## 19.1. INTRODUCTION

Steam pressure acting directly on pistons is utilised in reciprocating steam engines. Steam engines can even be operated without any pressure drop in the cylinder (non-expansive steam engine). But the operation of steam turbines wholly depends on the dynamic action of the steam expanded through nozzles.

In steam turbines, enthalpy of the steam is first converted into kinetic-energy in nozzles or blade passages. The high velocity steam impinges on the curved blades which change the flow direction of steam. The change in flow direction of steam causes a force to be exerted on the blades fixed on a rotor and power is developed due to the rotation of these blades.

The steam turbine is universally used as prime-movers in all steam power plants.

## 19.2. ADVANTAGES OF STEAM TURBINES OVER RECIPROCATING STEAM ENGINES

Steam Turbines offer many advantages over steam engines as prime-movers. Some of these are listed below :

(1) The thermodynamic efficiency of a turbine plant is higher than that of the steam engine plant because it works on Rankine cycle whereas steam engine works on modified Rankine cycle. The expansion of steam can be continued up to the lowest pressure possible in steam turbines, thus obtaining more work for the same pressure ratio.

(2) The machine is highly simplified in construction and operation as parts like piston, piston rod, cross-head and connecting rod are not required due to the absence of reciprocating motion. Therefore, the maintenance cost is reduced and life of the prime-mover is increased considerably.

(3) Condensate can be used directly in the boiler without pretreatment as there is very little chance of lubricating oil mixing with steam.

(4) There is no condensation loss as the parts of the steam turbines are subjected to constant temperature and pressure at constant loads.

(5) The vibrations and noise are greatly minimised due to the absence of reciprocating parts. This fact also eliminates the need for a flywheel. It is also possible to balance the turbine rotor very accurately. This also further reduces the cost of foundation.

(6) The steam turbine can take considerable over-load with only a slight reduction in its efficiency.

(7) Much higher speeds are possible as only rotary motion is involved. Therefore the weight of machine per unit power is considerably reduced.

(8) The specific steam consumption does not increase with years of service.

(9) Steam turbines can be designed in sizes ranging from a few kW to over 1000 MW in a single unit. Turbines are also suitable for very high steam pressures which is not possible with engines. This is another reason of the higher thermal efficiency of turbines.

Therefore steam turbines are well suited for use as prime-movers for large steam plants.

## 19.3. CLASSIFICATION OF TURBINES AND THEIR WORKING

The steam turbine was invented in the last decade of the nineteenth century. It has undergone several changes in design during the past nine decades. The steam turbine has been used predominantly as prime-mover in all thermal power stations. It is not likely to be replaced in the foreseeable future. Now-a-days,

single steam turbines of 1000 MW capacity are built in many countries and unit of 1500 MW capacity is planned in future power programmes. Continuing developments in materials and other areas promise even better performance and low cost.

Steam turbines are mainly divided into two groups :

(a) Impulse turbine

(b) Impulse-reaction turbine (in practice known as reaction turbine)

In both types of turbines, first the enthalpy of the steam at high pressure is converted into kinetic energy by passing it through nozzles.

In impulse turbines, the steam coming out at a very high velocity through the fixed nozzle as shown in Fig. 19.1. (a) impinges on the blades fixed on the periphery of a rotor. The blades change the direction of the steam flow without changing its pressure. The resulting motive force (due to the change in momentum) causes the rotation of the turbine shaft as shown in Fig. 19.1 (a).

The examples of impulse turbine are De-Laval, Curties and Rateau.

In pure reaction turbine, the high pressure steam from the boiler is passed through the nozzles as shown in Fig. 19.1 (b). When the steam comes out through these nozzles, the velocity of the steam increases relative to the rotating disc. The resulting reacting force of the steam on nozzle gives the rotating motion to the disc and the shaft. The shaft rotates in opposite direction to that of steam jet.

In practice, we hardly find any reaction turbine as described above. The common type is impulse-reaction turbine known as reaction turbine in practice.

In an impulse reaction turbine the steam expands both in fixed and moving blades continuously as the steam passes over them. Therefore, the pressure drop occurs gradually and continuously over both moving and fixed blades. The example of such a turbine is Parson's turbine.



