Cooling Technology: Why and How utilized in Food Processing and allied Industries

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Module No 10

Lecture 46

Centrifugal Compressor

Good afternoon, my dear boys and girls and my dear friends. We have completed in the previous classes compressors particularly your reciprocating type. Today, we are doing with centrifugal compression, right. Right from the word centrifugal compression, we can say, it is only on at the end. So, we can say that, what it is? It is also known as turbo compressors, and it belongs to the rotodynamic type of compressors. In the compressor class, previously, we have said, one type was rotodynamic, right.

So, that becomes, the centrifugal compressors. So, in these compressors, the required pressure rise takes place due to the continuous conversion of angular momentum imparted to the refrigerant vapour, or refrigerated vapour, by a high speed impeller, into static pressure. Unlike reciprocating compressors, centrifugal compressors are having steady flow devices. Hence, they are subjected to less vibration, as well as, less noise, right. So, if we show one such compressor, which is under centrifugal force, this is the two schematic views, right.

There is not schematic, it is schematic, and there are, as you see, from the refrigerant inlet, right, to the refrigerant outlet, it is divided into 6 parts. Number 1, is refrigerant inlet, and it is done through the eye of the centrifugal compressor. Then 2, is the impeller. This is the impeller, which, and the space of the impeller, which, we see that, this is here, 3, is the refrigerant passage. So, refrigerant from one to other impeller, between the impellers, it, yeah, from one to other impellers. So, it is passing through, then, 4, is the vaneless diffuser, this is, what we are referring to, is the vane less diffuser. 5 is the volute casing, right, sorry, 5 is the volute casing, and volute casing is this one, right, this one is the volute casing, right.

Obviously, there is no impeller in this, and 6, that is, this part, 5 is the volute casing, not that was 4, sorry, there was 4, this was 4, that is vaneless diffuser, and 5 is this, ok, that is, volute casing, and 6 is the discharge of the refrigerant, right. So, if we, if we analyze it, afterwards, we see that, the numbering, we have already said, the impeller, then number of blades, which is, made of the impeller, and which form the flow of the passages, that is, through the number 3, for refrigerant, right. Then, we can analyze it, that, from the eye, the refrigerant inlet, or refrigerant enters the flow passages, formed by the impeller blades, which, rotate at very high speed. As the refrigerant flows through the blade passages towards the tip of the impeller, it gains momentum, and its static pressure also increases. From the tip of the impeller, the refrigerant flows into a stationary diffuser, that is, which, we have shown as number 4.

In the diffuser, the refrigerant is decelerated. You know, acceleration, which is increasing, deceleration, which is decreasing the flow, or velocity, whatever we call. So, it is decelerated, and as a result, the dynamic pressure drop is converted into static pressure rise. Thus, increasing the volute casing, that is, content 5, where, further conversion of velocity into static pressure takes place, due to the divergent shape of the volute. Similarly, the pressurized refrigerant, that leaves the compressor from the volute casing, 6. The gain in momentum is due to the transfer of momentum from the high speed impeller blades to the refrigerant, confined between the blade passages, that is, 3, which we had shown.

Increase in static pressure is due to the shelf compression, caused by the centrifugal action, because under the centrifugal action, constantly it is rotating. This is analogous to the gravitational effect, where gravitational effect means, you have this much of material, so that, the pressure is also built up, because of the material, right, weight of the material, right, that is, how we call it to be gravity. So, this is analogous to the gravitational effect, which causes the fluid at a higher level to press the fluid, which causes the fluid at higher level, ok, which causes, or rather, press the fluid below which is already there, below it, due to the gravity, or by its weight. The static pressure produced in the impeller is equal to the static head, which would be produced by an equivalent gravitational column.

If we assume, the impeller blades to be radial, and the inlet diameter of the impeller to be small, then, the static head, that is, h developed in the impeller passage, for a single stage, can be written as h is equal to V square by g, right. So, if that be true, then, we have, of course, has to, we have to say, what is, what that is? h is the static head developed, that in terms of meter, right, like h rho g, then, V is the peripheral velocity of the impeller. So, we have said, angular velocities. Angular velocity can be converted into peripheral velocity, right. So, this peripheral velocity of the impeller will, or tip speed in terms of meter per second, and g is of course, the gravitational force due to the gravity, that is, meter per second square. So, we can increase in total pressure, that is, delta p, as the refrigerant flow through the passage, is given by delta p, is rho gh, and that is equal to rho V square, right.

