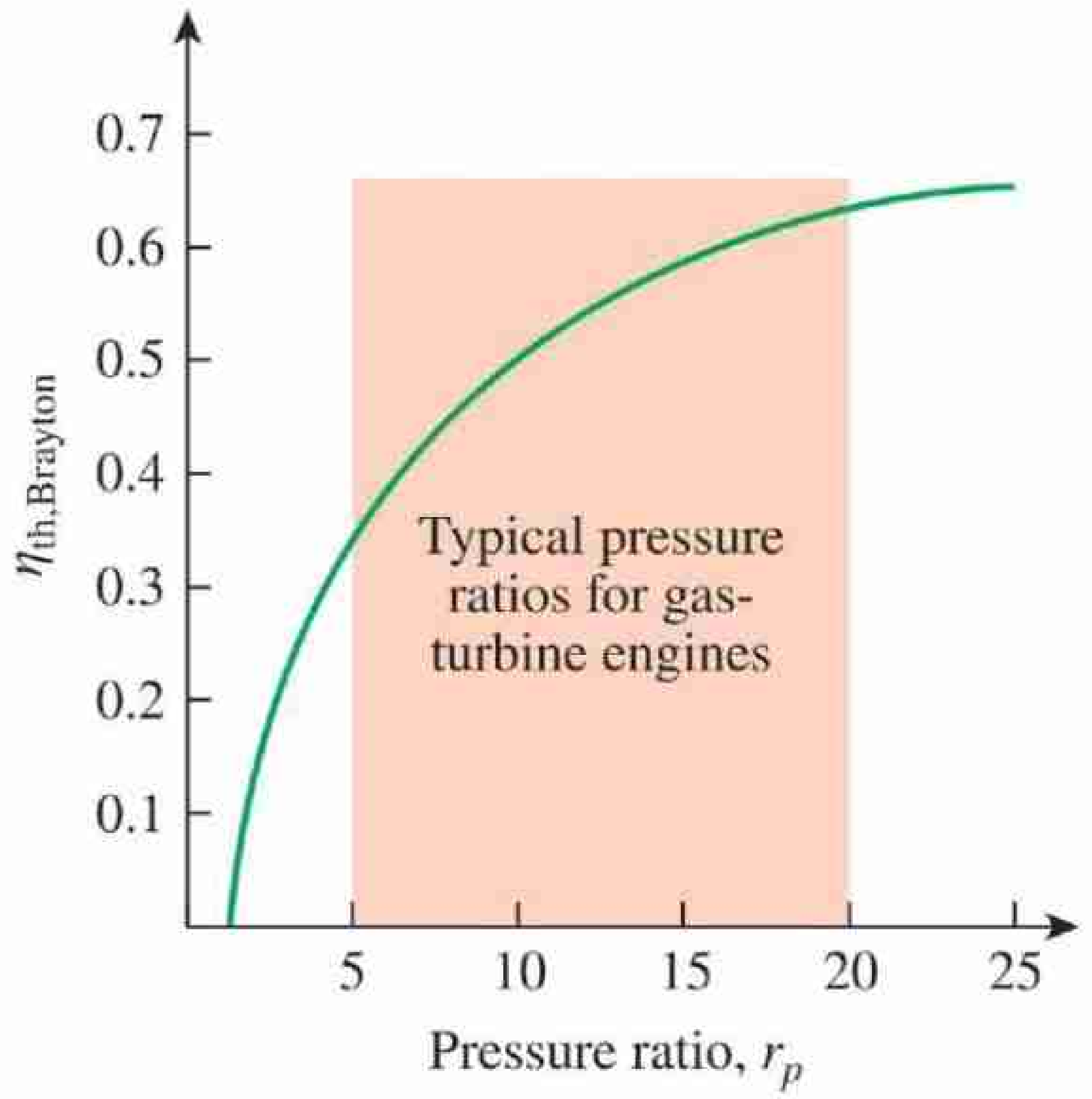


FIGURE
Thermal efficiency of the ideal Brayton cycle as a function of the pressure ratio.

Problem : *In an air standard Brayton cycle air at 1 bar, 20°C is supplied to a compressor where pressure ratio is 4.5. The maximum temperature is 1000 K. Determine. (a) $\eta_{thermal}$ (b) net work (c) work ratio*

Solution:

$$P_1 = 1 \text{ bar}$$

$$T_1 = 20°C = 293 \text{ K}$$

$$T_{max} = T_3 = 1000 \text{ K}$$

$$\frac{P_2}{P_1} = \frac{P_3}{P_4} = 4.5$$

1-2 isentropic process

$$\frac{T_2}{T_1} = \left(\frac{P_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_2 = 293 \times (4.5)^{0.4/1.4}$$

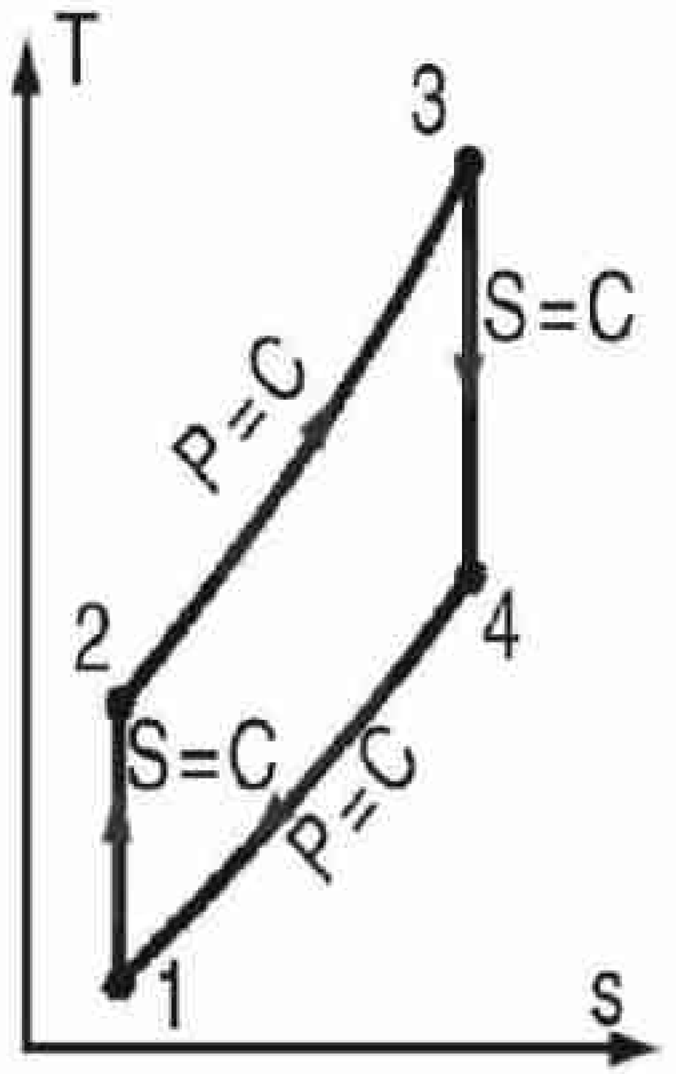
$$\mathbf{T_2 = 450.297 \text{ K}}$$

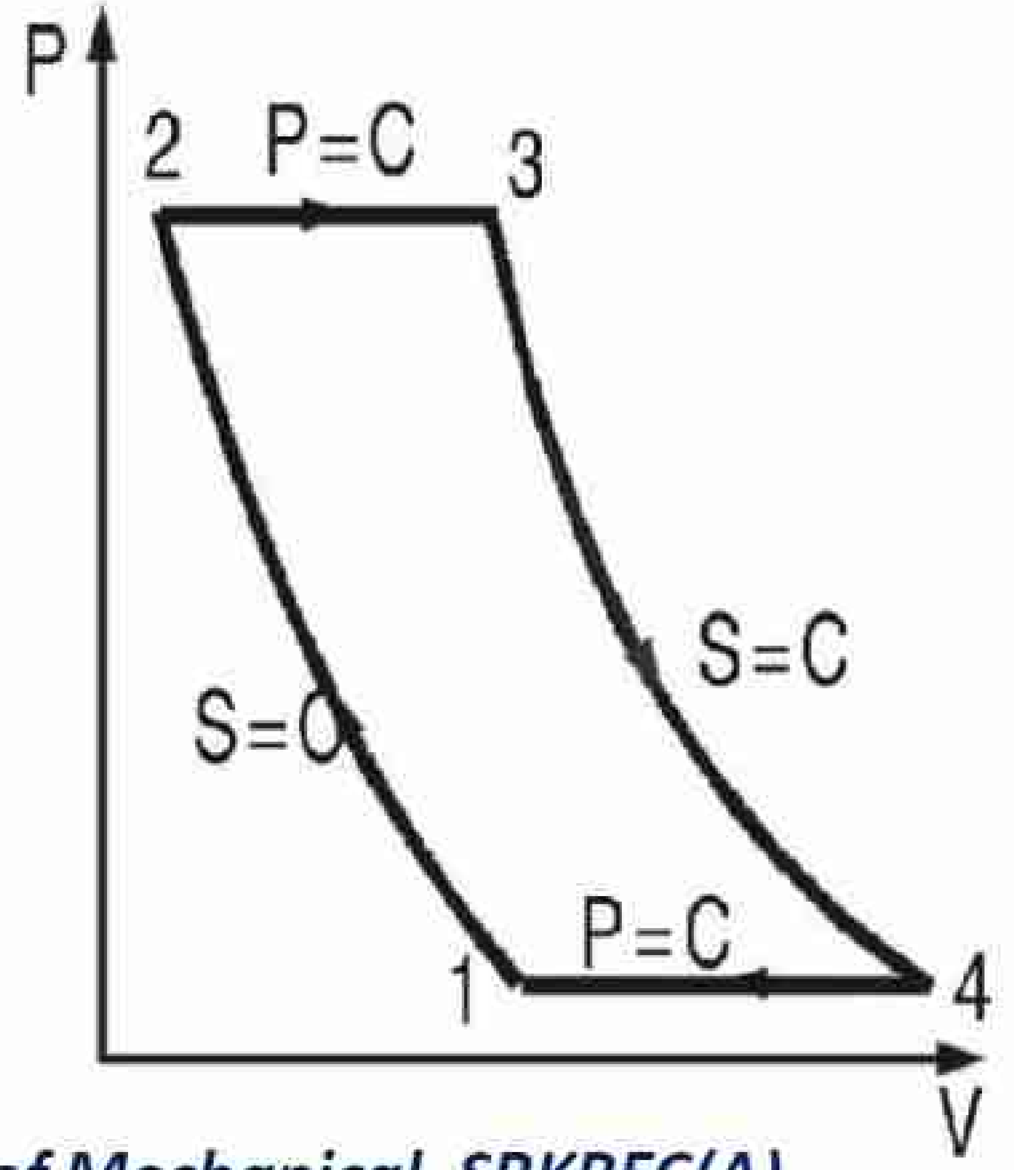
$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_{max}}{T_{min}} = \left(\frac{P_{max}}{P_{min}}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{1000}{T_4} = (4.5)^{0.4/1.4}$$

$$\mathbf{T_4 = 650.68 \text{ K}}$$





$$\eta_{thermal} = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}} = 1 - \frac{1}{(4.5)^{0.4/1.4}} = 0.35$$

$$\eta_{thermal} = 0.35$$

@V.Manikanth, Asst. Prof., Dept. of Mechanical, SRKREC(A).

(b) Turbine work $W_T = C_p\,(T_3 - T_4)$

$$= 1005\ (1000 - 650.58) = 1005\ (349.32)$$

$$\mathbf{W_T = 350891.94\ \ J/kg}$$

Compressor work

$$W_c = C_p\,(T_2 - T_1) = 1005\ (450.297 - 293)$$

$$\mathbf{W_c = 158005.5695\ \ J/kg}$$

$$W_{\text{net}} = (W_T - W_c) = 158005.5695 - 350891.94$$

$$\mathbf{W_{net} = 192886.3705\ \ J/kg}$$

$$\text{Work ratio} = \frac{W_{\text{net}}}{W_T} = \frac{192886.3705}{350891.84} = 0.54956$$

Work ratio = **0.54956**

Problem 4.3: *In an air standard joule cycle air at 300 K is supplied to a compressor where pressure ratio is 6. Max. temp is 1000 K.* $\dot{m}_a = 2.5\ kg/s$, $CV = 42$ MJ/kg. *Determine (a) Air fuel ratio; (b) net power; (c) Work ratio; (d)* η_{thermal}

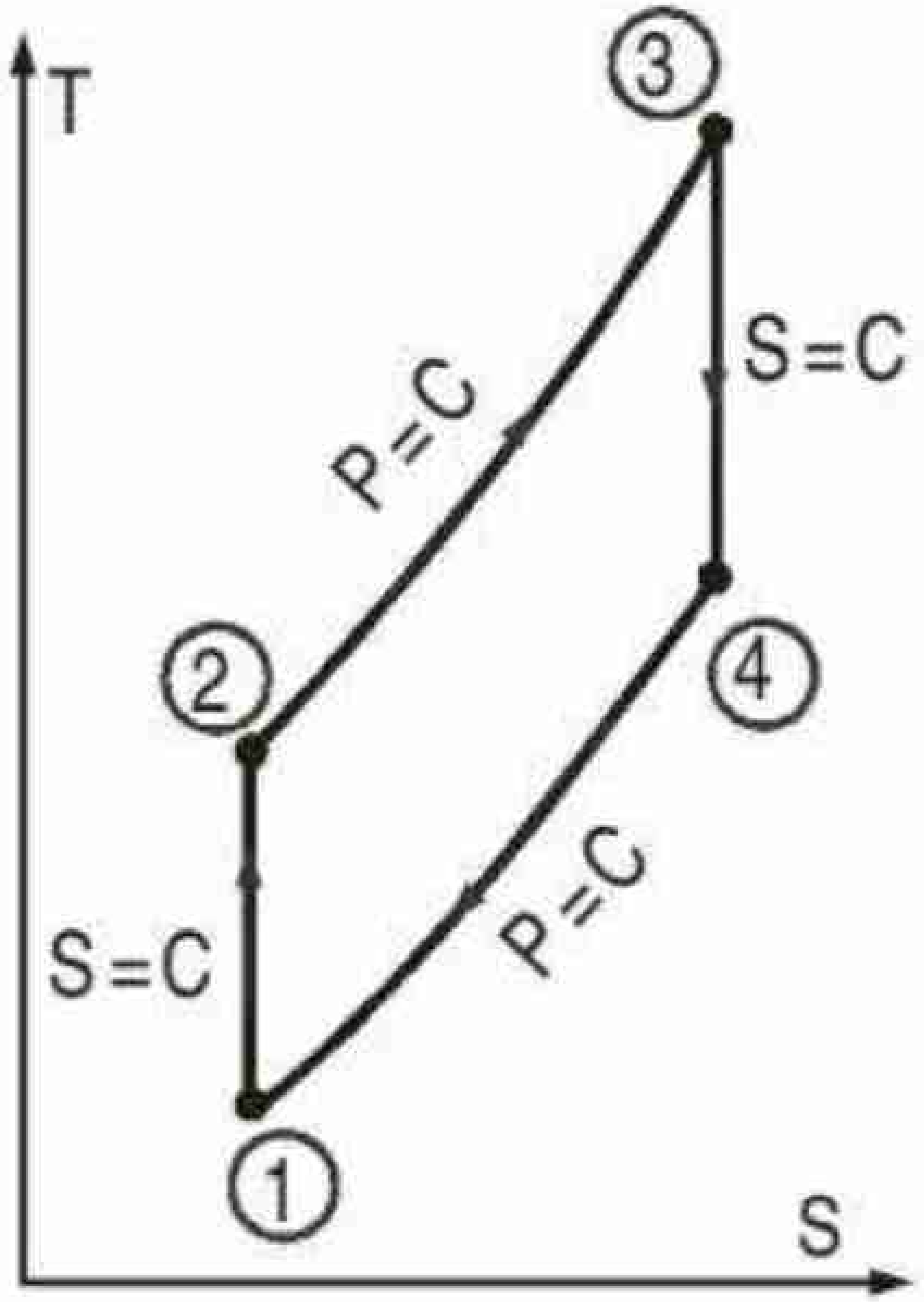
$$T_1 = 300 \text{ K}$$

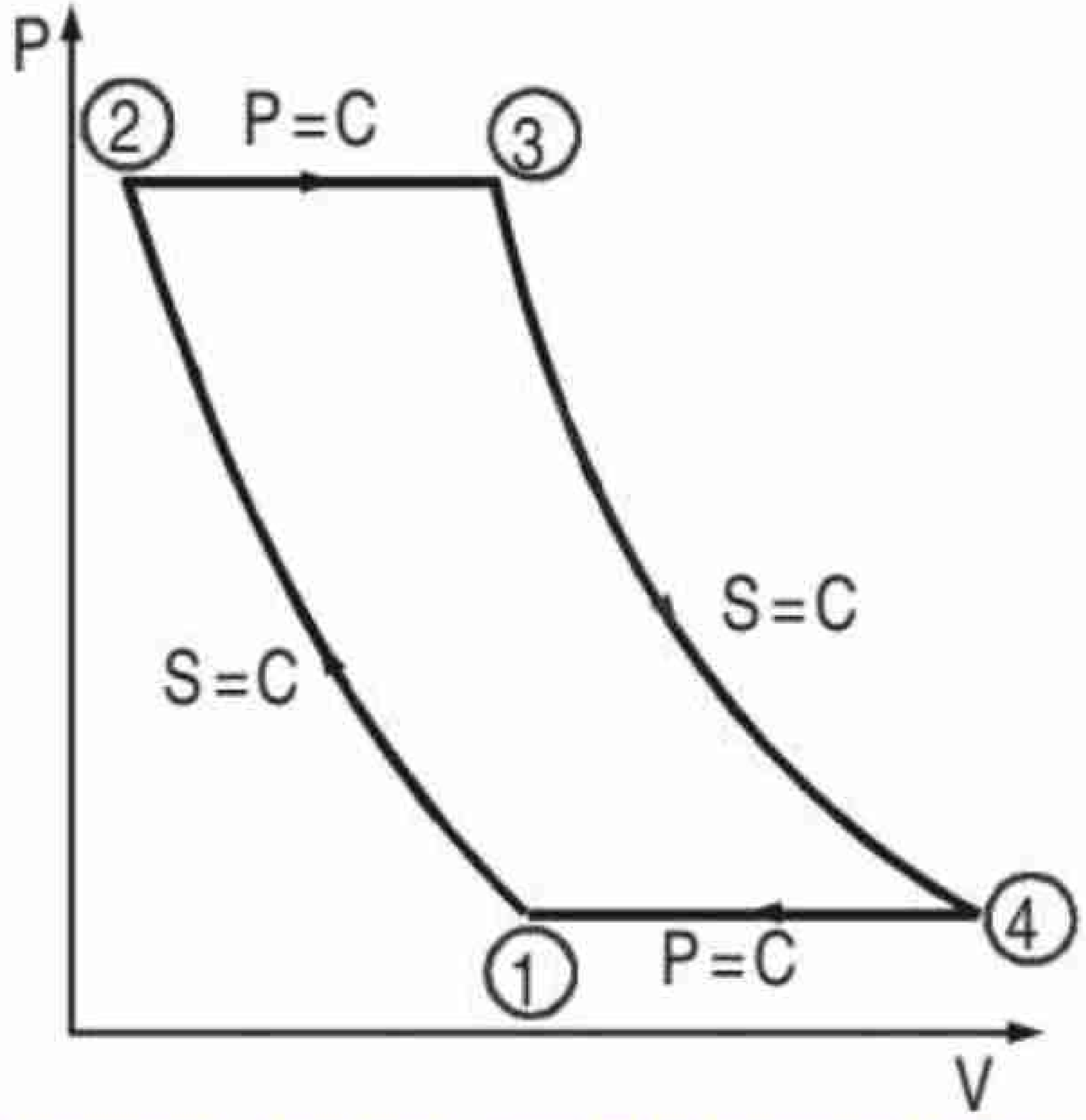
$$r_p = \frac{P_2}{P_1} = \frac{P_3}{P_4} = 6$$

$$T_3 = T_{\max} = 1000 \text{ K}$$

$$\dot{m}_a = 2.5 \text{ kg/s}$$

$$CV = 42 \text{ MJ/kg}$$





@V.Manikanth, Asst. Prof., Dept. of Mechanical, SRKREC(A).

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

$$\frac{T_2}{300} = (6)^{0.4/1.4}$$

$$T_2 = 500.55 \text{ K}$$

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} \quad \text{(or)} \quad \frac{1000}{T_4} = (6)^{\frac{0.4}{1.4}}$$

$$T_4 = 599.340 \text{ K}$$

$$W_T = C_p (T_3 - T_4)$$

$$= 1005 \ (1000 - 599.34)$$

$$W_T = 402462.97 \text{ J/kg}$$

$$W_c = C_p (T_2 - T_1)$$

$$= 1005 \ (500.55 - 300)$$

$$W_c = 201452.475 \text{ J/kg}$$

$$W_{\text{net}} = W_T - W_c = 201010.495 \text{ J/kg}$$

$$\text{Work ratio} = \frac{W_{\text{net}}}{W_T} = \frac{201010.495}{402462.97} = 0.4994$$

Work ratio **= 0.4994**

Air fuel ratio

$$Q_s = \dot{m}_a C_p (T_3 - T_2) = \dot{m}_f (CV)$$

$$\dot{m}_f = \frac{2.5 \times 1005\ (1000 - 500.55)}{42 \times 10^6} = 0.0299\ \text{kg/s}$$

$$\boldsymbol{\frac{\dot{m}_a}{\dot{m}_f} = \frac{2.5}{0.0299} = 83.72}$$

$$W_T = \dot{m}_a C_p (T_3 - T_4) = 1006157.425\ \text{Watts}$$

$$W_c = \dot{m}_a C_p (T_2 - T_{1)} = 503631.187\ \text{Watts}$$

$$W_{\text{net}} = W_T - W_c = 502526.237\ \text{Watts}$$

$$\eta_{\text{th}} = 1 - \frac{1}{(r_p)^{\frac{\gamma - 1}{\gamma}}} = 1 - \frac{1}{6^{0.4/1.4}} = \mathbf{0.4006}$$

WHY ACTUAL GAS TURBINE CYCLE DEVIATES FROM IDEAL CYCLE ?

In actual practice, it is difficult to attain the ideal conditions due to the various losses in different components.

The assumptions for actual gas turbine cycle are

(i) Since the fluid velocities are increased in nature in the plant, the change in K.E between the entry and exit of each elements are considered.

(ii) The compression and expansion process is not isentropic, resulting a increase in entropy.

(iii) Pressure loss gets vicinity due to fluid friction in combustion chamber, heat exchanger and the entry and exit passage.

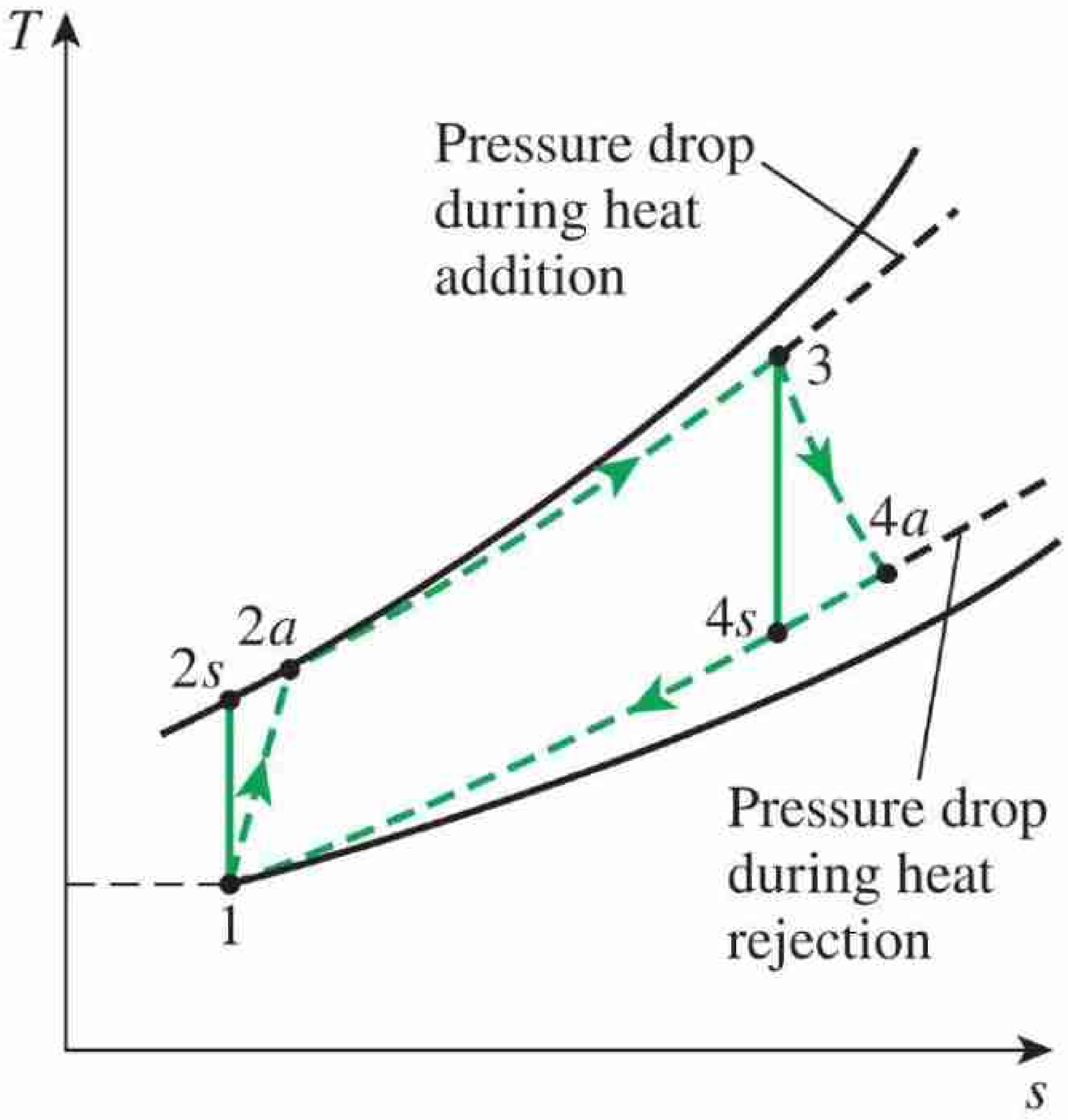
(iv) Perfect exchange of heat will not be performed in heat exchanger.

(v) Mass flow rate is assumed to be constant, even though fuel is added in combustion chamber.

(vi) The specific heats of the working substance vary throughout the cycle due to variation in imperative and chemical composition.

(vii) Slightly higher work than that required for compression process will be required to subside the bearing and windage friction losses.

Deviation of Actual Gas-Turbine Cycles from Idealized Ones



FIGURE

The deviation of an actual gas-turbine cycle from the ideal Brayton cycle as a result of irreversibilities.

The actual gas-turbine cycle differs from the ideal Brayton cycle on several accounts. The actual work input to the compressor is more, and the actual work output from the turbine is less because of irreversibilities. The deviation of actual compressor and turbine behavior from the idealized isentropic behavior can be accurately accounted for by utilizing the isentropic efficiencies of the turbine and compressor as

Compressor isentropic efficiency, η_{comp}

$$= \frac{\text{Work input required in isentropic compression}}{\text{Actual work required}}$$

Turbine isentropic efficiency, $\eta_{turbine}$

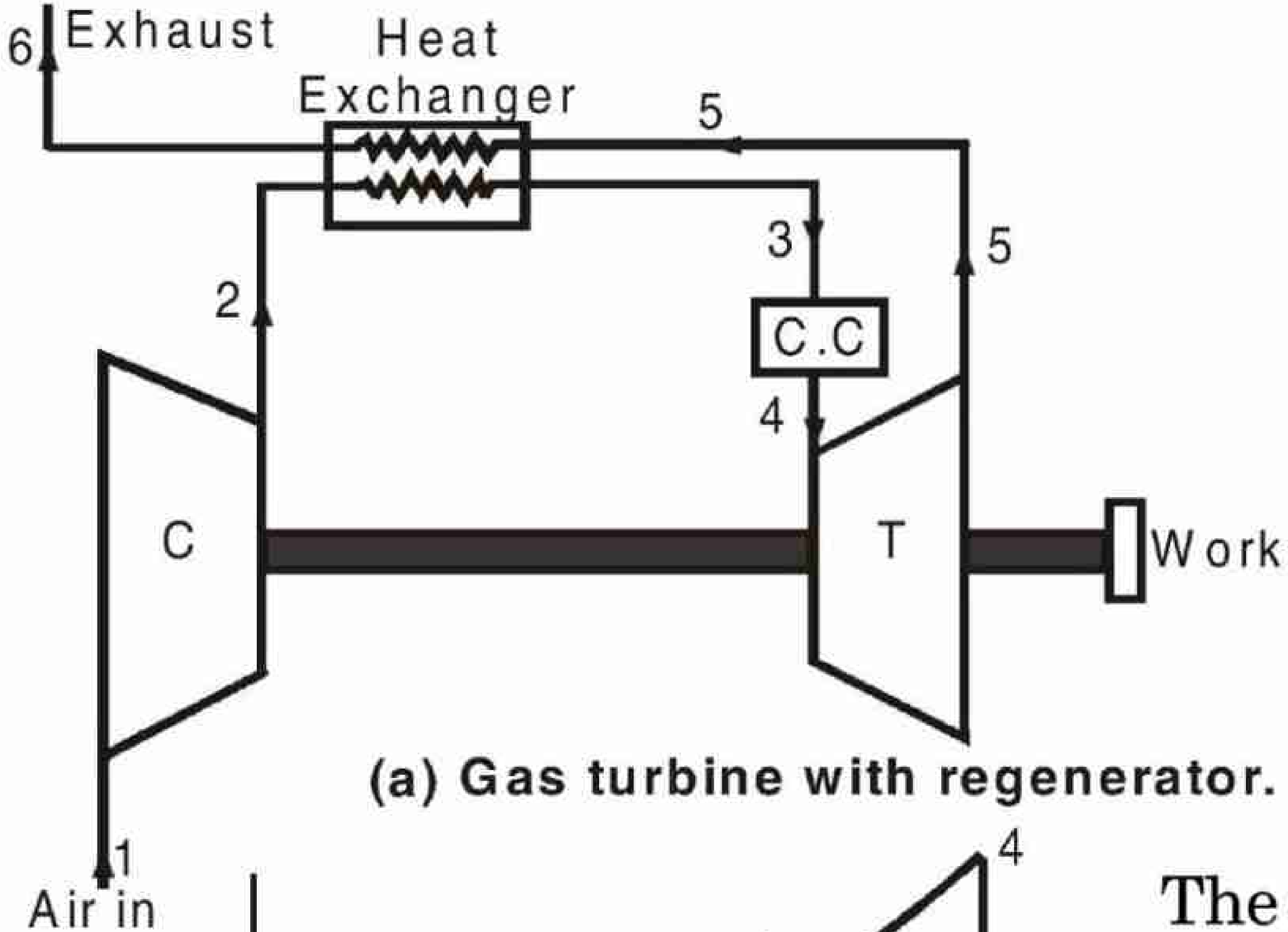
$$= \frac{\text{Actual work output}}{\text{Isentropic work output}}$$

The net work of a gas-turbine cycle is the difference between the turbine work output and the compressor work input, and it can be increased by either decreasing the compressor work or increasing the turbine work, or both. To increase the efficiency of the Brayton cycle, the following methods are employed.

1. Regeneration.
2. Reheating
3. Inter cooling

1.Regeneration:

The efficiency of the Brayton cycle is increased by regeneration. The large quantity of heat energy possessed by the exhaust gases leaving the turbine will be utilized to heat up the air leaving the compressor. This heating is done in a heat exchanger called a regenerator. So, the amount of heat supplied (Qs) from the external source is reduced. The amount of heat rejected is also reduced. So, the efficiency of the turbine plant (Brayton cycle) is increased.



(a) Gas turbine with regenerator.

