Applied Thermodynamic for Engineers
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Lecture – 27
SSS Cycles and Refrigerants

Hello friends, welcome again for the second lecture on refrigeration cycles. Now, in the yesterday's lecture you have been introduced the concept of refrigeration and also, the need of refrigeration, where we have learnt that the objective of refrigeration is not only to lower the temperature of a space than the immediate surrounding but also to maintain that low temperature. And while it is not very difficult to produce a low temperature zone but, it is much more difficult to maintain that low temperature.

Because, naturally heat always flows from high temperature to low temperature and therefore from the low temperature space we continuously have to extract the heat and deposit it back to the surrounding. And for that purpose, we need some work input according to the second law of thermodynamics which leads to the concept of the reverse heat engine, which is also the refrigeration or heat pump cycles. Yesterday, we have also discussed about the applications of refrigeration, different ways of producing the low temperature zone and out of which the application of phase-change processes has been found to be the most common one.

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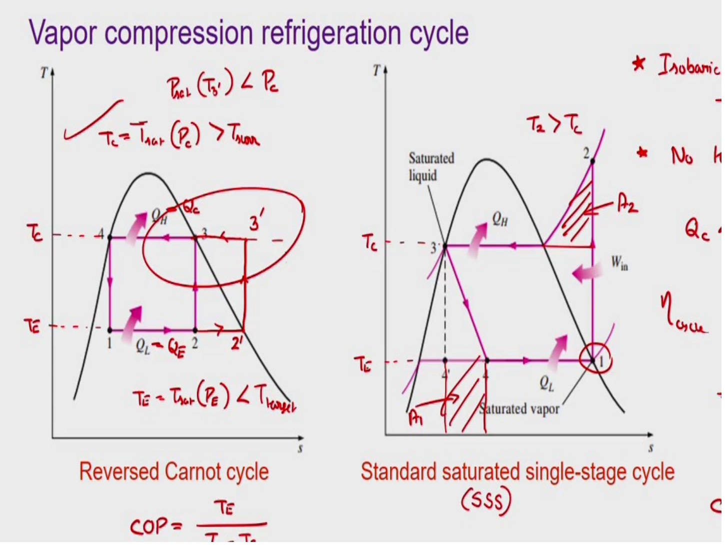

Vapor compression refrigeration cycle

Reversed Carnot cycle

Standard saturated single-stage cycle

In fact, the other processes that you have discussed all are in practical use to some degree. But the one that is application of the phase-change process probably is the most common way of establishing refrigeration. In fact, I should not use probably, because in 90% cases you will find that the refrigeration systems that you are dealing with or you are experiencing is based upon some kind of phase change process.

And even there also it is mostly the one that is retained here is the vapour compression refrigeration cycle. Where, we use a vapour to get compressed using a compressor and it operates between two pressure levels: one is the evaporator pressure and other is the condenser pressure. The most ideal cycle there is the reverse Carnot cycle. Which, is just opposite to the conventional Carnot cycle, here the directions of just become opposite.

Directions of all heat and work transfer and also the direction of each of the processes we have just reversed it. Accordingly, it is able to extract heat from low temperature zone. Let us say the lowest possible temperature that we are talking about here is the evaporator temperature T_E, the highest temperature is the condenser temperature T_C, so from that condenser the refrigerants will be able to extract Q_E amount of energy from the low temperature space. Whereas in the condenser the refrigerants will be depositing Q_C amount to the surrounding and in between the compressor, this is the compression process. Here, the purpose of the compressor is to change the pressure level of the refrigerant from the evaporator pressure level to the condenser pressure level such that, the saturation temperature corresponding to the evaporator pressure that is P_{sat} or I should write the other way T_E, which is nothing but the saturation temperature corresponding to your evaporator pressure has to be lower than the target temperature. Similarly, T_C, the condenser temperature, which is the saturation pressure corresponding to the condenser pressure has to be greater than the surrounding temperature. So, the purpose of the compressor is to operate between these two pressure levels or to change the pressure level of the difference between this P_E and P_C suitably.

Now, while the reverse Carnot cycle is the most ideal reverse heat engine we can have giving the best possible performance of this possible *COP,* but there are several practical problems with it. If you just write the *COP* for this, we know, we have already derived yesterday that *COP* for a reverse Carnot cycle will be equal to:

$$COP_{rev\ Carnot\ \ cycle} = \frac{T_E}{T_C - T_E}$$

However, the problem with the practical application of the reverse Carnot cycle is with the two isentropic processes.

While the two heat transfer processes are isothermal but they also isobaric in nature because both are happening inside the phase change zone and so achieving them is practice is not very difficult. But it is very difficult to have this compression process starting from the saturated mixture. And also, it is very difficult to have the expansion process hardly any work out be are going to get if we start is saturated liquid and continuous expand to a very low-quality mixture.

To eradicate that problem, particularly on the compressor side of an expansion, often this evaporation process is allowed to continue up to the saturated vapour line, let us say point 2' from there we go for the compression process up to the temperature level of T_C which is this to this point 3'. Now having the evaporation process from 1 to 2' and then the compression from 2 to 3' is again not too much problematic we can easily get them.

However, the problem happens during the condensation process because if we want to move from this 3' towards this point 4, then this 3' to 3, this part is the most problematic because here we are trying to have isothermal heat transfer, while the system is constantly moving to higher pressure zone. So basically, you have to design a compressor which will also act as an isothermal heat exchanger which is not possible in practice.

And for that purpose, we move to this cycle which we have turned as a standard saturated single stage cycle or the SSS cycle. And as we have discussed, if we compare this cycle with the previous one where the evaporation process is continuing up to the saturated vapour state. Then we have one additional part here. Let me just summarise the two changes that you are having in this SSS cycle compared to the reverse Carnot cycle. One is, here the heat transfer processes are not isothermal rather heat transfer process there isobaric. So, both evaporation and condensation process and isobaric in nature and therefore during the compression the final temperature, this T_2 is not T_C rather it is greater than T_C where T_C refers to this particular temperature, just being consistent with the notation used for the reverse Carnot cycle where this is our T_E.

Lowest temperature though remains T_E by highest temperature is higher than T_C now, so that here this choice of T_2 is such that T_2 becomes the saturation temperature corresponding to this T_C rather being corresponding saturation temperature, here we are going back to the notation that is:

$$T_2 = T_{sat}(P_c)$$

Because here, if we compare the Carnot cycle notation this particular notation here, we wrote this when our entire process was restricted in the saturation dome. But once your restricting this outside the saturation dome then this T_3' is greater than T_C or I should say it is greater than T_{sat} corresponding to P_C. Because the final pressure or rather if you write that the saturation pressure corresponding to T_3' will be less than P_C. So, we have to move towards a continuously high-pressure zone which problem is not here, as the entire heat transfer process is taking place at constant pressure.

And the second change that we have done, we are not having any turbine which has been replaced by a valve, a throttling device. So, we are not having any kind of work output prior to the process 3 to 4 is a plane throttling process. Let me first talk about the condensation process, in the condensation process this additional amount of heat has to be rejected. So, there is an increase in the total condensation amount of heat loss whereas in evaporation process because of the presence of the throttling this is the amount of heat transfer that is getting reduced. So, your Q_E there is reduction and if we call this area to be A_2 and call this area to be A_1 to be consistent with our notation that used in the previous lecture. Then you may have seen that the η for this cycle which has been defined as:

$$\eta_{cycle} = \frac{COP_{SSS}}{COP_{Carnot}}$$

that was found to be:

$$\eta_{cycle} = \frac{1 - \left({A_1}/{Q_{E,Carnot}} \right)}{1 + \left(\frac{A_1 + A_2}{W_{net\ in\ Carnot}} \right)}$$

So, any increase in A_1 and A_2 will reduce the efficiency of the cycle and accordingly the COP for this SSS cycle will always be less than COP for the Carnot cycle. But still this is practically much more relevant or much more possible because we do not have to deal with any compression during the heat transfer. Here, both the heat transfer as taking place at constant pressure. And also, we do not have to deal with that problematic turbine expansion rather we are replacing that with a throttling valve. Of course, that turbine going to work output that we could have got from that conceptual reverse Carnot cycle that you are not getting but still it is much more durable in practice.

