Applied Thermodynamics

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Module - 05

Refrigeration and Air-Conditioning System

Lecture - 38

Vapour Compression Refrigeration System – II

Dear learners, welcome to this course Applied Thermodynamics module 5 Refrigeration and Air Conditioning.

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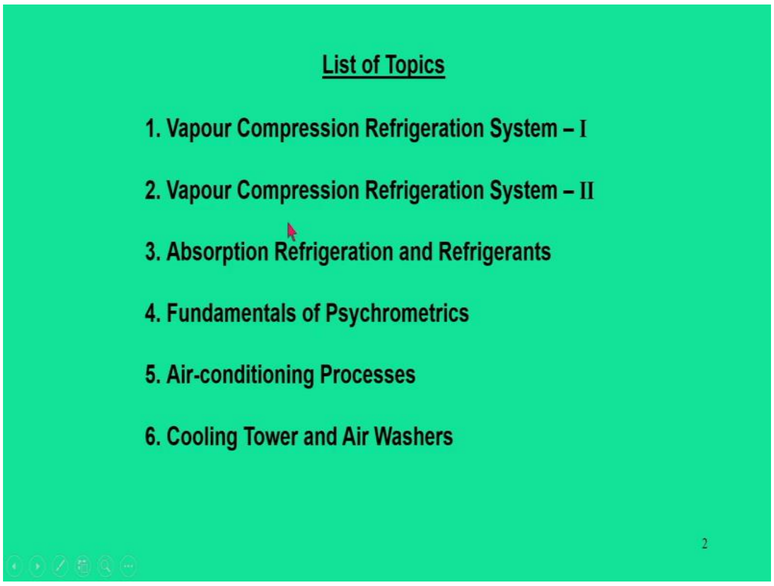

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So, as I discussed in the last class there are 6 lectures and first 2 lectures will be on vapour compression refrigeration systems. And you are in lecture number 2 that is Vapour Compression Refrigeration System part II.

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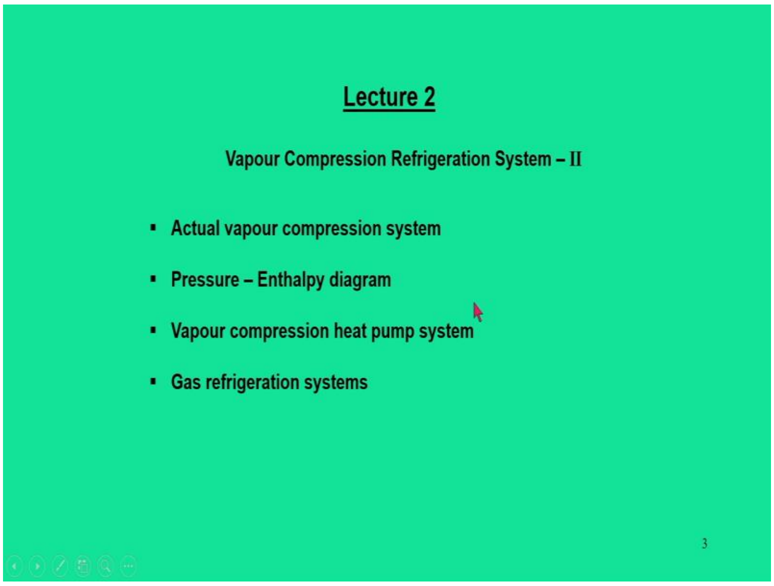

Lecture 2

Vapour Compression Refrigeration System – II

- Actual vapour compression system
- Pressure – Enthalpy diagram
- Vapour compression heat pump system
- Gas refrigeration systems

So, in this lecture our main intention would be to discuss about actual vapour compression systems. In our previous class we have analyzed a ideal vapour compression systems consisting of condenser, evaporator, compressor and a throttling valve and that was the ideal case of operations. Now, we will look into the actual vapour compression systems in which we will introduce irreversibilities into the components of the vapour compression system.

Apart from that we need to discuss something on pressure enthalpy diagram. This is similar to the idea of properties of pure substance. And here we are using a refrigerant and this refrigerant, if you consider it as a pure substance we can draw its thermodynamic diagrams. And one such diagram which is used in a refrigeration systems more commonly is known as a pressure enthalpy diagrams.

Apart from that there are other diagram also temperature-entropy and these are very commonly used in a refrigeration systems. This is similar to the fact that when you use otto cycle, PV diagram and Ts diagrams are very common, even for diesel cycle, for Brayton cycle, PV and Ts diagrams are very common. But here the common diagrams are pressure enthalpy and temperature entropy diagrams.

Apart from this we will also discussed about vapour compression heat pump systems. So, all these things we talk about vapour compression refrigeration systems. Our next segment would be vapour compression heat pump systems in which it is similar to a refrigeration system, but desired effect is little bit different.

And towards the end of the lecture we will discussed about gas refrigeration systems in which working fluid is taken as air and it operates on a reverse Brayton cycle. This is the entire summary for this lectures. So, let me start with the ideal vapour compression systems which we stopped in the last class.

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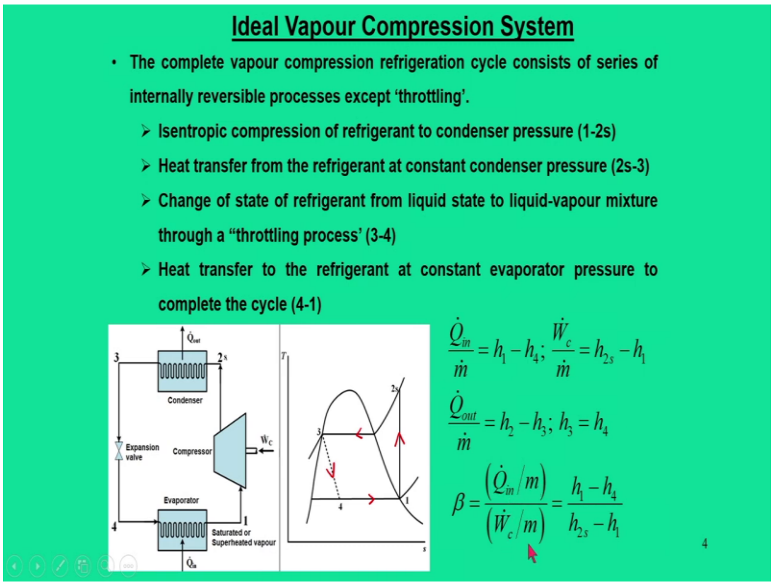

Ideal Vapour Compression System

- The complete vapour compression refrigeration cycle consists of series of internally reversible processes except 'throttling'.
  - Isentropic compression of refrigerant to condenser pressure (1-2s)
  - Heat transfer from the refrigerant at constant condenser pressure (2s-3)
  - Change of state of refrigerant from liquid state to liquid-vapour mixture through a "throttling process' (3-4)
  - Heat transfer to the refrigerant at constant evaporator pressure to complete the cycle (4-1)

$$\frac{\dot{Q}_{in}}{\dot{m}} = h_1 - h_4; \quad \frac{\dot{W}_c}{\dot{m}} = h_{2s} - h_1$$

$$\frac{\dot{Q}_{out}}{\dot{m}} = h_2 - h_3; \quad h_3 = h_4$$

$$\beta = \frac{\left(\dot{Q}_{in}/m\right)}{\left(\dot{W}_c/m\right)} = \frac{h_1 - h_4}{h_{2s} - h_1}$$

So, a complete vapour compression refrigeration cycle consist of series of internally reversible processes except throttling. So, these components are evaporator in which heat is added from this space to be cooled and in which refrigerant changes its phase from liquid state to saturated or super heated vapour states.

Then it expands in the compressors where work is being fed by external agency. And after compression the pressure of refrigerant rises to condenser pressure and then and the when the refrigerant enters to the condenser it releases heat to the ambient, as a result it again changes its phase from vapour to liquid.

Now, it already changes its phase, but still it is at condenser pressure we have to reduce this pressure of this refrigerant back to the evaporator pressure. So, we require an expansion valve, or simply we call this as a throttling valve. Then it comes back from state 3 to state 4 and again the cycle continues. The thermodynamic diagrams in T s plot refers 1- 2s compression process, 2s to 3 condenser process, 3 to 4 throttling process, 4 to 1 is evaporator process.

And for this we can calculate cooling load from in the evaporator that is $\frac{\dot{Q}_{in}}{\dot{m}} = h_1 - h_4$, compressor load $\frac{\dot{W}_c}{\dot{m}} = h_{2s} - h_1$. Condenser load, $\frac{\dot{Q}_{out}}{\dot{m}} = h_2 - h_3$ and in a throttling process it is $h_3 = h_4$.