BLADES    DISK

BEARING

STEAM FROM BOILER

MOVING NOZZLES

DISK

BEARING
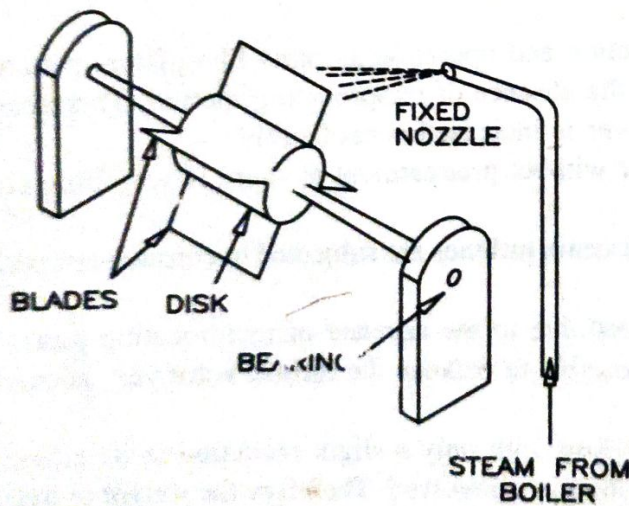
STEAM FROM BOILER

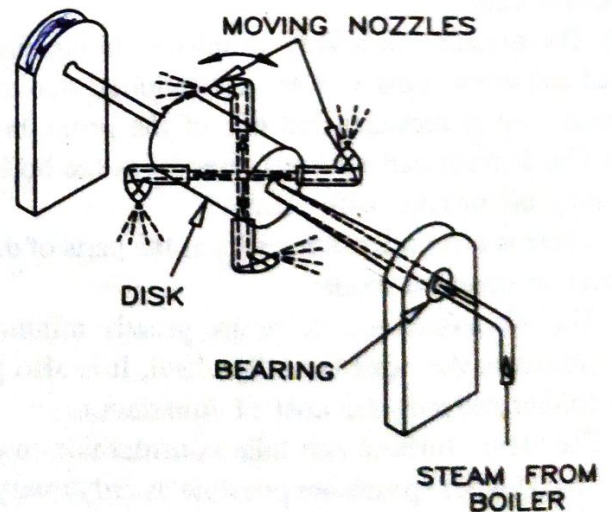FIXED NOZZLE

Fig. 19.1. (a) Impulse turbine                    Fig. 19.1. (b) Reaction turbine.

The impulse blades are symmetrical whereas the reaction blades are asymmetrical and are thicker at one end as shown in Fig. 19.2. The shape of the blade used in reaction turbine provides a suitable passage for steam to expand (like that of a nozzle).
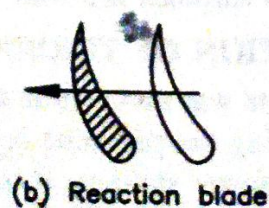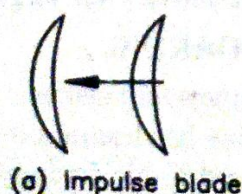


(a) Impulse blade                                     (b) Reaction blade

Fig. 19.2. Turbine blades.

The blade arrangements for impulse and impulse-reaction turbines are shown in Fig. 19.1. The variation in pressure and velocity of steam passing through the turbine is also shown in Fig. 19.1.

Most turbines are of axial flow type except Ljungstrom turbine which is radial flow. Maximum efficiency in a simple impulse turbine is secured by having a velocity of steam jet as it leaves from the expanding nozzle equal to twice the speed of the blades upon which it impinges. In order to meet this requirement,
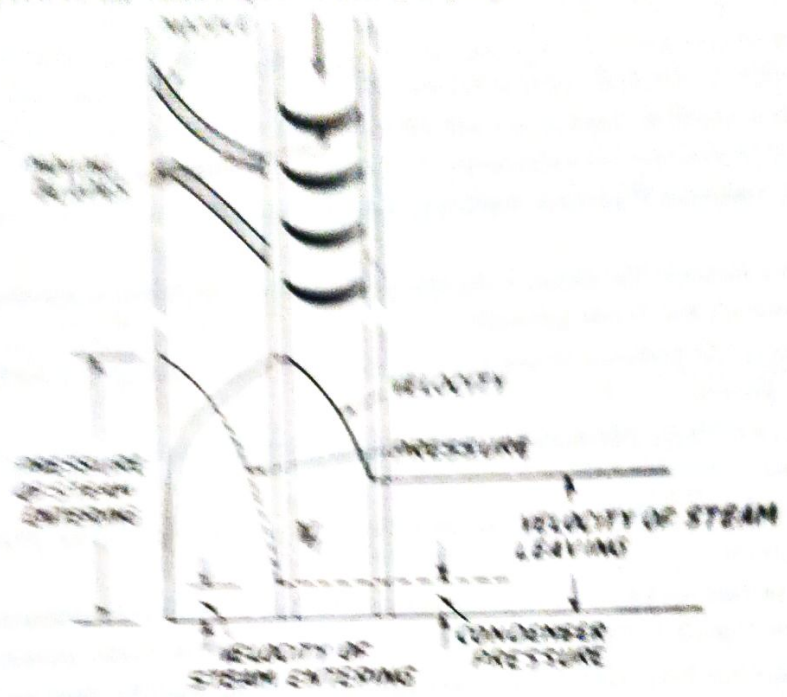