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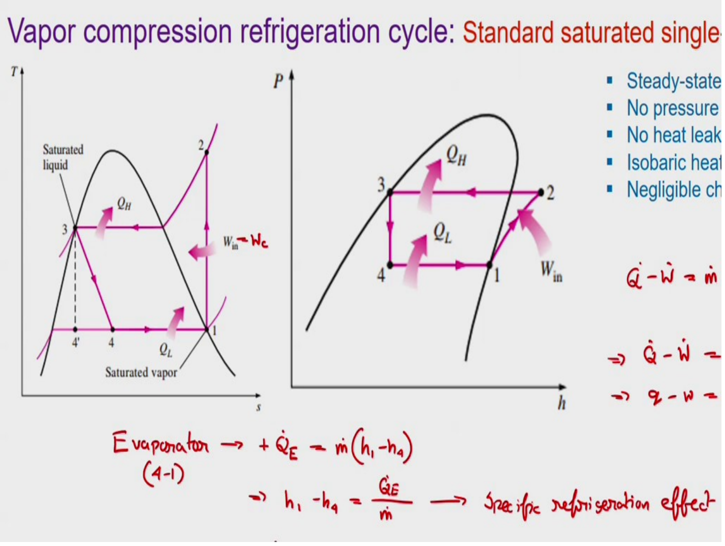

Vapor compression refrigeration cycle: Standard saturated single

- Steady-state
- No pressure
- No heat leak
- Isobaric heat
- Negligible ch

$\dot{Q} - \dot{W} = \dot{m}$

$\Rightarrow \dot{Q} - \dot{W} =$

$\Rightarrow q - w =$

Evaporator (4-1) $\rightarrow + \dot{Q}_E = \dot{m}(h_1 - h_4)$

$\Rightarrow h_1 - h_4 = \frac{\dot{Q}_E}{\dot{m}} \longrightarrow$ Specific refrigeration effect

And therefore, we use the standard saturated single-stage cycle for practical operations. In fact, its use is so common that quite often this is the one that is referred as a vapour compression refrigeration cycle only. So that *Ts* and also the *Ph* diagram both are shown here. The throttling process being a constant enthalpy process, so this cycle actually comprises of two constant pressure and one constant enthalpy process.

And therefore, it is quite convenient to represent this one on this *Ph* diagram. If you want to analyse this theoretically then these are the assumptions that you are considering. Here, there are four components evaporator, compressor, condenser and throttling valve. We are assuming all of them to be steady state steady flow devices, then no pressure drops and heat leakage from the pipelines because all these components will be connected by pipeline, so we are neglecting any kind of pressure drop and heat leakage from the pipelines. And also, I should at that here assuming both the compression process to be isentropic and we are neglecting any kind of pressure drop that may take place when the refrigerant passes through that evaporator and condenser. The heat addition and heat rejection are isobaric and no pressure drop during them and we are neglecting any changes in kinetic and potential energies. Then if we try to analyse this particular situation now for thermodynamics, for any kind of steady state steady flow process. If we want to apply the first law of thermodynamics for a single inlet single outlet device, we know that:

$$\dot{Q} - \dot{W} = \dot{m}[(h_e + \frac{1}{2}{c_e}^2 + gz_e) - (h_i + \frac{1}{2}{c_i}^2 + gz_i)$$

while *i* refer to the inlet state and as we are assuming to that kinetic and potential energy changes to negligible. So, this reduces to:

$$\dot{Q} - \dot{W} = \dot{m}\,(h_e - h_i)$$

or

$$q - w = \dot{m}\,(h_e - h_i)$$

So, now let us try to analyse each of the components separately. First the evaporator, so for the evaporator where we are talking about the process 4 to1, here it is isobaric and there is a heat transfer going on. But as the pressure is not changing during the process and this being a steady flow process, I hope you remember that for a steady flow process *w* can be written as:

$$w = -\int_1^2 vdp$$

as there is no pressure change this can be equal to zero in this particular situation. So, this can be written as:

$$\dot{Q}_E = \dot{m}\,(h_1 - h_4)$$

Q *dot E* is the amount of heat is added to the system. So, that is positive to be equal to:

$$(h_1 - h_4) = \frac{\dot{Q}_E}{\dot{m}}$$

this particular thing is known as specific refrigeration effect. *Q* *dot E* is the refrigeration effect because you are talking about the refrigeration cycle where the amount of heat extracted is the objective. So, *Q* *dot E* is our objective and once we are dividing it by the mass flow rate we are getting the specific refrigeration effect.

Now, if we move that to the compressor, for the compressor whereever process is 1 to 2, we are assuming this to be isentropic. So, no heat transfer is taking place rather work is being given to the system. So, let us write this W_{in} to be equal to W_C. In that case,

$$-(-\dot{W}_C) = \dot{m}(h_2 - h_1)$$

where *W* *dot C* is the work input given to the system which is work being done on the system being it is negative. That is,

$$\dot{W}_C = \dot{m}(h_2 - h_1)$$

and if there is any heat transfer going on, then we also have to add this but your assuming this to be isentropic and therefore this compression work is given as:

$$= \dot{m}(h_2 - h_1)$$

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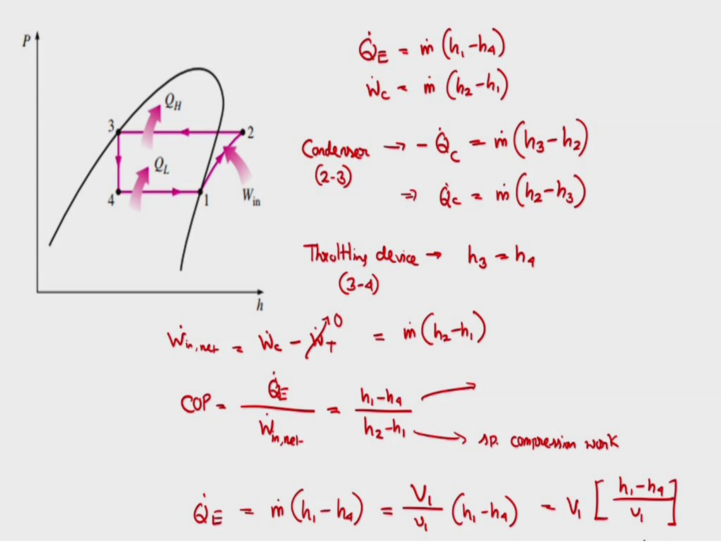

$$\dot{Q}_E = \dot{m}(h_1 - h_4)$$
$$\dot{W}_c = \dot{m}(h_2 - h_1)$$

Condenser (2-3) $\rightarrow -\dot{Q}_c = \dot{m}(h_3 - h_2)$

$\Rightarrow \dot{Q}_c = \dot{m}(h_2 - h_3)$

Throttling device (3-4) $\rightarrow h_3 = h_4$

$$\dot{W}_{in,net} = \dot{W}_c - \cancel{\dot{W}_T}^{\,0} = \dot{m}(h_2 - h_1)$$

$$COP = \frac{\dot{Q}_E}{\dot{W}_{in,net}} = \frac{h_1 - h_4}{h_2 - h_1}$$

$h_2 - h_1 \rightarrow$ sp. compression work

$$\dot{Q}_E = \dot{m}(h_1 - h_4) = \frac{V_1}{v_1}(h_1 - h_4) = V_1\left[\frac{h_1 - h_4}{v_1}\right]$$