And finally, we can calculate the COP that is beta which is nothing but $\beta = \frac{\left(\dot{Q}_{in}/m\right)}{\left(\dot{W}_c/m\right)} = \frac{h_1 - h_4}{h_{2s} - h_1}$.

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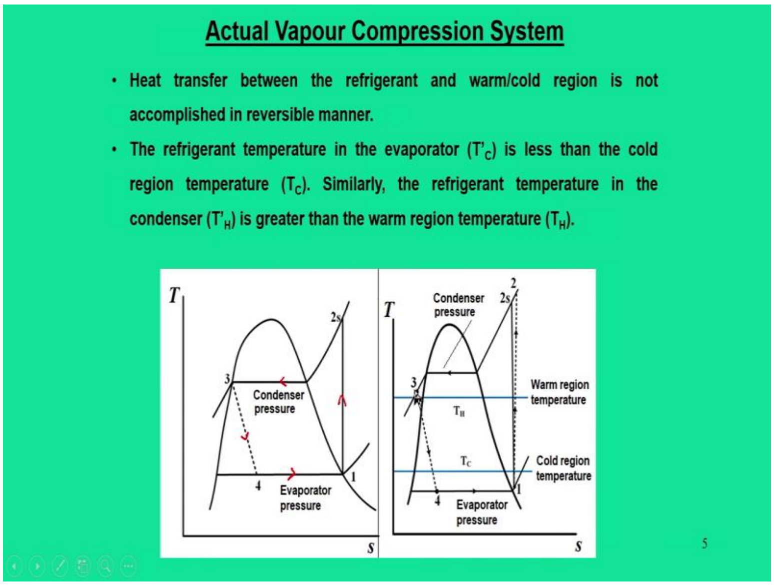

Actual Vapour Compression System

- Heat transfer between the refrigerant and warm/cold region is not accomplished in reversible manner.
- The refrigerant temperature in the evaporator (T'$_C$) is less than the cold region temperature (T$_C$). Similarly, the refrigerant temperature in the condenser (T'$_H$) is greater than the warm region temperature (T$_H$).

Now, we will relax this ideality and try to introduce some kind of irreversibility into the systems. So, that is what we say it is a actual vapour compression systems. In actual compression system I mentioned the first thing it differs is that your condenser pressure as well as temperature is higher than that of the warm region temperatures.

So, that means, condenser temperature is higher than the warm region temperature and evaporator temperature is lower than the cold region temperatures, then only the vapour compression system can work efficiently. So, in a ideal system process 1- 2s is isentropic process, 2-3 is a constant pressure process, throttling process 3 to 4 and finally, 4 to 1 is the evaporator process.

But if you see this particular diagram here when you refer the Ts diagram for an actual process we can say there is a dotted line here it seems that we have introduced the efficiency for the compressors. And we call this as an isentropic efficiency of the compressor.

The second other important aspects of this diagram is that in a ideal diagram, the refrigerant has to leave at state 3 from the condenser and this state 3 is the saturated liquid state. But it may so happen that we can extend it further, the refrigerant can come back to sub cooled regions where in actual diagram it is represented in the state 3 here.

So, in this way that refrigerant can be a sub cooled situations. So, this is basically done just to enhance the refrigerating effect because the point 4 will be accordingly located on this diagram and, evaporator load also increases. Then another important aspect is that when the refrigerant leaves from the evaporator, in a ideal system it is exactly at saturated vapour state1, but in a actual situation the point 1 may lie in a close to super heated regions.

So, these are the some features that actual vapour compression system differs from an ideal system.

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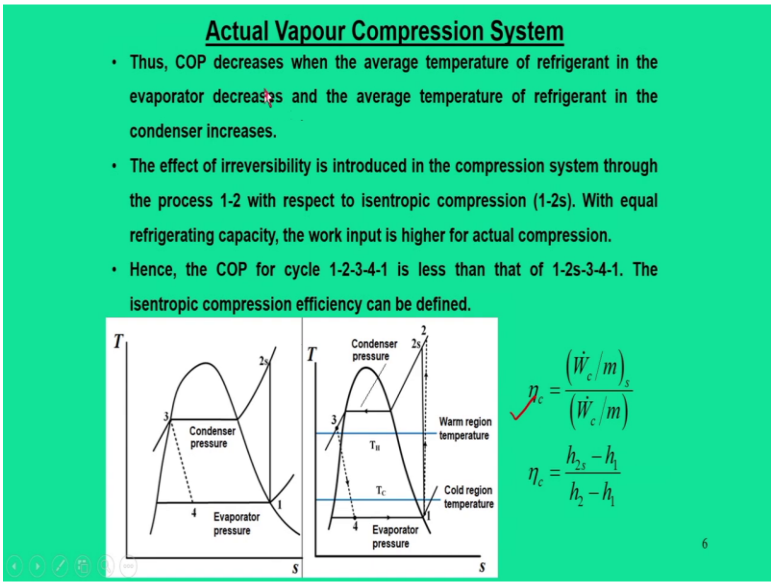

Actual Vapour Compression System

- Thus, COP decreases when the average temperature of refrigerant in the evaporator decreases and the average temperature of refrigerant in the condenser increases.
- The effect of irreversibility is introduced in the compression system through the process 1-2 with respect to isentropic compression (1-2s). With equal refrigerating capacity, the work input is higher for actual compression.
- Hence, the COP for cycle 1-2-3-4-1 is less than that of 1-2s-3-4-1. The isentropic compression efficiency can be defined.

$$\eta_c = \frac{\left(\dot{W}_c/m\right)_s}{\left(\dot{W}_c/m\right)}$$

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

And based on that one can introduce here is the isentropic efficiency of the compressor. That is isentropic work which is fed by compression system to the ideal work that is $\eta_c = \frac{\left(\dot{W}_c/m\right)_s}{\left(\dot{W}_c/m\right)}$.

So, in a sense that the isentropic efficiency of the compressor can be represented in the form of enthalpies $\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$. And as you see in this diagram; obviously, the denominator number is higher because point 2 lies above 2s. So, this is how we say the isentropic efficiency of the compressor is introduced in an actual systems.

And moreover, in this process of introducing re irreversibilities, the COP decreases, because the average temperature of refrigerant in the evaporator decreases and average temperature of refrigerant in the condenser increases. And both these effects ultimately leads to the decrease in the COP.

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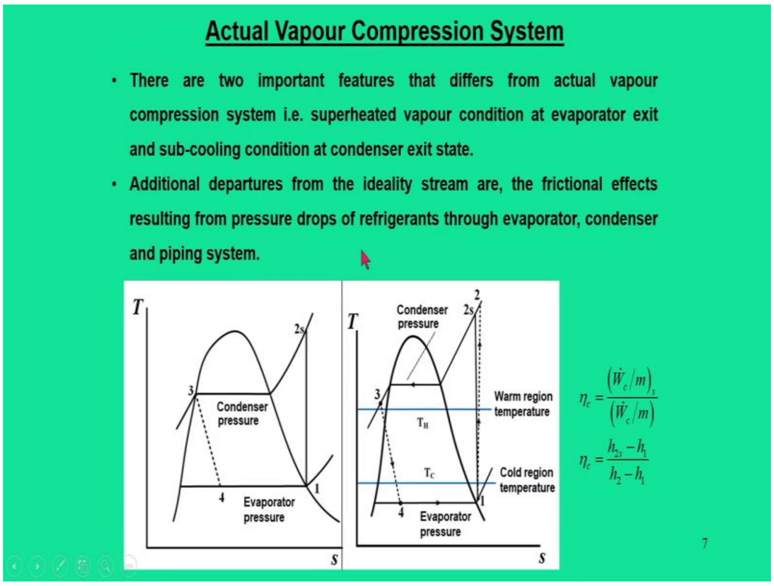

Actual Vapour Compression System

- There are two important features that differs from actual vapour compression system i.e. superheated vapour condition at evaporator exit and sub-cooling condition at condenser exit state.
- Additional departures from the ideality stream are, the frictional effects resulting from pressure drops of refrigerants through evaporator, condenser and piping system.

$$\eta_c = \frac{\left(\dot{W}_c/m\right)_s}{\left(\dot{W}_c/m\right)}$$

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1}$$

And this also I have explained the two other important aspects that is super heated vapour condition at the evaporator exit and sub cooling condition at the condenser exit states. Apart from this there are additional departures from the ideality of stream are, frictional effects resulting from pressure drop of refrigerant through the evaporators, condenser and the piping systems.