(a) An Impulse Turbine
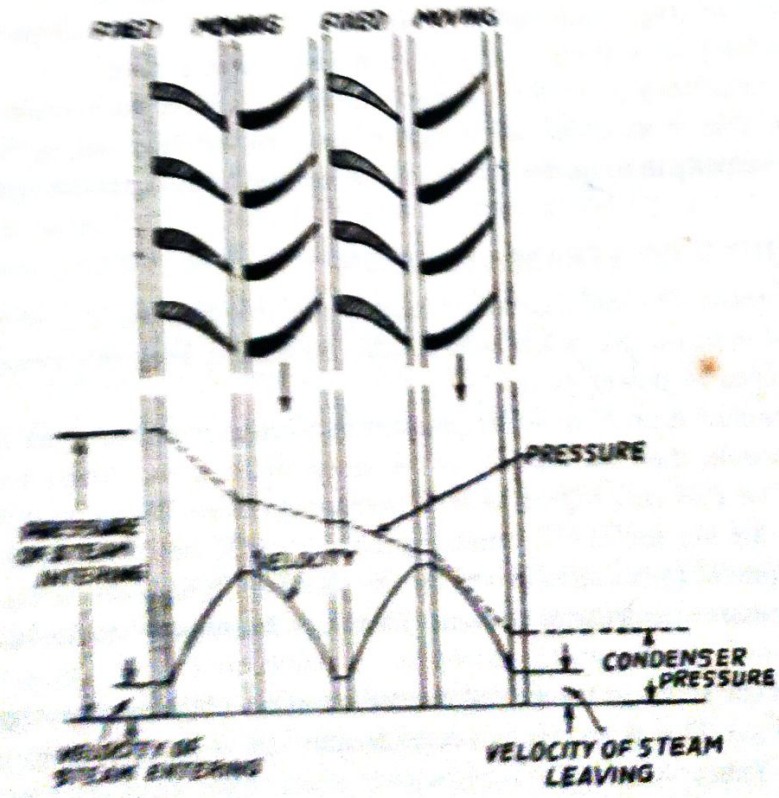


(b) A Reaction Turbine

Fig. 19.1. Arrangements of blades for impulse and reaction turbines.

the turbines must either operate at extremely high speed or the pressure drop must be divided into a number of stages. Since it is desirable to have a direct connection for driving the usual, 50-cycle or 60 cycle turbine generator, most turbine generator sets operate at 3000 or 3600 r.p.m. and have a number of stages. A few of the largest units are built to operate at 1800 r.p.m. as the blade area required to pass the large volume of expanded steam in the last stage before the condenser requires such long blades that the blades cannot withstand the centrifugal forces at 3600 r.p.m. The steam pressure also has a bearing on the number of stages.

Most of the power plant turbines are of the impulse reaction type. But due to a number of reasons, like the low volume at the high inlet pressure, the first one or two stages are of the impulse type.

After the first impulse stage, there are usually a number of impulse-reaction stages. Nearly 20 to 30 stages are common in practice with reaction turbines. The details about these are discussed in the next articles.

**Impulse vs. Impulse Reaction Turbine.** The differences between the impulse and reaction turbines are listed below :

1. In impulse turbine, the steam completely expands in the nozzles and its pressure remains constant during its flow through the blade passages.