If you just write the terms again from the previous slide, so have got

$$\dot{Q}_E = \dot{m}\,(h_1 - h_4)$$

and

$$\dot{W}_C = \dot{m}(h_2 - h_1)$$

Now we have the condenser, in the condenser we having the process 2 to 3 so and it is again isobaric, so no work transfer but there is heat transfer. So, this heat is being rejected by the system so it is negative and is equal to:

$$-\dot{Q}_C = \dot{m}\,(h_3 - h_2)$$

which gives you

$$\dot{Q}_C = \dot{m}\,(h_2 - h_3)$$

And finally, the throttling device where you are having the process 3 to 4 know we talking about the throttling process during is there is no heat transfer, there is no work transfer. And so, we know that throttling is an isenthalpic process, we can directly write:

$$h_3 = h_4$$

So, with this we have to evaluate the performance of the system in terms of its *COP*. So,

$$\dot{W}_{in,net} = \dot{W}_C - \dot{W}_T$$

as turbine is not present in this case so that will become:

$$\dot{W}_{in,net} = \dot{W}_C - \cancel{\dot{W}_T}$$

$$= \dot{m}(h_2 - h_1)$$

Then your *COP* becomes your desired output which is:

$$COP = \frac{\dot{Q}_E}{\dot{W}_{in,net}}$$

it becomes now:

$$= \frac{h_1 - h_4}{h_2 - h_1}$$

here this numerator is the specific refrigeration effect that have written and this denominator, it is often called the specific compression work, i.e., the amount of compressor work that you have to supply by unit mass flow rate.

So just from the knowledge of the enthalpy of these three points 1, 2 and 4 we are able to identify the value of the *COP* from this. And if you want to calculate any of the heat transfer, work transfer those also known just for knowledge of this enthalpies. There is another way sometimes we write this the refrigeration effect, *Q* *dot E*, if you just repeat this

$$\dot{Q}_E = \dot{m}\,(h_1 - h_4)$$

ṁ is the mass flow rate which remains constant at all the state points for a during steady state steady flow operation and ṁ can also be this is being the mass flow rate so we can also write this one as:

$$\dot{Q}_E = \dot{m}\,(h_1 - h_4) = \frac{V_1}{v_1}(h_1 - h_4) = V_1\left[\frac{h_1 - h_4}{v_1}\right]$$

Then this bracketed quantity is sometimes called the volume refrigeration effect. So, while the refrigeration effect is represent in terms of kilowatt, this is kilowatt per unit specific or terms of refrigeration per unit specific volume.

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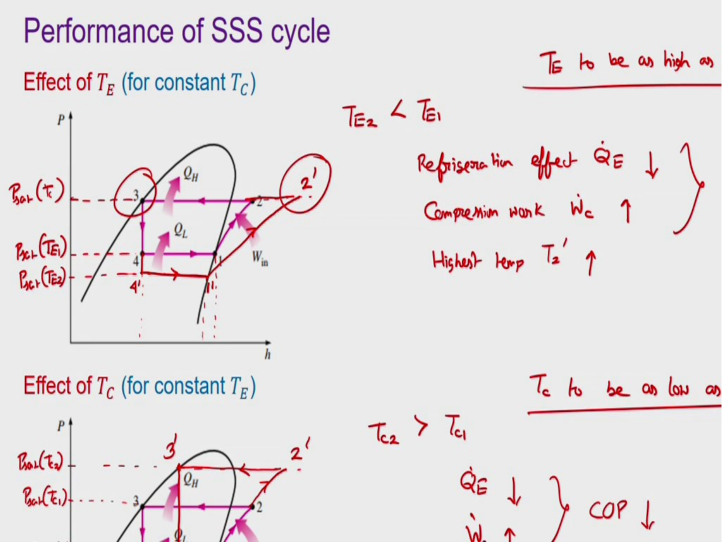

Performance of SSS cycle

Effect of T_E (for constant T_C)

T_E to be as high as

$T_{E2} < T_{E1}$

Refrigeration effect $\dot{Q}_E$ ↓

Compression work $\dot{W}_c$ ↑

Highest temp T_2' ↑

Effect of T_C (for constant T_E)

T_c to be as low as

$T_{c2} > T_{c1}$

$\dot{Q}_E$ ↓

$\dot{W}$ ↑

COP ↓

Now, to evaluate the performance of these SSS cycle, let us check the effect of the two end temperatures: T_C and T_E. So first let us check the evaporation temperature for constant T_C. From the previous slide we have seen how to calculate the corresponding performance parameter and also the *COP*. Now let us check the effect of T_E. So, I have plotted the *Ph* diagram for a particular combination of T_C and T_E. Let us say initially this is the value of T_C which remains constant, this is your TE for case one. Let us now choose another T_E for case number two. T_{E2} where this T_{E2} is less than T_{E1}. Now, if you want to plot the cycle then what do we have to do? For when the evaporator temperature is T_{E2} then this point 4 will get expanded to this point to become from some 4', then your evaporation process starts from here and proceed up to the saturated vapour line to give the end state as 1' and then we have to go to the final pressure to the compression. So, this being a constant, we have to expand this. Actually, I should not have written this T_C and T_E this way, it will be better if we write in terms of the pressures. This is the saturation pressure corresponding to P_C or maybe we can write this on as P_{sat} at corresponding to your T_C. This is P_{sat} corresponding to T_{E1} and the second one is P_{sat} corresponding to T_{E2}, the lower temperature.

Now for compression we have to somehow see that P_{sat} corresponding to T_C, the same final pressure. So, your compression line, may be somewhat like this, to reach the final temperature of 2' and from there it will proceed along the constant pressure line to reach the same state point 3.

So, three of the state points are changing, only one state point that is point number 3 at the end of condensation that is remaining the same. So, if we check the effects now, what about the refrigeration effect? Your refrigeration effect which is given in terms of this $Q_{dot\ E}$. Just compare the magnitude of the enthalpy change, point 4 and 4' they are similar because this defined as a throttling process. But if we compare 1' with 1, definitely you can see 1' is getting shifted towards left. That is the enthalpy value there is less than point 1 so, refrigeration effect decreases for this T_2. What about the compression work $W_{dot\ C}$? Here we are going from a lower pressure level to higher pressure level because the evaporator pressure has reduced while the final pressure the condenser pressure remains the same, so this will increase. And the combination of this two being COP which is nothing but:

$$COP = \frac{\dot{Q}_E}{\dot{W}_C}$$

for a throttling related device, this will of course reduce. And another parameter of importance is the highest temperature. So, highest temperature, which is this T_2' that is also increasing because of a reduction in the evaporation temperature, which is also not always the desirable thing.

The highest temperature associates the cycle is increasing, refrigeration effect is decreasing, compression work is increasing, leading to a reduction in overall *COP*. So, the recommendation is, we want our T_E to be as high as possible. Now, if you want to check the effect of T_C for a given value of T_E. Again, we have the same Ph diagram, let us identify the states. Here the evaporator side condition is not changing, so evaporator pressure remains the same which is this one P_{sat} at corresponding to T_E.

Let us say this one is your T_{sat} corresponding to T_1 and we take another pressure for higher value of condenser temperature, which is P_{sat} corresponding to T_{C2}. So, for this particular cycle what will be the condition where should be the point 3? If the point 3 has to be saturated liquid then the point has to be someone here. So, this is your 3', throttling process will go to this leading to 4', 4' to 1 will remain the same or at the end point of evaporation there is one will be remaining same.