Of course, in our calculation we normally do not take into account for solving the problems unless and until it is very elaborately specified. Now, we will move to the thermodynamic property evaluations.

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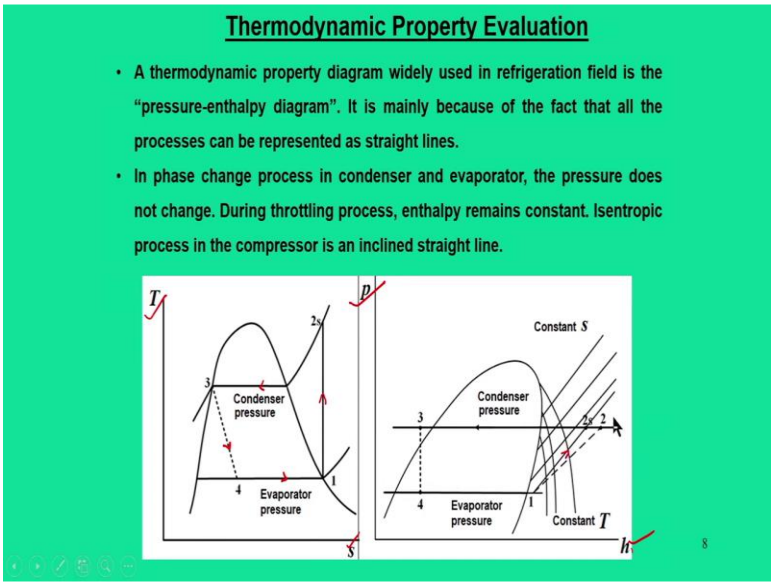

Thermodynamic Property Evaluation

- A thermodynamic property diagram widely used in refrigeration field is the "pressure-enthalpy diagram". It is mainly because of the fact that all the processes can be represented as straight lines.
- In phase change process in condenser and evaporator, the pressure does not change. During throttling process, enthalpy remains constant. Isentropic process in the compressor is an inclined straight line.

So, as I mentioned in our earlier analysis that whether it is a ideal vapour compression system or actual vapour compression systems, in order to know the performance of each component like condenser, evaporator, throttling valve, compressor, it requires the evaluation of properties values at state points.

And in particular the properties values that are important to us are enthalpy and entropy. And of course, condenser pressure or condenser temperature, evaporator pressure and evaporator temperatures. Considering these aspects there are two important property diagrams that are explained here, one is temperature entropy diagram which is also very common in all our thermodynamic study.

Apart from that another diagram that becomes very important for a refrigeration system that is pressure enthalpy diagrams. So, we will try to say that what is the significance or why these diagrams are more important. First thing we know that since we are talking about temperatures of cold region and hot region or warm region then we must have one axis which talks about temperature.

So, that is what we can say we say it is a temperature in one of the axis. Other part is side by side when you talk about temperatures, we also have talked about the pressure because it is the property of refrigerant. When you deal with states saturated liquids, liquid vapour mixture, saturated vapour then super heated vapour and to cater all these state points we require information for both pressure and temperatures that is what in one of the axis the pressure and temperature comes into pictures. And other aspect is that now when you deal with the thermodynamic processes the other important property that becomes vital is entropy.

Because compression process first thing we say its isentropic process and also in a sense that when you talk about throttling. So, enthalpy becomes a significant properties where enthalpy does not change. So, if you take all those parameter in their respective axis; that means, then it becomes easy for us to locate the state points.

So, that is the reason these two phase, temperature entropy and pressure enthalpy are taken for discussions. So, if you look at the temperature entropy diagrams 1-2s is isentropic compression, 2s to 3 is constant pressure heat rejection in a condenser. 3 to 4 is a constant enthalpy process, 4 to 1 is evaporator process or constant temperature heat rejections.

Now, if you look within the dome we can say irrespective of the whether you take Ts diagram or ph diagram, this line is a straight line within the dome. And also if you look at temperature and entropy diagrams constant pressure line also falls in the same straight line. And even if you look within the dome of pressure enthalpy diagram also, here evaporator temperature and pressure are straight lines.

So, it is easy to locate those properties so; that means, whether you talk about pressure or temperature, based on the property table from saturated state, one can locate the properties, that is one aspect.

Other aspect is that if you look at temperature entropy diagrams, the process 3-4 is a constant enthalpy process, but interestingly it is a inclined straight line. And; that means, in a temperature entropy diagram, this enthalpy lines is a inclined straight line. And of course, it is a irreversible process, but it is a inclined straight line and 1-2s is isentropic process, so it is a vertical straight line.

Now, if you come back to pressure enthalpy diagram, the process 1 to 2s which was supposed to be a vertical straight line now it becomes an inclined straight line as you can see in a ph diagram. And of course, 2s to 3 it is a constant pressure process, so you can draw a straight line from 2 s to 3.

And 3 to 4 again it is a constant enthalpy process and from 4 to 1 it is a evaporation process which is again a constant pressure process. So, if you look at this diagram most of the lines are almost straight line and because of this reason, these two diagrams are normally preferred ok.

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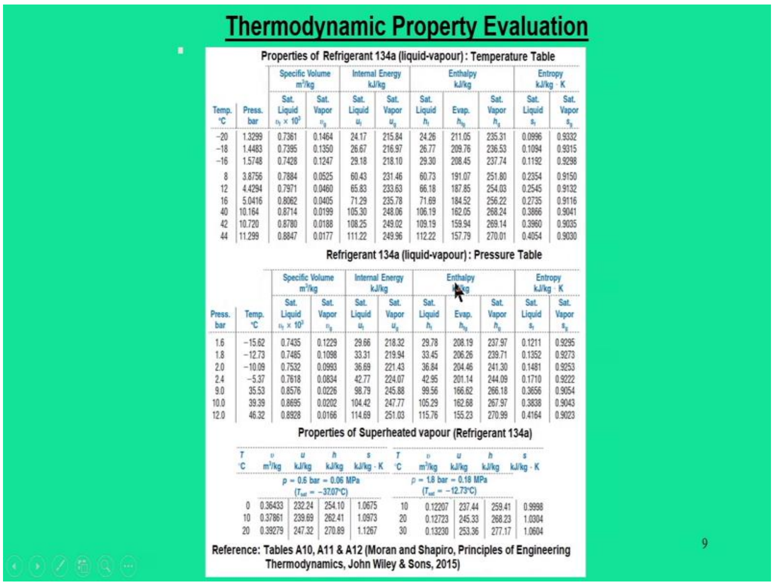

Thermodynamic Property Evaluation

Properties of Refrigerant 134a (liquid-vapour) : Temperature Table

		Specific Volume m³/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy kJ/kg · K	
Temp. °C	Press. bar	Sat. Liquid $v_f \times 10^3$	Sat. Vapor v_g	Sat. Liquid u_f	Sat. Vapor u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Sat. Vapor s_g
−20	1.3299	0.7361	0.1464	24.17	215.84	24.26	211.05	235.31	0.0996	0.9332
−18	1.4483	0.7395	0.1350	26.67	216.97	26.77	209.76	236.53	0.1094	0.9315
−16	1.5748	0.7428	0.1247	29.18	218.10	29.30	208.45	237.74	0.1192	0.9298
8	3.8756	0.7884	0.0525	60.43	231.46	60.73	191.07	251.80	0.2354	0.9150
12	4.4294	0.7971	0.0460	65.83	233.63	66.18	187.85	254.03	0.2545	0.9132
16	5.0416	0.8062	0.0405	71.29	235.78	71.69	184.52	256.22	0.2735	0.9116
40	10.164	0.8714	0.0199	105.30	248.06	106.19	162.05	268.24	0.3866	0.9041
42	10.720	0.8780	0.0188	108.25	249.02	109.19	159.94	269.14	0.3960	0.9035
44	11.299	0.8847	0.0177	111.22	249.96	112.22	157.79	270.01	0.4054	0.9030