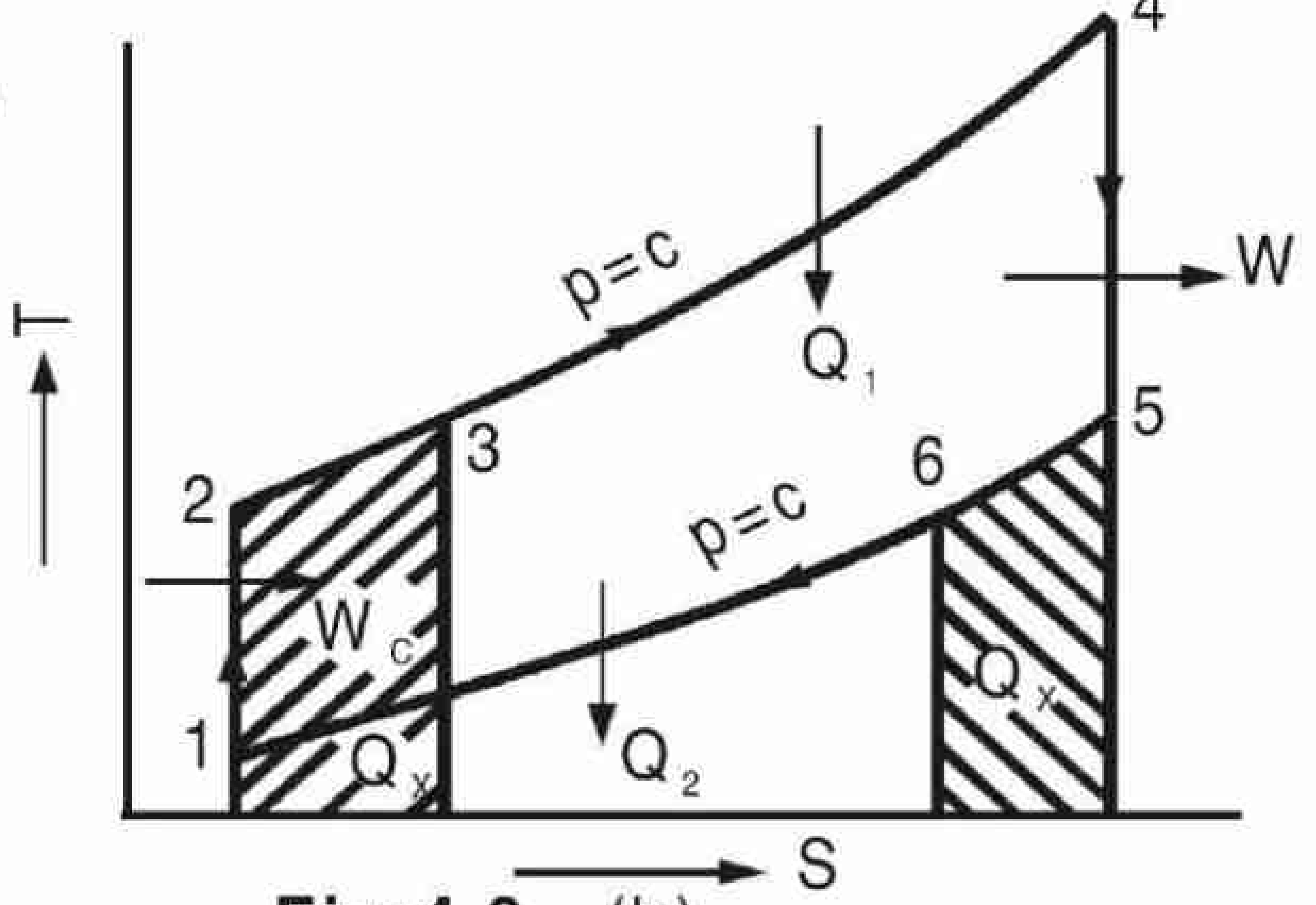


Fig:4.6 (b)
Effect of regeneration on Brayton cycle

Heat supplied $Q_s = C_p\,(T_4 - T_3)$

Heat rejected $Q_r = C_p\,(T_6 - T_1)$

$$\text{Efficiency of Brayton cycle} = 1 - \frac{Q_r}{Q_s}$$

$$= 1 - \frac{(T_6 - T_1)}{(T_4 - T_3)}$$

The work output of the cycle remains unchanged. So,

$$W_T = C_p\,(T_4 - T_5)$$

$$W_C = C_p\,(T_2 - T_1)$$

Regenerator effectiveness

$$= \frac{\text{Actual heat transfer}}{\text{Maximum possible heat transfer}}$$

Problem: *In gas turbine plant, working on Brayton cycle with a regenerator of 75% effectiveness, the air at the inlet to the compressor is at 1 bar and 30°C. The pressure ratio is 6 and the maximum cycle temperature is 900°C. If the turbine and compressor have each an efficiency of 80%, find the percentage increase in the cycle efficiency due to regeneration.*

Given

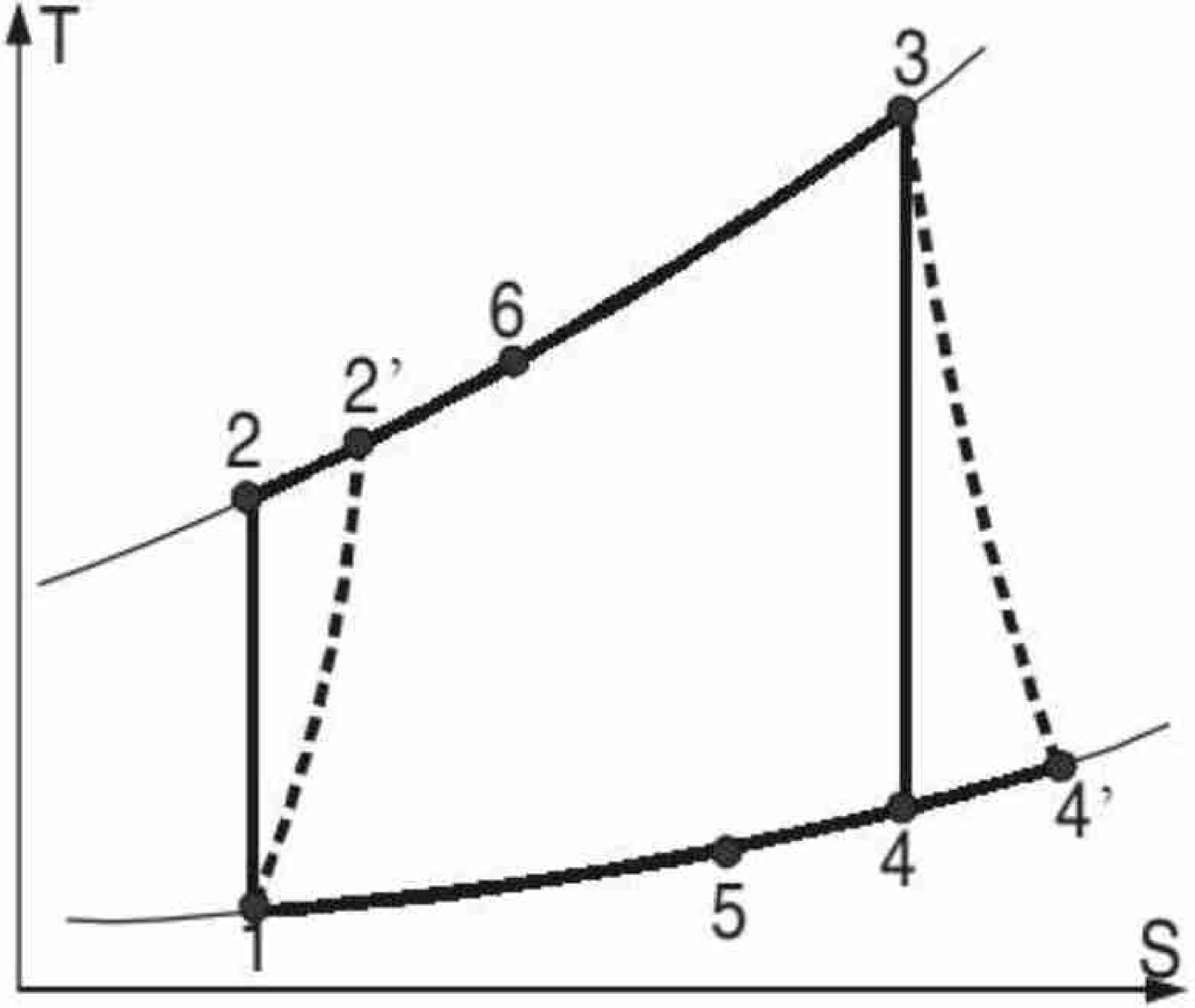
$$P_1 = 1 \text{ bar}$$

$$T_1 = 30 + 273 = 303 \text{ K}$$

$$r_p = \frac{P_2}{P_1} = 6$$

$$T_3 = 900 + 273 = 1173 \text{ K}$$

$$\eta_{\text{Turbine}} = \eta_{\text{comp.}} = 0.8$$



Without a Regenerator

To Find T_2

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}}$$

$$T_2 = 303\ (6)^{\frac{0.4}{1.4}} = 505\ \text{K}$$

To Find T_4

$$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{1}{6}\right)^{\frac{0.4}{1.4}}$$

$$T_4 = 1173 \left(\frac{1}{6}\right)^{\frac{0.4}{1.4}} = 703\ \text{K}$$

$$\text{Compressor efficiency } \eta_{comp} = \frac{\text{Isentropic compression}}{\text{Actual compression}}$$

$$= \frac{T_2 - T_1}{T_2' - T_1}$$

$$T_2' - T_1 = \frac{T_2 - T_1}{\eta_c}$$

$$T_2' = T_1 + \frac{T_2 - T_1}{\eta_c}$$

$$= 303 + \frac{505 - 303}{0.8}$$

$$= 555.5\ \text{K}$$

To Find T_4'

$$\text{Turbine efficiency } \eta_T = \frac{\text{Actual expansion}}{\text{Isentropic expansion}} = \frac{T_3 - T_4'}{T_3 - T_4}$$

$$T_3 - T_4' = \eta_T (T_3 - T_4)$$

$$T_4' = T_3 - \eta_T (T_3 - T_4)$$

$$= 1173 - 0.8\ (1173 - 703)$$

$$\mathbf{T_4' = 797\ K}$$

To Find Turbine Work (W_T) and Compressor Work (W_C)

$$W_T = C_p\ (T_3 - T_4') = 1.005\ (1173 - 797) = \mathbf{378\ kJ/kg}$$

$$W_C = C_p\ (T_2' - T_1) = 1.005\ (555.5 - 303) = \mathbf{253.8\ kJ/kg}$$

To Find Heat Supplied Q_s

$$Q_s = C_p\ (T_3 - T_2') = 1.005\ (1173 - 555.5)$$

$$= \mathbf{620.6\ kJ/kg}$$

To Find Efficiency η

$$\eta = \frac{\text{Net workdone}}{\text{Heat supplied}} = \frac{W_T - W_C}{Q_s}$$

$$= \frac{378 - 253.8}{620.6} = 0.2001$$

$$= \mathbf{20.01\%}$$

With regenerator

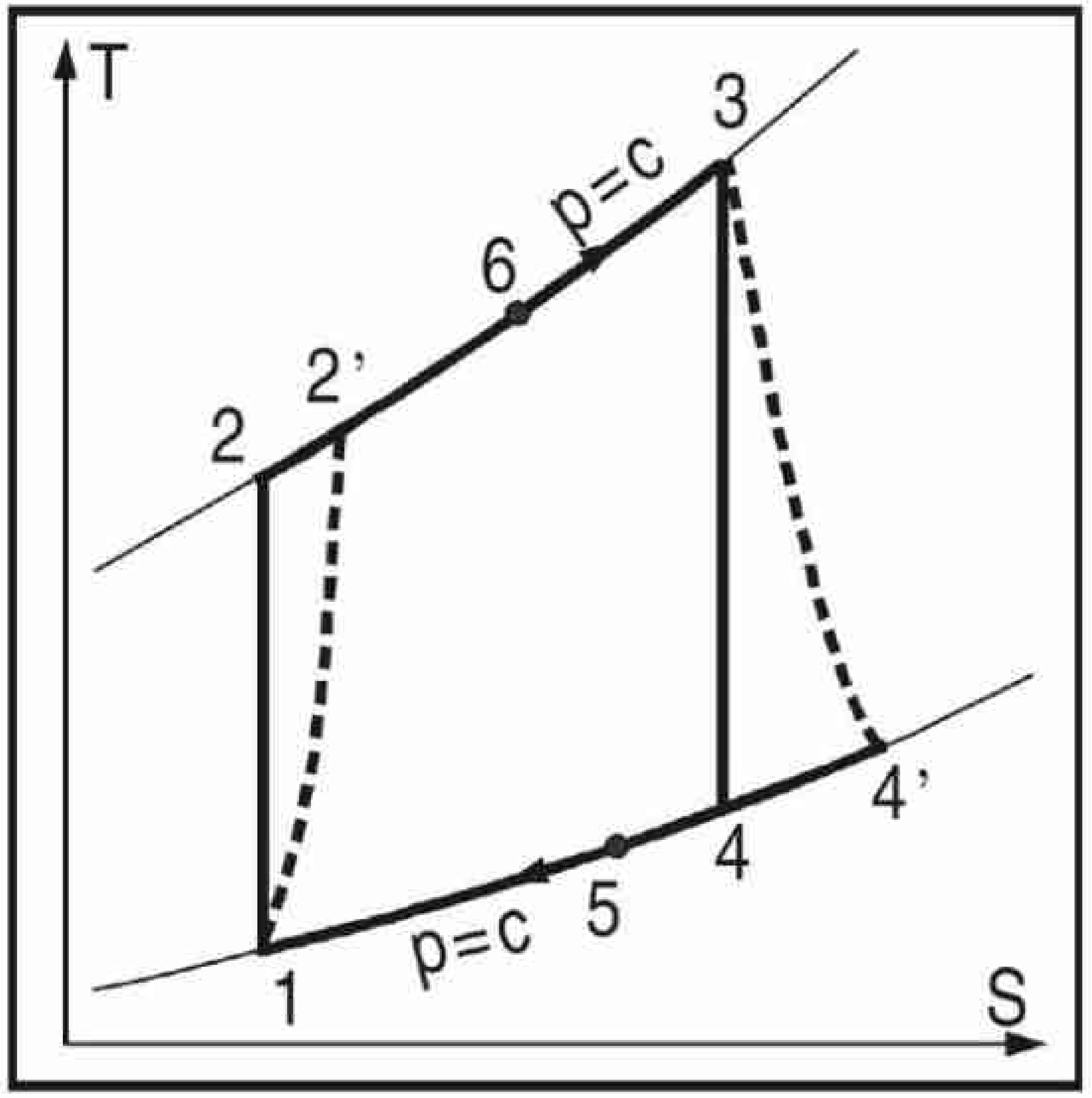
Regenerator effectiveness

$$= \frac{\text{Actual heat transfer}}{\text{Maximum possible heat transfer}} = \frac{(T_6 - T'_2)}{(T'_4 - T'_2)} = 0.75$$

$$T_6 = T'_2 + 0.75\ (T'_4 - T'_2)$$

$$= 555.5 + 0.75\ (797 - 555.5)$$

$$= \mathbf{736.63\ K}$$



Heat supplied with regenerator

$$Q_s' = C_p\ (T_3 - T_6)$$

$$= 1.005\ (1173 - 736.63)$$

$$= \mathbf{438.55\ kJ/kg}$$

Here W_{net} remain same.

$$= \frac{W_{net}}{Q_s'} = \frac{(378 - 253.8)}{438.55}$$

$$= 0.2832$$

$$= \mathbf{28.32}\ \%$$

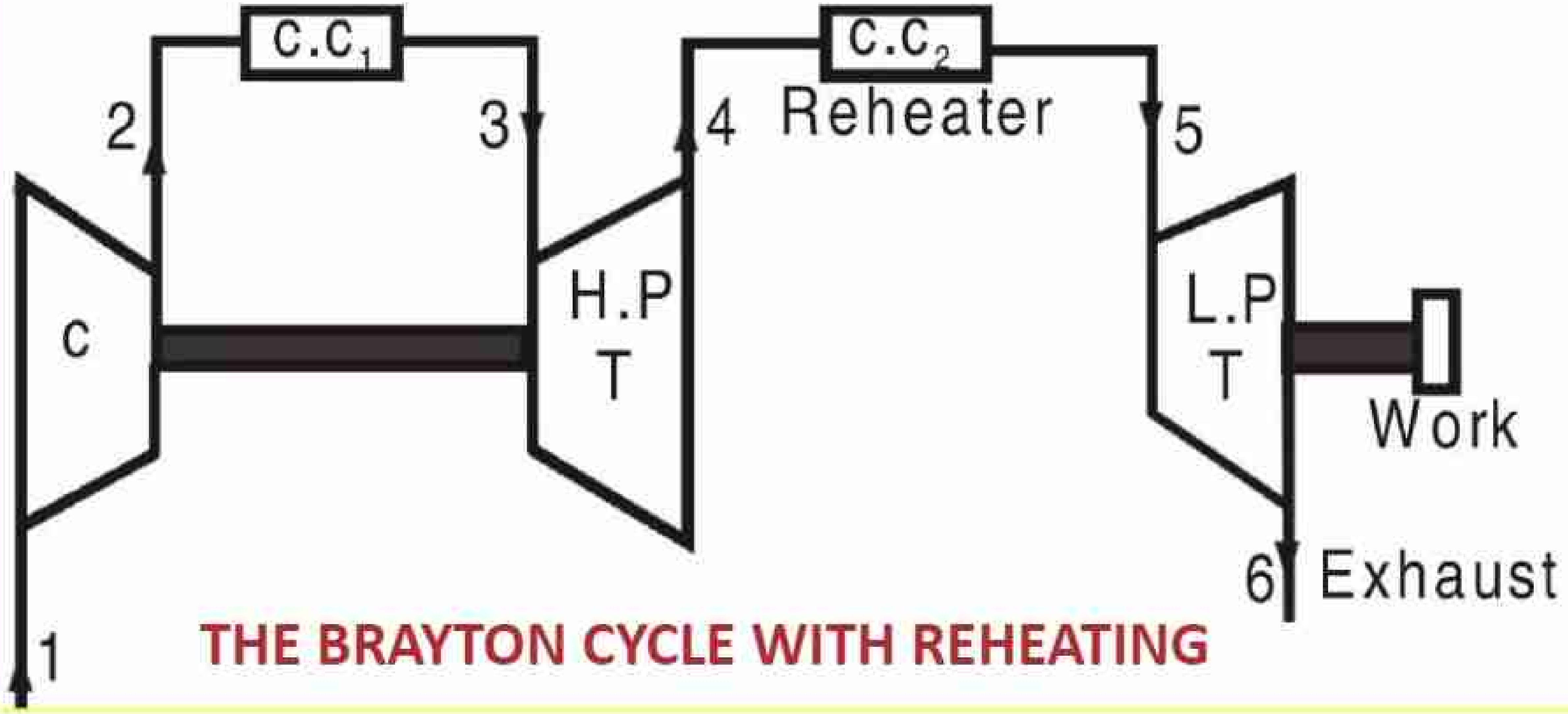
Percentage increase due to regeneration:

$$= \frac{0.2832 - 0.2001}{0.2001} = 0.4153$$

$$= \mathbf{41.53}\ \%$$

2. Reheating

The efficiency of the Brayton cycle can be improved by introducing reheater in between high pressure (HP) turbine and low pressure (LP) turbine. The gases are expanded in two turbines. The power from the HP turbine will be utilized to run the compressor and the power from LP turbine will give the useful power output.



THE BRAYTON CYCLE WITH REHEATING

The work output of a turbine operating between two pressure levels can be increased by expanding the gas in stages and reheating it in between—that is, utilizing multistage expansion with reheating. This is accomplished without raising the maximum temperature in the cycle. As the number of stages is increased, the expansion process becomes nearly isothermal.

The steady flow compression or expansion work is proportional to the specific volume of the fluid. Therefore, the specific volume of the working fluid should be as high as possible during an expansion process.

Combustion in gas turbines typically occurs at four times the amount of air needed for complete combustion to avoid excessive temperatures. Therefore, the exhaust gases are rich in oxygen, and reheating can be accomplished by simply spraying additional fuel into the exhaust gases between two expansion states.

$$\text{Network done} = W_{net} = W_{T\,(HP)} + W_{T\,(LP)} - W_C$$

If $\quad W_{T\,(HP)} = W_C, \quad$ then

$$W_{net} = C_p\,(T_5 - T_6)$$

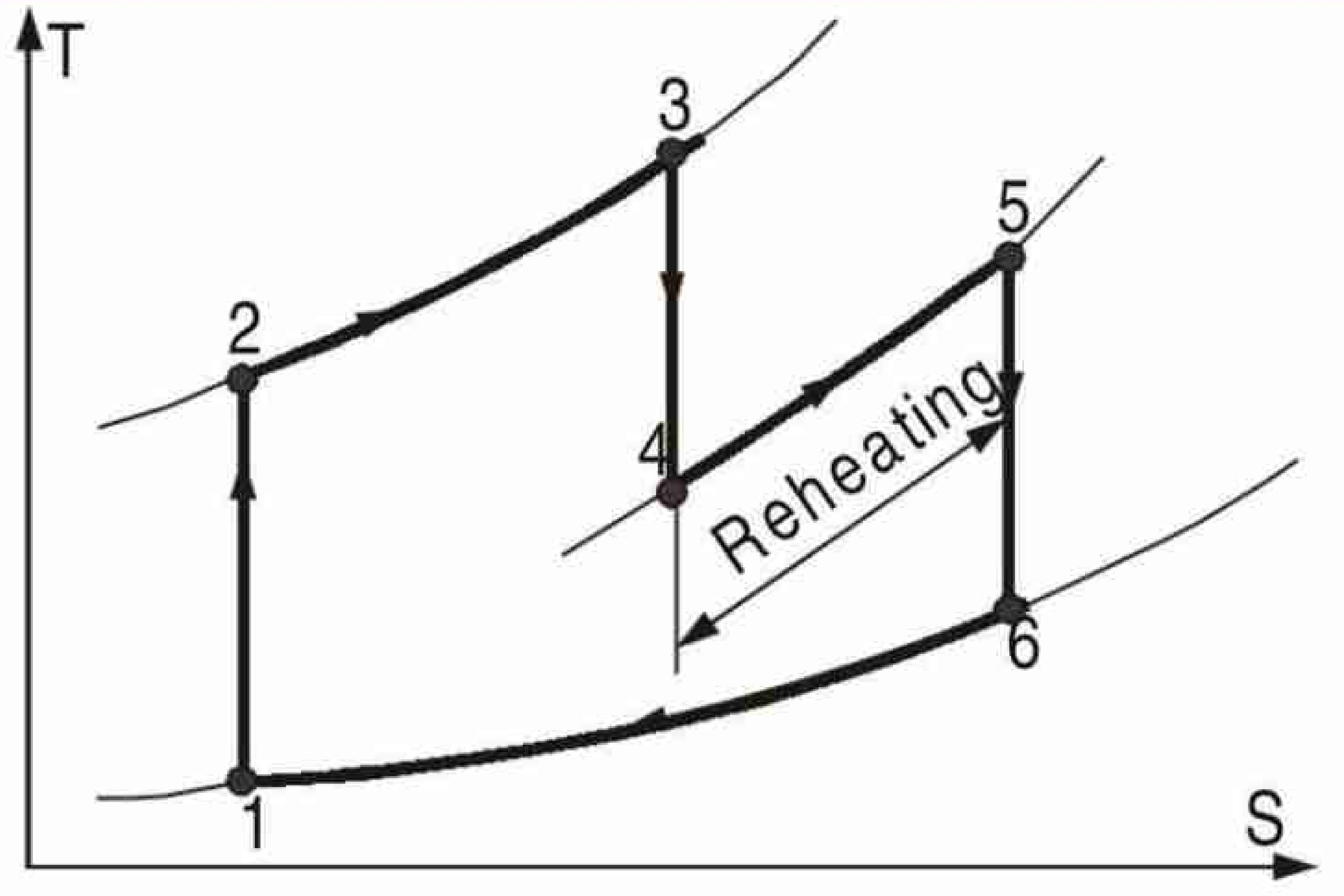


Fig Gas turbine with reheater.

when the net work is increased by reheater, the heat supplied is also increased. So the net effect may either increase or decrease the efficiency of the Brayton cycle.

$$\eta = \frac{W_{net}}{Q_{supplied}} = \frac{C_p\,(T_5 - T_6)}{C_p\,(T_3 - T_2) + C_p\,(T_5 - T_4)}$$

$$\eta = \frac{(T_5 - T_6)}{(T_3 - T_2) + (T_5 - T_4)}$$

Problem: *In a closed cycle gas turbine, there is a one stage compressor and a two stage turbine. And the components are mounted on the same shaft. The pressure and temperature at the inlet of the first stage compressor are 1.5 bar and 20°C. The maximum cycle temperature and pressure are limited to 750°C and 6 bar. A reheater is used between the two turbines. Gases are heated in the reheater to 750°C before entering into the L.P. turbine. Assuming compressor and turbine efficiencies as 0.82, calculate (i) the efficiency of the cycle. (ii) The mass of the fluid circulated if the power developed by the plant is 500 kW. The working fluid used in the cycle is air. For air,* $\gamma = 1.4$ *and* $C_p = 1.005\ \mathrm{kJ/kgK}$. *Take intermediate pressure as 3 bar.*

Solution:

@V.Manikanth, Asst. Prof., Dept. of Mechanical, SRKREC(A).

$$T_1 = 20 + 273 = 293 \ \ \mathrm{K} \ ; T_3 = T_5 = 750 + 273 = 1023 \ \mathrm{K}$$

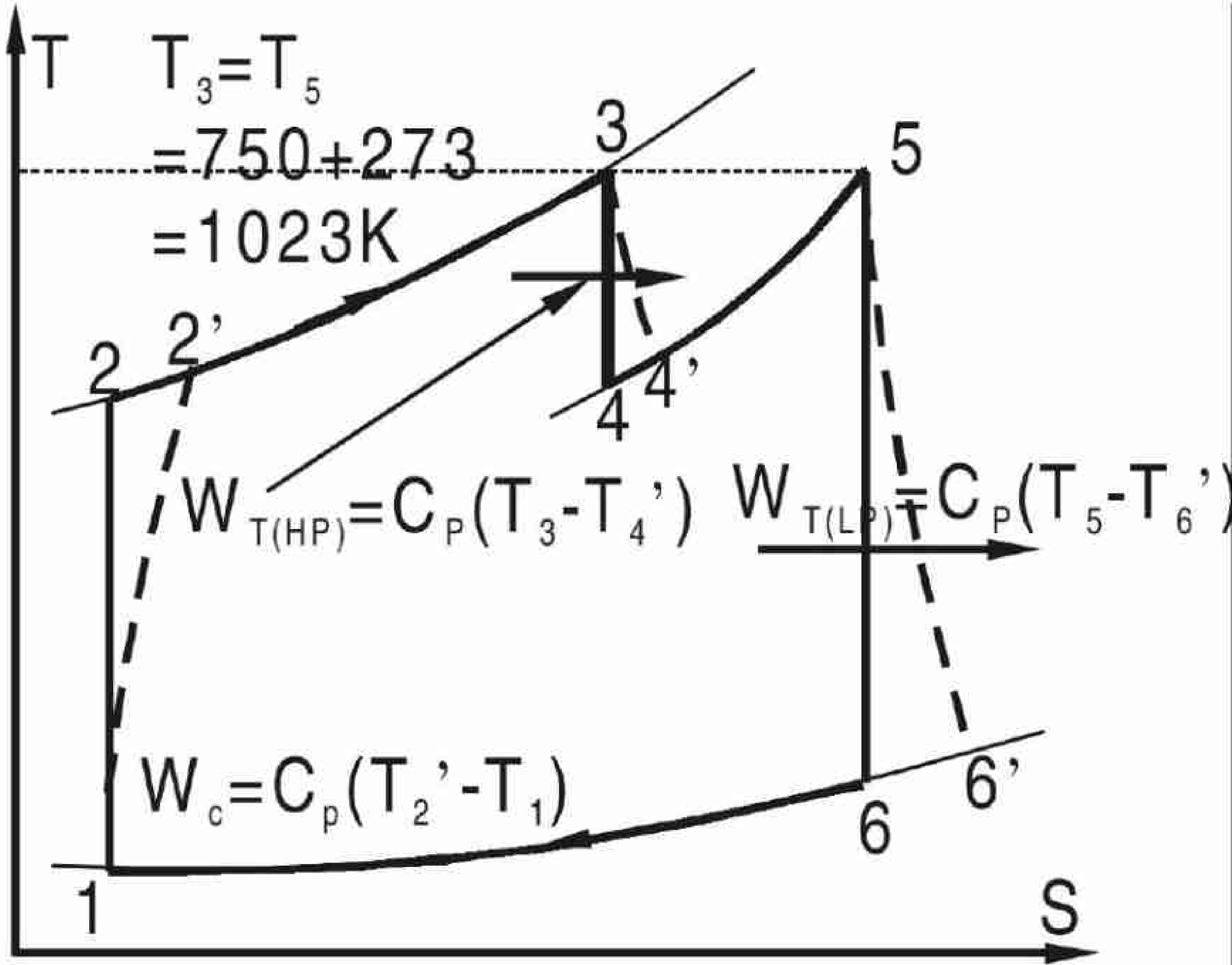
$$T_3 = T_5 = 750 + 273 = 1023 \ \mathrm{K}$$

$$P_1 = 1.5 \ \ \text{bar} \ ; P_2 = 6 \ \text{bar} \ ; \eta_c = \eta_T = 0.82$$

$$P_4 = P_5 = 3 \ \text{bar}$$

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

$$T_2 = 293\left(\frac{6}{1.5}\right)^{\frac{0.4}{1.4}} = 435.4 \ \mathrm{K}$$



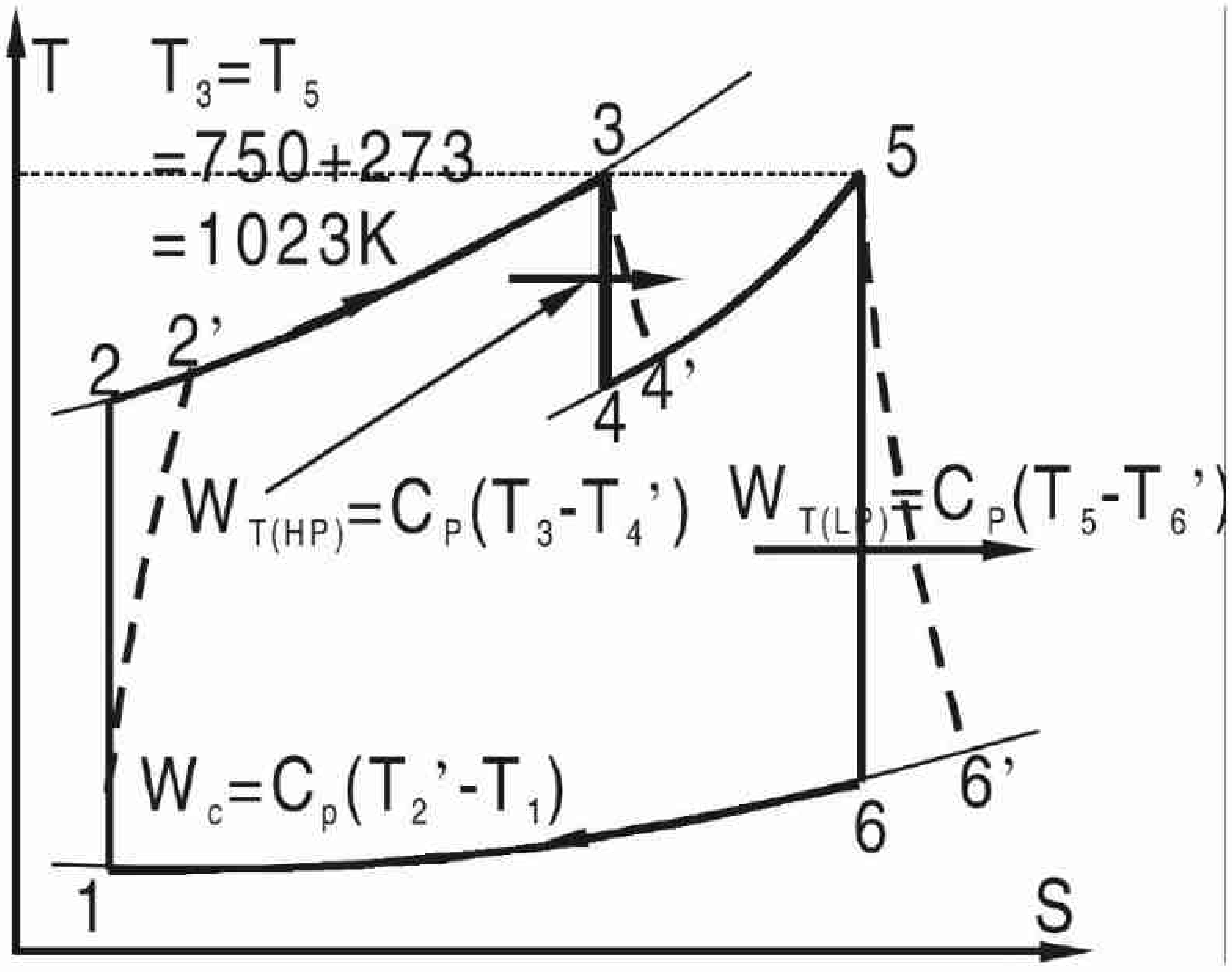
$$\eta_{comp} = \frac{T_2 - T_1}{T_2' - T_1}$$

$$T_2' - T_1 = \frac{T_2 - T_1}{\eta_c}$$

$$T_2' = T_1 + \frac{T_2 - T_1}{\eta_c}$$

$$= 293 + \frac{435.4 - 293}{0.82}$$

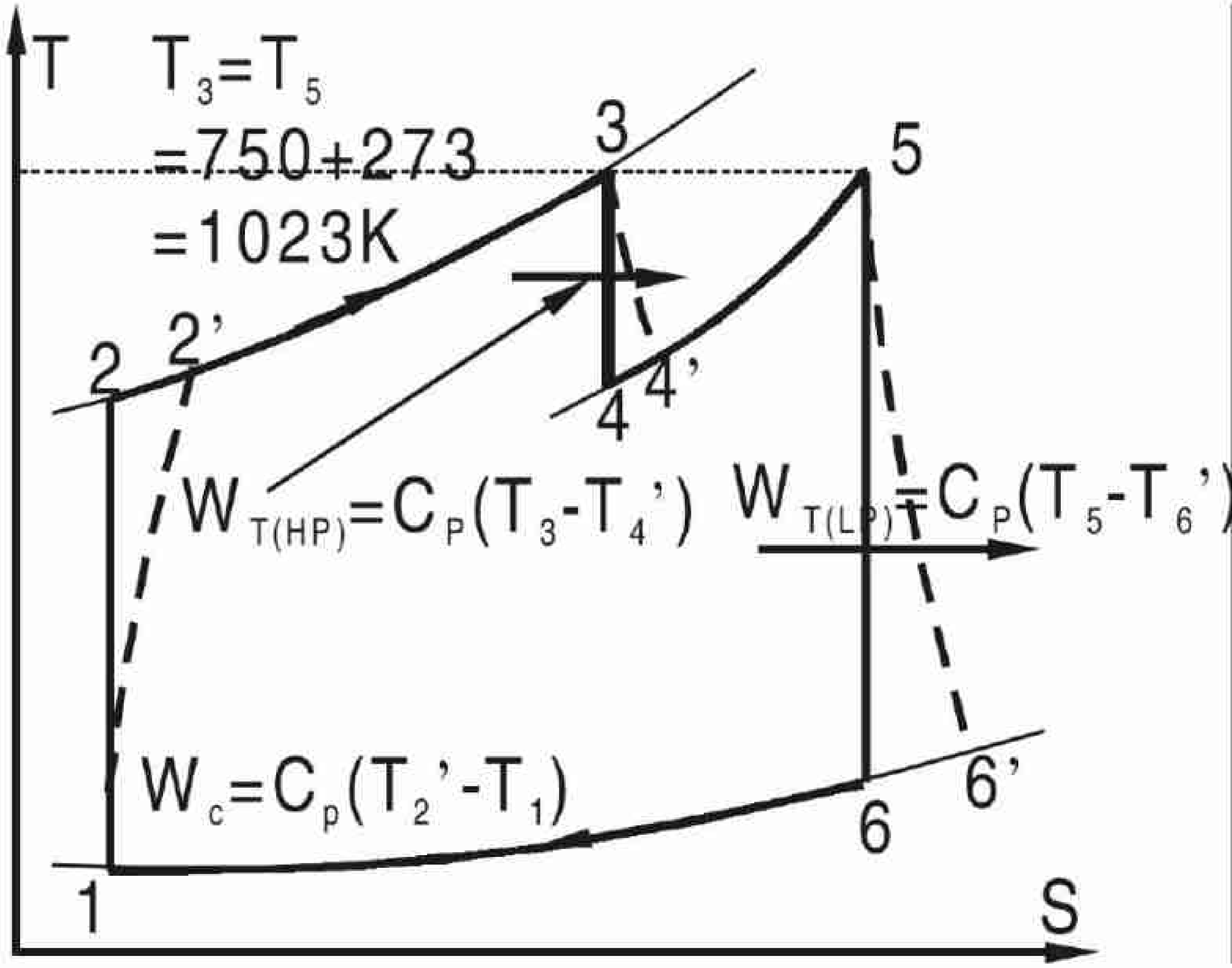
$$= 466.7 \text{ K}$$



Compressor work $W_c = C_p\,(T_2' - T_1)$

$$= 1.005\ (466.7 - 293) = 174.57$$

$$\boldsymbol{W_c = 174.57\ \ \text{kJ/kg}}$$



To find T_4, T_4' and $W_{T\,(HP)}$

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

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

$$= 1023\left(\frac{3}{6}\right)^{\frac{0.4}{1.4}} = \mathbf{839.2\ \ K}$$

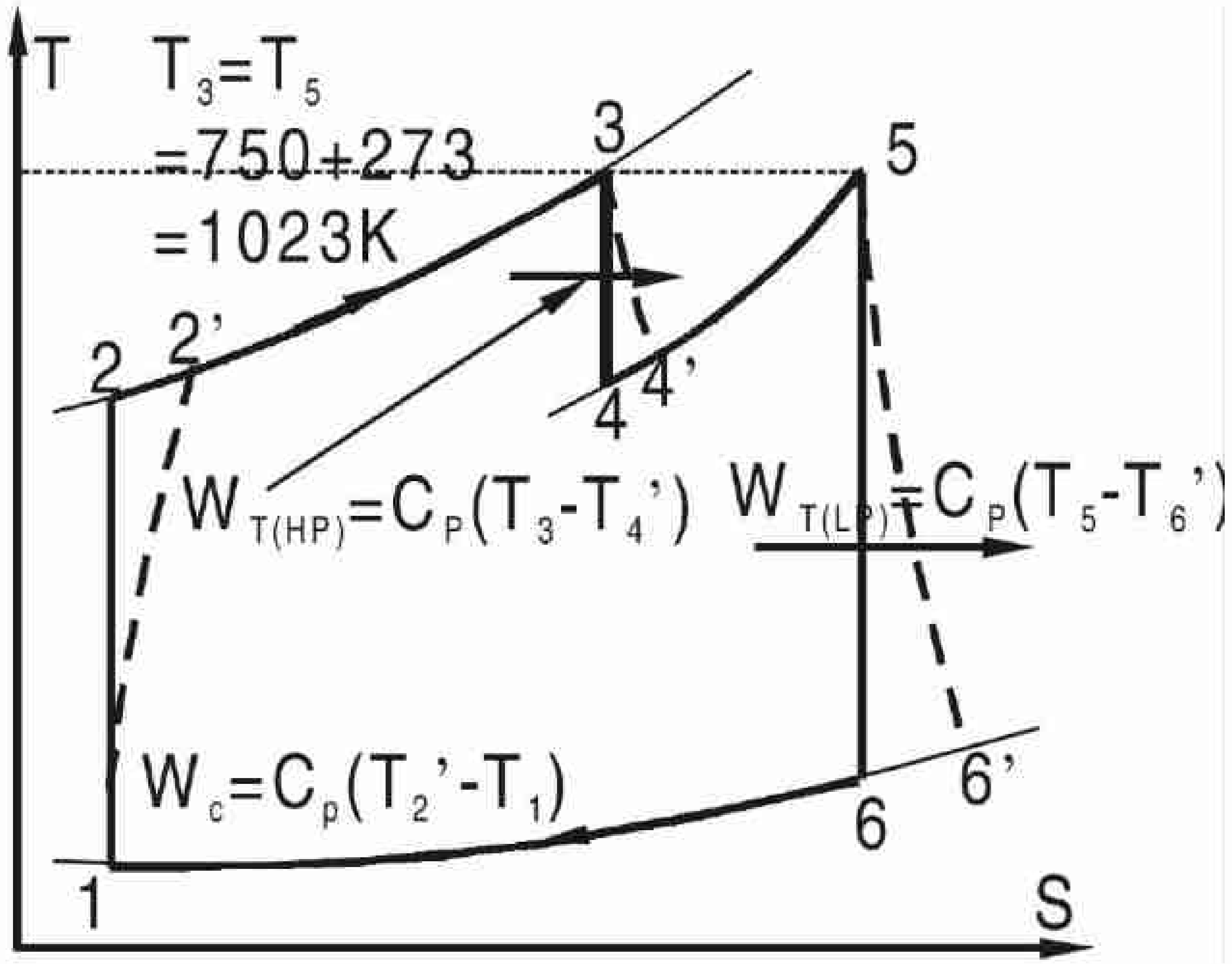
$$\eta_T = 0.82 = \frac{T_3 - T_4'}{T_3 - T_4}$$