In impulse reaction turbines steam expands partially in the nozzle and further expansion takes places in the rotor blade passage.

2. The relative velocity of steam passing over the blade of impulse turbine remains constant (assuming there is no friction).

The relative velocity of steam passing over the blade of reaction turbine increases as the steam passing over the blade expands.

3. The impulse turbine blade have symmetricals profile where the reaction turbine blades have aerofoil section as shown in Fig.19.2. The area of flow changes along the blade passage like that in a nozzle.

4. The pressure on both ends of the moving blade of an impulse turbine is same whereas different pressures exist on two ends of the moving blade of a reaction turbine.

5. The number of stages required for reaction turbine are more compared with impulse turbine for the same power developed as the pressure drop in each stage is small.

6. The blade efficiency curve for the reaction turbine is more flat compared with the efficiency curve of impulse turbine. This is an added advantage of reaction turbine, leading to part load economy.

7. The steam velocity in a reaction turbine is not very high and hence the speed of the turbine is relatively low.

## 19.4. COMPOUNDING OF STEAM TURBINES

In order to increase the plant thermal efficiency and to reduce the size of the plant, the steam pressure and temperature used in power plants has been steadily increasing. Presently pressures in the range of 100-140 bar are commonly used in power plants.

If the entire pressure drop from boiler pressure to condenser pressure (say 100 bar to 0.1 bar) is carried out in single stage nozzle, then the velocity of the steam entering the turbine blades will be very high. The turbine speed has to be also very high as it is directly proportional to the steam velocity, Such high R.P.M. of the turbine rotors are not useful for practical purposes and a reduction gearing is necessary between the turbine and the equipment (generator) driven by the turbine. There is also a danger of structural failure of the blades due to excessive centrifugal stresses. Therefore the velocity of the blades is limited to about 400 m/sec.

The velocity of the steam at the exit of the turbine is also high when a single stage of blades are used as shown in Fig. 19.3 (a). This gives rise to a considerable loss of kinetic energy (about 10 to 12%) reducing the efficiency of the unit.

These difficulties associated with the use of single stage turbines for large pressure drops can be solved by compounding. i.e., the use of more than one stage.
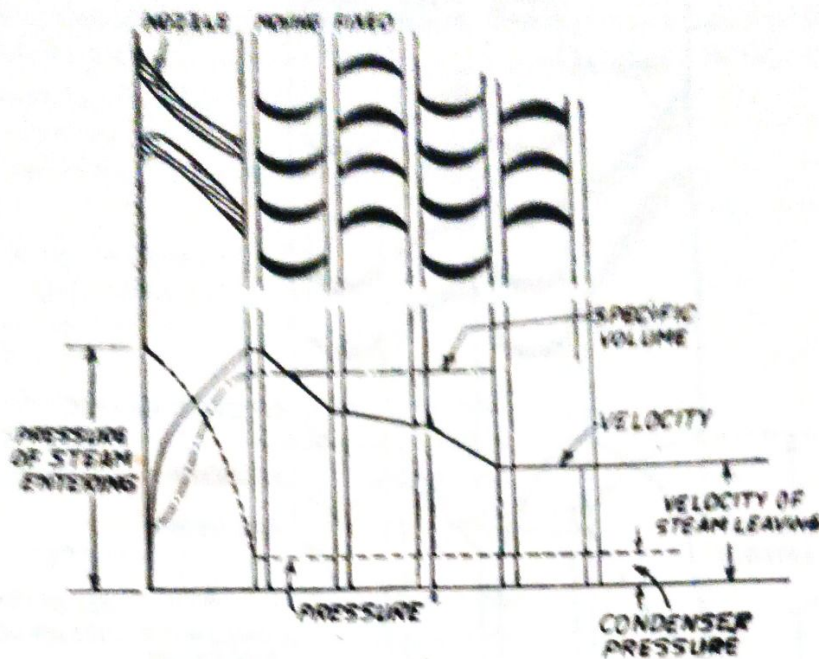
Fig. 19.4. Velocity compounded impulse turbine.