Refrigerant 134a (liquid-vapour) : Pressure Table

		Specific Volume m³/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy kJ/kg · K	
Press. bar	Temp. °C	Sat. Liquid $v_f \times 10^3$	Sat. Vapor v_g	Sat. Liquid u_f	Sat. Vapor u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Sat. Vapor s_g
1.6	−15.62	0.7435	0.1229	29.66	218.32	29.78	208.19	237.97	0.1211	0.9295
1.8	−12.73	0.7485	0.1098	33.31	219.94	33.45	206.26	239.71	0.1352	0.9273
2.0	−10.09	0.7532	0.0993	36.69	221.43	36.84	204.46	241.30	0.1481	0.9253
2.4	−5.37	0.7618	0.0834	42.77	224.07	42.95	201.14	244.09	0.1710	0.9222
9.0	35.53	0.8576	0.0226	98.79	245.88	99.56	166.62	266.18	0.3656	0.9054
10.0	39.39	0.8695	0.0202	104.42	247.77	105.29	162.68	267.97	0.3838	0.9043
12.0	46.32	0.8928	0.0166	114.69	251.03	115.76	155.23	270.99	0.4164	0.9023

Properties of Superheated vapour (Refrigerant 134a)

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg · K	T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg · K
p = 0.6 bar = 0.06 MPa (T_{sat} = −37.07°C)					p = 1.8 bar = 0.18 MPa (T_{sat} = −12.73°C)				
0	0.36433	232.24	254.10	1.0675	10	0.12207	237.44	259.41	0.9998
10	0.37861	239.69	262.41	1.0973	20	0.12723	245.33	268.23	1.0304
20	0.39279	247.32	270.89	1.1267	30	0.13230	253.36	277.17	1.0604

Reference: Tables A10, A11 & A12 (Moran and Shapiro, Principles of Engineering Thermodynamics, John Wiley & Sons, 2015)

And here also there are there is another way that we can evaluate the thermodynamic property table for the refrigerants. So, normally when you deal with the steam tables, the water was taken as the working fluid. And in fact, it is used in the steam power plants to locate the thermodynamic properties.

So, in that water steam property evaluation using steam tables, we have properties based on temperature table, properties based on pressure table that was done with reference to water, but here the working fluid is refrigerants. So, let us talk only one particular refrigerant and this particular refrigerant is a very common nowadays that is refrigerant 134a.

And for that refrigerant we can have the similar property tables which are available in the books.

So, these tables are exhaustive tables that gives properties in three sequences; one is liquid vapour table based on the temperature data. That means, if you know the temperature; corresponding saturation pressure is known. And accordingly all thermodynamic properties like specific volume, internal energy, enthalpy and entropies can be calculated.

The other variance for this is if you know the pressure range like if you know condenser pressure or evaporator pressure that then you can use this pressure table. And then it from the pressure table corresponding saturation temperature is known. So, now, from the pressure table, exactly in a similar manner we can calculate specific volume, internal energy, enthalpy and entropy.

So, this is how the properties are evaluated within the dome. Within the dome I mean, the saturated liquid, saturated vapour or we can say some cases we say it is a dryness fraction. And apart from that when you go to the super heated region, now in our thermodynamic states our super heated region comes in the state 2s; when the refrigerant comes out of the compression systems.

So, for those regions we can also use the super heated vapour. And for that we require two parameters or two data one is the pressure data, one of them is either specific volume, internal energy or enthalpy or entropy. But here since our entropy remains constant during the compression process, so the apart from pressure or temperature we also require entropy.

So, from pressure entropy we can evaluate the thermodynamic states of the refrigerant in the point 2s. So, this is how the property is going to be studied the either in the form of Ts diagram p h diagrams and in the way of determining the properties using the tables. And similarly, this table is done for refrigerant 134 a.

But the similar property tables can also be possible for other refrigerants. Other refrigerant could be R 11, R 22, carbon dioxides; for all of them the property tables are available in the book. But from our course we will basically stick to one particular refrigerant and for that refrigerant we are going to use the property values using that table.

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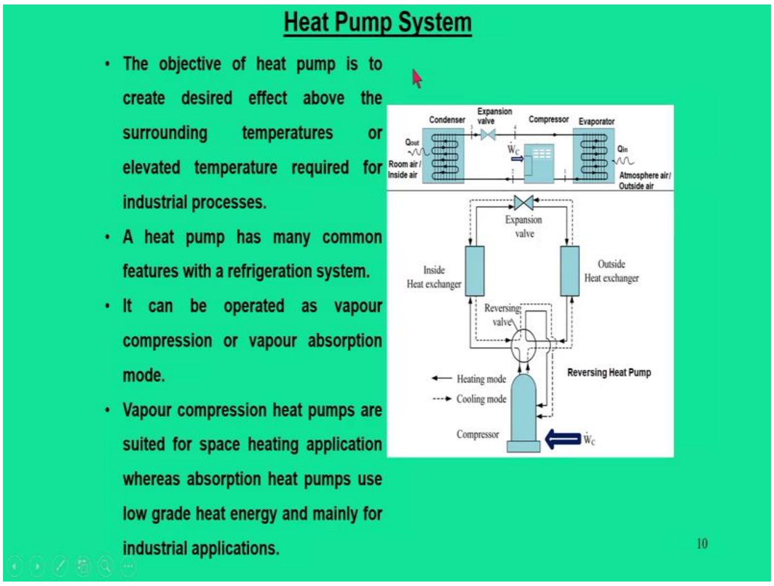

Heat Pump System

- The objective of heat pump is to create desired effect above the surrounding temperatures or elevated temperature required for industrial processes.
- A heat pump has many common features with a refrigeration system.
- It can be operated as vapour compression or vapour absorption mode.
- Vapour compression heat pumps are suited for space heating application whereas absorption heat pumps use low grade heat energy and mainly for industrial applications.

Now, we will move to the heat pump systems. So, the heat pump systems are normally operate in a opposite manner as that of refrigeration system; by opposite manner I mean that, your desired effect comes from the condenser side not evaporator side. So; that means, the heat load that comes out of the condenser is getting used by some kind of apartments or rooms which is needs to be heated.

So, that is the main intention of this. So, if you look at this particular diagram, almost it operates in a same manner as is done, but the desired effect is different. So, the objective of the heat pump is to create desired effect above the surrounding temperature or elevated temperature required for industrial processes.

And it has many common features with the refrigeration systems and more importantly the heat pump systems can be operated either in a vapour compression mode or also it can be used as a vapour absorption mode. And in later part we will see that how it can be used as a absorption mode. And in most application you will see that when you deal with the heat pump systems normally they are integrated with vapour absorption systems.

Because we can tap the low grad energy for its usage in this heat pump applications. But whereas, mostly when you talk about refrigeration and air conditioning plants they are based on the vapour compression mode. And normally vapour compression heat pumps are suited for space heating applications whereas, absorption heat pumps use lower grade energy mainly used for industrial applications.

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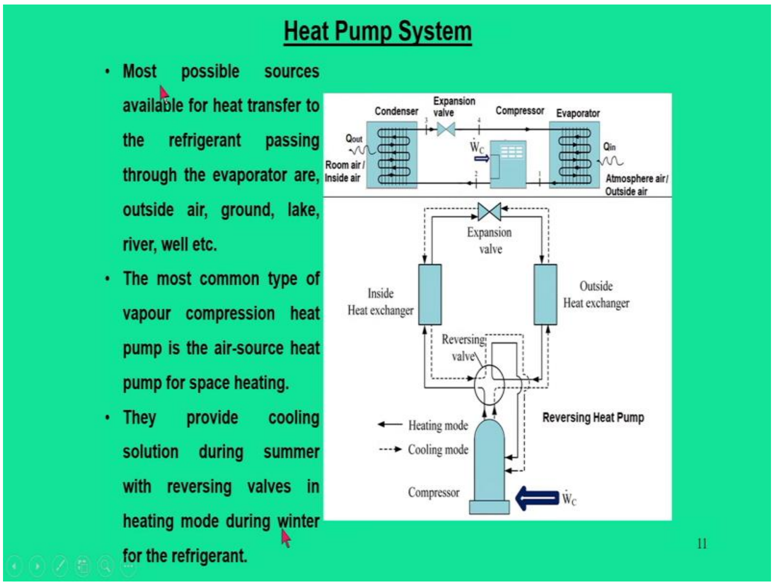

Heat Pump System

- Most possible sources available for heat transfer to the refrigerant passing through the evaporator are, outside air, ground, lake, river, well etc.
- The most common type of vapour compression heat pump is the air-source heat pump for space heating.
- They provide cooling solution during summer with reversing valves in heating mode during winter for the refrigerant.

The most possible sources of heat transfer to the refrigerant passing through the evaporator are outside air, ground, lake, river and well. Normally, in this evaporator we talk about atmospheric air or outside air and there are other ways that we can take out from the evaporator side. We can also used we can ground, lake, river, well; all these things are possible sources of heat transfer.