$$h = \frac{V^2}{g} \quad \dots (1)$$

Where, h = static head developed, m; V = peripheral velocity of the impeller wheel or tip speed, m/s; g = acceleration due to gravity, m/s^2

So, we can say, that can be seen that, for a given refrigerant, with a fixed density, the pressure rise, that depends only on the peripheral velocity, or tip speed of the blade. This, please keep in mind that, it can be seen that, for a given refrigerant, with a fixed density, the pressure rise depends only on the peripheral velocity, or tip speed of the blade. The tip speed of the blade is proportional to the rotational speed, that is, rpm of the impeller, and the impeller diameter, the maximum permissible tip speed is limited by, of course, the strength of the structural material of the blade. Usually, it is made of high speed chrome nickel steel. Obviously, if the structure gets burst, then no point. So, it can go to that, where, the peripheral speed could be as high.

$$\Delta P = \rho g h = \rho V^2 \quad \dots (2)$$

So, that the limiting condition is the strength of the structural material, and the sonic velocity of the refrigerant. Since, we are not having time, but sonic velocity, if you go to my another course, which, has already been given, and it is available in NPTEL, where, both fluid flow and heat transfer, I have covered a little mass transfer also. So, there you will see that, the sonic velocity is the velocity of sound in air, right. Velocity of sound in air. You have heard the siren, if you are nearby of any industry, right, or of course, there is no such fighting, or I should not say fighting, there is no such war, where, earlier this siren used to be blown, to warn the people. However, so that is the sonic velocity.

So, these are the two controlling limit, that how much peripheral speed can go up? Under these limiting conditions, the maximum achievable pressure rise? So, hence, it can be said to be maximum achievable temperature lift, because, pressure is also going up, temperature is also going up for a given volume of the single stage centrifugal compressor, and that is, limited for a given refrigerant. Hence, multistage centrifugal compressors are used for large temperature lift, or rise of the, lift means, rise applications. In multistage centrifugal compressors, the discharge of the lower stage compressor is fed to the inlet of the next stage compressor, right, and so on.

In the multistage centrifugal compressors, the impeller diameter of all stages remain same, but the width of the impeller becomes progressively narrow in the direction of flow, as refrigerant density increases progressively, right. Then, we come to, the blades of the compressor are either forward curved, or backward curved, or radial. So, backward curved blades were used in the older compressors, whereas, the modern centrifugal compressors use mostly radial blades. The stationary diffuser can be vane, or vane less, as the name implies in vaned diffusers, vanes are used in the diffuser to form flow passages, the vanes can be fixed or it can also be adjustable. Vaned diffusers are compared, or it is compact, rather, compared to the vane less diffuser, and are commonly used for high discharge pressure applications, right.

So, vane less diffusers, vane diffusers, are much more compact than that of the vane less, or rather, that of the vane one, right. Vaned diffusers are compact, vanned diffusers are compact, compared to the vane less type, and are commonly used for high discharge pressure applications. However, the presence of vanes in the diffusers can give rise to shocks, as the refrigerant velocities at the tip of the impeller blade, could reach sonic velocities, right, in large. So, high speed centrifugal compressors, it could be up to the sonic velocities, that is why, it may lead to producing a shock. In vane less diffusers, the velocity of refrigerant in the diffuser, decreases, and static pressure increases, as the radius increases, as a result, for required pressure rise, the required size of the vane less diffuser could be large, compared to vanned diffuser.

However, the problem of shock, due to supersonic velocities at the tip, does not arise with vane less diffusers, at the velocity, as the velocity can be diffused smoothly. In this case also, another term has come up that is supersonic. So, I said sonic velocity that is the velocity of sound in air. Supersonic and subsonic, there are two types. So, supersonic is that, where the velocity of sound is higher than that of the sonic velocity, whereas, subsonic velocity is lower than that of the sonic velocity, and these are expressed in terms of Mach number, right. Let me write in terms of Mach number, M A C H, Mach number, and when, this is normally expressed as Mach number, Mach number is 1. It is sonic velocity, when Mach number is greater than 1, it is supersonic velocity, and when Mach number is less than 1, it is subsonic velocity, right.

So, if you again, as I said, if you are really interested, you can consult, by another, this thing. We can say that the presence of vanes in the diffusers can give rise to shocks as the refrigerant velocities, at the tip of the impeller blade, could reach sonic velocities, in large, high speed centrifugal compressors, right. In vane, in vane less diffusers, the velocity of refrigerant in the diffuser decreases, and static pressure increases, as the radius increases. As a result, for a required pressure rise, the required size of vane less diffuser could be large, compared to the vane diffusers. However, the problem of shock due to supersonic velocities, at the tip, does not arise with the vane less diffusers, as the velocity can be diffused smoothly.