$$T_3 - T_4' = 0.82\,(T_3 - T_4)$$

$$T_4' = T_3 - 0.82\,(T_3 - T_4)$$

$$= 1023 - 0.82\,(1023 - 839.2)$$

$$T_4' = \mathbf{872.3\ K}$$



$$W_{T\,(HP)} = C_p\,(T_3 - T_4') = 1.005\,(1023 - 872.3)$$

$$= 151.47\ \text{kJ/kg}$$

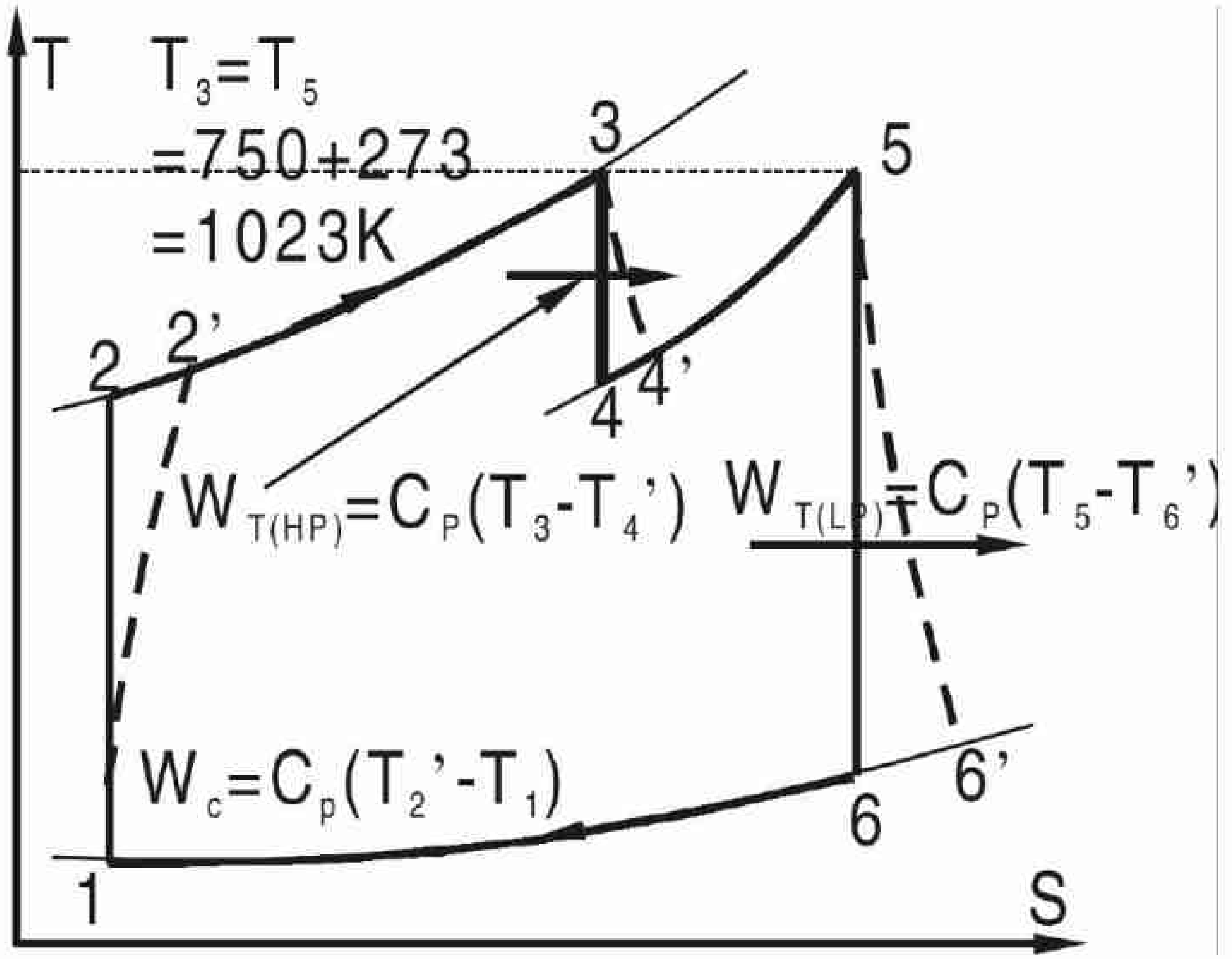
$$\boldsymbol{W_{T\,(HP)} = 151.47\ \text{kJ/kg}}$$

To find T'_6 and $W_{T\,(LP)}$

verify this by calculating T_6'

$T'_6 = T_4' = 872.3$ since $\eta_{T\,(HP)} = \eta_{T\,(LP)}$ and pressure ratio is same for both expansions.

So $\mathbf{W_{T\,(LP)} = 151.47}$ **kJ/kg**



$$\text{Net work done} = W_{T\,(HP)} + W_{T\,(LP)} - W_c$$

$$= 151.47 + 151.47 - 174.57$$

$$= \mathbf{128.37\ kJ/kg}$$

Heat supplied $= Q_s$

$$= C_p\,(T_3 - T_2') + C_p\,(T_5 - T_4')$$

$$= 1.005\,(1023 - 466.7 + 1023 - 872.3)$$

$$= \mathbf{710.54\ kJ/kg}$$

$$\eta_{thermal} = \frac{\text{Net work done}}{\text{Heat supplied}} = \frac{W_{net}}{Q_s}$$

$$= \frac{128.37}{710.54} = 0.181$$

$$= \mathbf{18.1\ \%}$$

To find ($\dot{m}$) mass of fluid circulated.

$$\text{Power} = \dot{m}\ (W_{net})$$

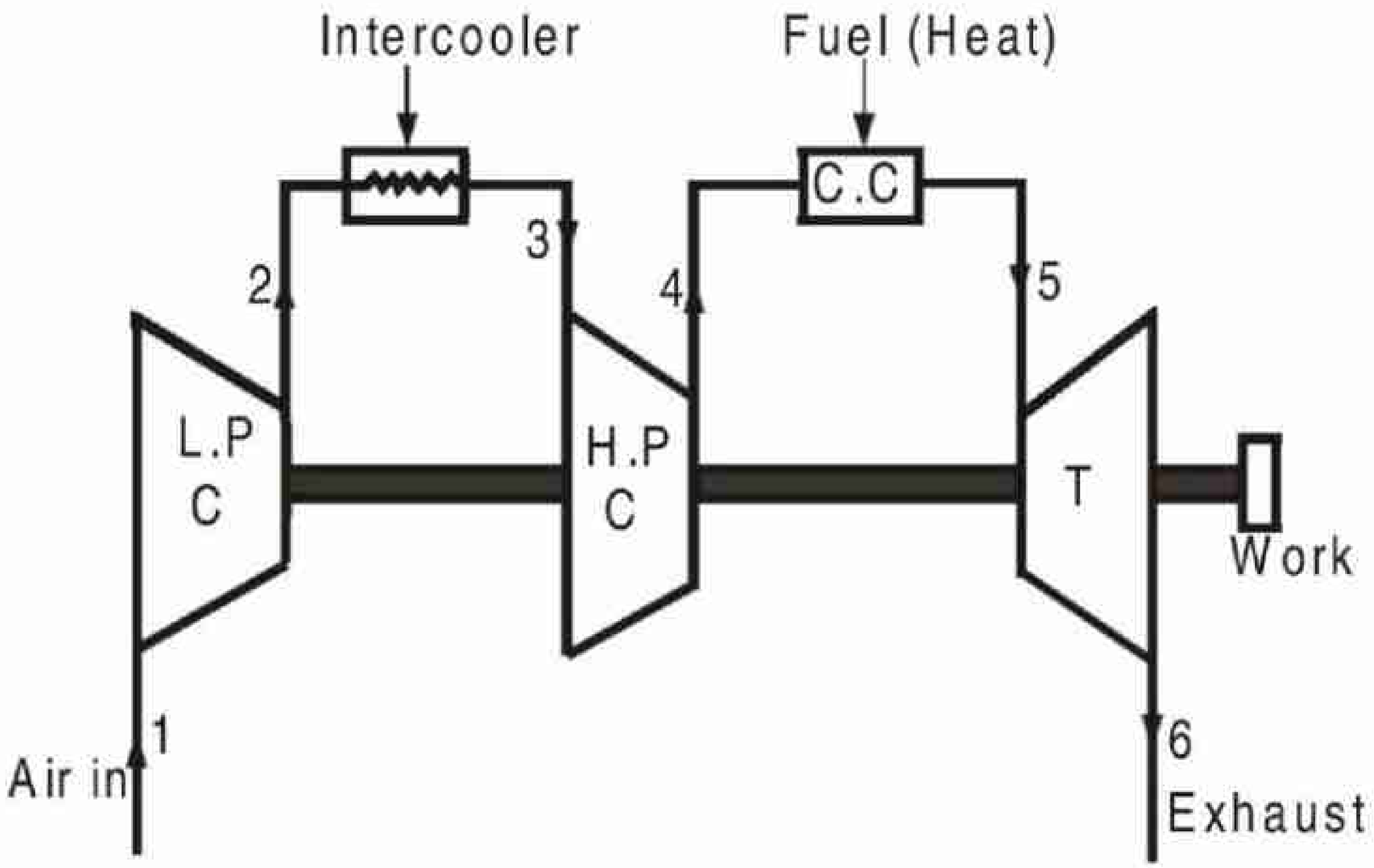
$$500 = \dot{m}\ (128.37) \Rightarrow \dot{m} = \mathbf{3.895\ kg/s}$$

3.Intercooling

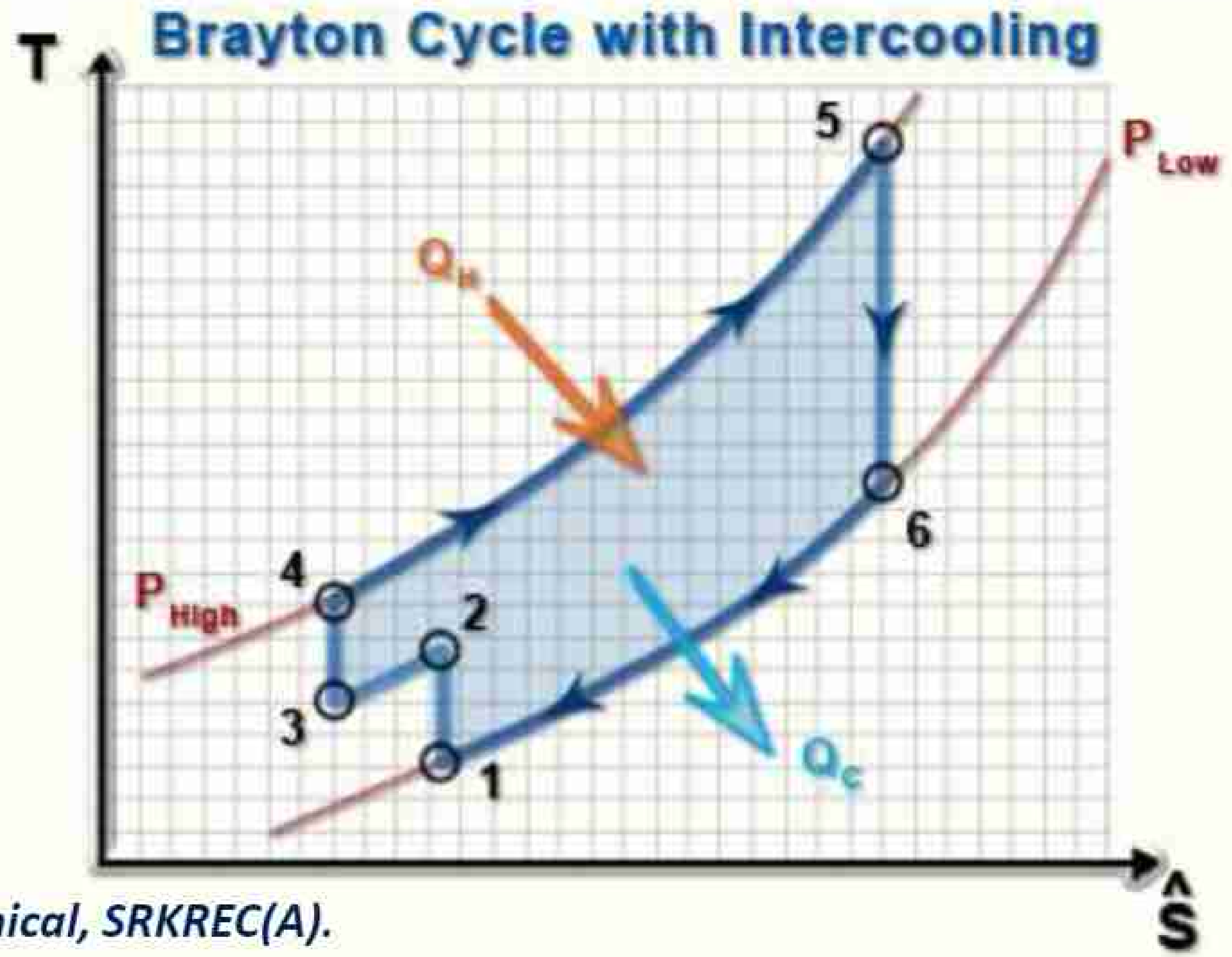
The work required to compress a gas between two specified pressures can be decreased by carrying out the compression process in stages and cooling the gas in between—that is, using ***multistage compression with intercooling***. As the number of stages is increased, the compression process becomes nearly isothermal at the compressor inlet temperature, and the compression work decreases.

The foregoing argument is based on a simple principle: *The steady flow compression or expansion work is proportional to the specific volume of the fluid. Therefore, the specific volume of the working fluid should be as low as possible during a compression process. This is precisely what intercooling accomplish.*

Air is compressed in the LP compressor and then cooled to the initial temperature in the intercooler and then further compressed in the HP compressor. Work required by the compressor is reduced by compressing the air in two stages LP and HP compressor.



Turbine plant with intercooler



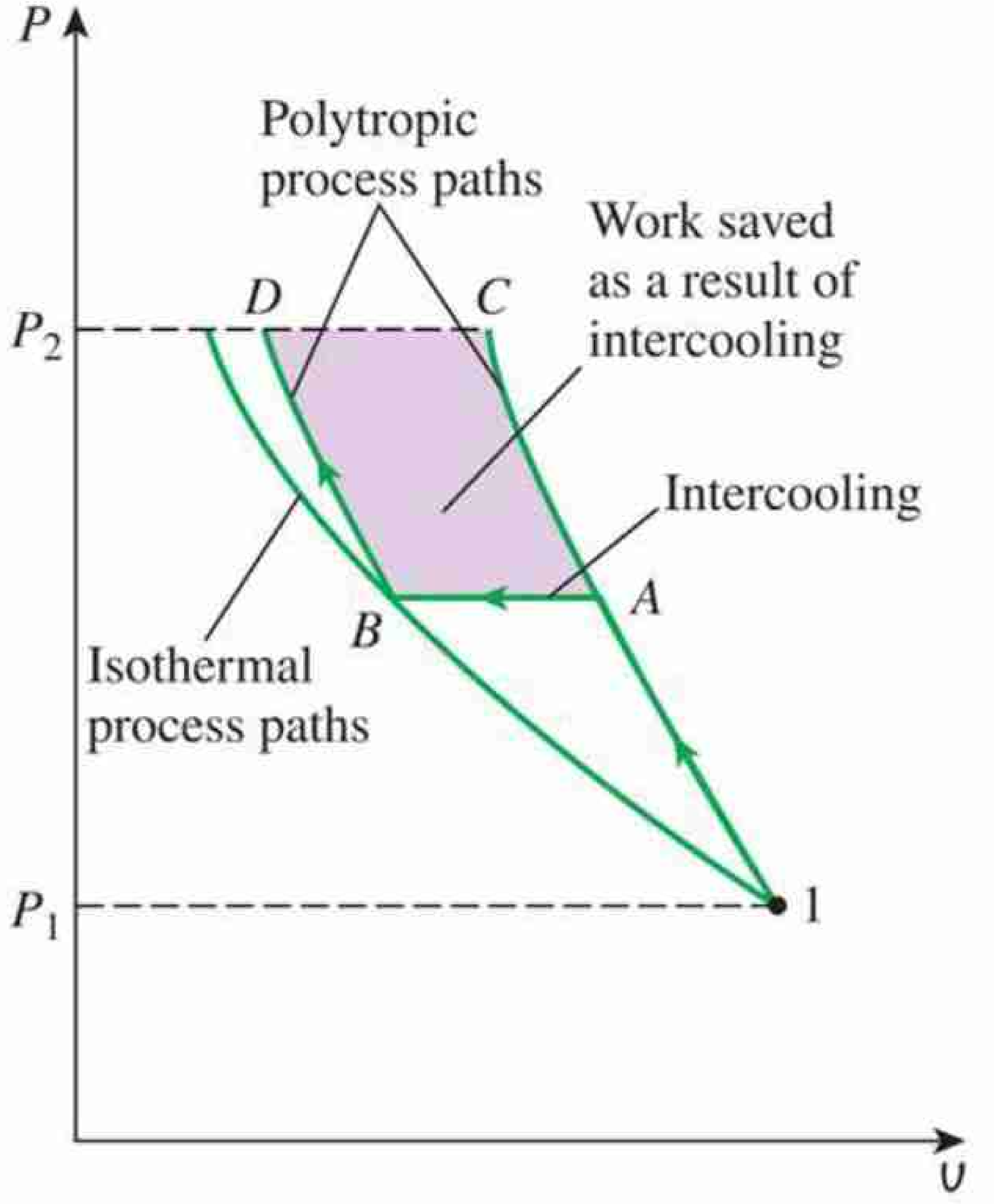
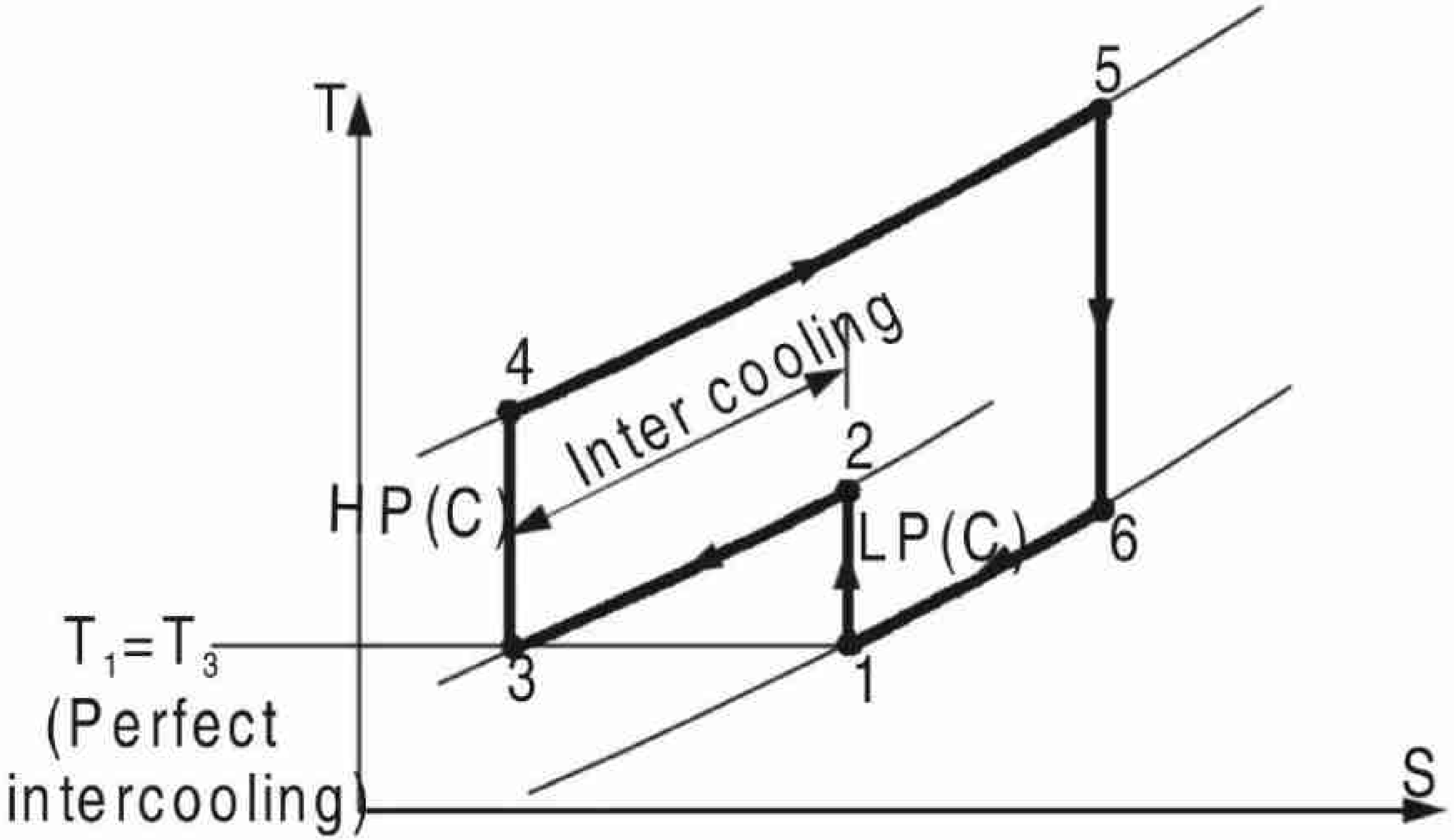


FIGURE
Comparison of work inputs to a single-stage compressor (1*AC*) and a two-stage compressor with intercooling (1*ABD*).



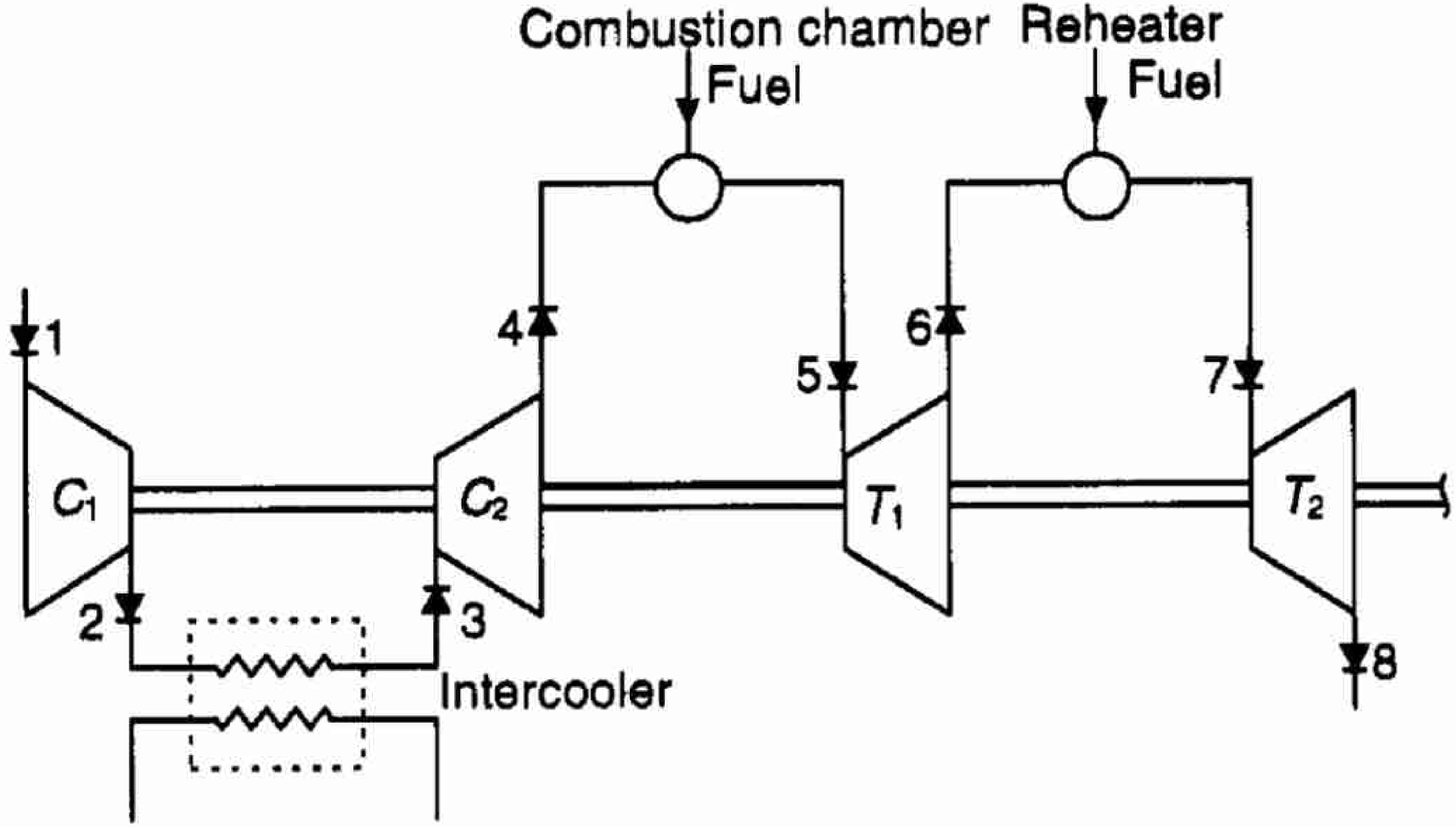
Work input with intercooling $\Big\}$ $W_c = C_p\,(T_2 - T_1) + C_p\,(T_4 - T_3)$

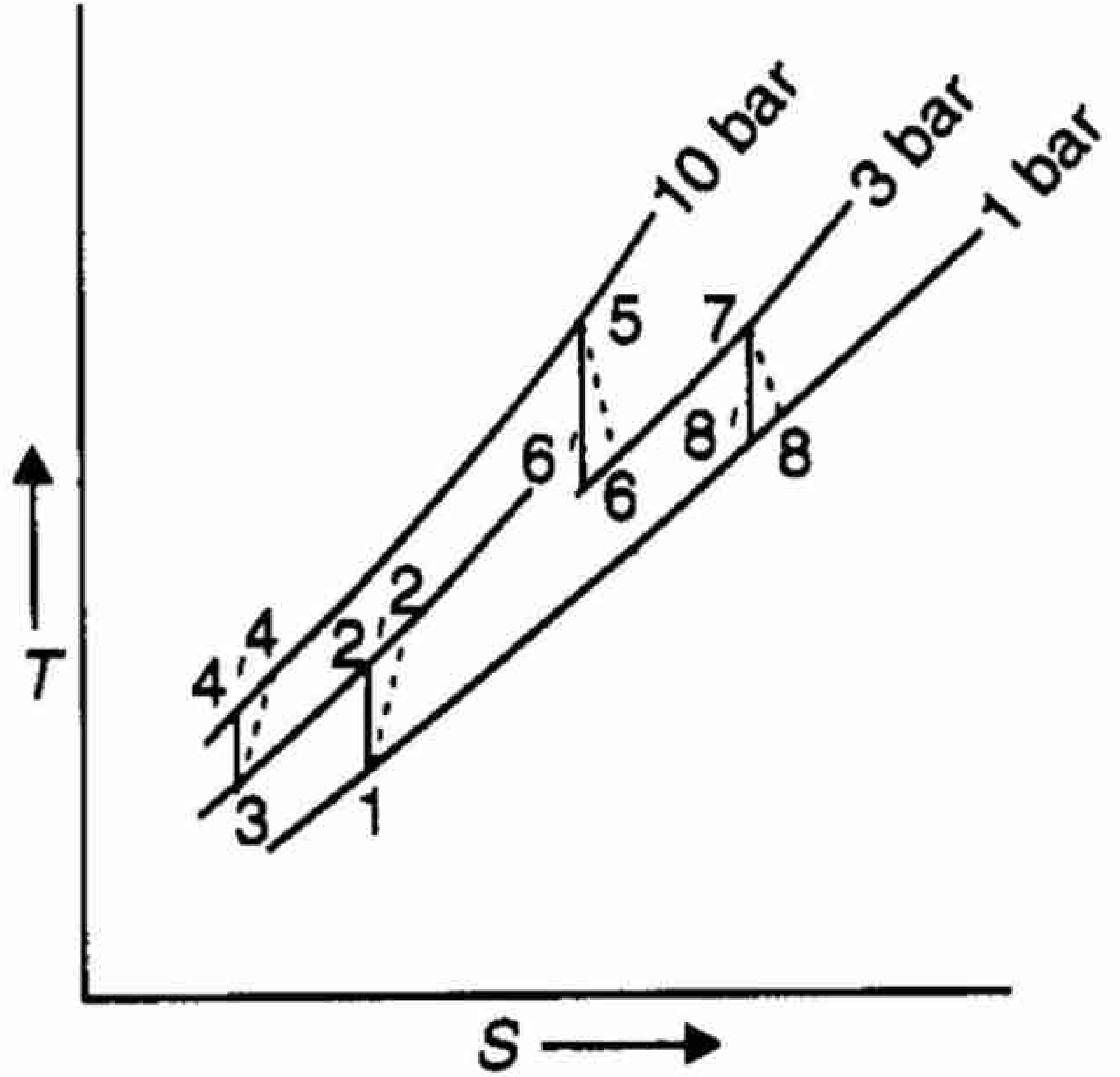
Heat supplied with intercooling $Q_2 = C_p\,(T_5 - T_4)$

Note: Here perfect intercooling means, the condition when the temperature of air leaving the intercooler equals the of air at the compressor intake.

When intercooler is introduced, work input may be reduced. But, the heat supply is increased. So by using the intercooler, the η efficiency of the Brayton cycle may increase or decrease.

A gas turbine plant has air supplied at 1 bar, 27°C for being compressed through pressure ratio of 10. Compression of air is achieved in two stages with perfect intercooling in between at optimum pressure. The maximum temperature in cycle is 1000 K and compressed air at this temperature is sent for expansion in two stages of gas turbine. First stage expansion occurs upto 3 bar and is subsequently reheated upto 995 K before being sent to second stage. Fuel used for heating in combustion chamber has calorific value of 42,000 kJ/kg. Considering c_p = 1.0032 kJ/kg. K throughout cycle determine, net output, thermal efficiency and air fuel ratio when air flows into compressor at 30 kg/s. Take isentropic efficiency of compression and expansion to be 85% and 90% respectively.