The other type of heat pump systems which are mainly common is shown here we call this as a reversing heat pump mode. When you deal with a industrial mode of heat pump, we also have to see that it can cater the need of heating as well as the cooling. So, whenever it is required we can switch it to a heating mode and when it is required which you can switch it to a cooling mode.

So, for that things the components are designed in a such a way that there are some kind of valves which operate in a reverse manner, so we call as a reversing valve. So, by reversing valve I mean, we can control or regulate the direction of flow of refrigerant within the components. And so, in a heating mode the refrigerant will follow in one sequence and their sequence will be reversed when you go for a cooling mode.

So, based on this it can the cater the need of both heating as well as cooling applications.

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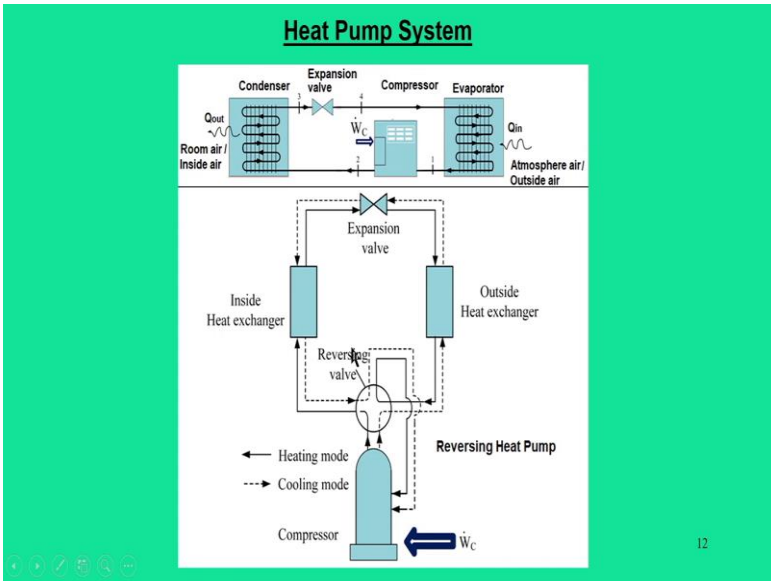

Heat Pump System

So, this is how this a complete heat pump systems is defined either it can be a space heating as it is in the first figure or it can be a heating as well as cooling mode by using a reversing heat pump. But we are basically interested in a heat pump systems and our main intention is that the desired effect that comes out is mainly on space heating.

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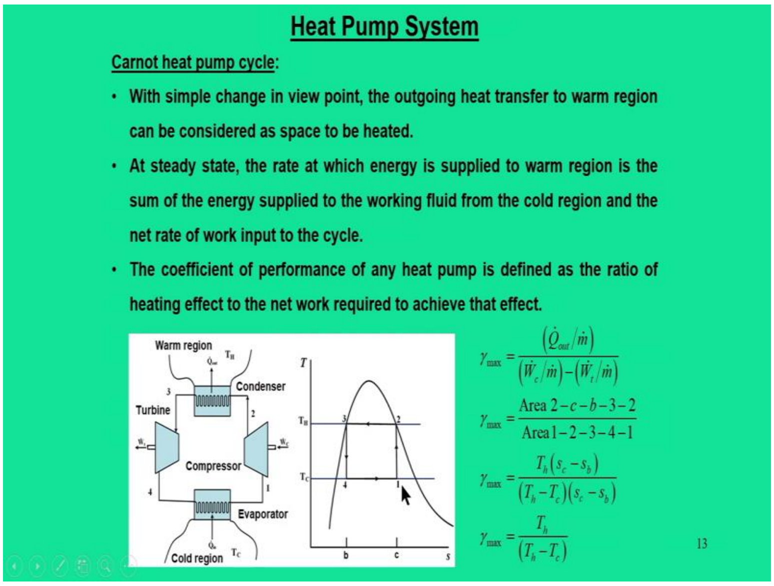

Heat Pump System

Carnot heat pump cycle:

- With simple change in view point, the outgoing heat transfer to warm region can be considered as space to be heated.
- At steady state, the rate at which energy is supplied to warm region is the sum of the energy supplied to the working fluid from the cold region and the net rate of work input to the cycle.
- The coefficient of performance of any heat pump is defined as the ratio of heating effect to the net work required to achieve that effect.

$$\gamma_{\max} = \frac{\left(\dot{Q}_{out}/\dot{m}\right)}{\left(\dot{W}_c/\dot{m}\right)-\left(\dot{W}_t/\dot{m}\right)}$$

$$\gamma_{\max} = \frac{\text{Area } 2-c-b-3-2}{\text{Area } 1-2-3-4-1}$$

$$\gamma_{\max} = \frac{T_h\left(s_c-s_b\right)}{\left(T_h-T_c\right)\left(s_c-s_b\right)}$$

$$\gamma_{\max} = \frac{T_h}{\left(T_h-T_c\right)}$$

And this is how for the heat pump basic theory that comes pops up and it comes from the basic Carnot heat pump cycles where we can draw again this Ts diagram. And from this

Ts diagram and with the components involving evaporator, compressor, turbine and the heat pump operating between warm region T_h and T_c one can calculate the COP. And here this COP is represented just to make a notation change that is you can say it is a γ.

And when I say heat pump your desired effect changes that is here it becomes Q out; that means, heat rejected to the warm region and in the denominator the your compression work remains same $\gamma_{\max} = \dfrac{\left(\dot{Q}_{out}/\dot{m}\right)}{\left(\dot{W}_c/\dot{m}\right)-\left(\dot{W}_t/\dot{m}\right)}$. Now, by locating the states point 1, 2, 3, 4 one can calculate the area on the diagrams and from this area diagrams we can calculate the maximum COP.

$$\gamma_{\max} = \frac{\text{Area } 2-c-b-3-2}{\text{Area} 1-2-3-4-1}$$

$$\gamma_{\max} = \frac{T_h\left(s_a - s_b\right)}{\left(T_h - T_c\right)\left(s_a - s_b\right)} \quad .$$

$$\gamma_{\max} = \frac{T_h}{\left(T_h - T_c\right)}$$

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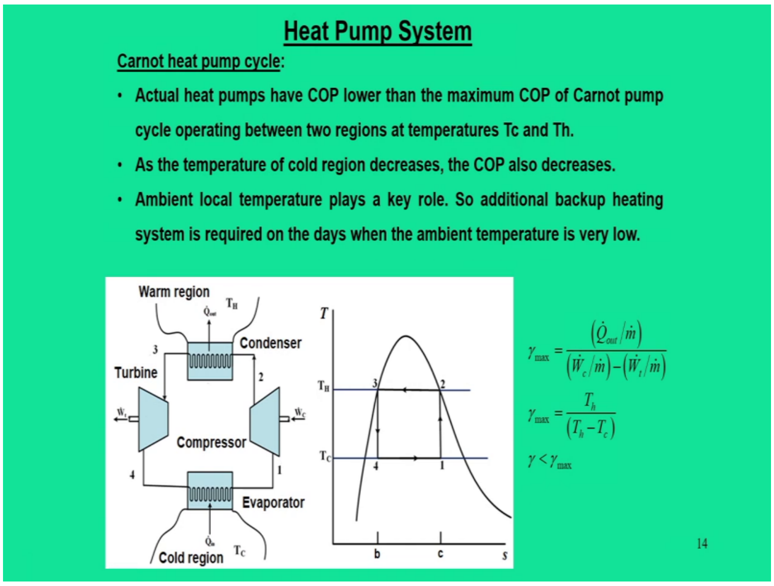

Heat Pump System

Carnot heat pump cycle:

- Actual heat pumps have COP lower than the maximum COP of Carnot pump cycle operating between two regions at temperatures Tc and Th.
- As the temperature of cold region decreases, the COP also decreases.
- Ambient local temperature plays a key role. So additional backup heating system is required on the days when the ambient temperature is very low.

$$\gamma_{\max} = \frac{\left(\dot{Q}_{out}/\dot{m}\right)}{\left(\dot{W}_c/\dot{m}\right)-\left(\dot{W}_t/\dot{m}\right)}$$

$$\gamma_{\max} = \frac{T_h}{\left(T_h - T_c\right)}$$

$$\gamma < \gamma_{\max}$$

And we cannot operate a Carnot heat pump because actual heat pumps have COP lower than the maximum COP. So, we have to switch over to a realistic versions, so we, come back again to a vapour compression mode heat pump systems. And this is done in a similar fashion as you did it for vapour compression refrigeration systems. And there we say the COP becomes gamma and this gamma is of course, less than gamma max that is Carnot COP ($\gamma < \gamma_{\max}$).