Now, this 1 to 2' the compression process will be extended to this elevated pressure giving this as 2' and then from there we will be getting this as your condensation process. So, here our T_{C2} is greater than T_{C1}. So, what is effect of $\dot{Q}_E$ in this case? Definitely there is a reduction in the amount of heat extracted in the evaporator, $\dot{Q}_E$ is decreasing the compression work lowest pressure is same, but highest pressure is increasing, the compression work is also increasing, leading to a reduction is *COP*. And the compressor exit temperature or the highest temperature of the cycle the T_2' that is also higher compared to the previous case, so again this is also detrimental. That is quite similar to the reduction in the evaporator temperature and in increase in the condenser temperature also has all possible detrimental effects and therefore we always want T_C to be as low as possible.

But of course we are handicapped by the environmental conditions T_C has to be at the surrounding temperature and therefore the T_C requirement during the winter seasons can be significantly lower than the T_C requirement in the summer season, and therefore the *COP* for the same evaporator temperature during the winter season can be much better compared to the summer season.

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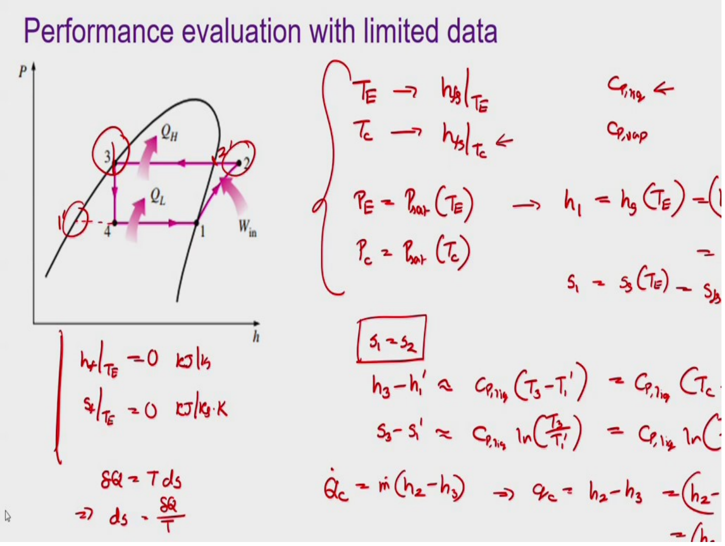

Performance evaluation with limited data

$T_E \rightarrow h_{fg}|_{T_E}$

$T_c \rightarrow h_{fg}|_{T_c} \leftarrow$

$c_{p,liq} \leftarrow$

$c_{p,vap}$

$P_E = P_{sat}(T_E) \rightarrow h_1 = h_g(T_E) = ($

$P_c = P_{sat}(T_c)$

$=$

$s_1 = s_g(T_E) = s_{fg}$

$\boxed{s_1 = s_2}$

$h_3 - h_1' \approx c_{p,liq}(T_3 - T_1') = c_{p,liq}(T_c$

$s_3 - s_1' \approx c_{p,liq} \ln\left(\frac{T_3}{T_1'}\right) = c_{p,liq} \ln($

$h_f|_{T_E} = 0$ kJ/kg

$s_f|_{T_E} = 0$ kJ/kg·K

$\delta Q = T ds$

$\Rightarrow ds = \frac{\delta Q}{T}$

$\dot{Q}_c = \dot{m}(h_2 - h_3) \Rightarrow q_c = h_2 - h_3 = (h_2 -$

$= (h$

Now, if we have to check out a few numerical examples, but before that let us see what are the minimum quantity of data that with which we can evaluate the performance of such a system. Let us see, for the cycle that is given we are given with the temperatures T_E and T_C, also we are given with the h_{fg} corresponding to T_E and h_{fg} corresponding to T_C. Because once we know that T_E and T_C and know nature of the refrigerants we can easily get the value of h_{fg}.

And we have also heard about the specific heat for both liquid and the vapour phases. Then what we can do? Of course, we know all the state points, as we know T_E and T_C at least from there we can identify others. So, as we know T_E so we can calculate the evaporator side pressure to be equal to the saturation pressure corresponding to T_E and condenser side pressure to be equal to the saturation pressure corresponding to T_C.

$$P_E = P_{sat}(T_E)$$

$$P_C = P_{sat}(T_C)$$

Remember here the T_C refers to the temperature from this point onwards, not over this zone. Because temperature is changing from this 2, till the saturation dome, but inside the temperature remains constant. So, you now know the pressures and you can plot the cycles. As we know the pressure, so, what else we can identify? From the cycle now as we know the evaporate side condition, we can get the value of h_1 where h_1 is equal to hg corresponding to the value of T_E. Now if you do not have any information about the value of hf at this pressure, then we can choose certain reference. Let us extend this 4 to 1 line extend to liquid side up to a point 1'. So, 1' correspond to saturated liquid at the evaporation temperature and let us assume $h_f$corresponding to T_E to be equal 0 kJ/kg. And s_f again corresponding the same T_E to be equal to 0 kJ/kgK.

$$hf|_{TE} = 0 \text{ kJ/kg}$$

$$sf|_{TE} = 0 \text{ kJ/kg}$$

These are some different values we are choosing. Any choice of reference will not hamper your calculation because we are ultimately dealing with the changes in enthalpy and entropy. So, what will be your h_1?

$$h_1 = h_g(T_E) = (h_f + h_{fg})|_{TE}$$

and we have said $h_f$to be equal to zero. So, this will be equal to:

$$= h_{fg}|_{TE}$$

and similarly s1 which should be equal to:

$$s_1 = s_g(T_E)$$

in the same way it now becomes

$$= s_{fg}|_{TE}$$

So, we know theentropy and enthalpy at state point 1, at compressor inlet. So, now for compressor exit what we can do? We know

$$s_1 = s_2$$

for isentropic compression. Now as this process is being isentropic or as we are you talking about a reversible process, so for any reversible process we know that:

$$\delta Q = Tds$$

or

$$ds = \delta Q/T$$

From there let us see what we can write. For the compression process we know,

$$s_1 = s_2$$

Then if we write say,

$$h_3 - h_1'$$

that is, we are taking out this point and this point, then how we can approximate this? If the specific heat for the liquid phase remains constant this can be approximated as:

$$h_3 - h_1' = C_{p,liq}\,(T_3 - T_1')$$

that is

$$= C_{p,liq}\,(T_C - T_E)$$

assuming the specific to be constant and the liquid phase to be incompressible then this we can easily do this.

Similarly,

$$(s_3 - s_1') \approx C_{p,liq}\; ln\left(\frac{T_3}{T_{1'}}\right) = C_{p,liq}\; ln\left(\frac{T_C}{T_E}\right)$$

So, as we know the temperatures and as we have said that $h_1'$and s_1' to be equal to 0, so from there, we know the values of h_3 and s_3. Now, what we can do for the condensation process? For the condensation process, we know that,

$$\dot{Q}_C = \dot{m}\,(h_2 - h_3)$$

or

$$q_C = (h_2 - h_3)$$

Now,

$$(h_2 - h_3) = (h_2 - h_2') + (h_2' - h_3)$$

Now for point 3 we know already values and what will be $h_2' - h_3$? That is $h_2' - h_3 = h_{fg}|_{TC}$

Substituting this in the above equation it becomes:

$$= (h_2 - h_2') + h_{fg}|_{TC}$$

and what is $h_2 - h_2'$? Here we are talking about superheated vapour. So, if we assume superheated vapour and to have constant specific heat, then we can approximate this one as in terms of C_P as:

$$= C_{p,\,vap}\,(T_2 - T_2') + h_{fg}|_{TC}$$

And we know that $T_2'$is nothing but T_c so, the equation is now:

$$= C_{p,\,vap}\,(T_2 - T_C) + h_{fg}|_{TC}$$

So only thing that remains is the point T_2', temperature point 2'. Now, how we can get that? These are the conditions that we have to use for that. During the compression process if you assume the vapour to be an ideal gas, we can easily with this s_1 and s_2 in each other and from there we can identify the value of temperature T_2.