Assuming that the pressure ratio in each stage is same, we have

$$\frac{p_2}{p_1} = \frac{p_4}{p_3} = \sqrt{\frac{p_4}{p_1}}$$

For perfect intercooling the pressure ratio of each compression stage = $\sqrt{10}$ = 3.16

For process 1–2′, $\frac{T_{2'}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_{2'} = (3.16)^{\frac{1.4-1}{1.4}} \times 300 = 416.76 \text{ K}$

Considering isentropic efficiency of compression,

$$\eta_c = 0.85 = \frac{T_{2'} - T_1}{T_2 - T_1} = \frac{416.76 - 300}{T_2 - 300}$$

For perfect intercooling, $T_3 = T_1 = 300$ K, $T_2 = 437.36$ K

For process 3–4,

$$\frac{T_{4'}}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{1.4-1}{1.4}} \Rightarrow T_{4'} = 300 \times (3.16)^{\frac{1.4-1}{1.4}} = 416.76 \text{ K}$$

Again due to compression efficiency, $\eta_c = 0.85 = \dfrac{T_{4'} - T_3}{T_4 - T_3}$

$$T_4 = 437.36 \text{ K}$$

Total compressor work, $W_C = 2 \times c_p \times (437.36 - 300) = 275.59$ kJ/kg

$T_5 = 1000$ K, For Expansion process 5–6′

$$\frac{T_{6'}}{T_5} = \left(\frac{P_6}{P_5}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_{6'} = 1000 \times \left(\frac{3}{10}\right)^{\frac{1.4-1}{1.4}} = 708.93 \text{ K}$$

Considering expansion efficiency, $0.90 = \dfrac{T_5 - T_6}{T_5 - T_{6'}}$

$$\Rightarrow \qquad T_6 = 738.04 \text{ K}$$
$$T_7 = 995 \text{ K}$$

For expansion in 7–8′,

$$\frac{T_{8'}}{T_7} = \left(\frac{P_8}{P_7}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\Rightarrow \qquad T_{8'} = 995 \left(\frac{1}{3}\right)^{\frac{1.4-1}{1.4}} = 726.95 \text{ K}$$

Considering expansion efficiency, $0.90 = \dfrac{T_7 - T_8}{T_7 - T_{8'}} \Rightarrow T_8 = 753.75 \text{ K}$

Expansion work output per kg air $= c_p\,(T_5 - T_6) + c_p\,(T_7 - T_8)$

$$W_T = 1.0032\ \{(1000 - 738.04) + (995 - 753.75)\}$$
$$W_T = 514.85 \text{ kJ/kg}$$

Heat added per kg air $= c_p(T_5 - T_4) + c_p(T_7 - T_6)$

$$q_{\text{add}} = 1.0032\ \{(1000 - 437.36) + (995 - 738.04)\}$$

$$q_{\text{add}} = 822.22 \text{ kJ/kg}$$

Fuel required per kg of air, $m_f = \dfrac{822.22}{42000} = 0.01958$

$$\text{Air-fuel ratio} = \frac{1}{0.01958} = 51.07$$

$$\text{Net output} = W_T - W_C = 239.26 \text{ kJ/kg}$$

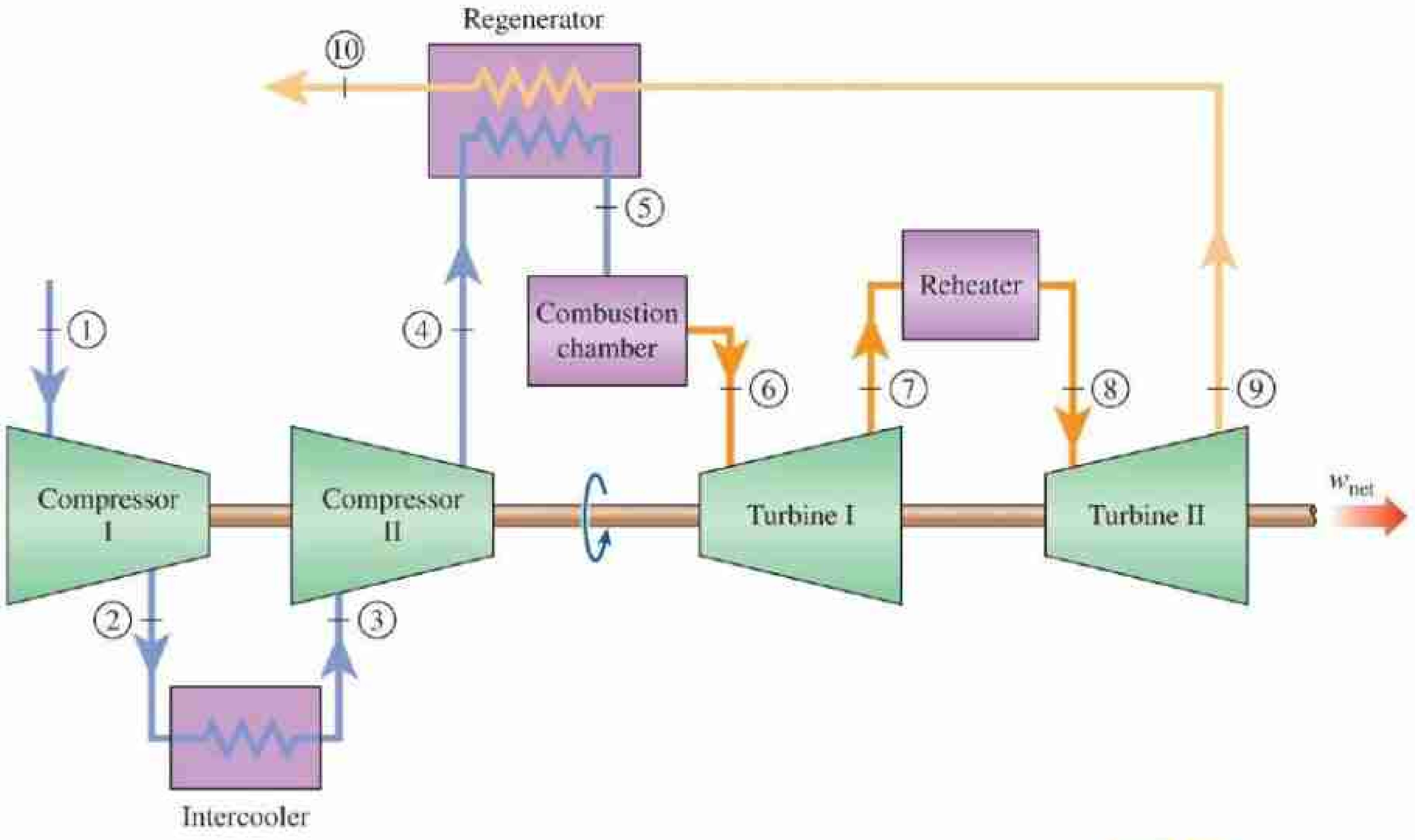
Output for air flowing at 30 kg/s, $= 239.26 \times 30 = 7177.8$ kW

$$\text{Thermal efficiency} = \frac{W_T - W_C}{q_{\text{add}}} = \frac{239.26}{822.22} = 0.2909 \text{ or } 29.09\%$$

$\boxed{\text{Thermal efficiency} = \mathbf{29.09\%},\ \text{Net output} = \mathbf{7177.8}\ \text{kW},\ \text{A/F ratio} = \mathbf{51.07}}$ **Ans.**

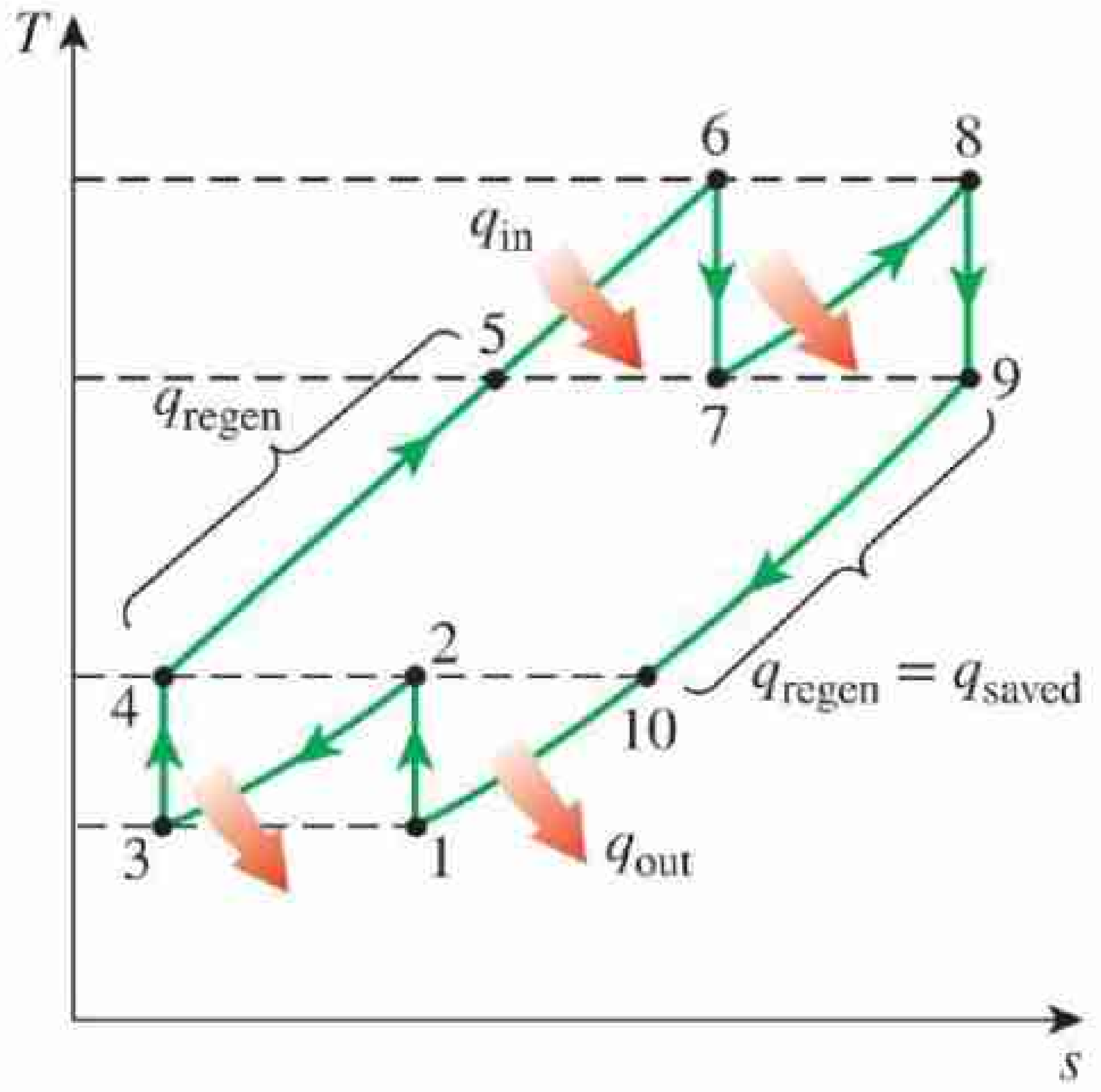
THE BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION

The working fluid leaves the compressor at a lower temperature, and the turbine at a higher temperature, when intercooling and reheating are utilized. This makes regeneration more attractive since a greater potential for regeneration exists. Also, the gases leaving the compressor can be heated to a higher temperature before they enter the combustion chamber because of the higher temperature of the turbine exhaust.



FIGURE

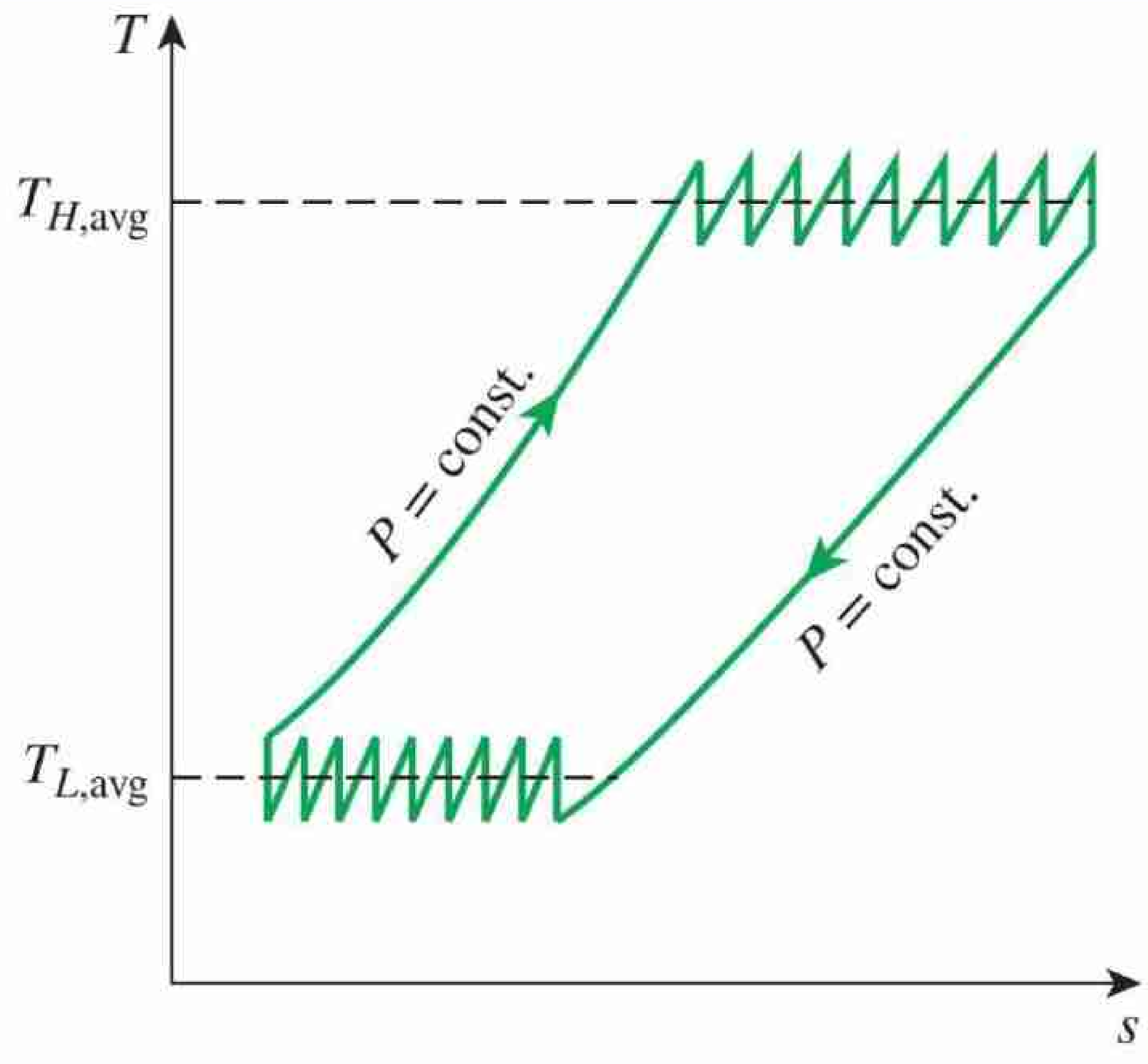
A gas-turbine engine with two-stage compression with intercooling, two-stage expansion with reheating, and regeneration.



FIGURE

T-s diagram of an ideal gas-turbine cycle with intercooling, reheating, and regeneration.

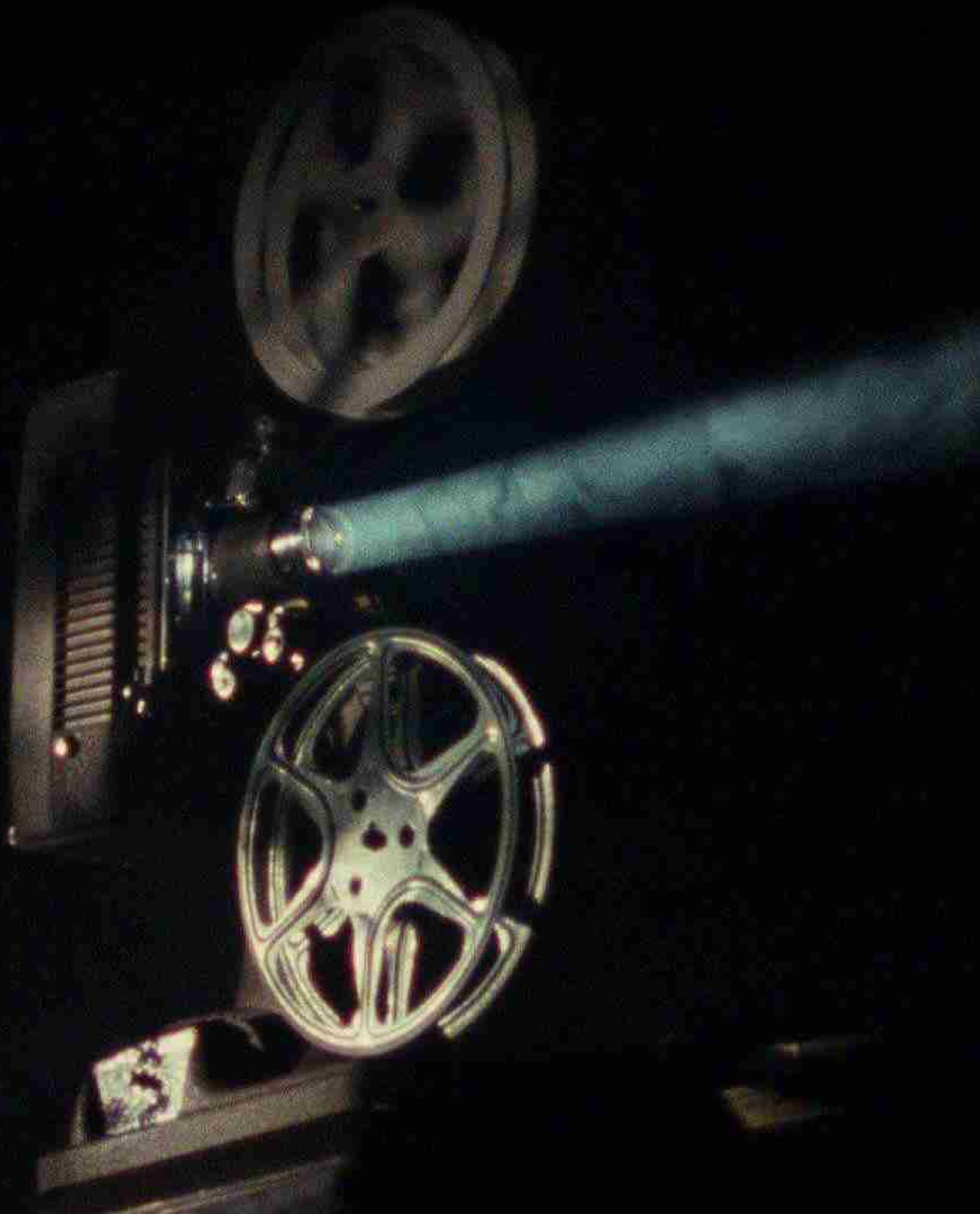
A discussion of ideal cycles for the gas turbine would not be complete without some reference to the theoretical cycle of Ericsson. If the number of compression and expansion stages is increased, the ideal gas-turbine cycle with intercooling, reheating, and regeneration approaches the Ericsson cycle, as illustrated in Fig., and the thermal efficiency approaches the theoretical limit (the Carnot efficiency). However, the contribution of each additional stage to the thermal efficiency is less and less, and the use of more than two or three stages cannot be justified economically.



FIGURE

As the number of compression and expansion stages increases, the gas-turbine cycle with intercooling, reheating, and regeneration approaches the Ericsson cycle.

Some Useful Documentaries



MS9001E


JET PROPULSION
BREITLING
NASA
NAVAIR
Raytheon
NATURAL GAS INDUSTRY'S
THE BLUE FLAME
REACTION DYNAMICS

The principle of jet propulsion involves imparting momentum to a mass of fluid in such a manner that the reaction of imparted momentum provides a propulsive force. It may be achieved by expanding the gas, which is at high temperature and pressure, through a nozzle due to which a high velocity jet of hot gases is produced (in the atmosphere) that gives a propulsive force (in opposite direction due to its reaction).

The propulsion system may be broadly classified as follows :

1. **Air stream jet engines**, (Air-breathing engines)

(*a*) *Steady combustion systems or* continuous air flow engines

Ex: Turbo-jet, turbo shaft, turbo prop, turbo fan, ramjet & scram jet.

(*b*) *Intermittent combustion system or* intermittent flow engines

Ex: Pulse jet or flying bomb.

2. **Self contained rocket engines** (Non-air breathing engines)

(*i*) Liquid propellent (*ii*) Solid propellent (iii) Hybrid propellent

In principle, any fluid can be used to achieve the jet propulsion. Thus water, steam or combustion gases can be used to propel a body in a fluid. But there are limitations in the choice of the fluid when the bodies are to be propelled in the atmosphere. Experience shows that only two types of fluids are particularly suitable for jet propulsion.

(I) *A* ***heated and compressed atmospheric air*** *–*The jet of this character is called a thermal jet and the jet propulsion engine using atmospheric air is called ***air breathing engines***.

(II) ***A jet of gas produced by the chemical reactions of fuel and oxidizer-*** A jet produced in this way is known as ***rocket jet*** and the equipment wherein the chemical reaction takes place is called a ***rocket motor.*** The complete unit including the propellant is called a ***rocket engine.***

SPECIFIC FUEL CONSUMPTION

It is the ratio between fuel consumption rate per unit thrust. Since the output is in the form of thrust, a thrust fuel consumption is

$$TSFC = \frac{\dot{m}_f}{F}$$

It is an important parameter to compare the engine performance of different types of aircraft propulsion systems.

Specific Thrust

It is defined as the thrust produced per unit mass flow rate through the propulsive device.

$$F_{sp} = \frac{F}{\dot{m}}$$

It is an another useful parameter for comparing the different types of propulsion devices.

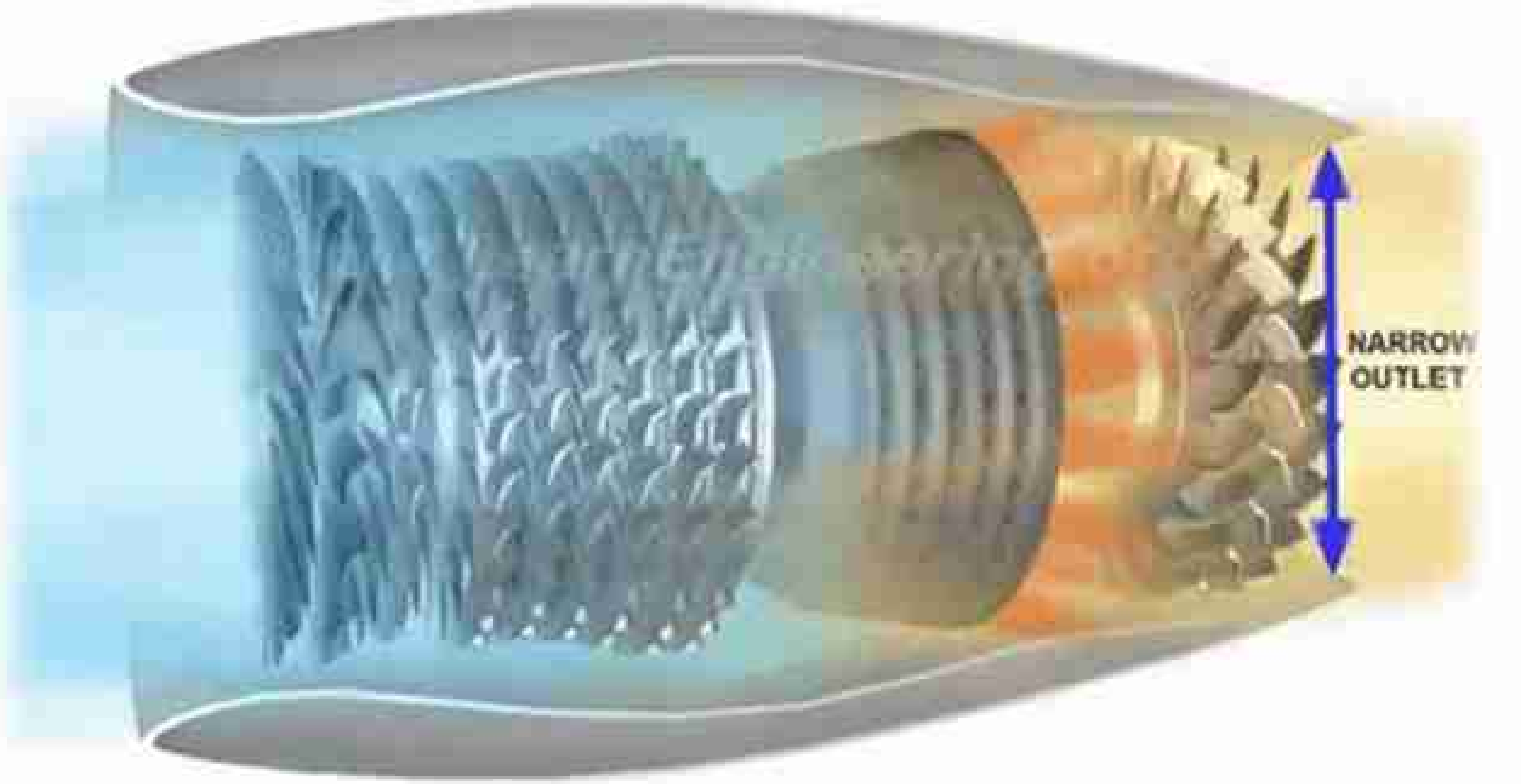
Specific Impulse

It is defined as the thrust produced per unit weight flow rate through the propulsive device. It is also an another useful performance parameter in aircraft propulsion devices.

$$I_{sp} = \frac{F}{\dot{W}}$$

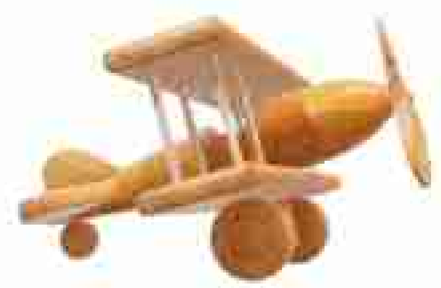
@V.Manikanth, Assistant Professor, Dept. of Mechanical, SRKREC(A).

Jet Propulsion



Air Breathing Engine
(Using atm air to produced Power)

- Jet Engines
  (Propulsive thrust is produced by jet)
  - Gas Turbine Engine
    (Available moving parts like Compressor and Turbine)
    - Turbojet
    - Turboprop
    - Turbofan
    - Turbo-shaft
  - Non-Gas Turbine Engine
    (No moving parts)
    - Ramjet
    - Scramjet
    - Pulsejet
- Reciprocating Engines



RECIPROCATING OR PROPELLER ENGINES

In early days, the source of power for an air breathing engine was a reciprocating internal combustion engine which used to drive a propeller connected to it. The extensive use of aircraft for military purposes led to a very rapid development of reciprocating internal combustion engines during the two world wars and it is now a highly developed piece of equipment as compared to its industrial counterpart.

For small aircraft flying at velocities less than about 500 km/h reciprocating engine is in an enviable position due to its excellent fuel economy and good take-off characteristics.

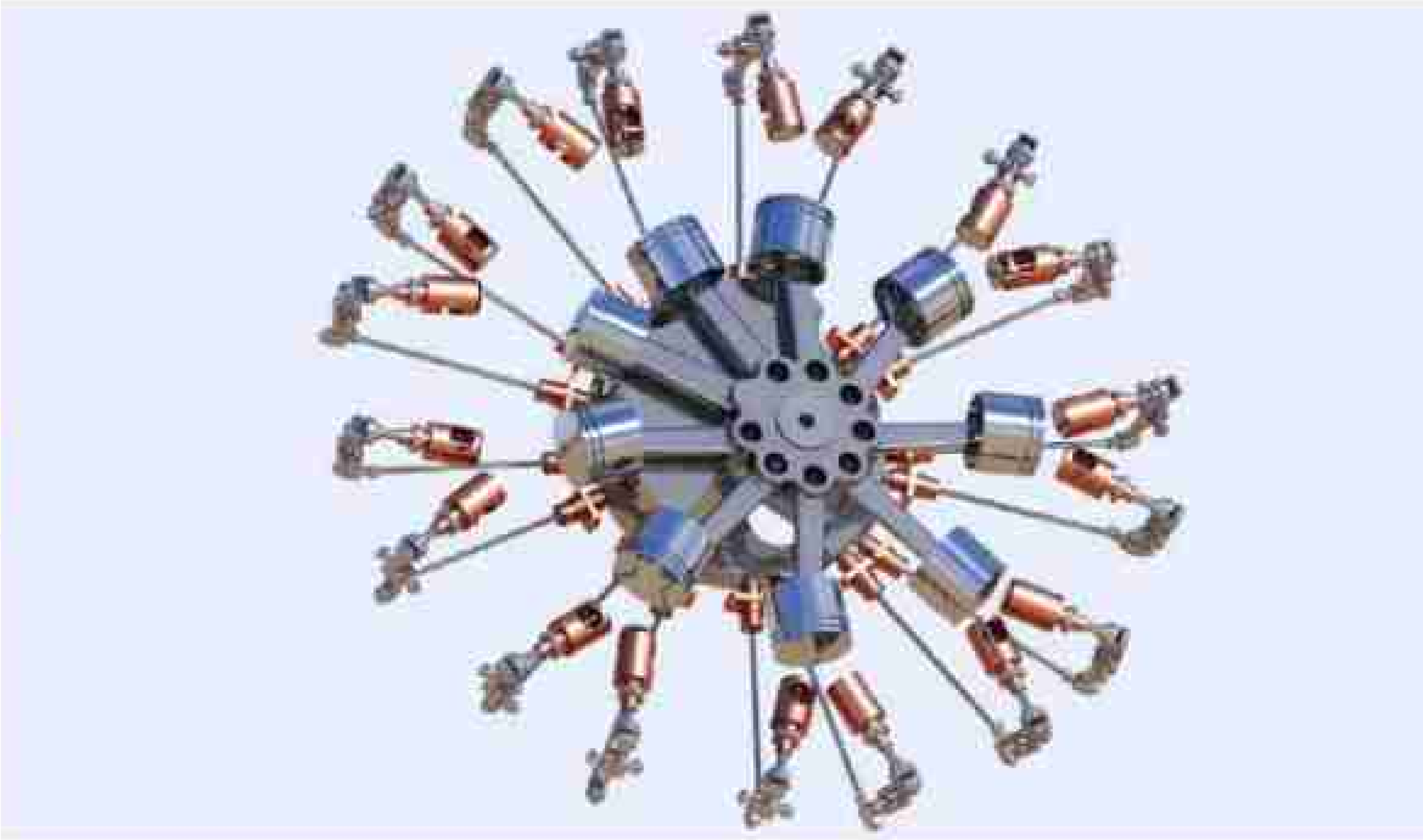
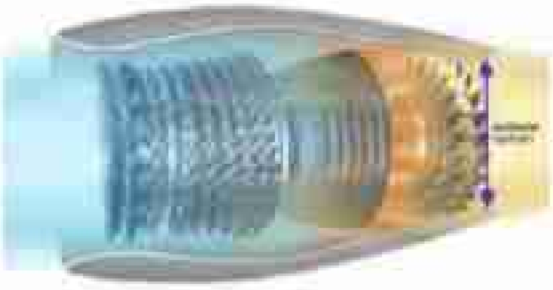


Fig: Radial engine

Due to comparatively large drop in power with altitude operation and the need of using high octane fuels, along with the difficult cooling and lubrication problems, high weight/power ratio, and larger frontal area these are being replaced by turbojets in higher speed ranges.

However, the use of reciprocating engines is continuously on the decline because its development has reached almost a saturation stage as far as maximum power is concerned. The demand of present-day aircraft, in terms of high flight speeds, long distance travels and high load carrying capacities, is soaring to new heights.

Unfortunately, all the methods of raising the output of the engine increase the engine size, frontal area of the aircraft, complexity and cost of the plant. The drag of the plane will also increase to critical values with increase in engine size. Rapid developments in design of turbojet and turboprop engines have started exploding the best fuel economy myth of the reciprocating engines. They are nearing the specific fuel consumption value of reciprocating engines. Hence, for aircraft propulsion, gas turbine engines are the ideal power plant.



Gas Turbine engines

The turbo-jet, turbo-prop, turbofan and turbo shaft are modified forms of simple open cycle gas turbine. A heated and compressed atmospheric air admixed with the products of combustion produced by burning fuel in that air can be used for jet propulsion. The thermochemical energy of the fuel is utilized for increasing the temperature of the air to the desired value. The jet of this character is called a *thermal jet* and the jet propulsion engine using atmospheric air is called *air breathing engines*.

World War II was the turning point for the development of gas turbine technology. All modern aircrafts are fitted with gas turbines. *In practice, the choice of the power plant will depend on the required cruising speed, desired range of the aircraft and maximum rate of climb.*

The details of various gas turbine engines mentioned above are discussed under two categories: *(i) pilotless operation, and (ii) piloted operation*. The ramjet and pulse jet engines come under the category of pilotless operation whereas the turboprop and turbojet engines are used for piloted operation.

i) Turbo jet engine:

The most common type of air breathing engine is the turbojet engine.