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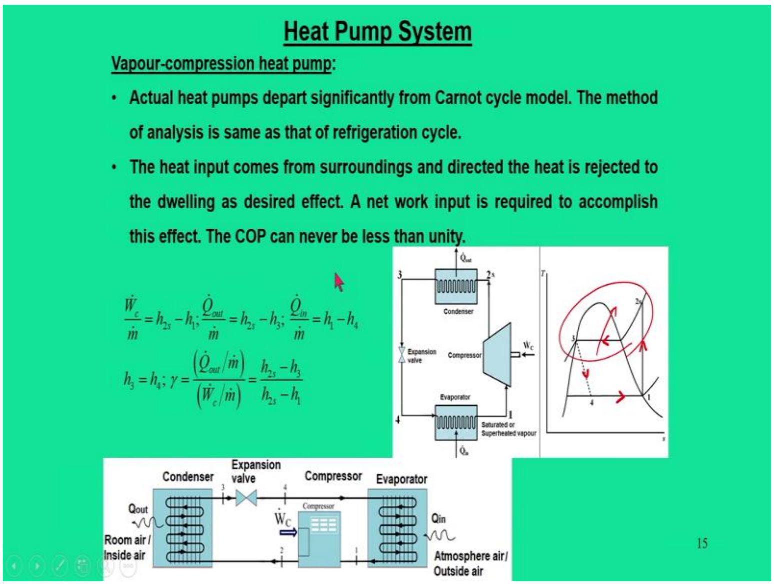

Heat Pump System

Vapour-compression heat pump:

- Actual heat pumps depart significantly from Carnot cycle model. The method of analysis is same as that of refrigeration cycle.
- The heat input comes from surroundings and directed the heat is rejected to the dwelling as desired effect. A net work input is required to accomplish this effect. The COP can never be less than unity.

$$\frac{\dot{W}_c}{\dot{m}} = h_{2s} - h_1; \frac{\dot{Q}_{out}}{\dot{m}} = h_{2s} - h_3; \frac{\dot{Q}_{in}}{\dot{m}} = h_1 - h_4$$

$$h_3 = h_4; \gamma = \frac{\left(\dot{Q}_{out}/\dot{m}\right)}{\left(\dot{W}_c/\dot{m}\right)} = \frac{h_{2s} - h_3}{h_{2s} - h_1}$$

And based on that, the vapour compression heat pumps are evolved. So, this is a this is the circuit diagram for a vapour compression heat pump systems involving condenser, expansion valve, compressor and evaporator. And moreover, the Ts diagram also represented as 1 to 2s compression process, 2s to 3 condenser process, 3 to 4 throttling process and 4 to 1 is the evaporator process. But the role of evaporator is not felt here rather this particular part of heat that goes out of from the condenser becomes very vital as a desired effects.

So, if you see here the expressions that are of interest to us is $\frac{\dot{W}_c}{\dot{m}} = h_{2s} - h_1$ and $\frac{\dot{Q}_{out}}{\dot{m}} = h_{2s} - h_3$. And this is the very vital part because that heat is used for space heating. Of course, when that heat is getting utilized, we have to also to calculate evaporator load as well which is $\frac{\dot{Q}_{in}}{\dot{m}} = h_1 - h_4$.

Again throttling process $h_3 = h_4$. And finally, COP for heat pump that is equal to $\gamma = \dfrac{\left(\dot{Q}_{out}/\dot{m}\right)}{\left(\dot{W}_c/\dot{m}\right)} = \dfrac{h_{2s} - h_3}{h_{2s} - h_1}$. So, here the net work input required to accomplish this effect comes from the electrical heating load. And here we can see that the COP can never be less than unity for a heat pump. So, we also mentioned that COP of heat pump is COP of refrigerator plus 1.

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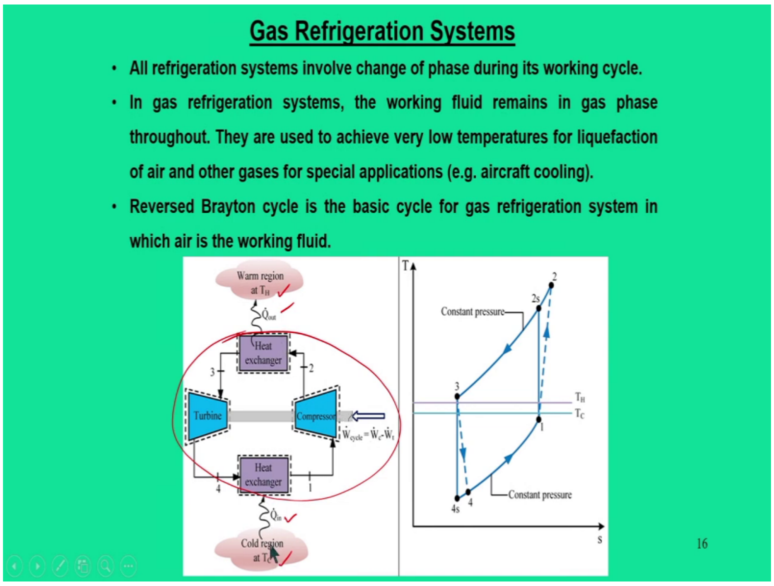

Gas Refrigeration Systems

- All refrigeration systems involve change of phase during its working cycle.
- In gas refrigeration systems, the working fluid remains in gas phase throughout. They are used to achieve very low temperatures for liquefaction of air and other gases for special applications (e.g. aircraft cooling).
- Reversed Brayton cycle is the basic cycle for gas refrigeration system in which air is the working fluid.

The next segment in our discussion is a gas refrigeration systems. So, here the main important aspects of a vapour compression systems or vapour absorption system with a gas refrigeration system is that here you are using gas as a working fluid in the refrigerations.

And in particular if gas is air, then we can say air is used as a cooling medium or in the working fluid in this thermal circuit. And here main assumption is that air does not change its the phase, but temperature of air can be lower by using a gas refrigeration cycles. And through this process we can achieve the refrigerating effects.

And we have to recall a Brayton cycle in a gas turbine engines when using it for power production systems, there you used to have a turbines. And our main intention is that that point of time it was power output from the turbine. But here we will operate that Brayton cycle in a reverse mode and we call this as a reverse Brayton cycle.

And it is considered as the basic gas refrigeration cycle with air as a working fluid. So, here the term that we are going to use of course, turbine, compressor, integrated systems, the air expands in the turbine is going to be used to drive this compressor. And side by side there are two exchangers involved and here you will not use the word evaporator or condenser because there is no change in the phase.

That is the reason we use the word heat exchanger. And of course, this particular cycle, we can say it is a reversed heat engine operating between the cold region T_c and warm region T_h where it takes some kind of work input into the systems as a results it is possible to extract heat from the cold region and send it back to warm regions.

And again when you deal with the Brayton cycle, the common diagram that is dealt it was the temperature entropy diagram. And if you look at this thermodynamic processes of each components involved in a gas refrigeration systems we can see, in a compression is ideally an isentropic process 1 to 2s. If isentropic process is relaxed a bit, so we can introduce some efficiency or introduce irreversibility, the process goes from 1 to 2.

So, you call this as isentropic efficiency for the compressors. And 2 to 3 process is a constant pressure process and it is a heat rejection in the heat exchanger. And this heat exchanger is integrated with the warm region, where heat is released from air to this warm region.

Now, the air that comes after this heat exchanger, it expands in a turbine in which power that comes out. Now, this power is typically used to drive the compressors. Now, 3 to 4 is a again expansion in the turbine process. We use the word turbine in a gas refrigeration systems but in a vapour compression system you use the word expansion valve that is the basic difference we must take a note of it. So, the process 3 to 4 ideally it has to be an isentropic process. So, 3 to 4s is as to is in isentropic process, but if you introduce efficiency in the turbine systems we can say it is an isentropic efficiency for the turbine.

So, actual process goes from 3 to 4. Now, again from 4 to 1, it is the second heat exchanger that is getting utilized by taking heat from the cold regions and from process goes from the 4 to 1.