So, we can see with just some knowledge of these limited data points we can evaluate the performance by suitable reference. So, let us now she we can see one complete numerical example.

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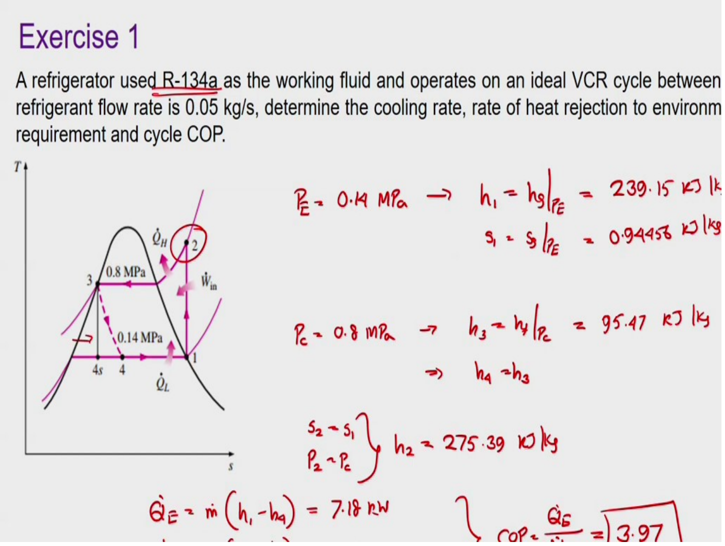

Exercise 1

A refrigerator used R-134a as the working fluid and operates on an ideal VCR cycle between
refrigerant flow rate is 0.05 kg/s, determine the cooling rate, rate of heat rejection to environm
requirement and cycle COP.

$P_E = 0.14$ MPa $\rightarrow$ $h_1 = h_g|_{P_E} = 239.15$ kJ/k

$s_1 = s_g|_{P_E} = 0.94456$ kJ/kg

$P_C = 0.8$ MPa $\rightarrow$ $h_3 = h_f|_{P_C} = 95.47$ kJ/kg

$\Rightarrow$ $h_4 = h_3$

$\left.\begin{array}{l} s_2 = s_1 \\ P_2 = P_C \end{array}\right\} h_2 = 275.39$ kJ/kg

$\dot{Q}_E = \dot{m}(h_1 - h_4) = 7.18$ kW

$COP = \frac{\dot{Q}_E}{} = \boxed{3.97}$

A very simple one where we are talking about a refrigeration system, which is using R134a as a refrigerant and is operating on an ideal vapour compression refrigeration cycle between a condenser and evaporator pressures which are given, refrigerator flow rate is given and we have to identify several things. So, this is the cycle, here we are using R134a as a working medium. So, we have to use the property table for that. And once we have the corresponding property table available, then we can easily for evaluate the performance. Like suppose your evaporator pressure

$P_E = 0.14$ MPa, the lower one.

then your h_1 will be equal to:

$$h_1 = h_g|_{PE} = 239.15 \text{ kJ/kg}$$

and

$$s_1 = s_g|_{PE} = 0.94456 \text{ kJ/kgK}$$

Now, look at the point number 3 we know,

$P_C = 0.8$ MPa

So,

$$h_3 = h_f|_{PC} = 95.47 \text{ kJ/kg}$$

and process 3 to 4 being throttling one we know that,

$$h_4 = h_3$$

the same value. So, only that we have to see this point number 2. So, we know that,

$$s_2 = s_1$$

and also

$$P_2 = P_C$$

Using this two for the superheated vapour table from there we can identify the value of h_2 to be 275. 39 kJ/kg.

We have already solved enough problems using water as a working medium for such kind of processes. So, you know how to get the value, you just need the R134a for the table. So, once you have all of them then we can easily calculate the required parameters. Like if you want to get the cooling rate:

$$\dot{Q}_E = \dot{m}\,(h_1 - h_4)$$

which is going to be in this case 7.18 KW.

Then the rate of heat rejection to environment which is:

$$\dot{Q}_C = \dot{m}\,(h_2 - h_3)$$

so, that will be coming as 9.0 kW. The compressor power requirement:

$$\dot{W}_C = \dot{m}(h_2 - h_1)$$

and this one also can be written as

$$= \dot{Q}_C - \dot{Q}_E$$

either you can calculate in both case you are going to get this one to be 1.81 kW. And if you combine this your *COP* which is:

$$COP = \frac{\dot{Q}_E}{\dot{W}_C} = 3.97$$

So, for every unit of compression work input you are going to get about 4 times refrigeration effect for this. So, and I am leaving one task to you instead of using a throttling valve, if you are using a turbine here, that is instead of having process 3 to 4 going by process 3 to 4*s*, where 4*s* refers to the exit state of the turbine. Then you have to repeat the calculation for all these parameters. The only information that you need in this case is the value of h_{4s}. So, the h_{4s} calculated as

$$s_3 = s_{4s}$$

So, the value of s_3 you can get from here and calculate the quality at point 4*s*. And from that quality you can get the h_{4s}, then you can repeat the calculation of all these parameters. Only thing I can tell you that the *COP* for this case will be about 5.07 which is about 25 to 30% increase in the final value if you are using a turbine instead of a throttling device but as I have mentioned practically that is very difficult.

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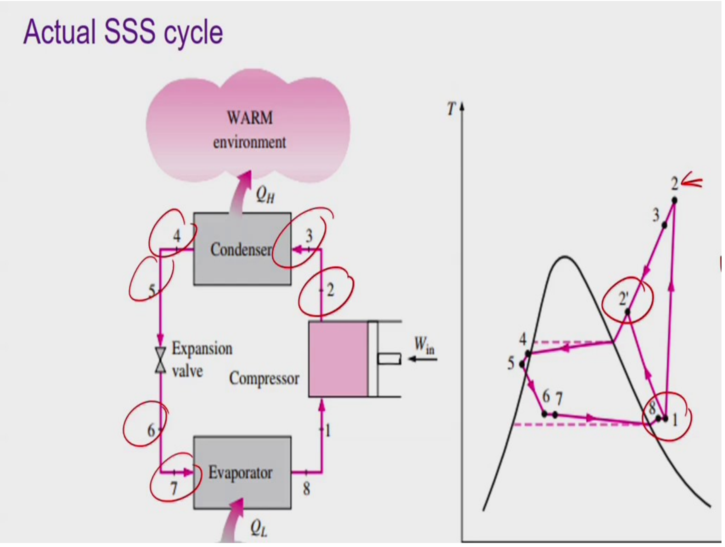
Actual SSS cycle

Now, in a real situation there can be several irregularities or irreversibilities that can be present in your SSS cycle just like shown here. Here, we can have several things like the compressor can have isentropic efficiencies and therefore instead of starting from point 1 if you have reaching point 2, we may reach somewhere here 2'. So, we need to define an isentropic efficiency for the compressor. So isentropic efficiency for the compressor is η_c.

In this case will be equal to the minimum possible work input required which is following the isentropic process. That is $h_2 - h_1$ divided by the actual working input requirement which requires $h_2' - h_1$. So here of course, we are talking on the pressure level to be the same and accordingly you can calculate the value of this h_2' or, locate the point 2'.