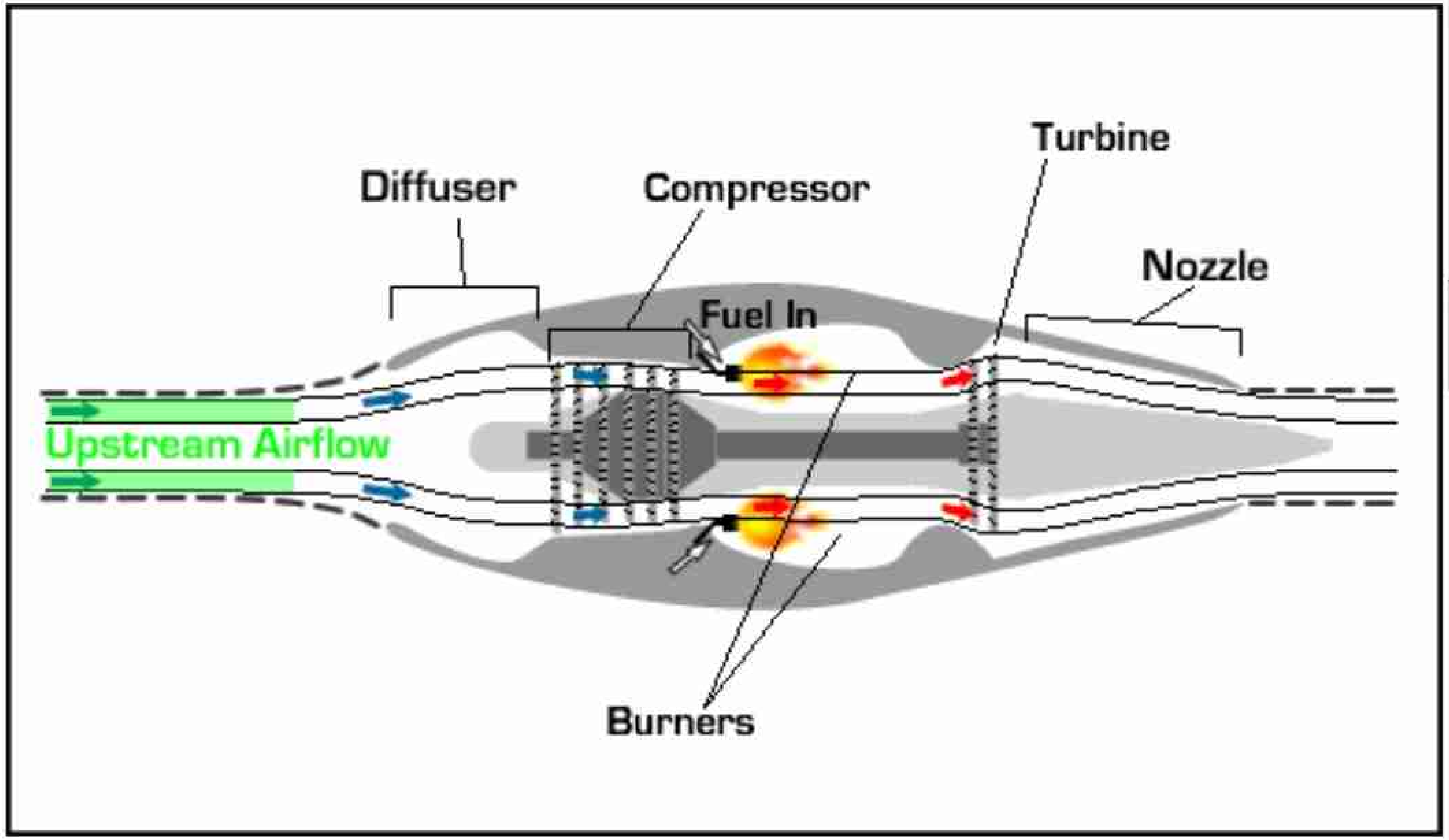
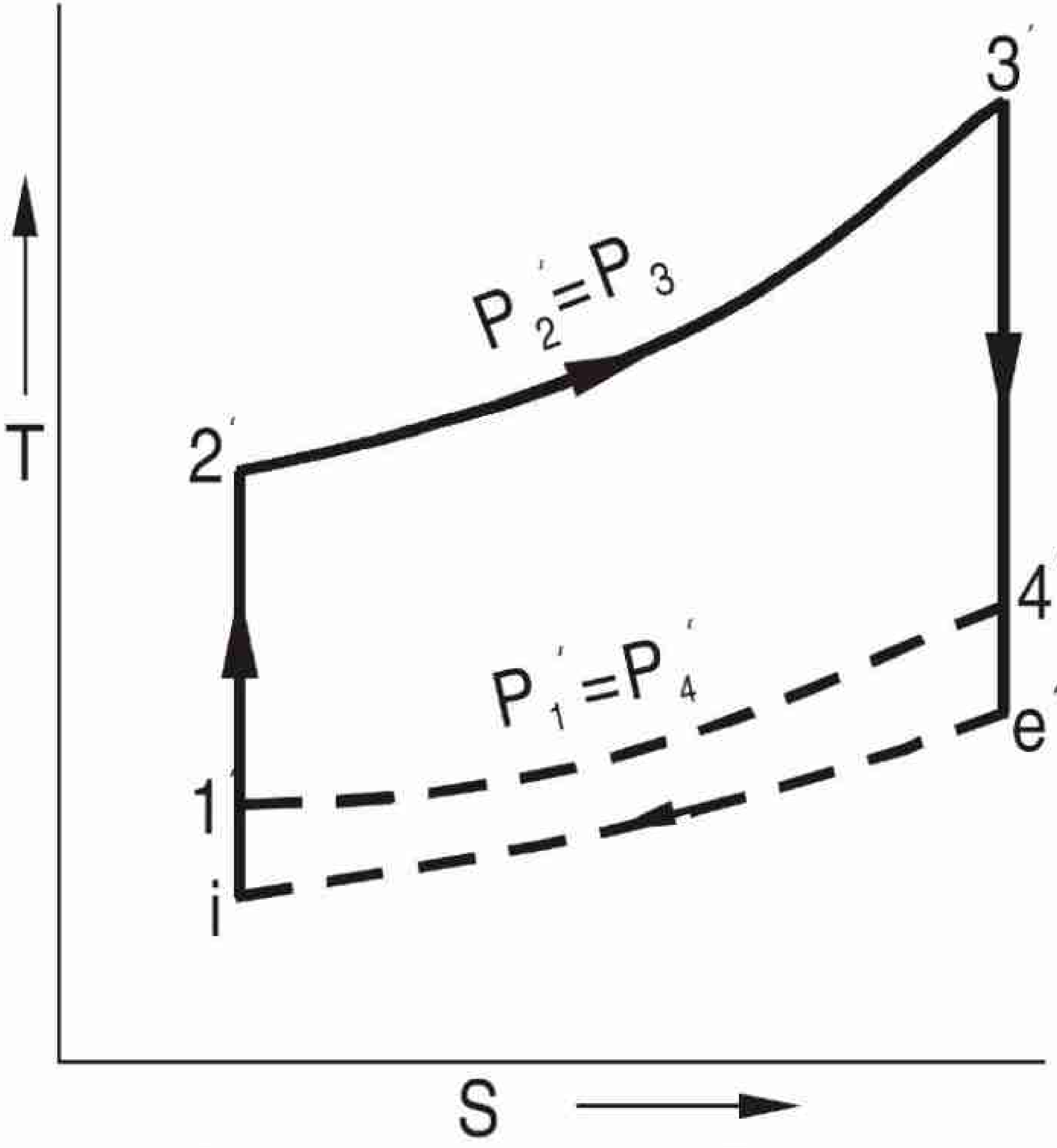


Fig: General Electric J85-GE-17A Turbojet Engine, Cutaway | National Air and Space Museum

This engine consists of
I. inlet diffuser 2. compressor
3.combustion chamber
4.turbine 5.exhaust nozzle.



Ideal Joule or Brayton Cycle

Actual Joule or Brayton Cycle

$i - 1 \Rightarrow$ Inlet diffuser

$1 - 2 \Rightarrow$ Air compressor

$2 - 3 \Rightarrow$ Combustion chamber

$3 - 4 \Rightarrow$ Turbine

$4 - e \Rightarrow$ Nozzle or tail pipe

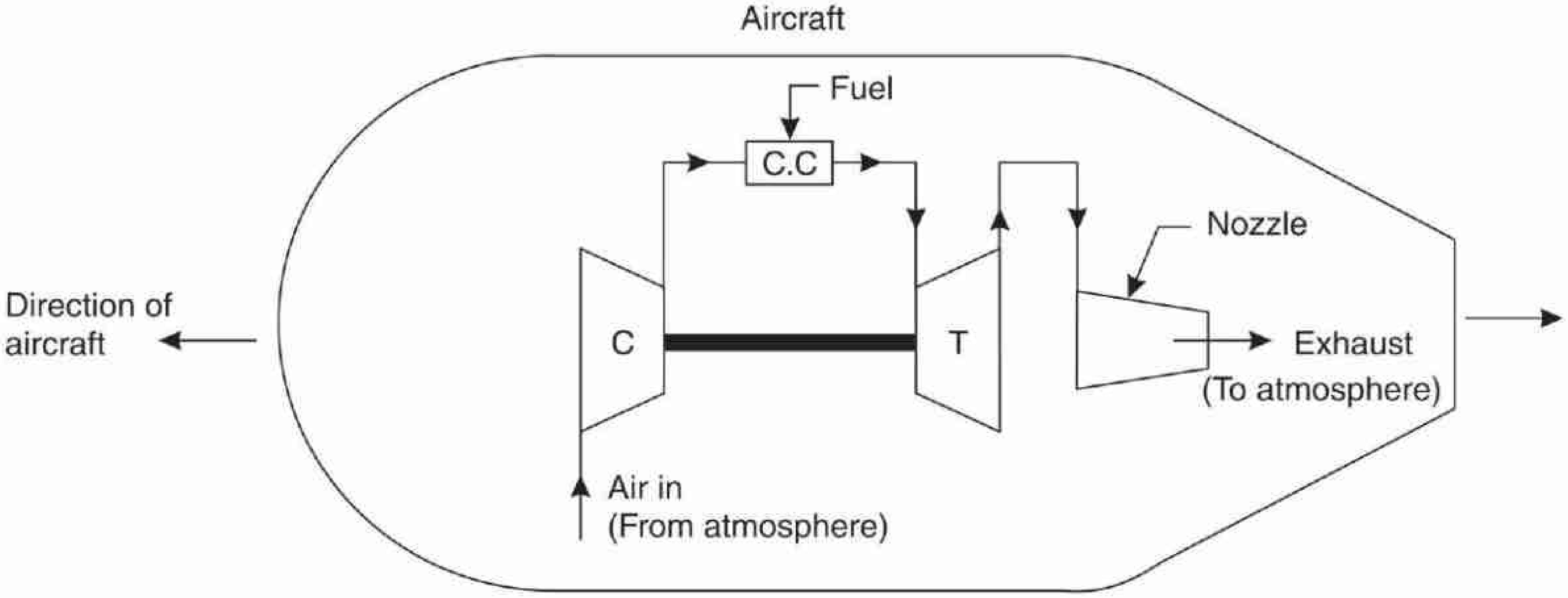


Fig. Gas turbine plant for turbo-jet.

Needs and Demands Met by Turbo Jet Engine

1. Lower frontal area due to the absence of fan. Therefore, the drag is less.
2. Suitable for long distance flights at higher altitudes and speeds.
3. Since this engine has a compressor, it can operate under static conditions.
4. Reheat can be possible to increase the thrust.
5. Lower weight per unit thrust at design speed and altitude.
6. Since a diffuser is at the inlet, part of the compression is done by it without any work input.

Disadvantages

1. Propulsive efficiency and thrust are lower at lower speeds.
2. Thrust specific fuel consumption is high at low speeds and altitudes.
3. It is not economical for short distance flights.
4. Long runway is required due to slower acceleration.
5. Sudden decrease of speed is difficult to achieve.

APPLICATIONS:

Turbo jet engines are used in military aircrafts, guided missiles and piloted aircrafts, etc.

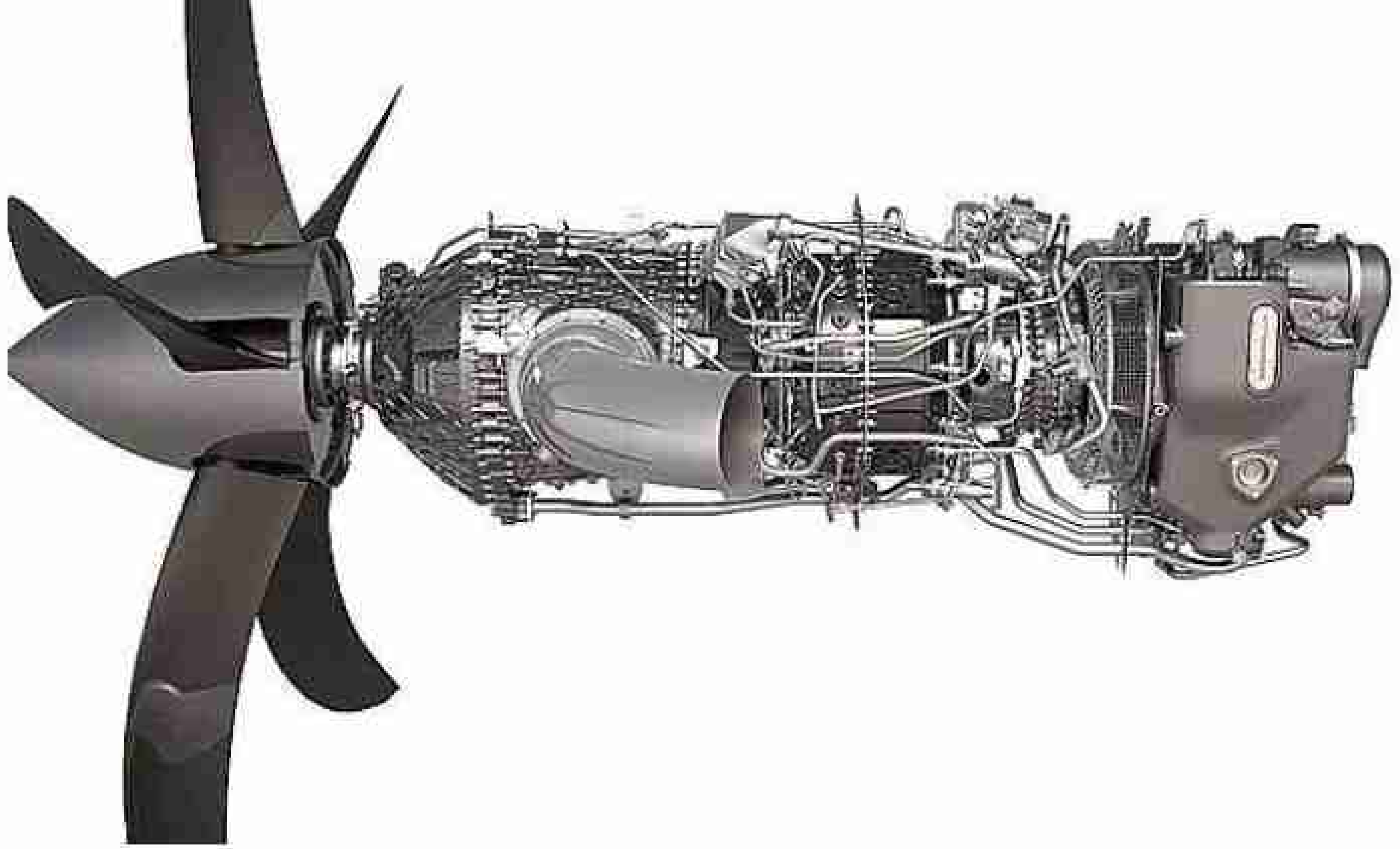
Tu-144, the world's first commercial supersonic transport aircraft

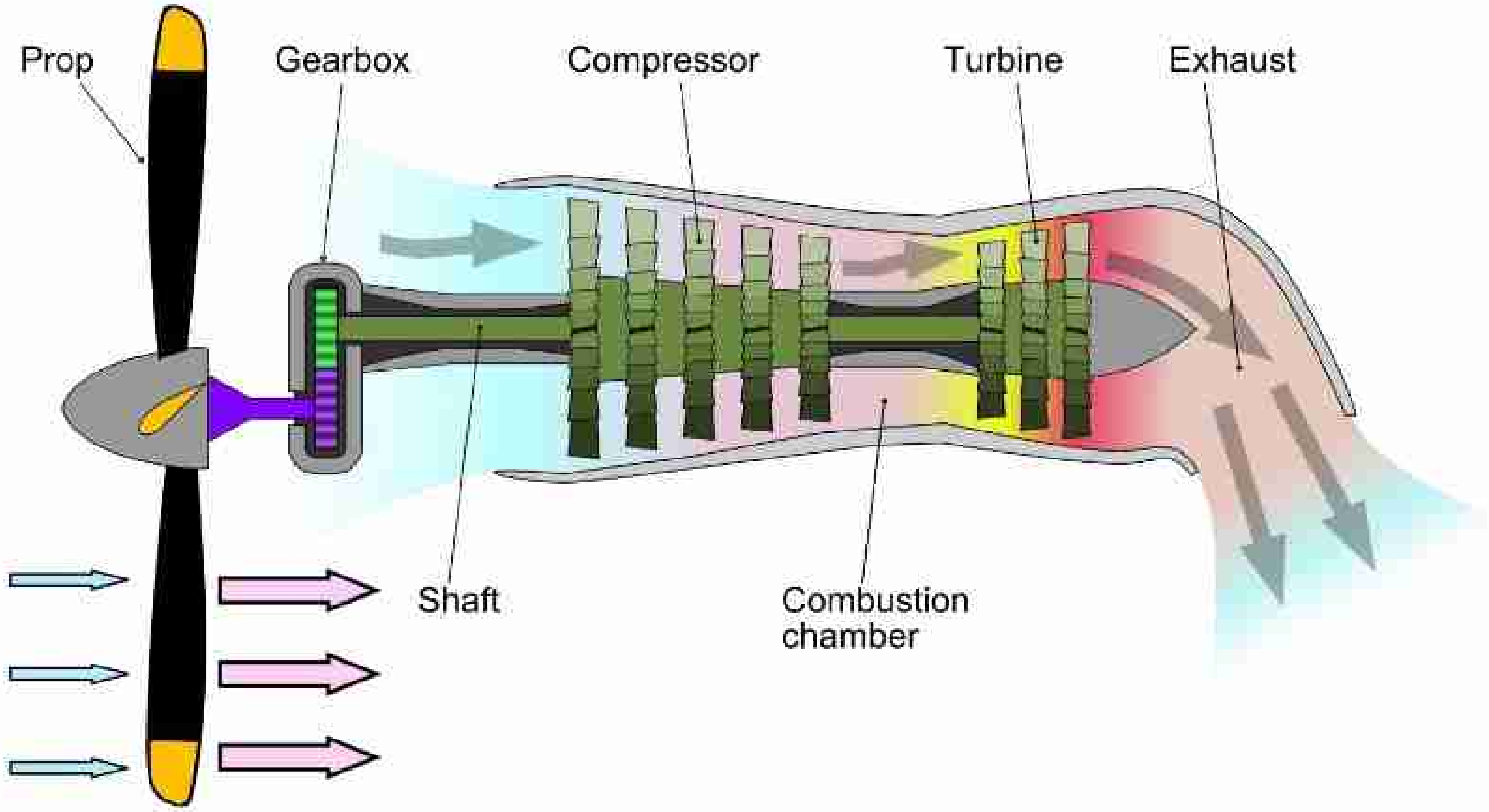
United States Tomahawk cruise missile

MiG-25PU two-seat super sonic interceptor

ii) Turbo prop engine:

A turboprop engine is a turbine engine that drives an aircraft propeller. In its simplest form a turboprop consists of an intake, compressor, combustor, turbine, and a propelling nozzle. Air is drawn into the intake and compressed by the compressor. Fuel is then added to the compressed air in the combustor, where the fuel-air mixture then combusts. The hot combustion gases expand through the turbine. Some of the power generated by the turbine is used to drive the compressor. Thrust is obtained by the combusting gases, pushing toward a (vectored) surface in front of the expanding gas.





The rest is transmitted through the reduction gearing to the propeller. Further expansion of the gases occurs in the propelling nozzle, where the gases exhaust to atmospheric pressure. The propelling nozzle provides a relatively small proportion of the thrust generated by a turboprop.

In contrast to a turbojet, the engine's exhaust gases do not generally contain enough energy to create significant thrust, since almost all the engine's power is used to drive the propeller.

Advantages

1. Propulsive efficiency is very high.
2. The TSFC based on thrust is low.
3. High acceleration at lower speed enables to a shorter runway.
4. Thrust reversal is possible by varying the blade angle, this gives the advantage of decreasing the speed drastically.
5. Used for shorter distance travels. ($C < 600$ Kmph)

Disadvantages

1. Heavier propeller, compressor and turbine decreases pay load capacity.

2. A reduction gear is required to transmit the power from the turbine shaft to the propeller shaft.

3. If the speed of the engine increases above 600 Kmph, the efficiency drastically decreases.

4. The frontal area is being blocked on account of large diameter propeller which increases the co-efficient of drag.

5. Engine is heavier and more complicated.


CESSNA 425


Beechcraft King Air 350i

Piaggio Avanti Evo

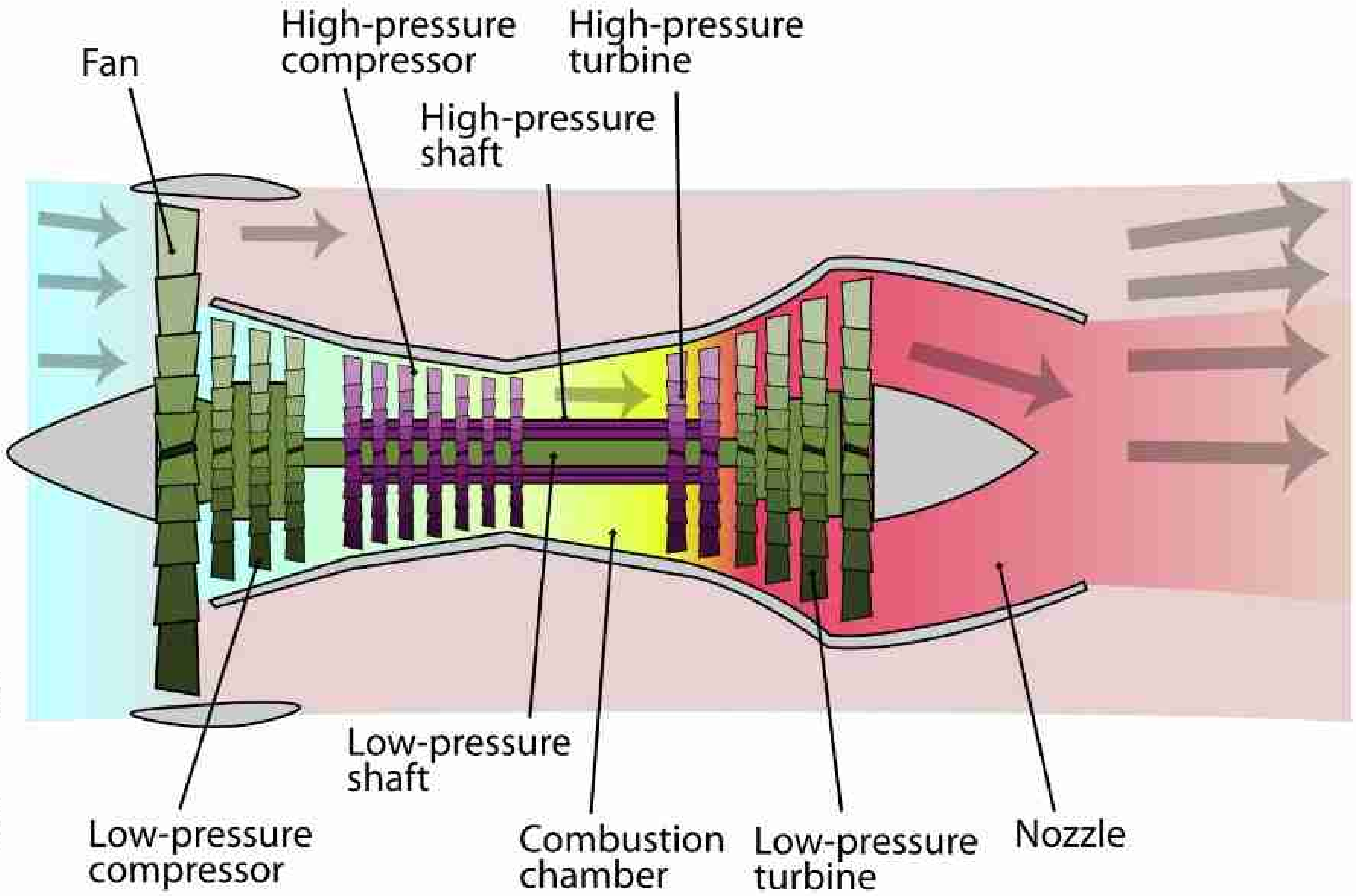
SAB340


Merlin III

iii) Turbo fan:

The turbofan or fanjet is a type of airbreathing jet engine that is widely used in aircraft propulsion. The word "turbofan" is a portmanteau of "turbine" and "fan“.

The ratio of the mass-flow of air bypassing the engine core divided by the mass-flow of air passing through the core is referred to as the bypass ratio.

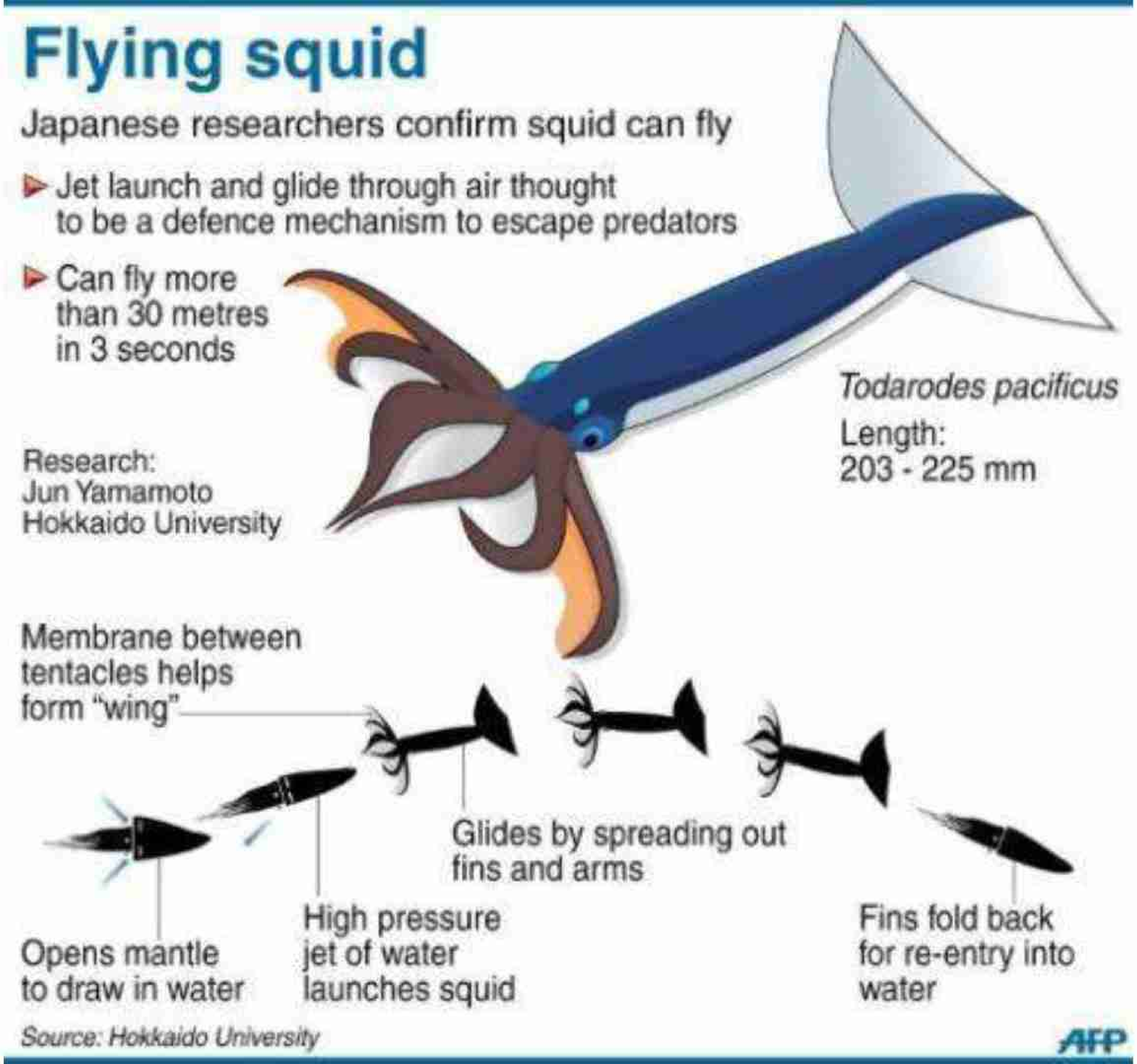


The engine produces thrust through a combination of these two portions working together; engines that use more jet thrust relative to fan thrust are known as low-bypass turbofans, conversely those that have considerably more fan thrust than jet thrust are known as high-bypass.

Most commercial aviation jet engines in use today are of the high-bypass type, and most modern military fighter engines are low-bypass. Afterburners are not used on high-bypass turbofan engines but may be used on either low-bypass turbofan or turbojet engines. Modern turbofans have either a large single-stage fan or a smaller fan with several stages.

Chevrons on an Air India Boeing 787 GE GEnx engine

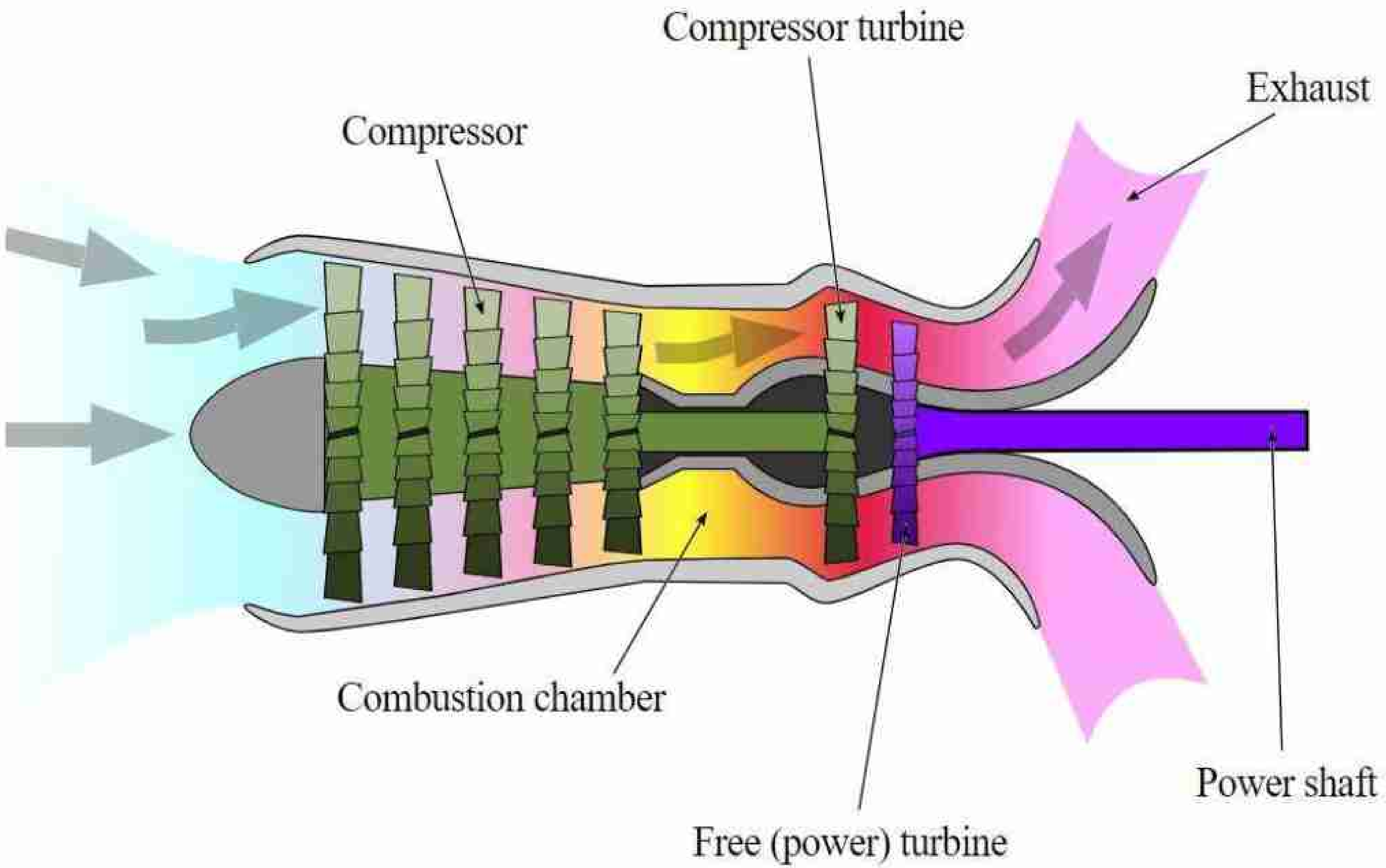
Soloviev D-30


Flying squid
Japanese researchers confirm squid can fly
Jet launch and glide through air thought to be a defence mechanism to escape predators
Can fly more than 30 metres in 3 seconds
Research: Jun Yamamoto Hokkaido University
Todarodes pacificus
Length: 203 - 225 mm
Membrane between tentacles helps form "wing"
Glides by spreading out fins and arms
Opens mantle to draw in water
High pressure jet of water launches squid
Fins fold back for re-entry into water
Source: Hokkaido University
AFP

iv) Turbo shaft:

A turboshaft engine is a form of gas turbine that is optimized to produce shaft power rather than jet thrust. In concept, turboshaft engines are very similar to turbojets, with additional turbine expansion to extract heat energy from the exhaust and convert it into output shaft power. They are even more like turboprops, with only minor differences, and a single engine is often sold in both forms.