**(1) Velocity Compounding.** The arrangement of this type of compounding is shown in Fig. 19.4. There is only one set of nozzles and two or more rows of moving blades. There is a row of fixed guide blades in between the moving blades as shown in Fig. 19.4. The function of fixed blades is only to direct the steam coming from first moving row to the next moving row. These are also known as guide blades. The enthalpy drop takes place only in the nozzle at the first stage and it is converted into kinetic energy. The kinetic energy of the steam gained in the nozzle is successively absorbed in stages by the rows of moving blades and finally the steam is exhausted from last row of the blades. A turbine working on this principle is known as velocity compounded impulse turbine. The Curtis turbine is an example.

The variation of the velocity, pressure and specific volume of the steam along the axis are also shown in Fig. 19.4. The specific volume of the steam remains constant as the steam flows along the axis of the turbine therefore the blade height is small in all rows.

**(2) Pressure Compounding.** A number of simple impulse turbine stages arranged in series as shown in Fig. 19.5 is known as pressure compounding. In this arrangement, the turbine is provided with one row of fixed blades (works as nozzles) at the entry of each row of moving blades. The total pressure drop of the steam does not take place in a single nozzle but is divided among all the rows of fixed blades which work as nozzles.

The velocity, pressure and specific volume variations of steam along the axis of the turbine are shown in Fig. 19.5. As the pressure of the steam gradually decreases, the specific volume of the steam gradually increases therefore the blade height has to be increased towards the low pressure side.

**(3) Pressure and Velocity Compounding.** This is a combination of pressure and velocity compounding. The total pressure drop of the steam from boiler to condenser pressure is divided into a number of stages as done in pressure compounding and the velocity obtained in each stage is also absorbed in several stages. The arrangement is shown in Fig. 19.6.

This arrangement requires fewer stages and a compact turbine can be designed for a given pressure drop.

The velocity, pressure and specific volume variations of steam along the axis of the turbine are shown in Fig. 19.6. The blade height in the second stage must be greater than the first stage as the specific volume of steam is higher in the second stage.
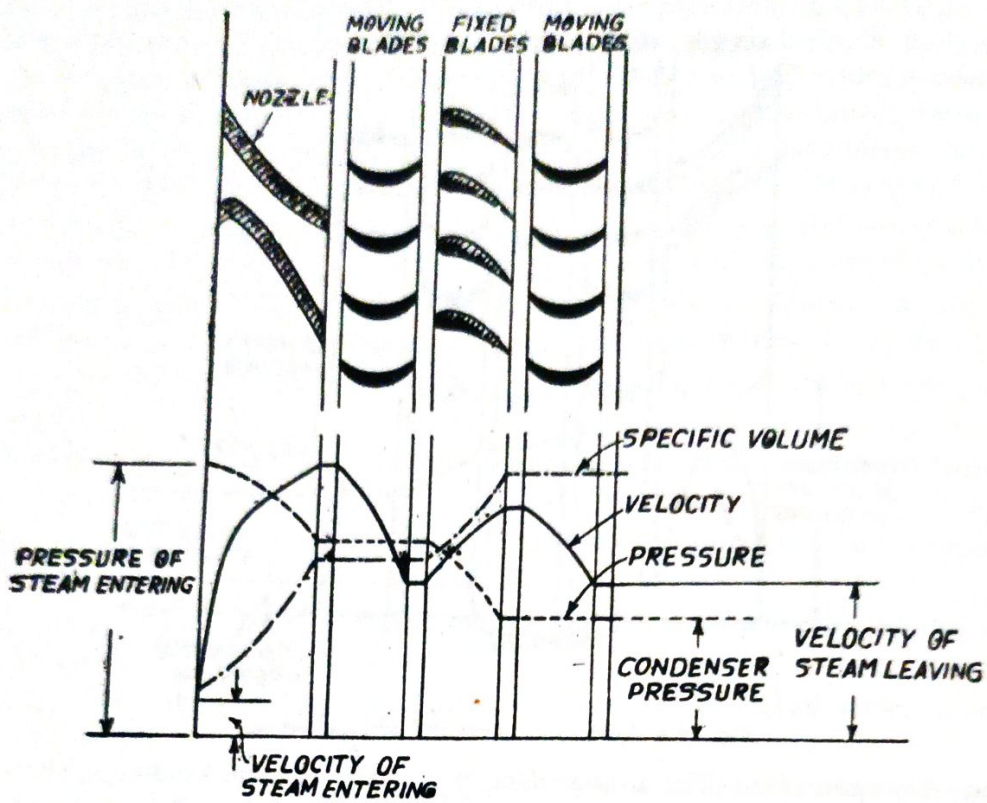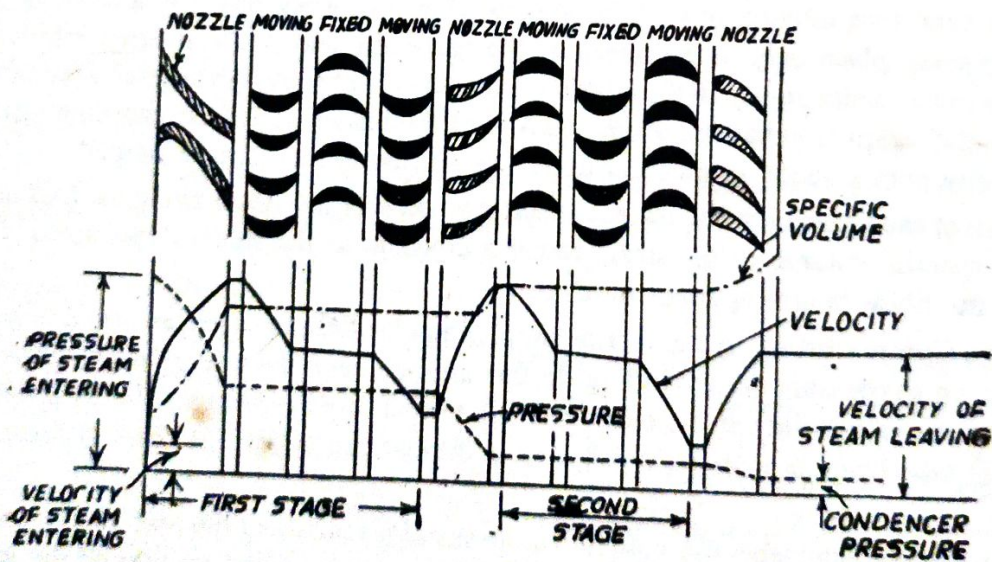
Fig. 19.5. Pressure compounded steam turbine.