We can also a significant amount of pressure losses like if the refrigerant is moving from the compressor exit and it is reaching the condenser between the point 2 and 3, we may have some amount of pressure loss. Similarly, from condenser to the throttling valve from throttling valve exit to the evaporator, we can a significant amount of pressure losses. So, all these pressure losses need to be taken into account while analysing a real cycle.

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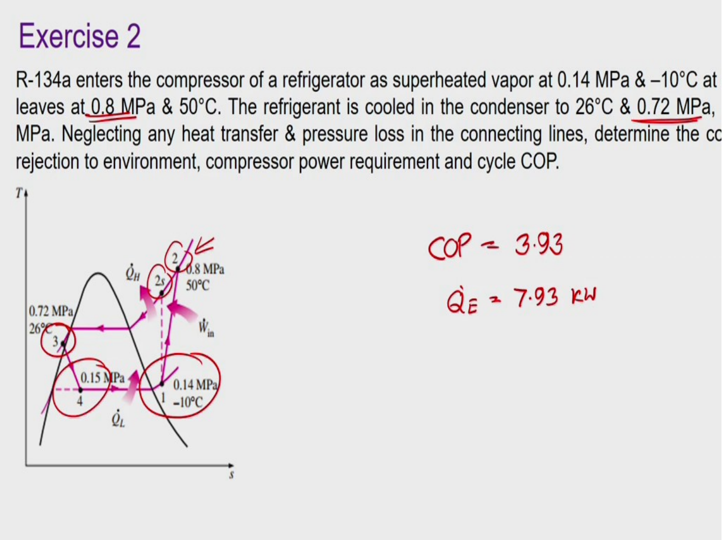

Exercise 2

R-134a enters the compressor of a refrigerator as superheated vapor at 0.14 MPa & –10°C at leaves at 0.8 MPa & 50°C. The refrigerant is cooled in the condenser to 26°C & 0.72 MPa, MPa. Neglecting any heat transfer & pressure loss in the connecting lines, determine the cc rejection to environment, compressor power requirement and cycle COP.

COP = 3.93

$\dot{Q}_E = 7.93$ kW

Just like the example I have here, I am talking about the same refrigerator that I used in the previous exercise. But here the superheated vapour, here my cycle is like this, where the superheated vapour at 0.14 MPa and −10 ^{0}C is entering the compressor. So, the compressor inlet our state is not saturated vapour rather superheated vapour. Quite often we want to do this in order to completely avoid any kind of droplet in the compressor.

And instead of reaching point 2*s* it is actually reaching point 2 when the conditions are 0.8 MPa in 50 ^{0}C. The refrigerant is cooled in the condenser to 26 ^{0}C and 0.72 MPa. Look at this, at the condenser inlet, pressure is the 0.8 MPa and outlet it is 0.72 MPa. So about 0.08 MPa close to 0.8 bar amount of pressure loss has taken place inside the condenser.

And also, the condenser exit condition is not of saturated liquid rather it is subcooled liquid. That is also sometimes you would like to extract some more amount of refrigeration effect from this. And it is throttled to 0.15 MPa which is this, so when it passes from the evaporator inlet to the evaporator exit there is a 0.01 MPa of pressure loss. Neglecting any heat transfer in pressure loss in the connecting pipes we have to determine the cooling rate, rate of heat rejection involved in it, compressor power requirement and the cycle *COP*. So, we need to get the values for all the state points 1, 2, 3 and 4, then the enthalpy from the table and then you can easily identify the required parameters and I am giving the final value. *COP* for this case will be equal to 3.93. So, *COP* has decrease marginally, but the $Q_{dot\ E}$ that is the evaporation load or I should say the cooling capacity that has also come to 7.93 kW. So, that is also changed compared the previous one. You please do this simple calculation to get this numbers on your own.

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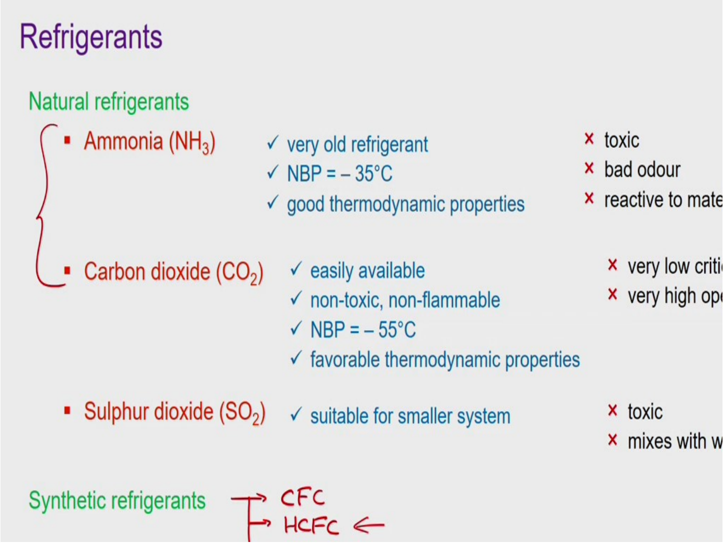

Refrigerants

Natural refrigerants

- Ammonia (NH_3)
  - ✓ very old refrigerant
  - ✓ NBP = – 35°C
  - ✓ good thermodynamic properties
  - × toxic
  - × bad odour
  - × reactive to mate
- Carbon dioxide (CO_2)
  - ✓ easily available
  - ✓ non-toxic, non-flammable
  - ✓ NBP = – 55°C
  - ✓ favorable thermodynamic properties
  - × very low criti
  - × very high op
- Sulphur dioxide (SO_2)
  - ✓ suitable for smaller system
  - × toxic
  - × mixes with w

Synthetic refrigerants → CFC

→ HCFC ←

Now, I would like to talk little bit about 10 minutes on the refrigerant that we commonly use in vapour compression refrigeration cycle. Refrigerants can broadly of three types: the first one is natural refrigerant. Natural refrigerants, there can be several ones of course, but I am mentioning about three of them one is ammonia. Ammonia is an age-old refrigerant one of the earliest refrigerants to be used and that is why it is quite well known. Its characteristic normal boiling point is very advantageous is $-35\ {}^{0}$C. We can easily go to quite low temperatures using ammonia is the working medium, and it has very good thermodynamic properties as well. Quite well known and quite different thermodynamic properties. So, as a refrigerant ammonia has these advantages. But it has its disadvantages also, like ammonia is toxic, ammonia has very bad odour. So, not very useful if it gets leaked to the surroundings and it is reactive, very important point is reactive to certain materials particularly to copper, which is used in the heat exchangers, in the condenser and evaporators.

So, if we are using ammonia as a refrigerant, we cannot use copper like material. Theoretically speaking, any kind of substance which can evaporate can act as a refrigerant. But there are certain refrigerants that we popularly keep on using historically so ammonia is the first one of them. Second is the very popular one for the modern times and also for near future that is carbon dioxide. Again, it is very easily available, it is non-toxic, non-flammable. So, there is no issue of toxicity like in ammonia.

Carbon dioxide has a normal boiling point of − 55 ^{0}C so capable of going to very low temperature and it has very favourable thermodynamic properties under certain situations even better than ammonia are much better than ammonia. But problem with carbon dioxide is its critical temperature is very low. Its critical temperature is only around the order of 32 ^{0}C. So, it can go through phase change process only below this temperature or well below this temperature.