Turboshaft engines are commonly used in applications that require a sustained high-power output, high reliability, small size, and light weight. These include helicopters, auxiliary power units, boats and ships, tanks, hovercraft, and stationary equipment

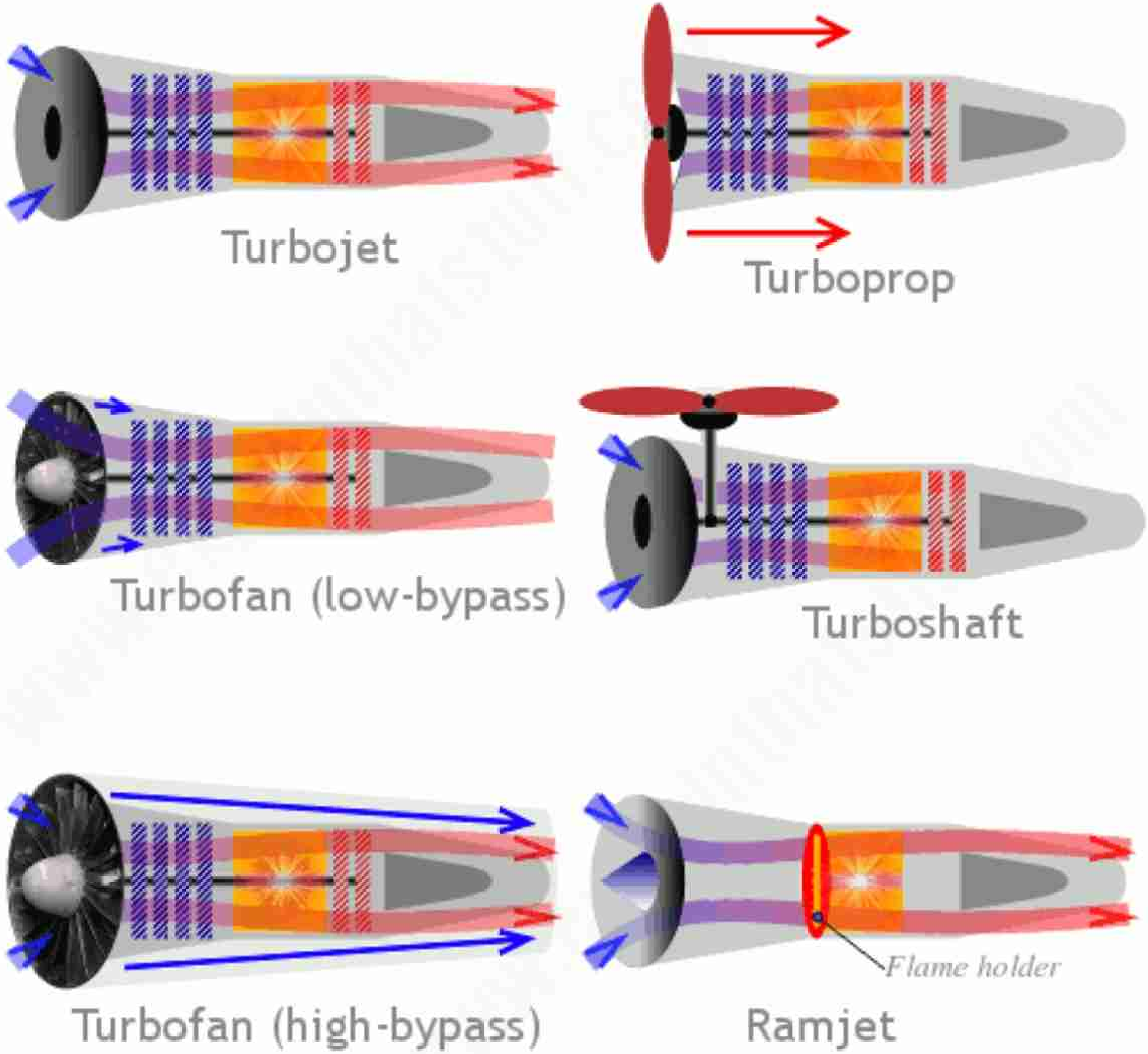


Boeing AH-64E Apache Guardian

Russian Air Force Mi-26

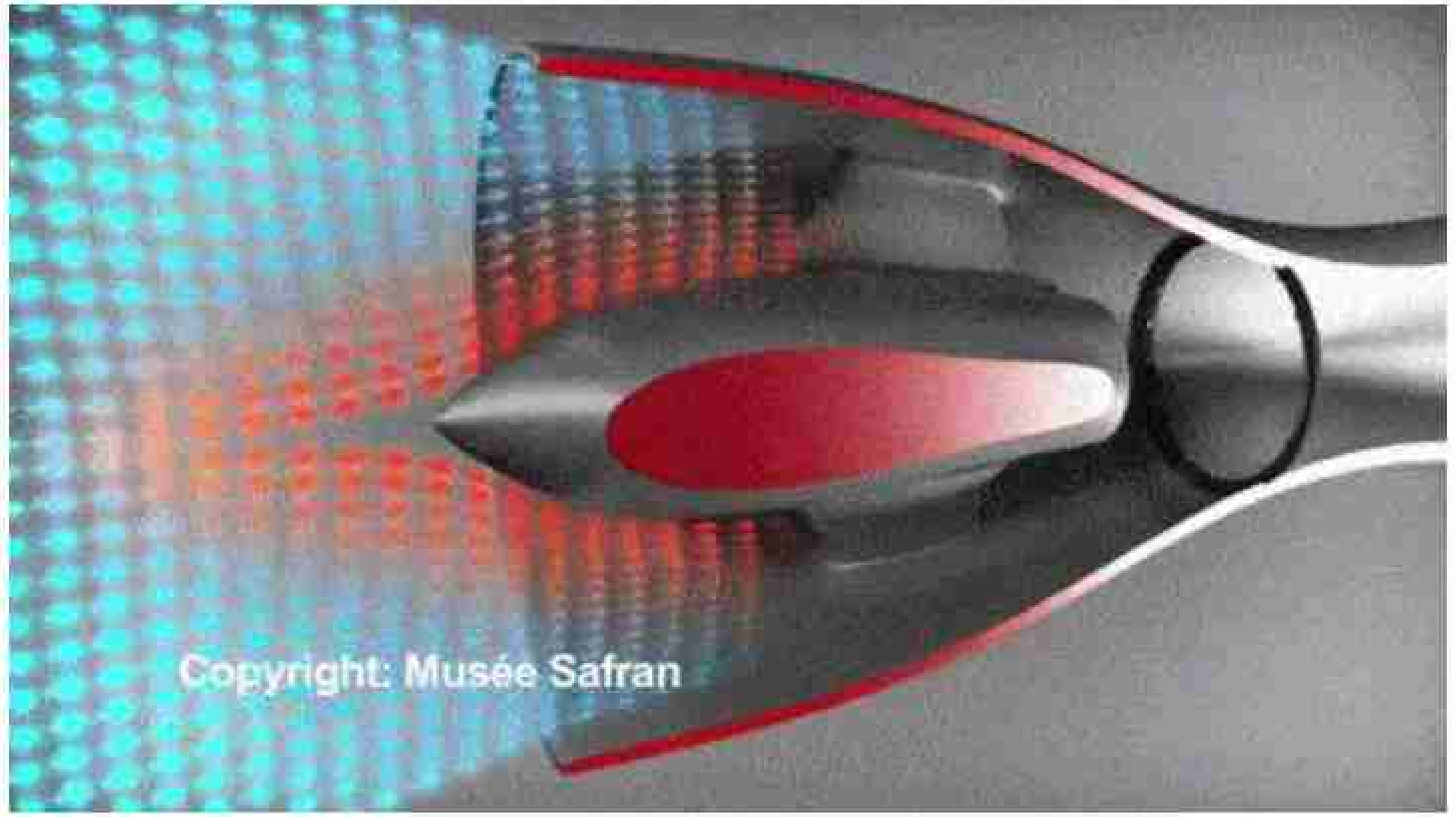
Leopard 2A7-boevoy tank

U.S. Navy LCAC Hovercraft


Turbojet
Turboprop
Turbofan (low-bypass)
Turboshaft
Flame holder
Turbofan (high-bypass)
Ramjet

RAMJET ENGINE

A ramjet, sometimes referred to as a flying stovepipe or an athodyd (aero thermodynamic duct), is a form of airbreathing jet engine that uses the engine's forward motion to compress incoming air without an axial compressor or a centrifugal compressor. Because ramjets cannot produce thrust at zero airspeed, they cannot move an aircraft from a standstill.



A ramjet-powered vehicle, therefore, requires an assisted take-off like a rocket assist to accelerate it to a speed where it begins to produce thrust. Ramjets work most efficiently at supersonic speeds around Mach 3 (2,300 mph; 3,700 km/h). This type of engine can operate up to speeds of Mach 6 (4,600 mph; 7,400 km/h).

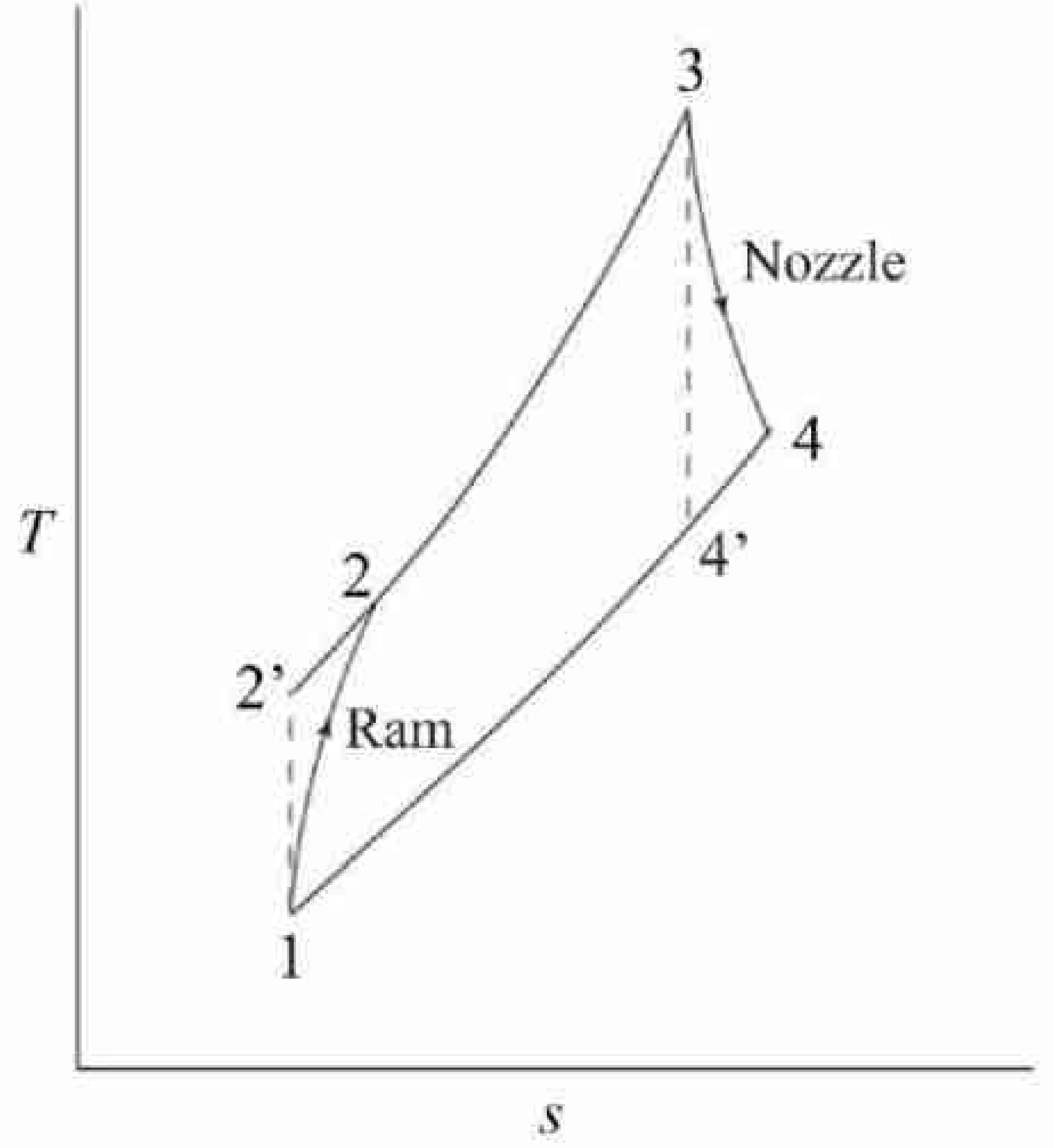
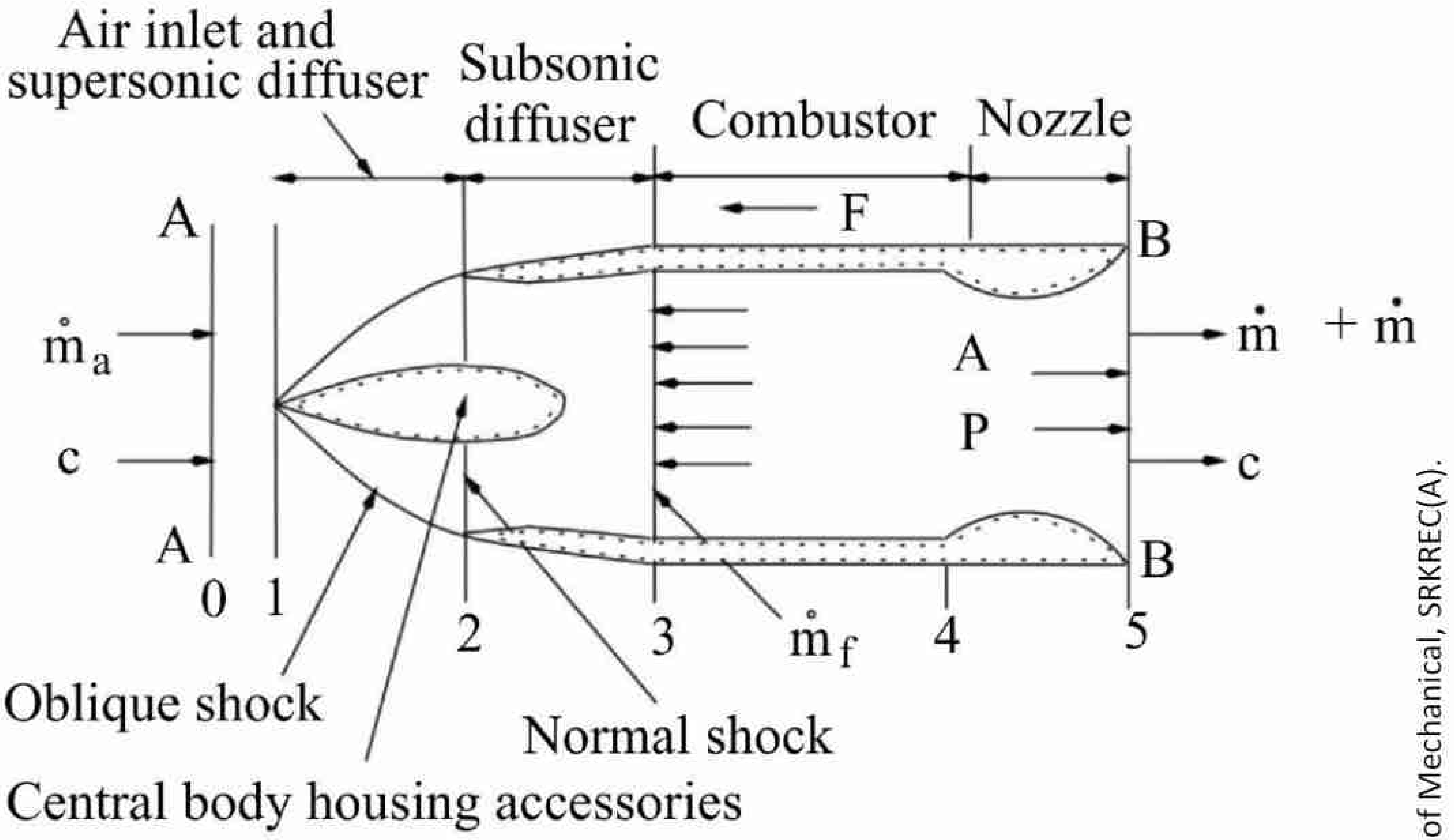
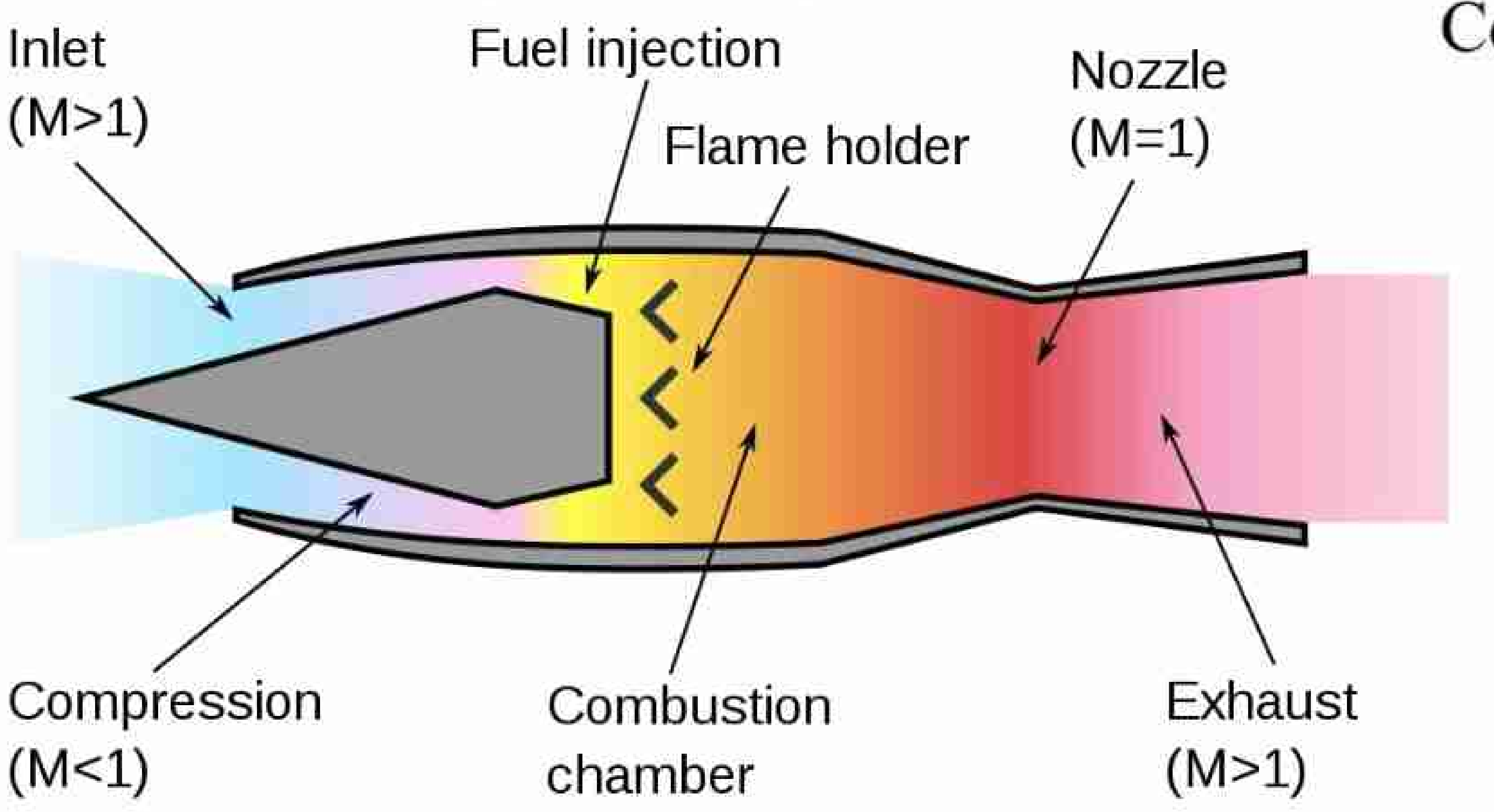


Fig. Ramjet cycle on *T-s* diagram





The engine consists of

(i) supersonic diffuser (1–2),

(ii) subsonic diffuser section (2–3),

(iii) combustion chamber (3–4), and

(iv) discharge nozzle section (4–5).

@V.Manikanth, Assistant Professor, Dept. of Mechanical, SRKREC(A).

Advantages

1. Pay load capacity is very high due to the absence of fan, compressor and turbine.

2. Its fuel consumption decreases with flight speed and approaches reasonable values when the flight mach number is between 2 to 4 and therefore, it is suitable for propelling supersonic missiles.

3. Since the frontal area is less, the co-efficient of drag is low.

4. It increases the mechanical efficiency due to the absence of sliding and moving parts.

5. High temperature and pressure can be employed.

Disadvantages

1. A starting device is required to propel ram jet upto supersonic speed.

2. Altitude limitation is there.

3. It has low thermal efficiency and high TSFC.

4. Due to high temperature of gas coming out from the nozzle, erosion occurs at the exit of the nozzle.

Applications

Used in guided missiles and high supersonic speed aircrafts.

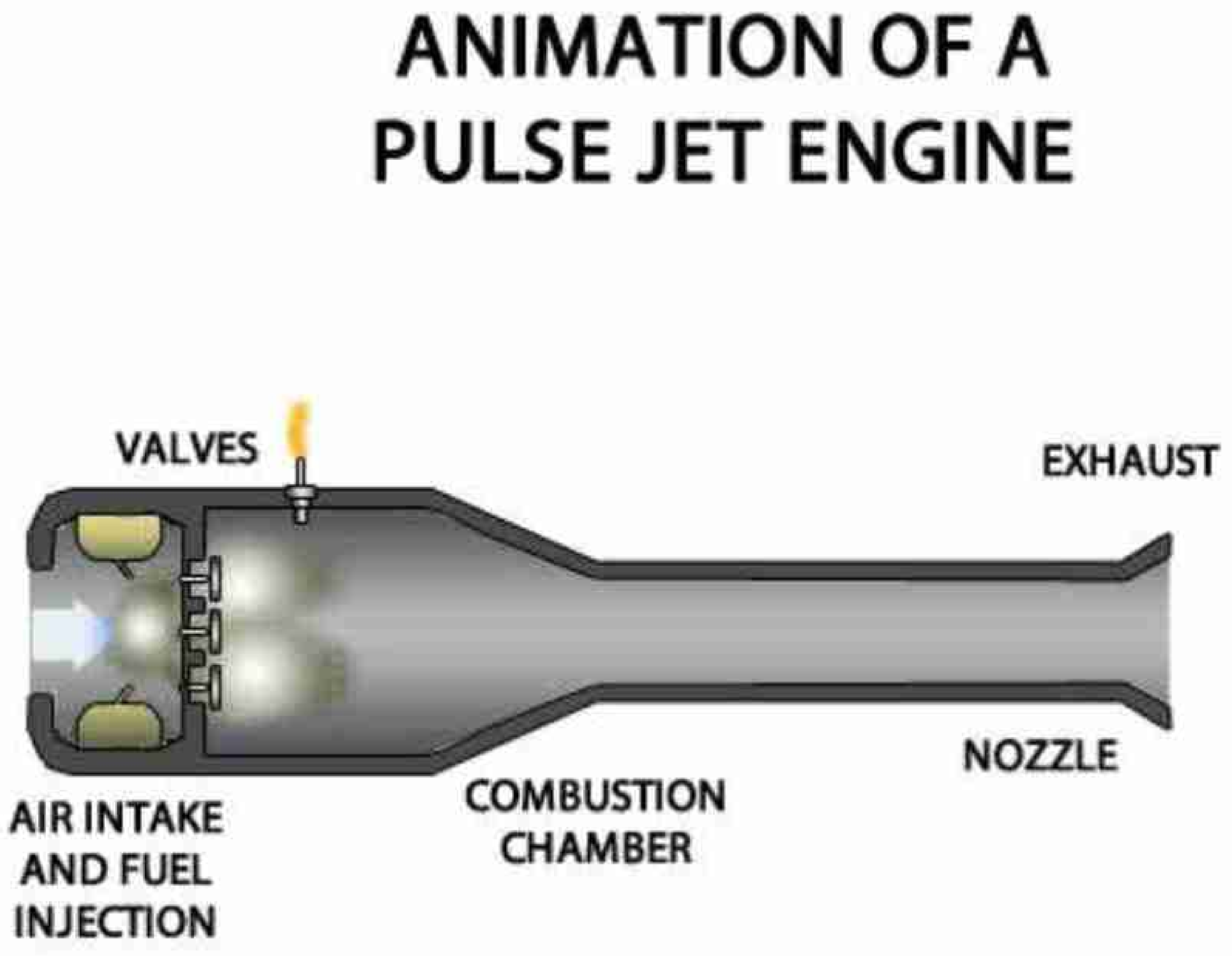
AQM-60 Kingfisher, the first production ramjet to enter service with the US military

Cruise missile BrahMos on display at IMDS-2007

The 'Akash' surface to air missile

Pulse Jet (or) Flying Bomb Engine

Pulse jet engine is very similar to ramjet engine in construction except that in addition to the diffuser at intake, combustion chamber and exhaust nozzle, it has mechanically operated flapper valve grids which can allow or stop air flow in the combustion chamber. Thus, pulse jet is an intermittent flow, compressor less type of device with minimum number of moving parts. Pulse jet was the power plant of German V-1 bomb popularly known as ‘Buzz Bomb’ first used in World War II in 1944.



The fig. shows a pulse jet engine which consists of an inlet diffuser, valve grid (contains springs that close on their own spring pressure), combustion chamber, spark plug and a discharge nozzle.

The function of the diffuser is to convert the kinetic energy of the entering air into static pressure rise by slowing down the air velocity.

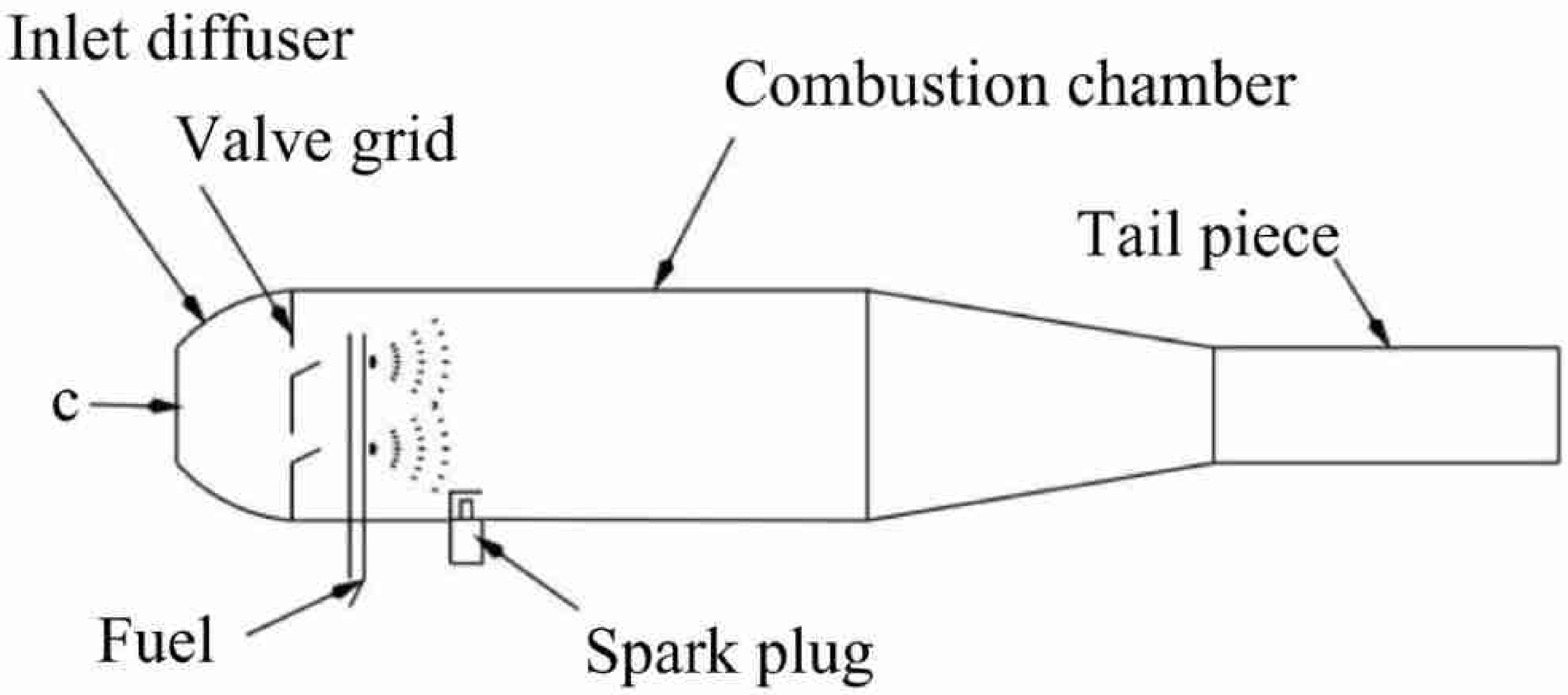


Fig. The pulse jet engine

When a certain pressure difference builds up across the valve grid, the valves will open. This makes the fresh air to enter the combustion chamber, where fuel is mixed with the air and combustion starts. To start the combustion initially the spark plug is used.

Since firing in the combustor is intermittent and therefore intermittent thrust is produced. The pulse-jet engine is a simple, cheap for subsonic flights and well adopted to pilotless aircrafts.

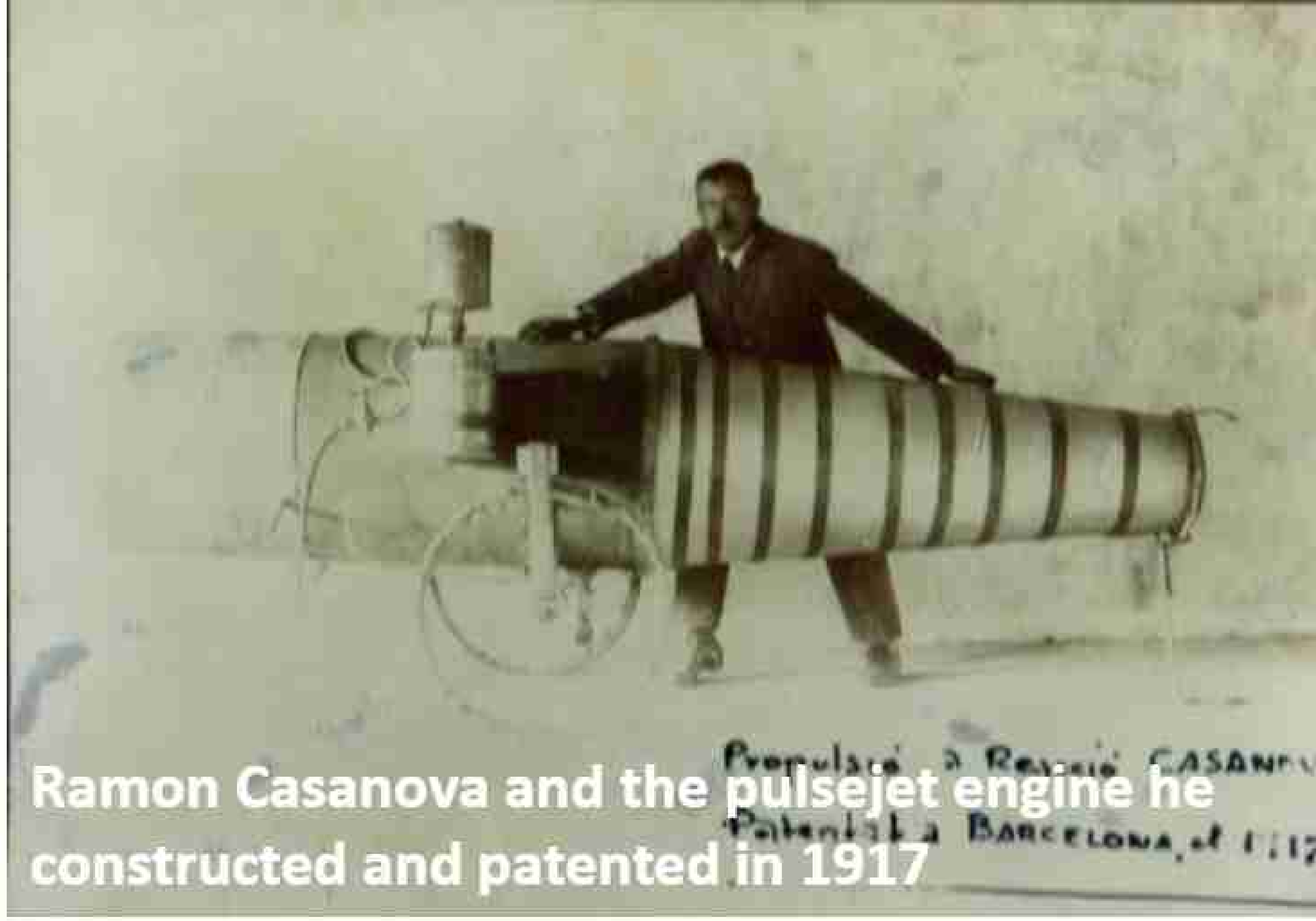
Advantages

1. It gives higher pay load capacity due to the absence of compressor, propeller and turbine.
2. It is simple in construction and cheap. It is suitable for subsonic flights.
3. Drag co-efficient is less due to smaller frontal area.
4. Due to the absence of sliding and moving parts mechanical efficiency is very high.

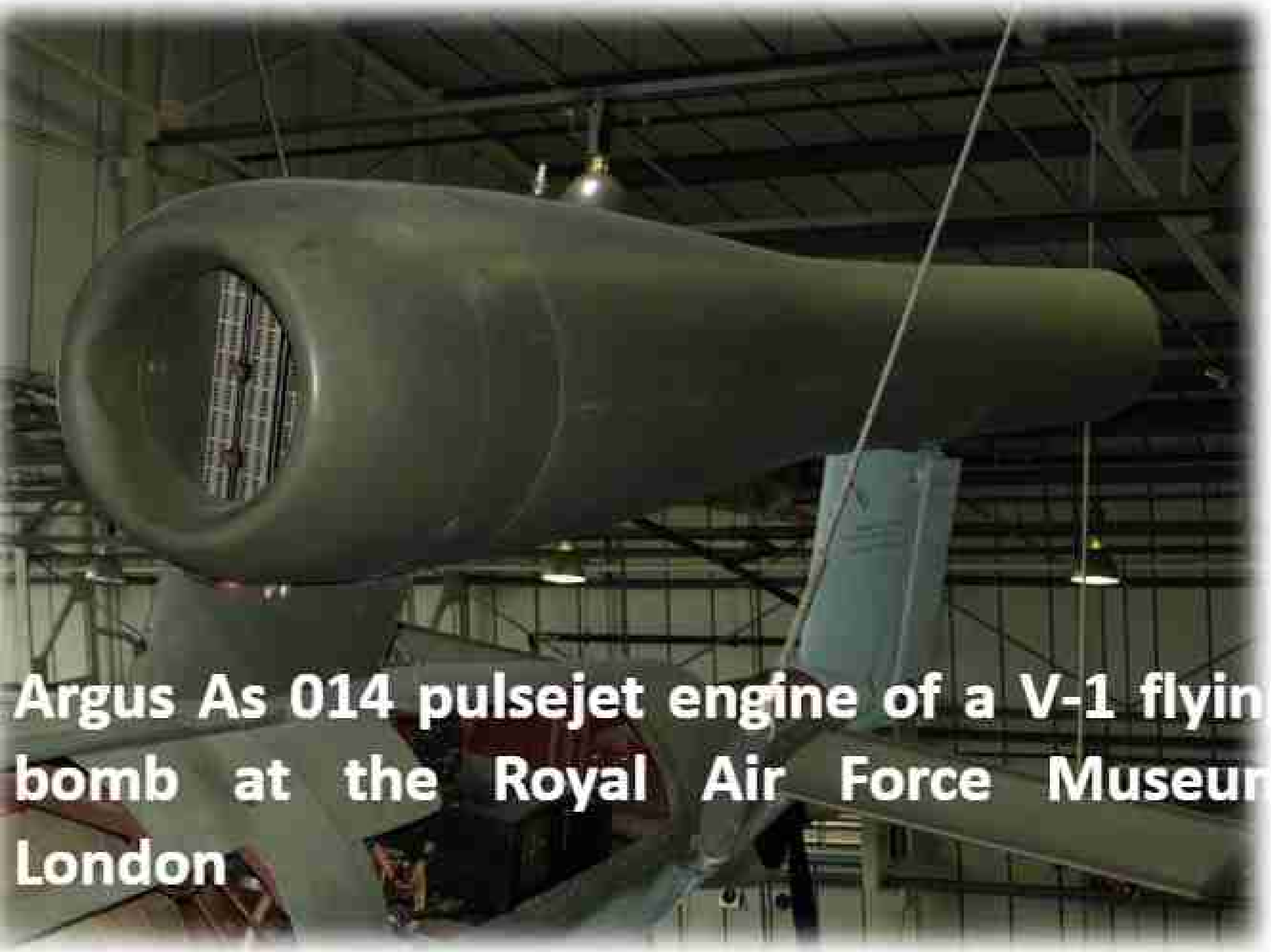
Disadvantages

1. Limited flight speed and altitude.
2. Severe vibrations and high intensity of noise due to intermittent combustion.
3. Nozzle erosion occurs, due to the high temperature of gases coming out from the nozzle.