Then, we can say, generally, as adjustable guide, vanes or pre rotation vanes are added at the inlet, that is, at the eye of the centrifugal compressor of the impeller for capacity control. And if we analyze the centrifugal compressor, then, we can say that, applying energy balance to the compressor, right, we obtain from steady flow energy equation. We obtain steady flow energy equation. So let us look at the compressor, which we are saying. So we are doing analysis of this, this is the centrifugal compressor, right this is the centrifugal compressor, and this is the inlet.

So, this is the exit, so, Q quantity of heat is rejected and W c quantity of work is done on the system. We know this arrow is on the system, and this arrow is by the system, right, that we have seen earlier. So, we can say that, from this figure, analyzing the figure, we can say that minus Q plus m into h i plus v i square by 2 plus g z I, that is equal to minus W c plus, rather, minus W c plus m into h e, exit, plus v e square by 2 plus g z e. If you remember, we had earlier said that, this type of equation, we said it to be Bernoulli's equation, right. So, we can say, then, that, of course, the terms are Q is heat transfer rate from the compressor, W c is the work transfer rate from the compressor, m is the mass flow rate of the refrigerant, v I, v e are inlet and outlet velocities of the compressor refrigerant, and z i and z e are height above a determinary gravitational force field.

$$-Q + m(h_i + \frac{V_i^2}{2} + gZ_i) = -W_c + m(h_e + \frac{V_e^2}{2} + gZ_e)$$

… (3)

where Q = heat transfer rate from the compressor
W = work transfer rate to the compressor
m = mass flow rate of the refrigerant
V_i, V_e = Inlet and outlet velocities of the refrigerant
Z_i, Z_e = Height above a datum in gravitational force field at inlet and outlet

So, equating, if it is horizontal, then, obviously, we do not have this g z, and this, ok. If this is also can be neglected, so, we can write minus Q plus m h i, that is, equal to minus W c plus m h e right. So, in that case, we can write that, in a centrifugal compressor, the heat transfer rate Q is normally negligible, as the area available for heat transfer is small compared to other energy terms. So, the rate of compressor work input for adiabatic compression, that can be given as W c equal to m into h e minus h i, exit, and inlet enthalpies. So, the above equation is valid for both reversible as well as irreversible adiabatic compression, provided the actual enthalpy is used at the exit in case of irreversible compression.

$$-Q + mh_i = -W_c + mh_e \quad \dots (4)$$

In case of reversible adiabatic compression, the power input to the compressor is given by W c isentropic, is equal to m into h e minus h I, isentropic. Then using that thermodynamic relation, earlier we have done, T d S equal to d h minus V d p, the isentropic work of compression, that can be given by W isentropic is equal to, W c isentropic is equal to h e minus h I, isentropic, that is equal to integral of V d p within the limit p i to p e and under isentropic condition. Thus, the expression for reversible isentropic work of compression is same, for both reciprocating as well as centrifugal compressors. If you go back to reciprocating, you will see the same. However, the basic difference between actual reciprocating compressors, and actual centrifugal compressors lie in the source of irreversibility.

$$W_c = m(h_e - h_i) \quad \dots (5)$$

$$W_{c,isen} = m(h_e - h_i)_{isen} \quad \dots (6)$$

Tds=dh–vdp

$$w_{c,isen} = (h_e - h_i)_{isen} = \int_{Pi}^{Pe} vdp \Big|_{isen} \quad \dots (7)$$

In case of reciprocating compressors, the irreversibility is mainly due to heat transfer and pressure drops across the bulbs, and connecting pipelines. However, in case of centrifugal compressors, since the refrigerant has to flow at very high velocities, through the impeller blade passages, for a finite pressure rise, the major source of irreversibility is due to the viscous shear stresses at the interface, between the refrigerant and the impeller blade surface, right. So, in reciprocating compressors, the work is required to overcome the normal forces acting against the piston. While in centrifugal compressors, work is required to overcome both normal pressure forces, as well as, viscous shear forces.

$$\eta_{pol} = \frac{w_{pol}}{w_{act}} = \frac{\int_{Pi}^{Pe} vdp}{(h_e - h_i)} \quad \dots (8)$$

That means, if this is equal to vdP integral, under Pi to Pe over h e minus h I, where, obviously, W poly and W act, are the polytrophic, and actual work of compression respectively, right. So, we have come to the end of the centrifugal compressor, and our time is also over. So, next, perhaps, we will go over the other, ok. Thank you so much.