So, in order to achieve phase change with carbon dioxide your condenser temperature also has to be much lower than this. Similarly, it has very high operating pressure because the triple point pressure for carbon dioxide is about 5 bar. So, we have to operate at a pressure level above this in order to have the liquid phase of the carbon dioxide.

Remember below the triple point pressure we cannot have the liquid phase. So, we have to work above this pressure level. But still carbon dioxide is quite useful and with recent modifications carbon dioxide is increasingly becoming more popular for large-scale affiliations. One issue with both ammonia and carbon dioxide are their latent heat of vaporization or enthalpy of vaporization is a quite high, therefore more suitable for large scale systems. Sulphur dioxide on the contrary is something that is most suitable for small scale system because it has a lesser latent heat of vaporization and therefore gives high mass flow rate. But sulphur dioxide is highly toxic so it has been used very less in refrigeration industry. We can also use water as one of the refrigerant, but water has not very favourable property as refrigerant. Particularly, you know that water freezes at 0 ^{0}C and therefore we cannot use water as refrigerant when the evaporator temperature is around 0 ^{0}C. Hydrocarbons are also been used a bit. Sulphur dioxide has another big disadvantage that when it mixes with water it can form sulphuric acid which is highly corrosive.

Next kind of refrigerants we have are synthetic refrigerants. Now, synthetic refrigerants you must have heard the names, there are three kinds of very popular names in use. One is CFC, CFC refers to chlorofluorocarbon. That is, there is chlorine and there is fluorine. We are talking about hydrocarbons only, but there the hydrogen atom has been replaced in a molecule by chlorine and fluorine atoms. Then we can also have HCFC, that is hydro chlorofluorocarbon where we have hydrogen also along with chlorine and fluorine in that hydrocarbon.

So, it is HCFC and finally we can have HFC, where we do not have chlorine at all, it is hydrofluorocarbon. Hydrofluorocarbon talks about the complete removal of chlorine from the molecule. It is only hydrogen and fluorine along with carbon. So, these are synthetic refrigerants, CFCs and HCFCs have been extensively used in domestic refrigerators, also in large-scale applications. But because of environmental issues, as we shall shortly be seeing, they are gradually being phased out in favour of HCFC and more in favour of HFCs. So, in future, we may not be having any refrigeration system running CFCs and HCFCs, but more on HFCs and other natural refrigerants. The idea is to remove chlorine based synthetic refrigerants and that is why CFCs and HCFCs are being phased out. We can also have mixtures, we can have azeotropic or non-azeotropic mixtures that are also used.

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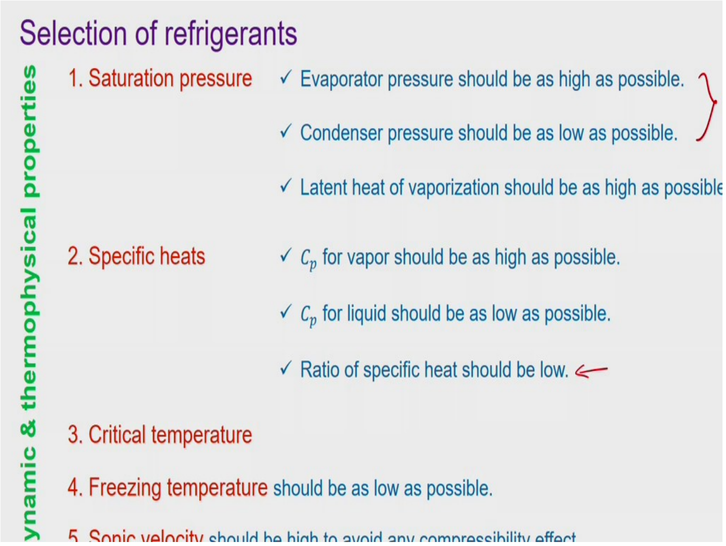

Selection of refrigerants

ynamic & thermophysical properties

1. Saturation pressure
   - ✓ Evaporator pressure should be as high as possible.
   - ✓ Condenser pressure should be as low as possible.
   - ✓ Latent heat of vaporization should be as high as possible
2. Specific heats
   - ✓ C_p for vapor should be as high as possible.
   - ✓ C_p for liquid should be as low as possible.
   - ✓ Ratio of specific heat should be low.
3. Critical temperature
4. Freezing temperature should be as low as possible.
5. Sonic velocity should be high to avoid any compressibility effect

Now, for selection of a refrigerant, there are several properties of the refrigerant to bother about. One kind of properties are thermodynamic and thermophysical properties, and there first is the saturation pressure levels. So, the evaporator pressure should be as high as possible. As we have seen the evaporation temperature has to be as high as possible. Correspondingly the evaporator pressure also has to be as high as possible because if the evaporator pressure is high close to atmospheric level then the problem of air leaking into the evaporator line that will be much lesser and leak detection also will be lower. And also, higher the pressure, lowering the specific volume of the corresponding gas, so we have deal with lesser volume of gas and so the size of the equipment will be smaller.

Then the condenser pressure should be as low as possible, just the opposite to this condenser pressure is always higher than evaporator pressure. But higher the pressure more thicker pipes that we need and therefore if the condenser pressure can be kept to some manageable level, we can use much thinner pipes, much thinner valves, heat exchangers and also it is much safer to operate. And another advantage is that, if the evaporate pressure and condenser pressure are close to each other then, the compression work requirement that will also keep on coming down leading to an increase in the efficiency.

Another point that is related to the saturation pressure that is the latent heat of vaporization should be as high as possible. That is very logical because, the higher the latent heat of vaporization, higher will be the amount of energy extracted during the phase change process, particularly at the evaporation side. So, we can have higher amount of cooling capacity. So, higher the latent heat of vaporization we can have higher amount of $Q_{dot\ E}$.

Second point is the specific heat, but before that I have to mention that these three points are not compatible with each other. Quite often we have to go for some kind of trade-off between that three to check which one is more meritorious compared to the other. For specific heat, the specific heat for the vapour should be as high as possible, so that you do not have to go for too much of super heating. Whereas specific for liquid should be as low as possible because in the condenser when we go for subcooling then we can go for significant temperature change for this liquid and the temperature change in throttling that becomes smaller.

Ratio of specific it should be low, because the lower the ratio of the specific heat, the less will be the compressor work requirement. So, this is also very interesting point to remember. Now the critical temperature, of course should be well above the maximum temperature at which you would like to work. Like for carbon dioxide as I have just mentioned, that for carbon dioxide critical temperature is so low that under normal atmospheric condition the temperature may be higher than the critical temperature. And therefore, carbon dioxide can't be used as a phase change material under normal atmospheric condition. So critical temperature has to be high. Then there are several smaller points like freezing temperature should be as low as possible. One big point with water, one big disadvantage with water has a freezing temperature which is quite close to what we need in evaporator and therefore water cannot commonly be used as refrigerant.

Then the sonic velocity should be high to avoid any kind of compressibility effect, high thermal conductivity to have higher degree of heat transfer in heat exchanger, low viscosity to avoid frictional losses or pressure drop in the pipelines etc.