Fig. 19.6. Pressure and velocity compounded steam turbine.

This method has the advantage of pressure compounding to provide higher pressure drop in each stage and hence fewer stages and the advantage of velocity compounding to reduce the velocity of each blade row.

## 19.5. VELOCITY DIAGRAMS

It is necessary to determine the rate of change of momentum of steam across the moving blades in order to find the force on blades. The power developed can then be determined from a knowledge of the flow rate and the blade speed. It is also equally important that the steam should enter and leave the blade without any shock and hence suitable inlet and outlet angles of the moving blades should be evaluated.

This is made possible with the help of velocity diagrams at the inlet and outlet of the moving blade. The velocity diagrams at the inlet and outlet of the moving blades are shown in Fig. 19.7 (a).

The steam flows axially through the annulus formed by the blade rings and useful impulse force should be calculated in the plane of rotation of the blades, tangential to the turbine rotor.

The velocity diagrams shown in Fig. 19.7 (b) are only valid for the moving blade of an impulse turbine and these diagrams should not be used for reaction turbine blades [shown in Fig. 19.7 (c)], because in reaction turbines, the relative velocity of steam increases as the steam flows over the moving blades. In the case of impulse turbine the relative velocity of steam remains constant as steam flows over the blades.

The notations used are listed below :

$\alpha_1$ = Angle, the absolute velocity at inlet makes with the plane of moving blades (or nozzle angle or outlet angle of fixed blade).

$\alpha_2$ = Angle the absolute velocity at outlet makes with the plane of the moving blade (or inlet angle of fixed blade).



Fig. 19.7 (a) Velocity diagram for impulse blade.

$\beta_1$ = Blade angle at inlet.

$\beta_2$ = Blade angle at outlet.

$U$ = Circumferential or tangential velocity of the blade.

$V_{a1}$ = Absolute velocity of steam at inlet of moving blade.

$V_{a2}$ = Absolute velocity of steam at outlet of moving blade.

$V_{w1}$ = Velocity of whirl at the entry of moving blade, i.e., tangential component. of $(V_{a1}) = V_{a1} \cos \alpha_1$.

$V_{w2}$ = Velocity of whirl at the exit of moving blade, i.e., tangential component of $(V_{a2}) = V_{a2} \cos \alpha_2$.