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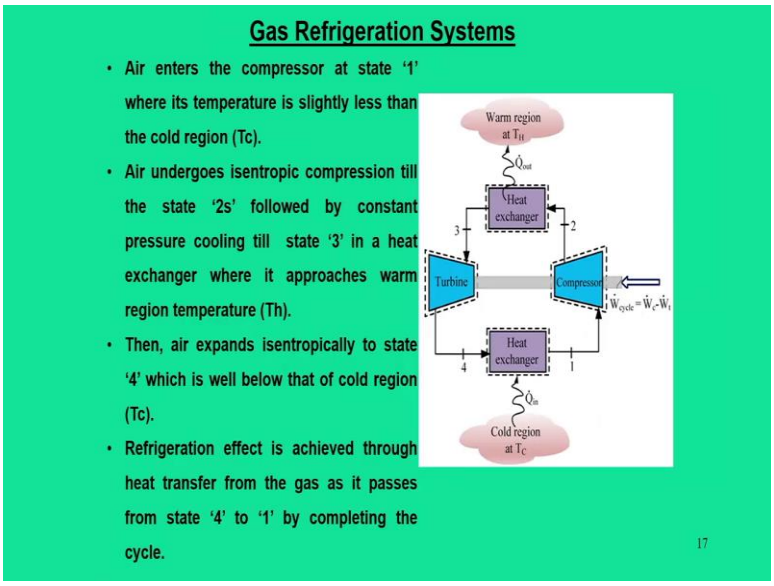

Gas Refrigeration Systems

- Air enters the compressor at state '1' where its temperature is slightly less than the cold region (Tc).
- Air undergoes isentropic compression till the state '2s' followed by constant pressure cooling till state '3' in a heat exchanger where it approaches warm region temperature (Th).
- Then, air expands isentropically to state '4' which is well below that of cold region (Tc).
- Refrigeration effect is achieved through heat transfer from the gas as it passes from state '4' to '1' by completing the cycle.

So, this is how a gas cycle system operates. So, I have elaborated it air enters the compressor as state 1 temperature is slightly less than the cold region, air undergoes isentropic compression till the state 2s is reached.

And from 2 to 3 it goes into an heat exchanger where it approaches the warm region temperatures, air expands isentropically to state 4 in which it is well below the temperature of cold region. And refrigeration effect is achieved through the heat transfer from the gas as it passes from 4 to 1 by completing the cycle in a second heat exchanger.

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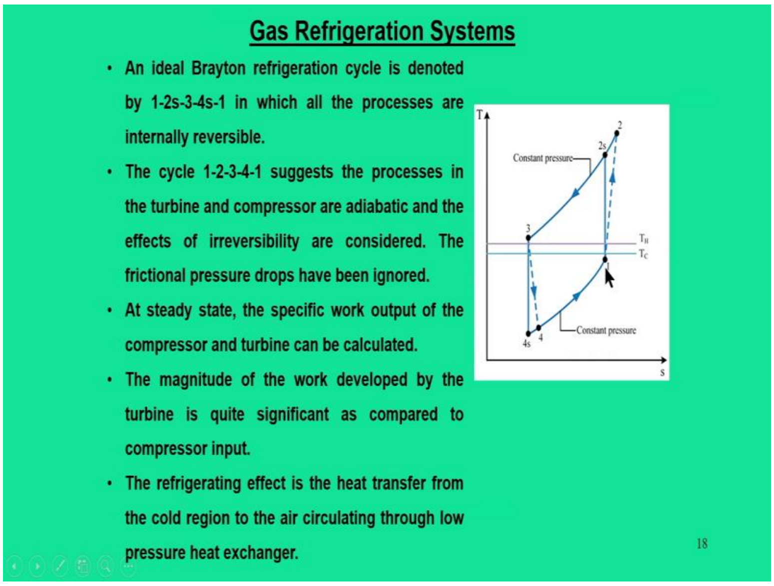

Gas Refrigeration Systems

- An ideal Brayton refrigeration cycle is denoted by 1-2s-3-4s-1 in which all the processes are internally reversible.
- The cycle 1-2-3-4-1 suggests the processes in the turbine and compressor are adiabatic and the effects of irreversibility are considered. The frictional pressure drops have been ignored.
- At steady state, the specific work output of the compressor and turbine can be calculated.
- The magnitude of the work developed by the turbine is quite significant as compared to compressor input.
- The refrigerating effect is the heat transfer from the cold region to the air circulating through low pressure heat exchanger.

So, if you look at the actual cycle, actual cycle is 1, 2, 3, 4 and if you say ideal cycle, it is 1, 2s, 3, 4s. And some important features are that at steady state specific work output from the compressor and turbine can be calculated. One can also calculate the magnitude of work developed by the turbine is quite significant and it is as compared to the compressor input. That is what the turbine work input is getting utilized to drive the compressors.

And the refrigerating effect is the heat transfer from the cold region to the air circulating through the low pressure at heat exchanger.

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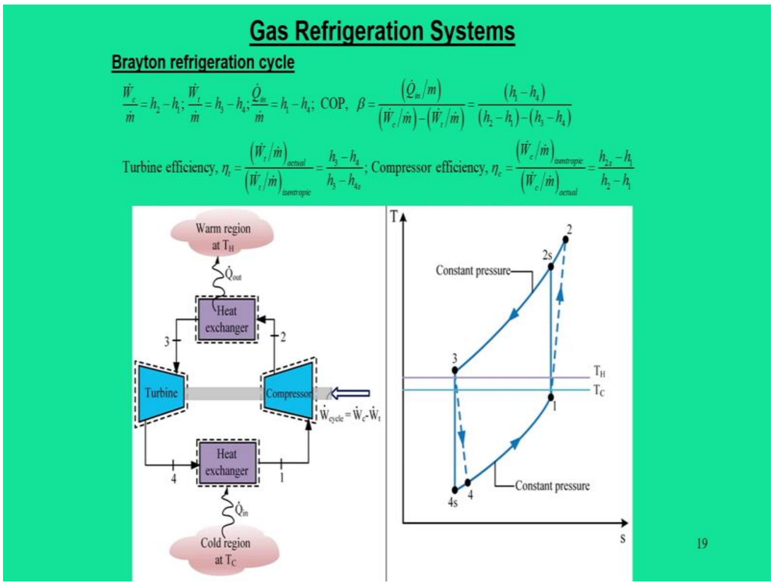

Gas Refrigeration Systems

Brayton refrigeration cycle

$$\frac{\dot{W}_c}{\dot{m}} = h_2 - h_1;\ \frac{\dot{W}_t}{\dot{m}} = h_3 - h_4;\ \frac{\dot{Q}_{in}}{\dot{m}} = h_1 - h_4;\ \text{COP},\ \beta = \frac{\left(\dot{Q}_{in}/m\right)}{\left(\dot{W}_c/\dot{m}\right) - \left(\dot{W}_t/\dot{m}\right)} = \frac{\left(h_1 - h_4\right)}{\left(h_2 - h_1\right) - \left(h_3 - h_4\right)}$$

$$\text{Turbine efficiency, } \eta_t = \frac{\left(\dot{W}_t/\dot{m}\right)_{actual}}{\left(\dot{W}_t/\dot{m}\right)_{isentropic}} = \frac{h_3 - h_4}{h_3 - h_{4s}};\ \text{Compressor efficiency, } \eta_c = \frac{\left(\dot{W}_c/\dot{m}\right)_{isentropic}}{\left(\dot{W}_c/\dot{m}\right)_{actual}} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

One important thing here I just want to point out that when you talk about air cycle refrigerations; of course, gives a desired refrigeration or cooling effect. It is very difficult to cater the need the sub cooled region temperature below 0°C using a gas cycle refrigerations.

So, it is normally required when the temperature in the cold region is not significantly low. Now, using the concept of Brayton refrigeration cycles one can calculate the compression work per unit mass of air in the form of enthalpy $\frac{\dot{W}_c}{\dot{m}} = h_2 - h_1$. Then turbine work is $\frac{\dot{W}_t}{\dot{m}} = h_3 - h_4$, heat input that is $\frac{\dot{Q}_{in}}{\dot{m}} = h_1 - h_4$ and your COP again we can calculate from the coordinates of enthalpies $\beta = \frac{(h_1 - h_4)}{(h_2 - h_1) - (h_3 - h_4)}$. And apart from that while talking about gas refrigeration we also talk about the efficiency of the components and here mainly isentropic efficiency of turbine and the compressors. And based on the state point calculations we can use the values based on their isentropic values as well as the actual values.

So, for instance if you take about turbine efficiency it is the actual turbine work to the isentropic work. And from this diagram we can actually calculate isentropic efficiency of turbine is $\eta_t = \frac{(\dot{W}_t / \dot{m})_{actual}}{(\dot{W}_t / \dot{m})_{isentropic}} = \frac{h_3 - h_4}{h_3 - h_{4s}}$. And for is compressors the isentropic efficiency of the compressor is $\eta_c = \frac{(\dot{W}_c / \dot{m})_{isentropic}}{(\dot{W}_c / \dot{m})_{actual}} = \frac{h_{2s} - h_1}{h_2 - h_1}$.