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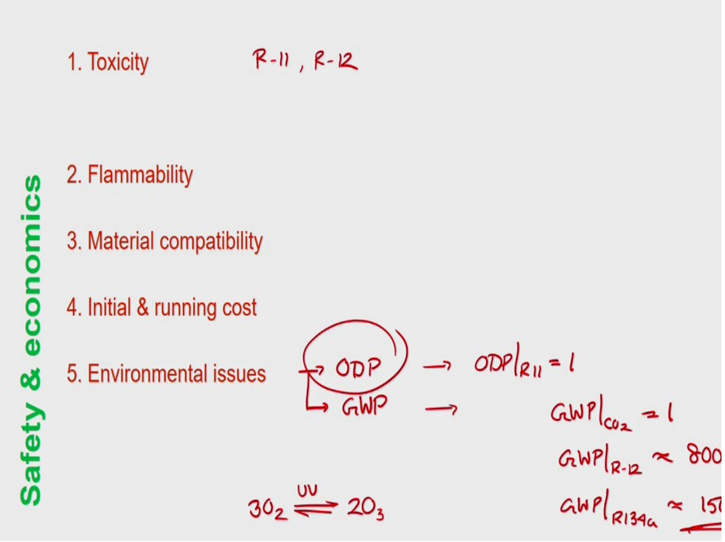

Safety & economics

1. Toxicity R-11 , R-12
2. Flammability
3. Material compatibility
4. Initial & running cost
5. Environmental issues

$\rightarrow$ ODP $\rightarrow$ $ODP|_{R11} = 1$

$\rightarrow$ GWP $\rightarrow$ $GWP|_{CO_2} = 1$

$GWP|_{R\text{-}12} \approx 800$

$GWP|_{R134a} \approx 15$

$3O_2 \xrightleftharpoons{UV} 2O_3$

Second group of properties are the safety and economic based properties. Here, that toxicity is very important. Natural refrigerants like ammonia and sulphur dioxide are highly toxic. Synthetic refrigerants like R-11 and R-12, they are non-toxic under normal condition, but when they come in close to some kind open flame, they produce highly toxic gases. So, you have to be careful while using them. Then flammability, refrigerants should be non-flammable at least within the working range it should not start to burn. The material compatibility is important, like we have talked about the compatibility or non-compatibility of ammonia with copper. Similarly the material that you are using for your pipeline should be compatible with your refrigerant. Initial and running cost should be manageable.

And finally the environmental issues. In terms of environmental issues. we talk about two parameters. One is ODP: Ozone Depletion Potential, other is GWP: Global Warming Potential. Both are highly talked about terms now a days. Here the global warming potential talks about the effect the refrigerants may have on the environment in increasing the temperature levels in environment. I am not going to the technical detail of how the GWP is calculated. But one information that you can have that, the GWP of carbon dioxide or, I should write the other way the GWP for carbon dioxide has been arbitrary set to value of 1 and based on which the other refrigerants are scaled. Certain refrigerants have significantly high of GWP compared to CO+ certain refrigerants may have a quite small. Like R-12, global warming potential of R-12 that is of the order of 8000. So, 8000 times higher global warming it will cause compared to carbon dioxide. Whereas, if we talk about Global Warming Potential and HFC which is the modern refrigerant R134a which is about 1500. So, though HFC does not have any kind of chlorine in this, so they do not have any effect on the ozone layer but that has significantly high global warming potential.

Now I go to the ODP, ODP refers to the Ozone Depletion Potential. For ODP, R-11 I should write the ozone depletion potential for R-11 is arbitrary set the value of 1. That is similar to GWP for carbon dioxide and everything is scaled based upon this. Certain refrigerants much higher ozone depletion potential, certain refrigerants quite less. Like that R134a that you are talking about which is HFC, its ozone depletion potential for this R134a is equal to an absolute zero. It does not have any effect on the ozone layer but can have a significant global warming potential. The idea of ozone deflection is that, in presence of UV rays the ozone can get converted to oxygen and this particular reaction gets quickened up by the presence of chlorine.

So, the kind of refrigerants which has chlorine in their molecular structure, when the refrigerants get released in the surroundings and when they are able to reach higher atmosphere, they generally decompose bleeding chlorine or open free chlorine in the surroundings. And that free chlorine can react with the ozone to get content converted back to oxygen thereby causing the depletion in the ozone concentration in the upper atmosphere. That is what we refer by this ozone depletion potential. Different refrigerants have different kind of ozone depletion potential.

And primary because of this consideration only, the CFCs and HCFCs which has chlorine in their molecule, i.e. chlorofluorocarbon and hydrochlorofluorocarbons has been or are being phased out. Whereas hydrofluorocarbon does not have chlorine in their molecular structure, so their ozone depletion potential is zero, but they still can have significant amount of effect on the global warming. So, these are the different parameters to be considered while selecting any refrigerant.

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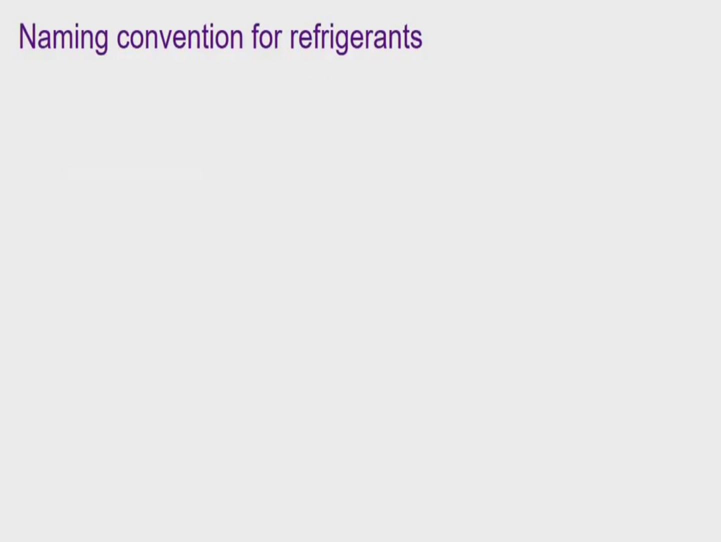

Naming convention for refrigerants

The next thing that we have to consider is the naming convention of the refrigerants, that is R-11, R-12, R-27 I am talking about. But today I have run out of time. I would like to stop here itself. In the next class I shall have talking about the naming convention and then I shall be moving to a few other aspects of vapour compression and subsequently the vapour absorption refrigeration system.

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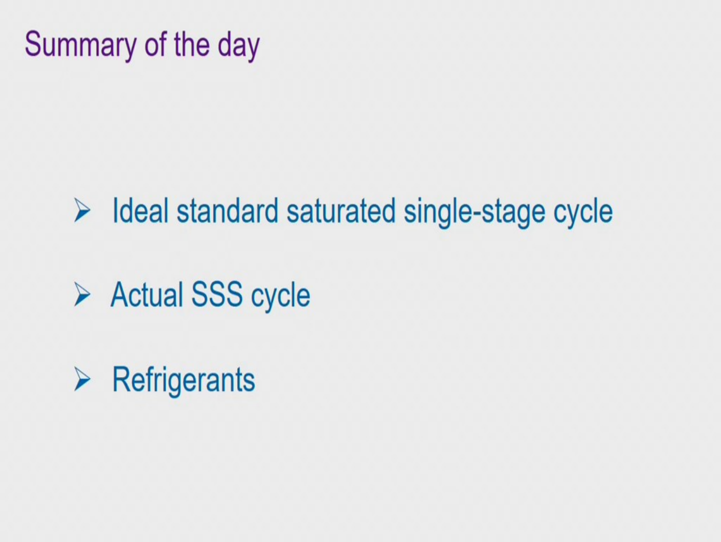

Summary of the day

- Ideal standard saturated single-stage cycle
- Actual SSS cycle
- Refrigerants

So, today we have talked about the ideal standard saturated single-stage cycle, we have seen how to analyse that using a complete set of data and also in limited data is it. Then we have talked about actual cycle, here we can have pressure losses, refrigerants etc. And finally, we have talked about the refrigerants, particularly the different categories of refrigerants and the selection of refrigerant. Naming convention of refrigerants is something that I wanted to talk about, but in the next class I shall be starting from that point onwards. So, till then just revise this lecture and if you have any query, write back to me. Thank you.