$V_{f1}$ = Velocity of flow at the entry of moving blade. i.e., axial – meridinal component of $(V_{a1}) = V_{a1} \sin \alpha_1$.

$V_{f2}$ = Velocity of flow at the exit of moving blade i.e., axial—meridinal component of $(V_{a2}) = V_{a2} \sin \alpha_2$.

$V_{r1}$ = Relative velocity of steam to moving blade at entrance.

$V_{r2}$ = Relative velocity of steam to moving blade at exit.

$m_s$ = Mass of steam flowing over blades.

$d$ = Mean diameter or blade drum.

$h$ = Height of blade.

$\rho$ (Speed ratio) = $\dfrac{U}{V_{a1}}$ .

$K$ (Blade velocity coefficient) = $\dfrac{V_{a2}}{V}$ (due to frictions)

The combined inlet and outlet velocity diagrams on the same base ($U$) are shown in Fig. 19.7 (b).
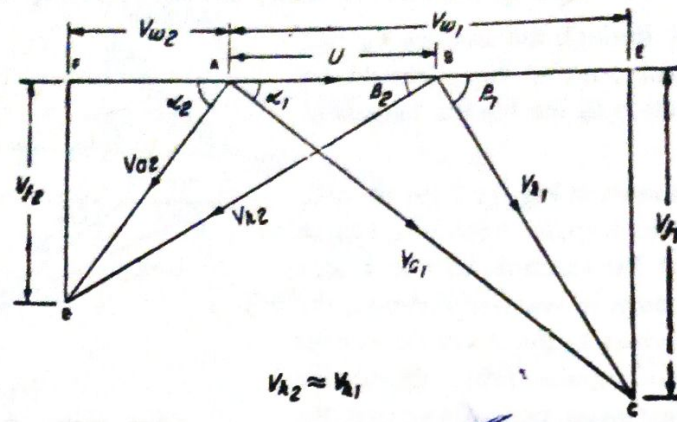


Fig. 19.7. (b). Velocity diagrams for impulse blades.

The work done on the blade is due to the force along the direction of its movement.

The change in velocity of whirl causes a force to act on the blades in the directions of motion. The velocity of flow determines the mass of steam flowing through the blade rows. The axil thrust force is also caused by any change in the flow velocity. The absolute velocity of the steam entering and leaving the turbine decides the inlet and exit blade angles when the turbine is running at a constant speed.

For blades with smooth surface, it can be assumed that the friction loss is very small or zero. For the above assumption, the condition $V_{r1} = V_{r2}$ is valid as no expansion takes place along the moving blades. There is always a certain loss of velocity during the flow of steam over the blade and this loss is taken into account by introducing a factor called blade velocity coefficient. It is given by

$$K = \frac{V_{r2}}{V_{r1}} \text{ where } V_{r2} < V_{r1}$$

The velocity diagram for the reaction turbine blade is shown in Fig. 19.7 (c).

It is already seen that the relative velocity of steam in the impulse turbine blade remains constant as the steam glides over the blades or is reduced slightly due to friction. In reaction turbine blades, the steam expands as it flows over the moving blades. The effect of the continuous expansion of steam during the flow over the blade is to increase the relative velocity of steam.
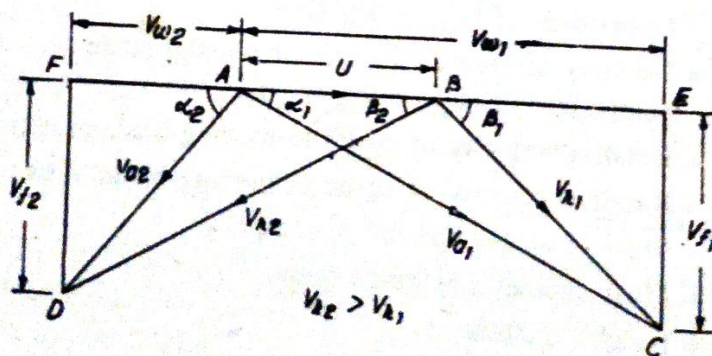


Fig. 19.7 (c) Velocity diagrams for reaction blade.

∴  $V_{r2} > V_{r1}$ for reaction turbine blade, and    $V_{r2} \leq V_{r1}$ for impulse turbine blade.