A Heinkel He 111H of Kampfgeschwader 53



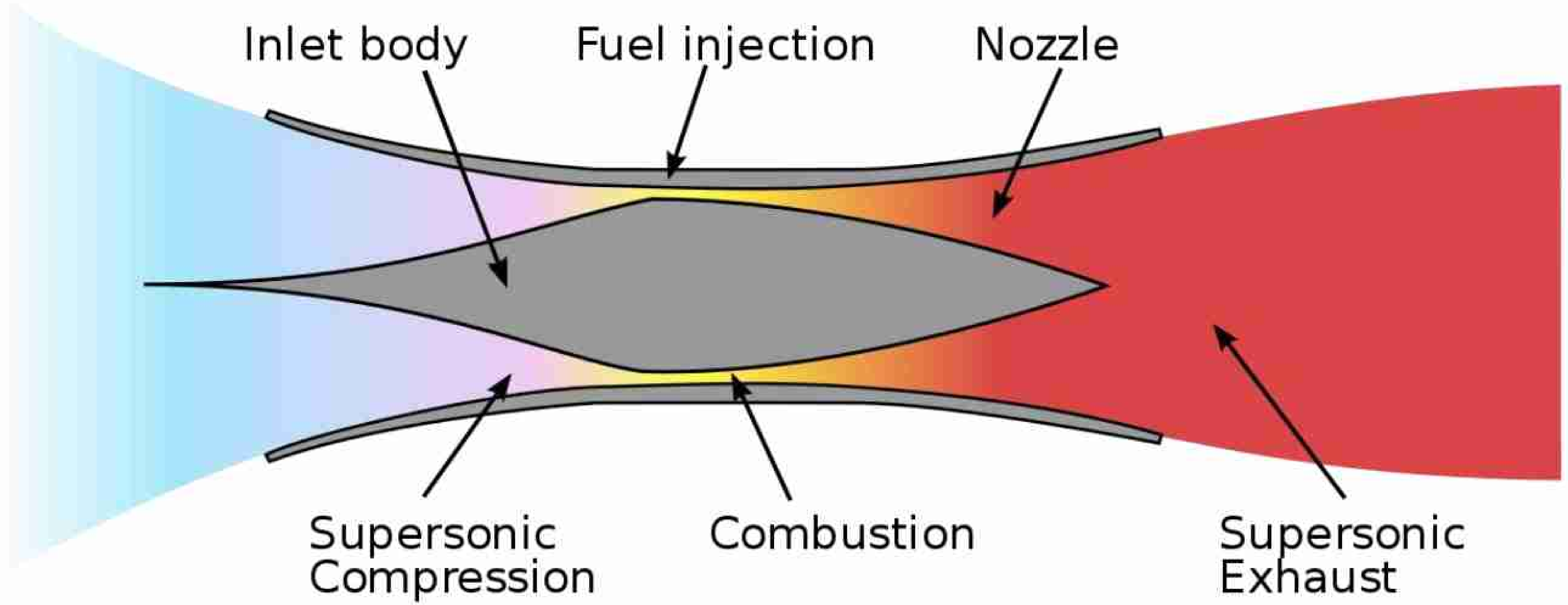
Ramon Casanova and the pulsejet engine he constructed and patented in 1917



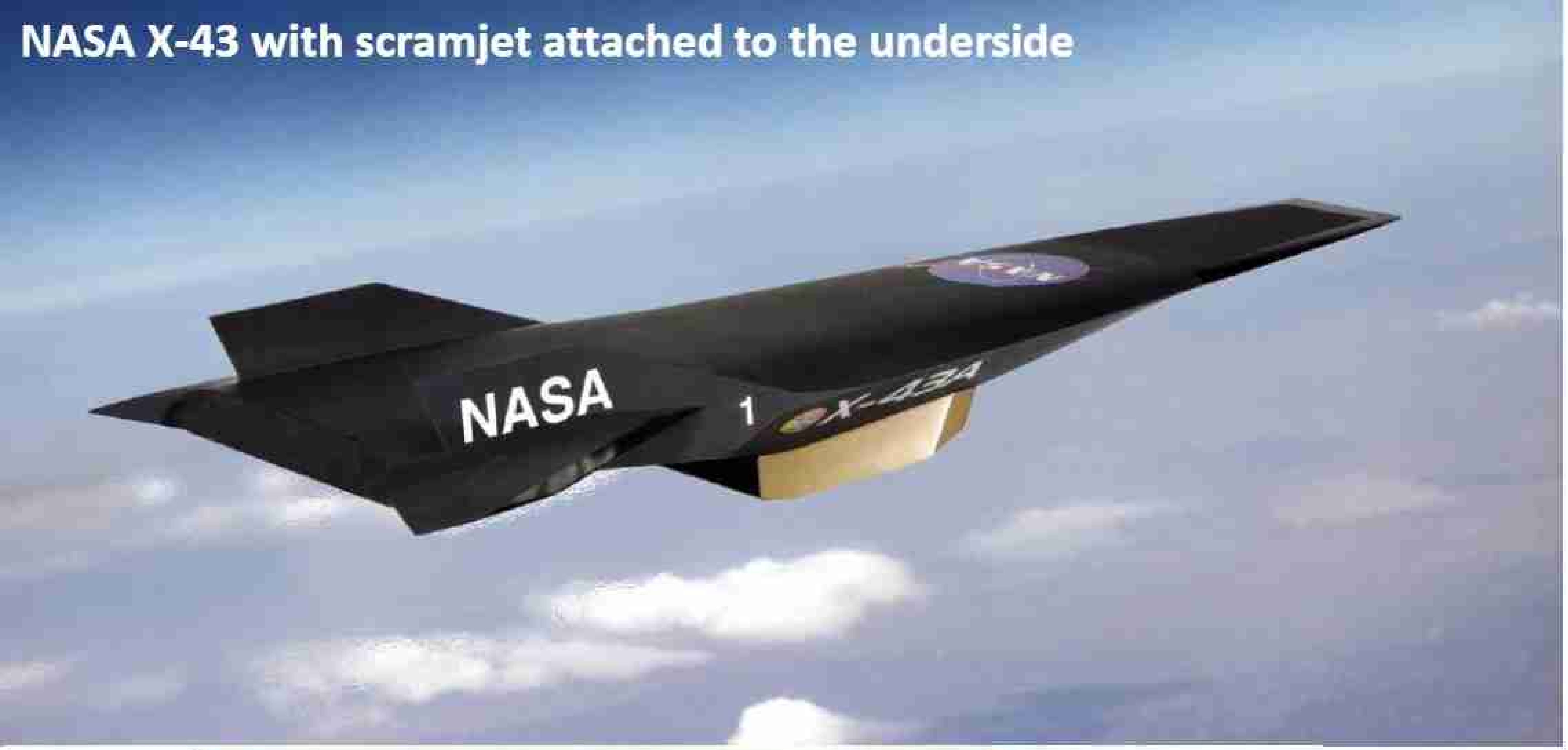
Argus As 014 pulsejet engine of a V-1 flyin bomb at the Royal Air Force Museu London

Scramjet

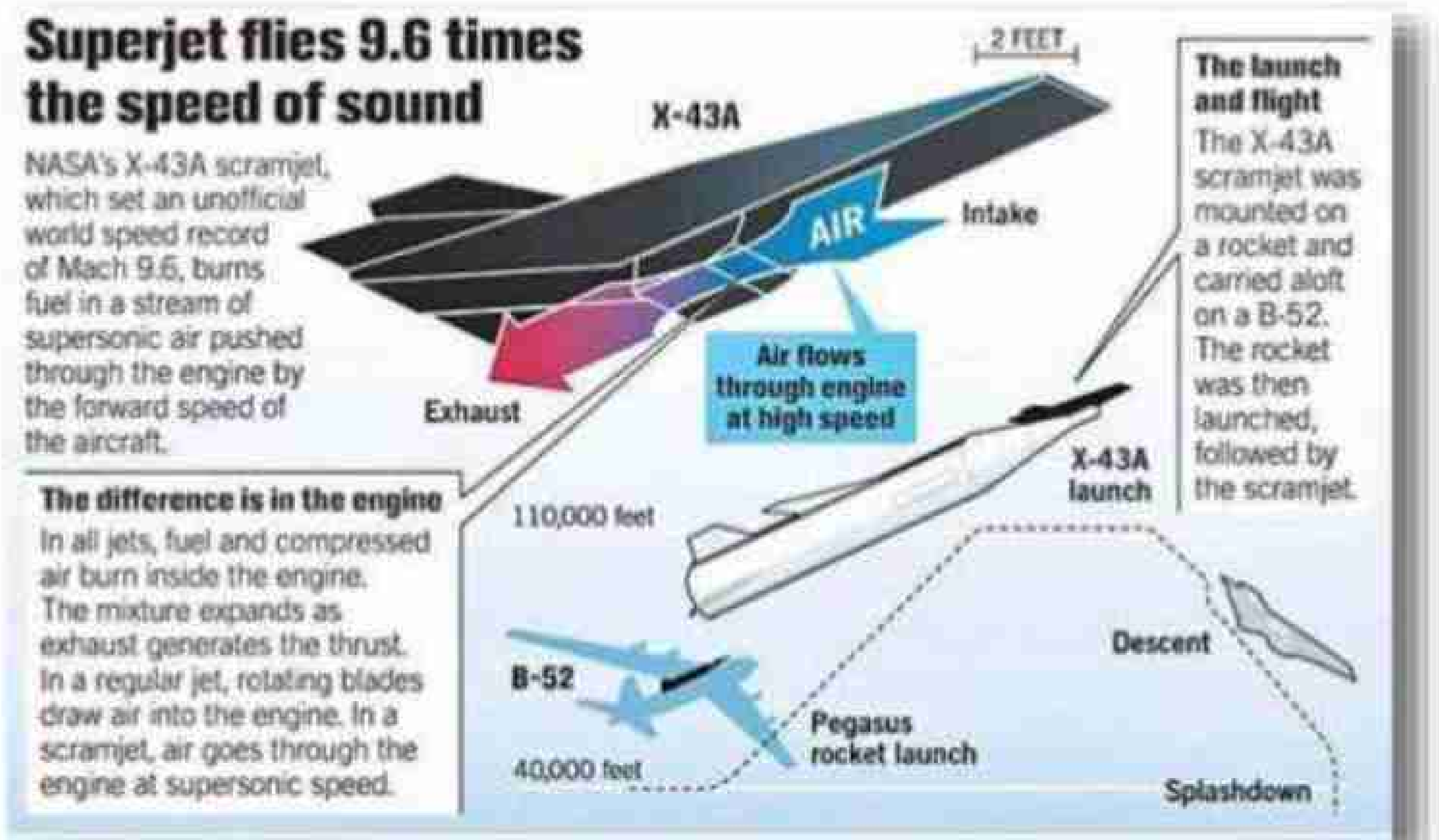
A scramjet (supersonic combustion ramjet) is a variant of a ramjet airbreathing jet engine in which combustion takes place in supersonic airflow. As in ramjets, a scramjet relies on high vehicle speed to compress the incoming air forcefully before combustion (hence ramjet), but whereas a ramjet decelerates the air to subsonic velocities before combustion, the airflow in a scramjet is supersonic throughout the entire engine. That allows the scramjet to operate efficiently at extremely high speeds.



NASA X-43 with scramjet attached to the underside



RECENT PROGRESS

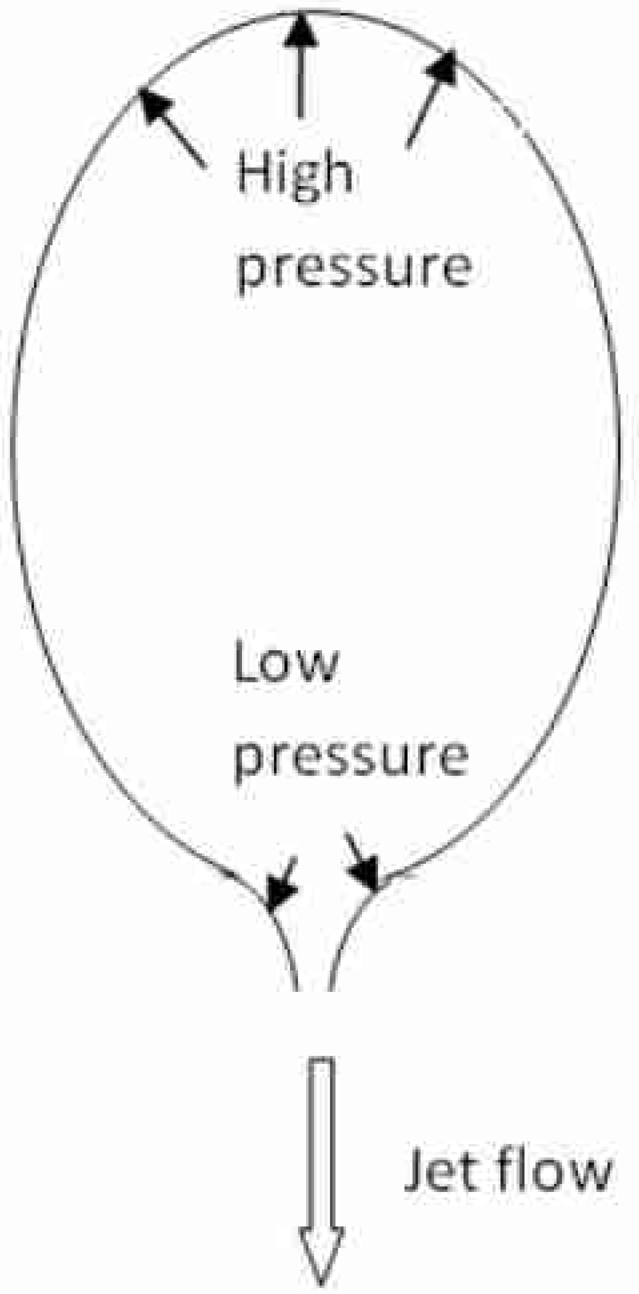


ROCKET PROPULSION

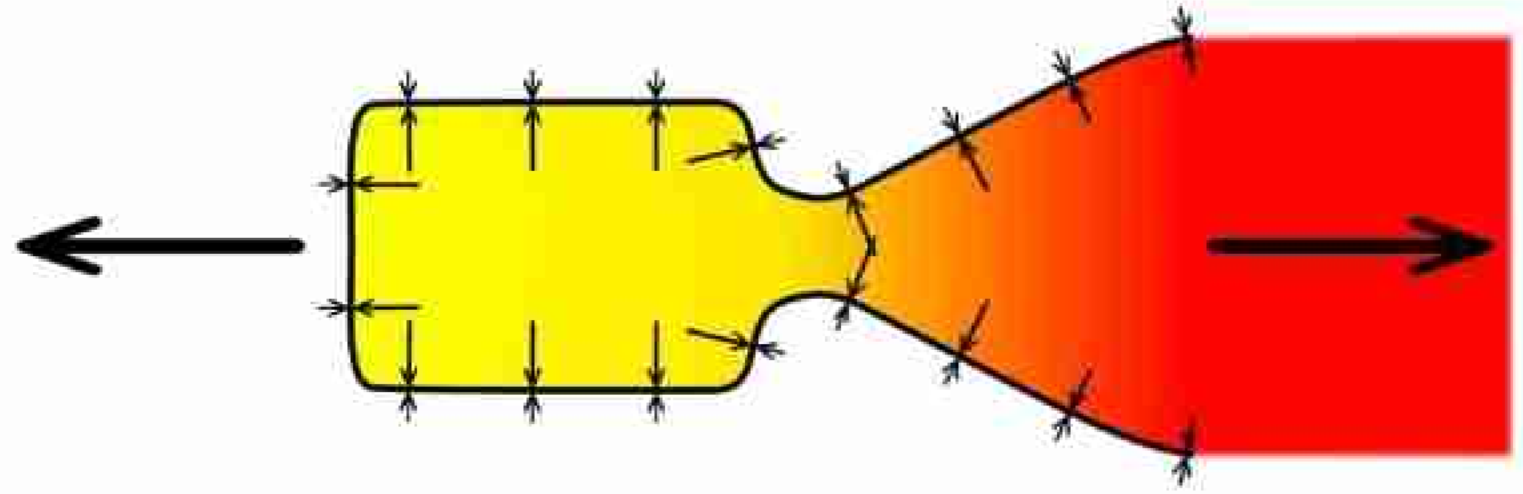
A lecture by
V.Manikanth
Assistant Professor
Dept. Of Mechanical Engineering
SRKR Engineering College (A)

A rocket (*from Italian: rocchetto*) is a missile, spacecraft, aircraft or other vehicle that obtains thrust from a rocket engine. Rocket engine exhaust is formed entirely from propellant carried within the rocket. Rocket engines work by action and reaction and push rockets forward simply by expelling their exhaust in the opposite direction at high speed and can therefore work in the vacuum of space.

The effect of the combustion of propellant in the rocket engine is to increase the internal energy of the resulting gases, utilizing the stored chemical energy in the fuel. As the internal energy increases, pressure increases, and a nozzle is utilized to convert this energy into a directed kinetic energy. This produces thrust against the ambient environment to which these gases are released.



A balloon with a tapering nozzle. In this case, the nozzle itself does not push the balloon but is pulled by it. A convergent/divergent nozzle would be better.



Rocket thrust is caused by pressures acting on both the combustion chamber and nozzle

The ideal direction of motion of the exhaust is in the direction to cause thrust. At the top end of the combustion chamber the hot, energetic gas fluid cannot move forward, and so, it pushes upward against the top of the rocket engine's combustion chamber. As the combustion gases approach the exit of the combustion chamber, they increase in speed. The effect of the convergent part of the rocket engine nozzle on the high-pressure fluid of combustion gases, is to cause the gases to accelerate to high speed. The higher the speed of the gases, the lower the pressure of the gas (Bernoulli's principle or conservation of energy) acting on that part of the combustion chamber.

In a properly designed engine, the flow will reach Mach 1 at the throat of the nozzle. At which point the speed of the flow increases. Beyond the throat of the nozzle, a bell-shaped expansion part of the engine allows the gases that are expanding to push against that part of the rocket engine. Thus, the bell part of the nozzle gives additional thrust. Simply expressed, for every action there is an equal and opposite reaction, according to Newton's third law with the result that the exiting gases produce the reaction of a force on the rocket causing it to accelerate the rocket.

The fields of application of rockets are as follows :

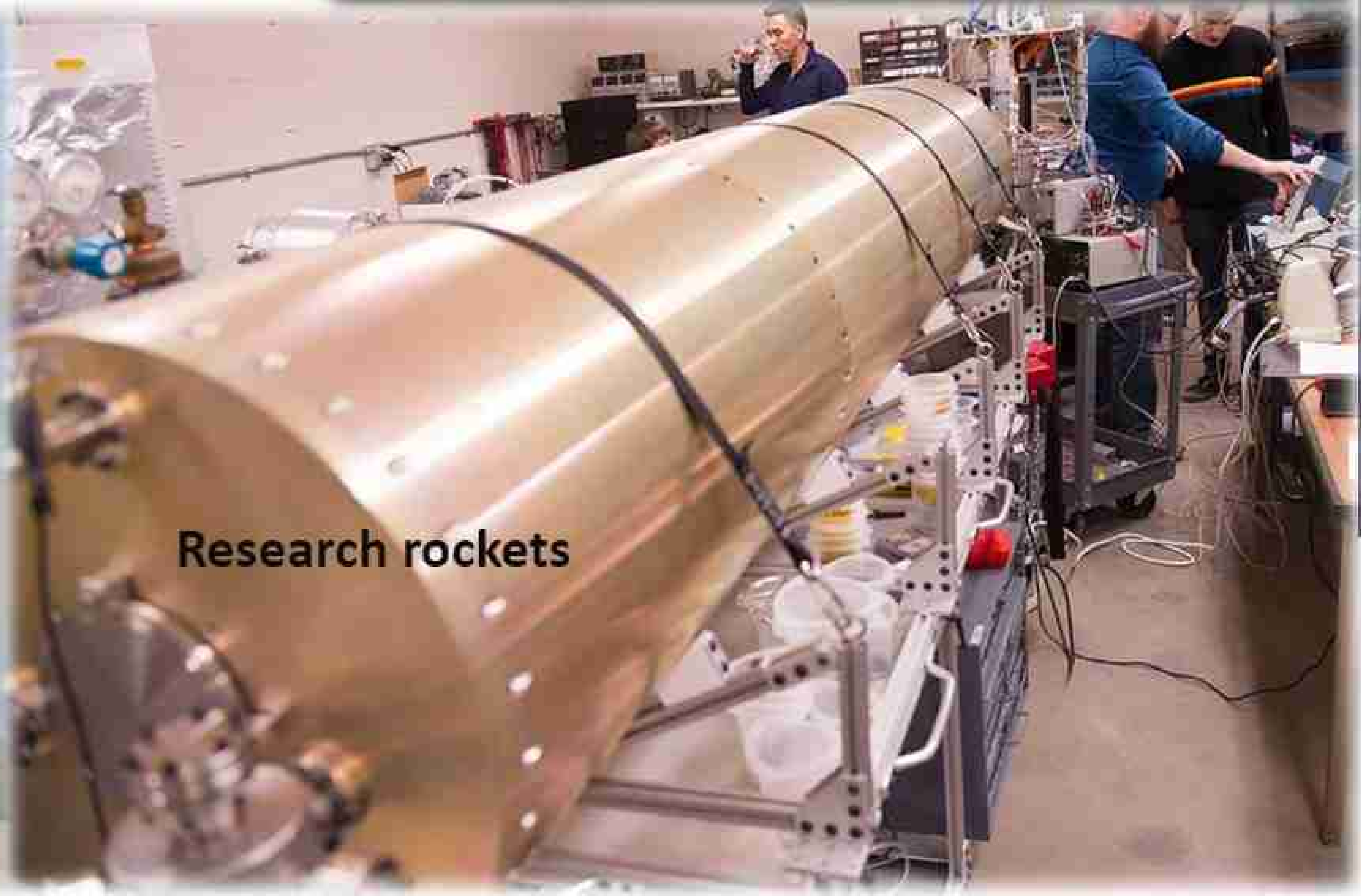

Lethal weapons


Illumination flares being used during military training exercises


Long range artillery


BQM-74E Chukar target drone using JATO


Launching vehicles


Research rockets

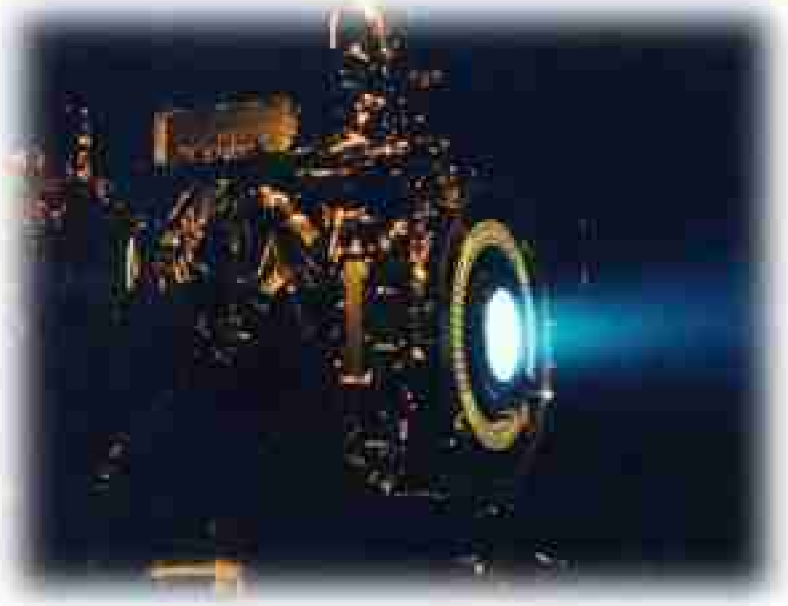


Table Performance Differences of the air-breathing engine and the rocket engines

Air-breathing engine	**Rocket engine**
Altitude limitation.	No altitude limitation, space travel possible.
Thrust decreases with altitude.	Thrust increases slightly with altitude.
Rate of climb decreases with altitude.	Rate of climb increases with altitude.
Engine ram drag increases with flight speed.	Engine has no ram drag; constant thrust with speed.
Flight speed always less than jet velocity.	Flight speed not limited, can be greater than jet velocity.
Reasonable efficiency and reasonable flight duration.	Low efficiency except at extremely high flight speed for small duration.

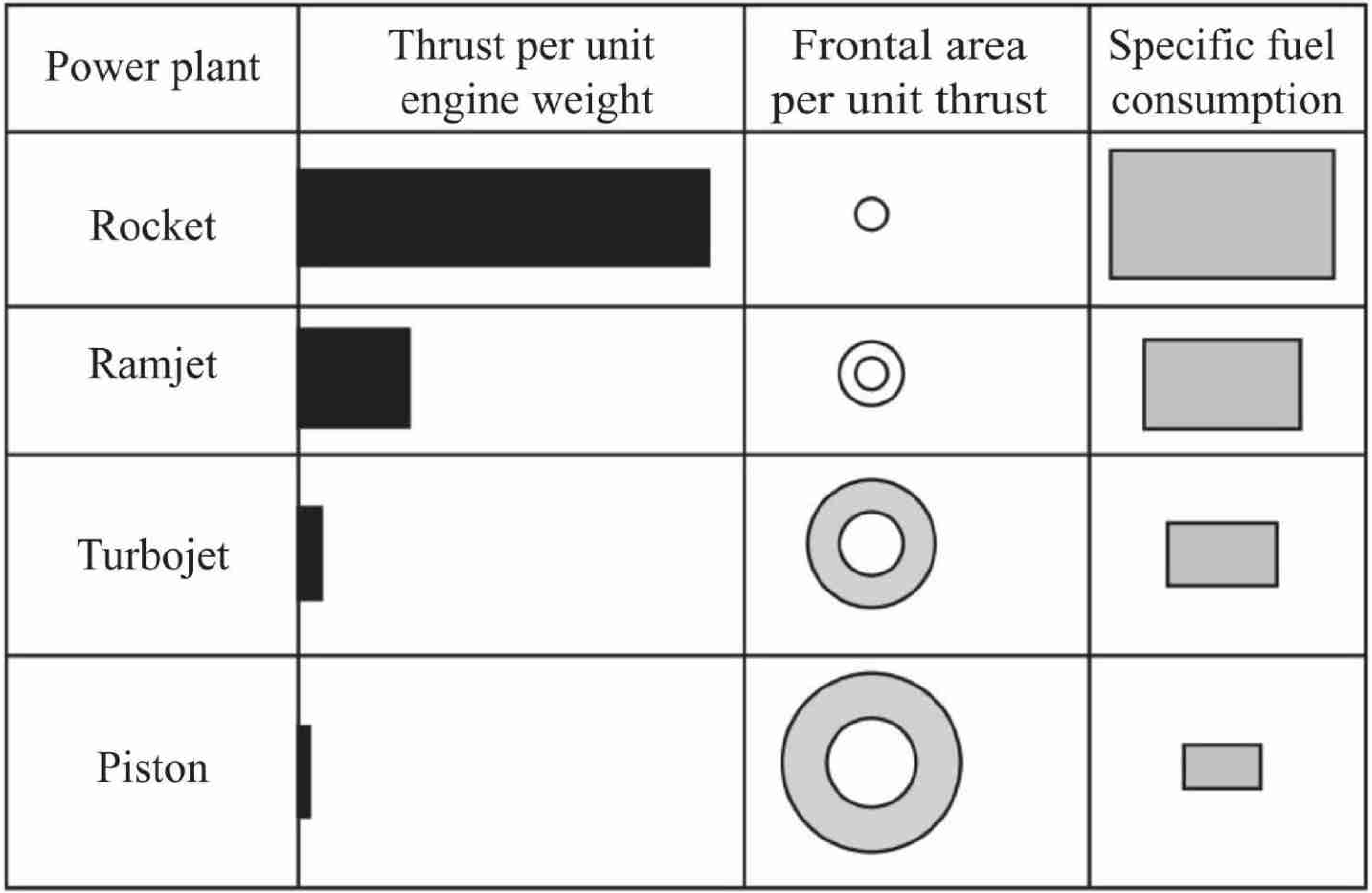


Fig. Performance details of different propulsive systems

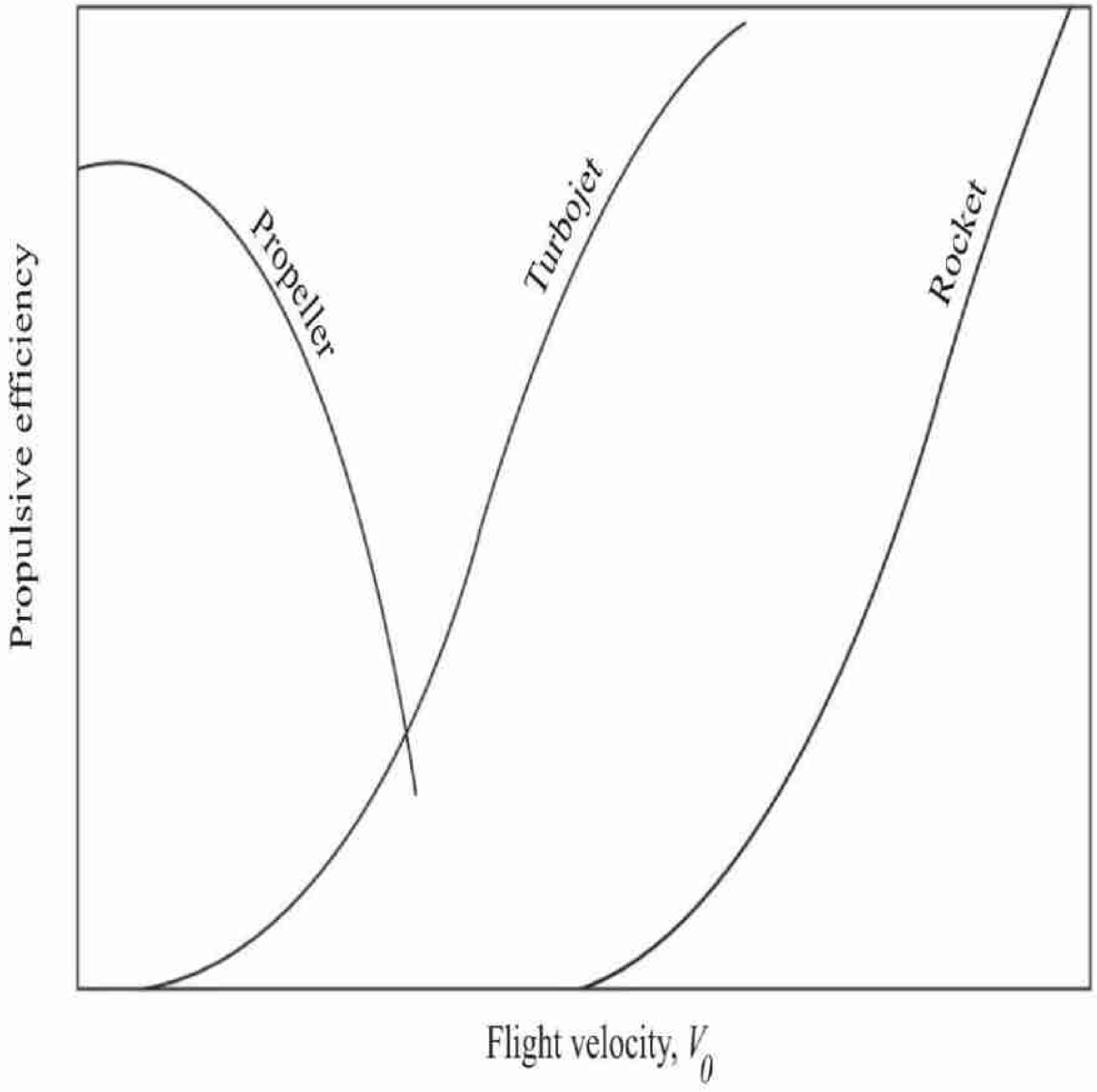


Fig. Propulsive efficiency at different speeds of various propulsion system

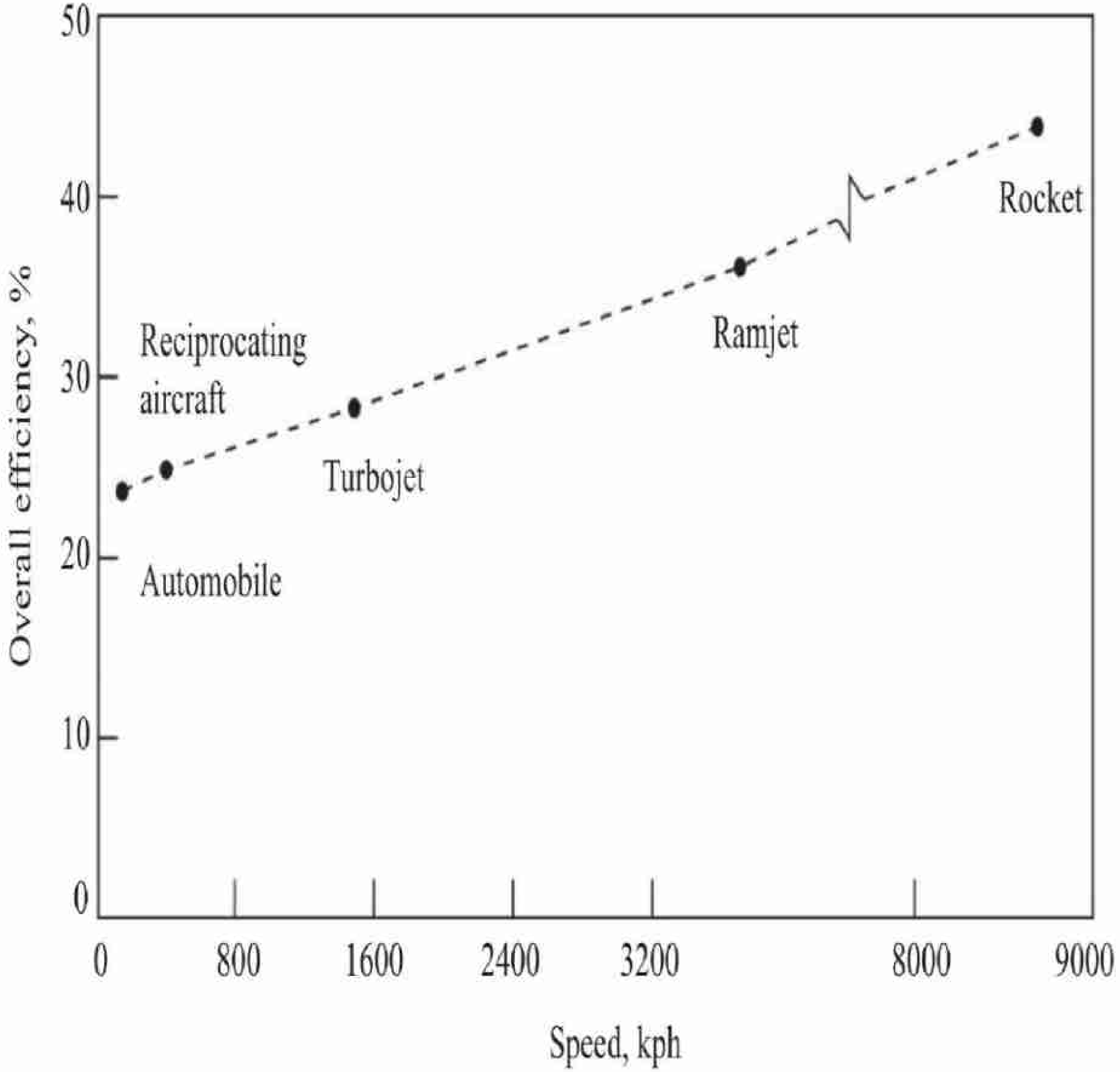
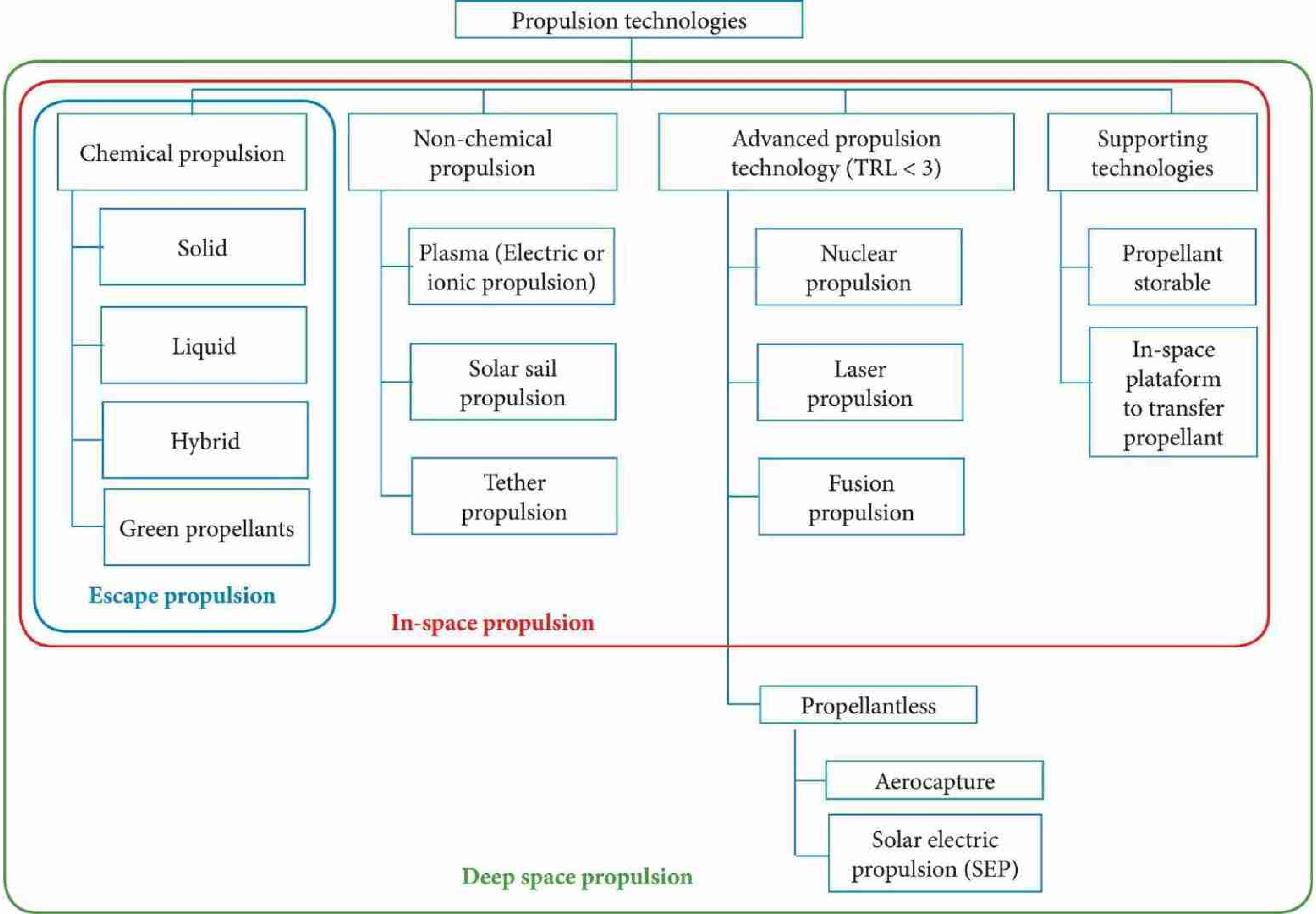


Fig. Efficiency of various propulsion systems at different speeds


Propulsion technologies
Chemical propulsion
Solid
Liquid
Hybrid
Green propellants
Escape propulsion
Non-chemical propulsion
Plasma (Electric or ionic propulsion)
Solar sail propulsion
Tether propulsion
In-space propulsion
Advanced propulsion technology (TRL < 3)
Nuclear propulsion
Laser propulsion
Fusion propulsion
Propellantless
Aerocapture
Solar electric propulsion (SEP)
Supporting technologies
Propellant storable
In-space plataform to transfer propellant
Deep space propulsion

An ideal rocket propellent should have the following characteristics/properties :

1. High heat value

2. Reliable smooth ignition

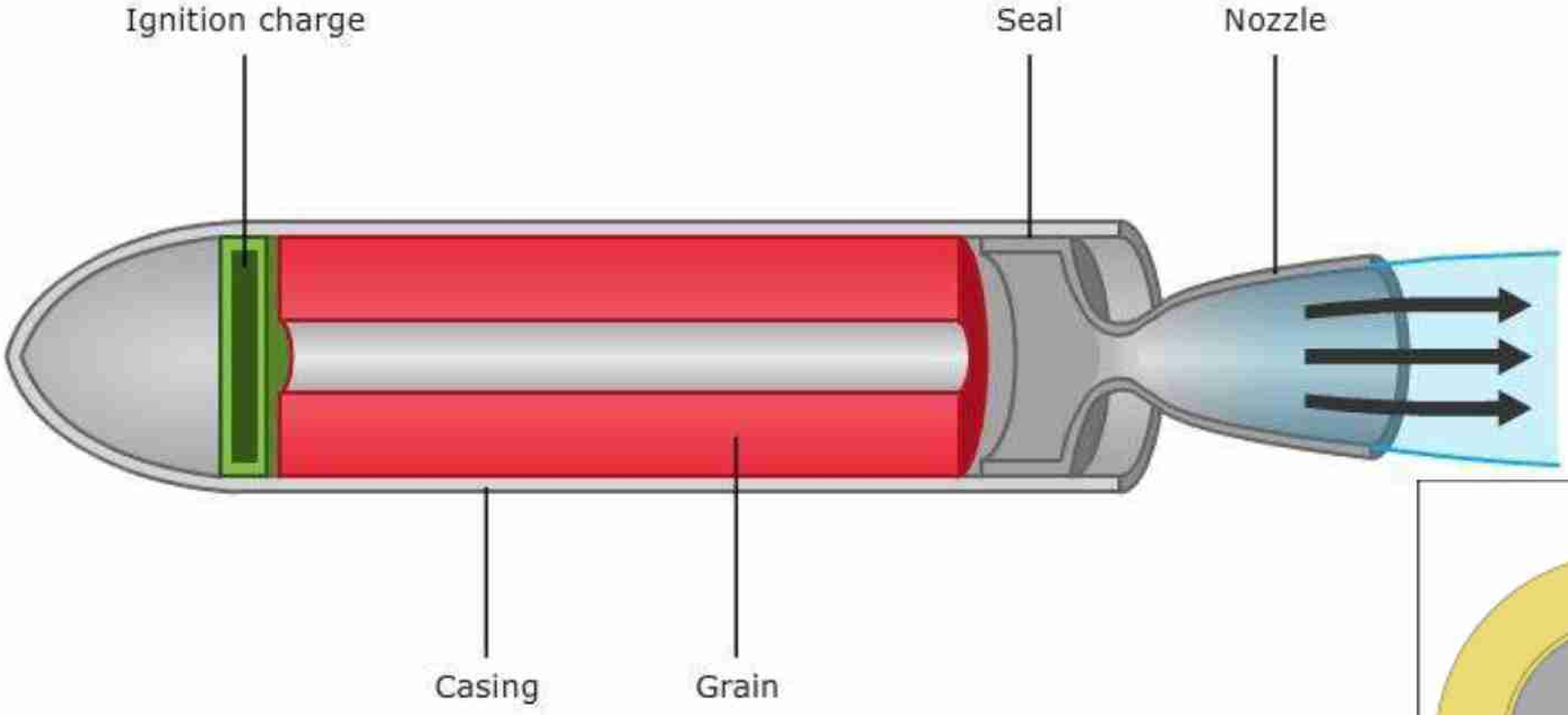
3. Stability and ease of handling and storing

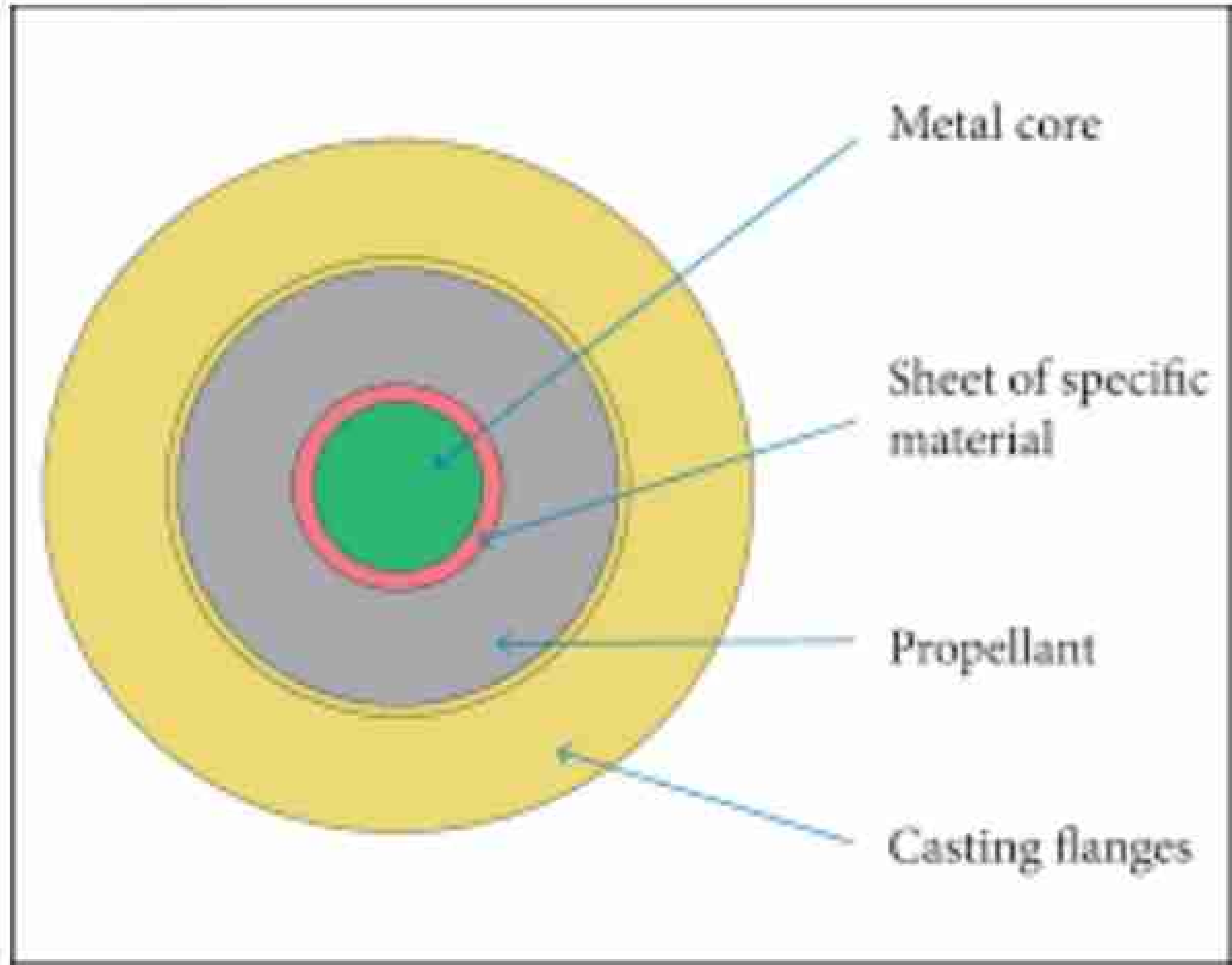
4. Low toxicity and corrosiveness

5. Highest possible density so that it occupies less space

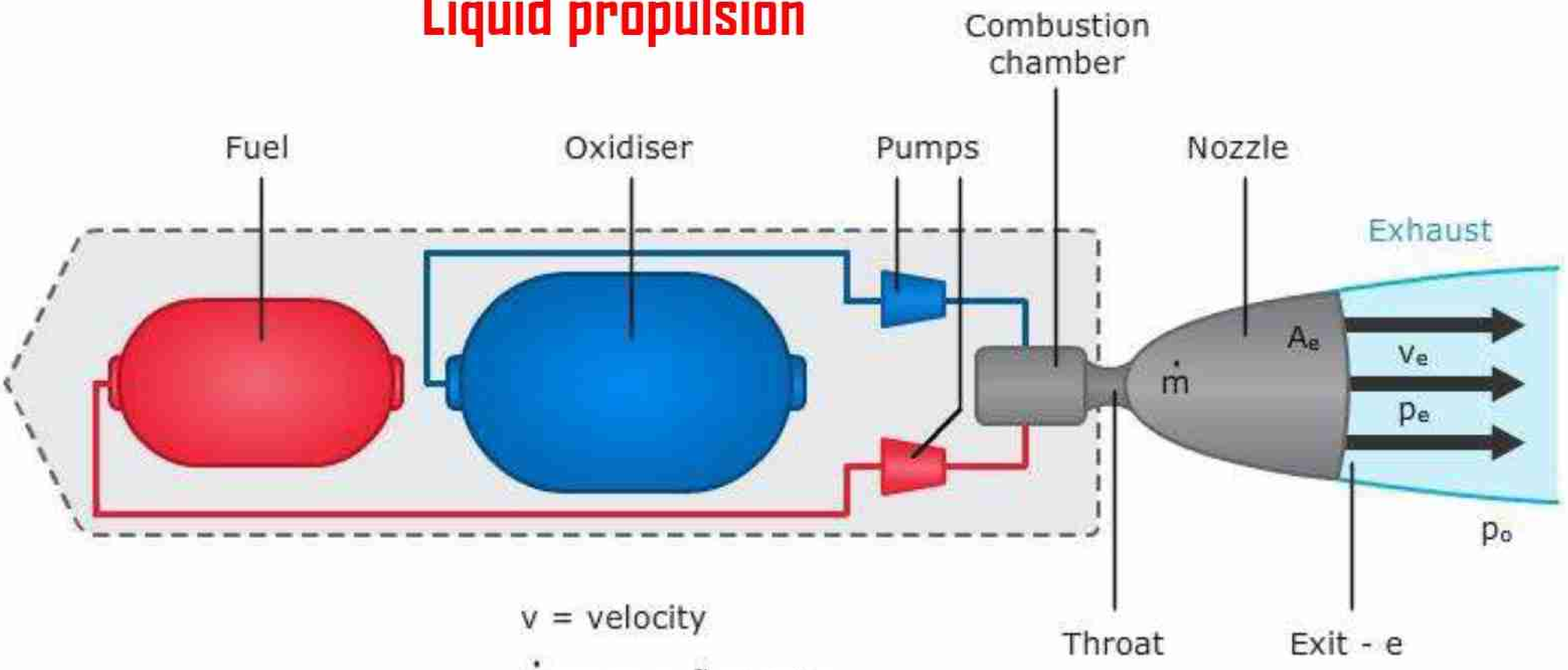
@V.Manikanth, Assistant Professor, Dept. of Mechanical, SRKREC(A).

Solid propulsion





Liquid propulsion



v = velocity

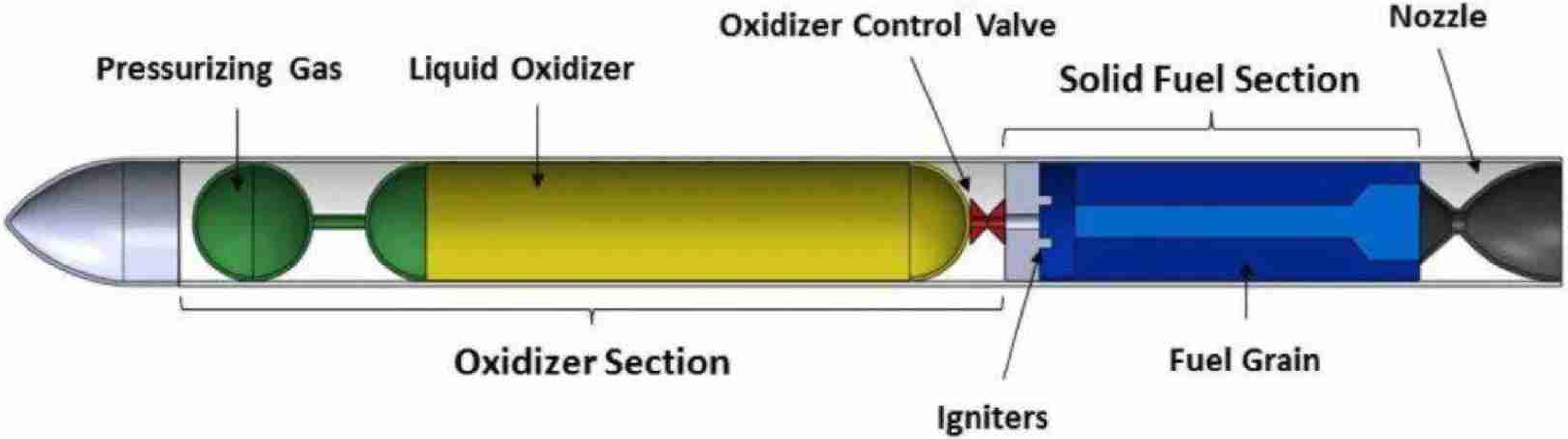
$\dot{m}$ = mass flow rate

p = pressure

A = area

$$\text{Thrust} = F = \dot{m}\, v_e + (p_e - p_o)\, A_e$$

Hybrid Rocket Engine

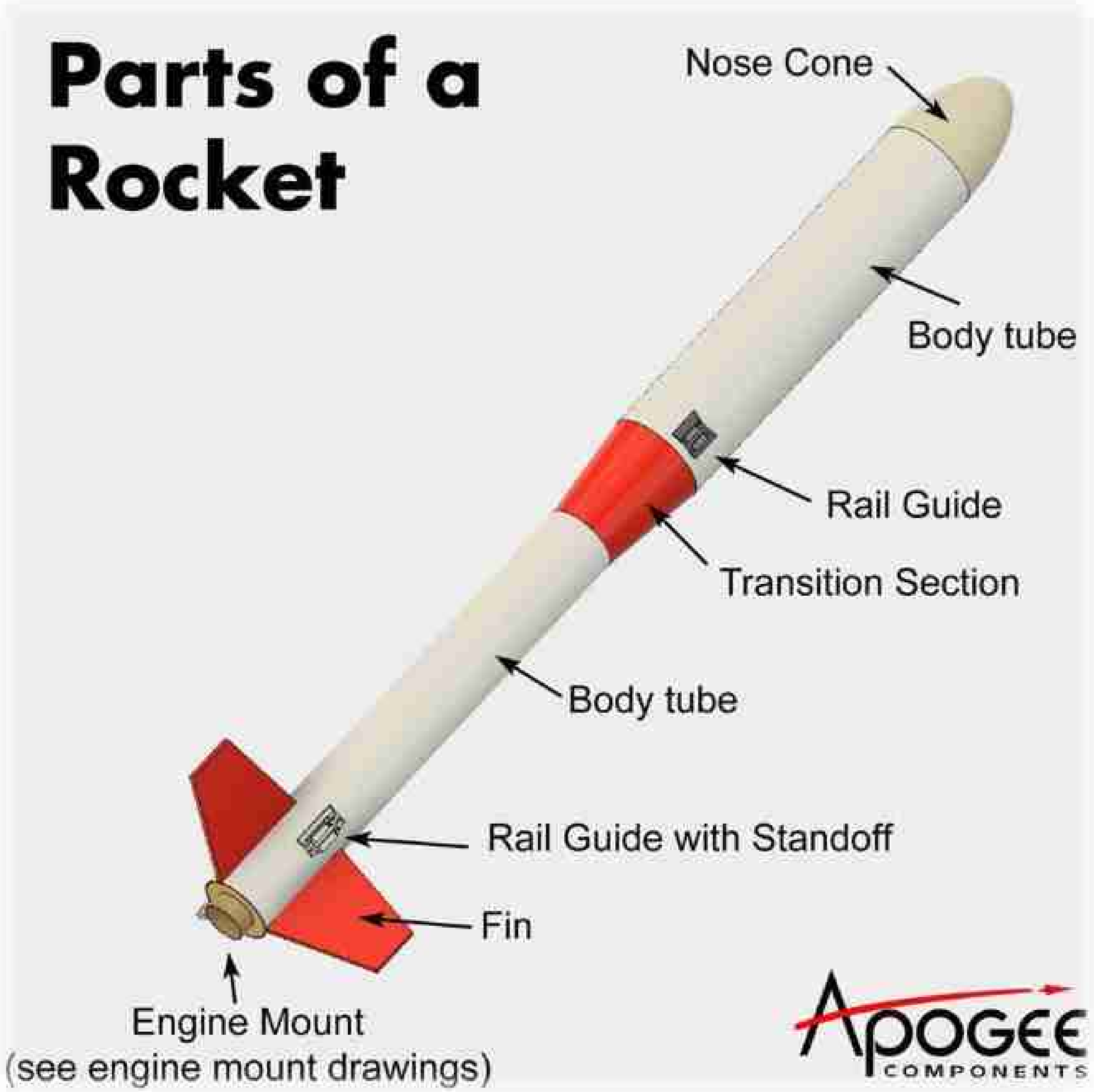


Basic Systems Illustration

Machine Information	
Created by	RDA
Used by	Humans
Usage	Interstellar travel between Earth and Pandora
Size	**length** 1,646 m **width** 330 m **height** 218.5 m
Speed	0.7 C = 130,200 mi/s = 209,537 km/s
Behind the scenes	
First appearance	*Avatar*

The Venture star has:

- Two matter-antimatter engines
- One photon sail
- One fusion PME (Planetary Maneuvering Engine)


Parts of a Rocket
Nose Cone
Body tube
Rail Guide
Transition Section
Body tube
Rail Guide with Standoff
Fin
Engine Mount
(see engine mount drawings)
Apogee
COMPONENTS

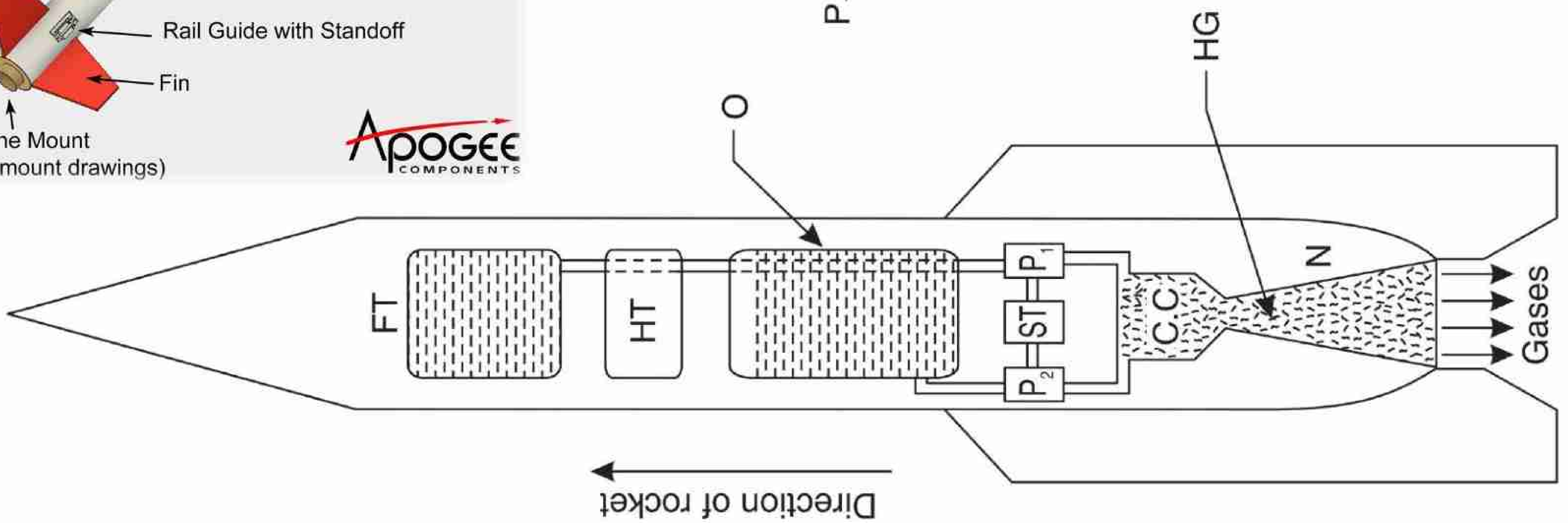

FT – Fuel tank
HT – Hydrogen peroxide tank
O – Oxidiser tank
ST – Steam turbine
P_1, P_2 – Pumps
C.C – Combustion chamber
HG – Hot gases
N – Nozzle
O
HG
FT
HT
P_1
ST
P_2
C.C
N
Gases
Direction of rocket

Fig. Rocket.

3 STAGE: CUS

Length : 9.9 m (32 ft)
Diameter : 2.8 m(9.2 ft)
Propellant mass : 15,000 kg
Engines : 1 CE-7.5
Thrust : 75 kN (17,000 lbf)
Specific impulse : 454 s (4.45 km/s)
Burn time : 846 seconds
Fuel : LOX/LH2

2 STAGE: GS2

Length : 11.9 m (39 ft)
Diameter : 2.8 m (9.2 ft)
Propellant mass : 42,500 kg
Engines : 1 GS2 Vikas 4
Thrust : 846.8 kN
Specific impulse : 295 s (2.89 km/s)
Burn time : 149 seconds
Fuel : N2O4/UDMH

1 STAGE

Length : 20.2 m (66 ft)
Diameter : 2.8 m (9.2 ft)
Propellant mass : 138,200 kg
Engines : 1 S139
Thrust : 4,846.9 kN
Specific impulse : 237 s (2.32 km/s)
Burn time : 100 seconds
Fuel :HTPB (solid)

Capacity

Payload to GTO : 2,700 kg

Manufacturer : ISRO
Country of origin : India
Height : 49.13 m (161.2 ft)[2]
Diameter : 2.8 m (9 ft 2 in)
Mass : 414,750 kg (914,370 lb)
Stages : 3
Cost per launch : US$47 million
Total launches :13 (6 Mk.I, 7 Mk.II)
Successes :8 (2 Mk.I, 6 Mk.II)

Boosters

No. boosters : 4 L40 Hs
Length : 19.7 m (65 ft)
Diameter : 2.1 m (6.9 ft)
Propellant mass : 42,700 kg each
Engines : 1 L40H Vikas 2
Thrust : 760 kN (170,000 lbf)
Total thrust : 3,040 kN (680,000 lbf)
Specific impulse : 262 s (2.57 km/s)
Burn time :154 seconds
Fuel : N2O4/UDMH

CAPACITY

Payload to GTO : 26,700 kg

Manufacturer : SpaceX
Country of origin : United States
Height : 70 m (230 ft)
Diameter : 3.66 m (12.0 ft)
Mass : 1,420,788 kg
Stages : 2+
Total launches : 2
Successes : 2

2 STAGE

Engines : 1 Merlin 1D Vacuum
Thrust : 934 kN
Specific impulse : 348 seconds
Burn time : 397 seconds
Fuel : LOX / RP-1

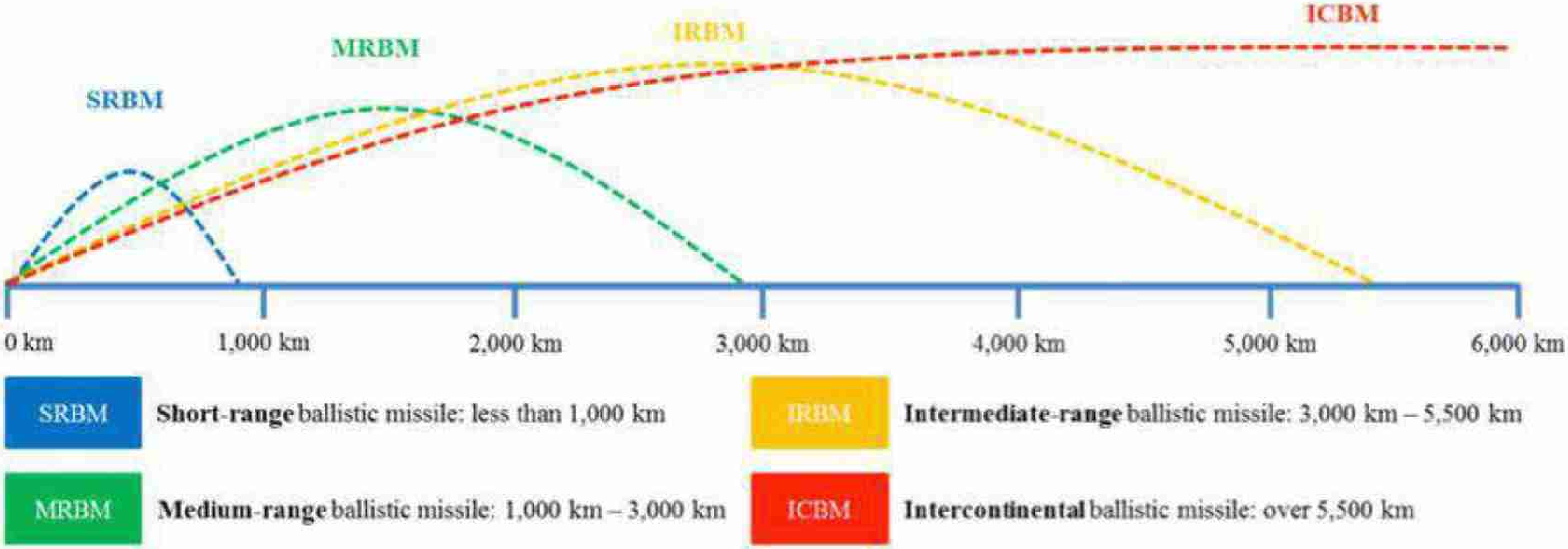
1 STAGE

Engines : 9 Merlin 1D
Thrust
Sea level : 7.6 MN
Vacuum : 8.2 MN
Specific impulse
Sea level : 282 seconds
Vacuum : 311 seconds
Burn time : 187 seconds
Fuel : Subcooled LOX / Chilled RP-1

BOOSTERS

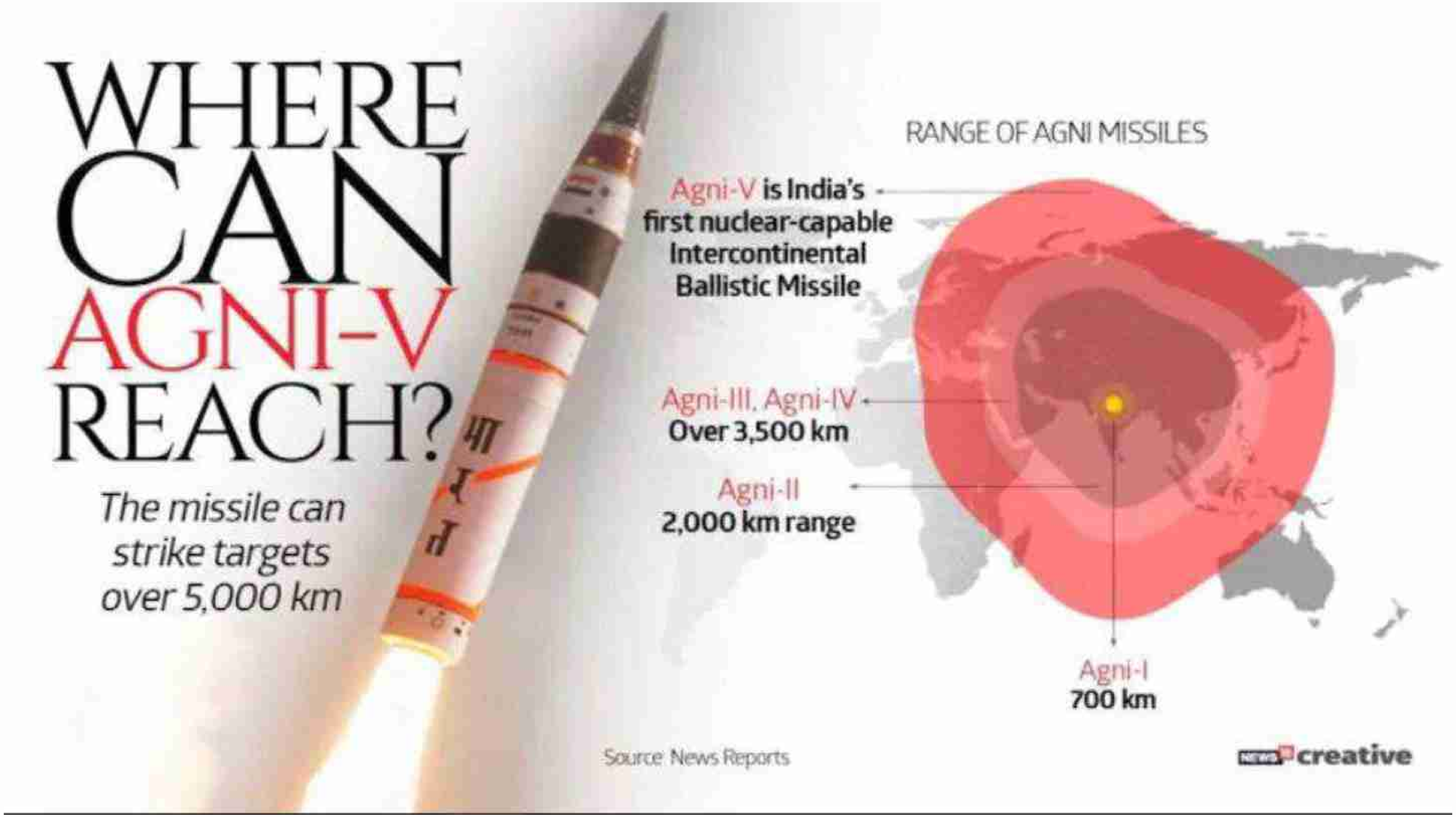
No. boosters : 2
Engines : 9 Merlin 1D per booster
Burn time : 154 seconds
Thrust
Sea level : 7.6 MN
Vacuum : 8.2 MN
Total Thrust
Sea level : 15.2 MN
Vacuum : 16.4 MN
Specific impulse
Sea level : 282 seconds
Vacuum : 311 seconds
Fuel : Subcooled LOX / Chilled RP-1

@V.Manikanth, Assistant Professor, Dept. of Mechanical, SRKREC(A).


ICBM
IRBM
MRBM
SRBM
0 km
1,000 km
2,000 km
3,000 km
4,000 km
5,000 km
6,000 km
SRBM
Short-range ballistic missile: less than 1,000 km
IRBM
Intermediate-range ballistic missile: 3,000 km – 5,500 km
MRBM
Medium-range ballistic missile: 1,000 km – 3,000 km
ICBM
Intercontinental ballistic missile: over 5,500 km

ICBM

@V.Manikanth, Assistant Professor, Dept. of Mechanical, SRKREC(A).


WHERE CAN AGNI-V REACH?
The missile can strike targets over 5,000 km
RANGE OF AGNI MISSILES
Agni-V is India's first nuclear-capable Intercontinental Ballistic Missile
Agni-III, Agni-IV Over 3,500 km
Agni-II 2,000 km range
Agni-I 700 km
Source: News Reports
creative