And moreover, another point I need to emphasize, since it is a air, we can use the simple equations like enthalpies can be written as c_pT for air. That means, instead of enthalpy it is also possible to evaluate the properties by knowing the temperatures.

So, of course, we all know that for air cycle refrigerations, the cardinal temperatures of each points in the cycle has to be noted first. So, once you note the temperature then when you multiply c_p value of air then we get the enthalpies. So, likewise this gas refrigeration cycle can be analyzed.

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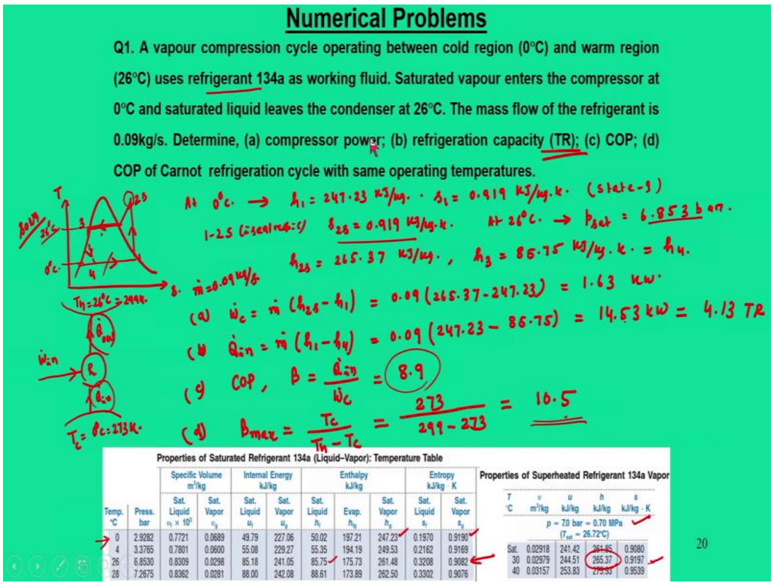

Numerical Problems

Q1. A vapour compression cycle operating between cold region (0°C) and warm region (26°C) uses refrigerant 134a as working fluid. Saturated vapour enters the compressor at 0°C and saturated liquid leaves the condenser at 26°C. The mass flow of the refrigerant is 0.09kg/s. Determine, (a) compressor power; (b) refrigeration capacity (TR); (c) COP; (d) COP of Carnot refrigeration cycle with same operating temperatures.

At 0°C. → $h_1 = 247.23$ kJ/kg. $s_1 = 0.919$ kJ/kg.K. (state-1)

1-2s (isentropic) $s_{2s} = 0.919$ kJ/kg.K. At 26°C. → $p_{sat} = 6.853$ bar.

$h_{2s} = 265.37$ kJ/kg., $h_3 = 85.75$ kJ/kg.K. $= h_4$.

$\dot{m} = 0.09$ kg/s

(a) $\dot{W}_c = \dot{m}(h_{2s} - h_1) = 0.09(265.37 - 247.23) = 1.63$ kW.

(b) $\dot{Q}_{in} = \dot{m}(h_1 - h_4) = 0.09(247.23 - 86.75) = 14.53\ kW = 4.13\ TR$

(c) COP, $\beta = \frac{\dot{Q}_{in}}{\dot{W}_c} = \boxed{8.9}$

(d) $\beta_{max} = \frac{T_c}{T_h - T_c} = \frac{273}{299 - 273} = 10.5$

Properties of Saturated Refrigerant 134a (Liquid–Vapor): Temperature Table

Temp. °C	Press. bar	Specific Volume m³/kg Sat. Liquid $v_f \times 10^3$	Sat. Vapor v_g	Internal Energy kJ/kg Sat. Liquid u_f	Sat. Vapor u_g	Enthalpy kJ/kg Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Entropy kJ/kg · K Sat. Liquid s_f	Sat. Vapor s_g
0	2.9282	0.7721	0.0689	49.79	227.06	50.02	197.21	247.23 ✓	0.1970	0.9190 ✓
4	3.3765	0.7801	0.0600	55.08	229.27	55.35	194.19	249.53	0.2162	0.9169
26	6.8530	0.8309	0.0298	85.18	241.05	85.75 ✓	175.73	261.48	0.3208	0.9082
28	7.2675	0.8362	0.0281	88.00	242.08	88.61	173.89	262.50	0.3302	0.9076

Properties of Superheated Refrigerant 134a Vapor

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg · K
$p = 7.0$ bar = 0.70 MPa ($T_{sat} = 26.72$°C) ✓				
Sat.	0.02918	241.42	261.85	0.9080
30	0.02979	244.51	265.37	0.9197 ✓
40	0.03157	253.83	275.93	0.9539

20

Now, we will try to solve some numerical problems based on the topics that we covered today. The first problem is based on a vapour compression cycle that operates a cold region that is 0°C and warm region 26°C and it uses refrigerant 134a as the working fluid. That means, temperature of cold region and warm region is given. Saturated vapour enters the compressor at 0°C; that means, your cold region temperature as well as the refrigerant they are at same temperature.

And saturated liquid leaves the condenser at 26°C, again at the condenser side also they are same and also other parameter is the mass flow of refrigerant is 0.09 kg/s and we have to calculate compressor power, refrigeration capacity, COP and of course, Carnot COP with same operating temperatures.

Now, first thing I need to emphasize that here we need the property data for refrigerant 134 a. So, that is what I have taken the extracts of the data from the book. And these extracts are nothing but the properties values that are required for us to solve the problems.

So, for example, in some situations you also require the super heated data when you deal with the compressor. So, that is the reason we have to use the data table from the book. Now, to solve the problem, the first thing that we need to do is that we have to draw the temperature entropy diagram.

For this we can first prepare a dome; so, in the dome we can say there are two straight lines which are horizontal and these are nothing, but your constant pressure lines. And, but outside the dome this constant pressure lines becomes in this fashion. So, we can locate the state points as 1, 2s, 3, 4; 1-2s is compressor, 2s to 3 is condenser, 3 to 4 is throttling, 4 to 1 is evaporator.

And the data that is given if you draw this line it is 0°C for evaporator and it is 26°C for condenser. And if you draw actually its circuit, so basically it is a refrigerator and the warm region temperature is T_h 26°C =299 K and in the cold region we have 0 °C that is 273 K and for this we also require W_{in}.

So, we take Q_{in} and you have Q_{out} and here Q_{out} is nothing but your condenser load, Q_{in} is nothing but your evaporator load. So, these things we are given with us, now to solve the problems you have to use this property table.

So, first thing to you, how to use the property table? So, at 0°C and for this refrigerant 134a if you a look at here, so, the point 1 is it saturated vapour. So, for saturated vapour we require enthalpy values h_g s_g. So, we can write as h_1 as 247.23 kJ/kg, s_1 as 0.919 kJ/kgK. Now, process 1 to 2s is isentropic and 2s is in the super heated region. So, for this region s_{2s} is same as s_1 that is 0.919 kJ/kgK.

So; that means, at 26°C one can find out what is the saturated pressure. So, we refer here, we can say saturated pressure is 6.853 bar. So basically, to calculate state 2s we have pressure 6.853 bar and entropy 0.919 kg kJ/kgK.

So, for that we have taken these super heated value close to a value of 7 bar and entropy value point 0.919. So, this gives the enthalpy value is h_{2s} is equal to 265.37 kJ/kg. And for state point 3, it is 26°C and enthalpy values at this point will be saturated liquid state.

So, 26°C and saturated liquid state, h_3 is 85.75 kJ/kgK. And this is also equal to h_4 because 3 to 4 throttling process. So, through this process we all know enthalpy at every point and we also know mass flow rate is equal to 0.09 kg/s.

$$\frac{\dot{W}_c}{\dot{m}} = \dot{m}\left(h_{2s} - h_1\right) = 0.09\left(265.37 - 247.33\right) = 1.63\text{kW}$$

$$\dot{Q}_{in} = \dot{m}\left(h_1 - h_4\right) = 0.09\left(247.33 - 86.75\right) = 14.53\text{kW} = 4.13\text{TR}$$

$$\beta = \frac{\left(\dot{Q}_{in}\right)}{\left(\dot{W}_c\right)} = 8.9$$

$$\beta_{\max} = \frac{T_c}{T_h - T_c} = \frac{273}{299 - 273} = 10.5$$

So, your maximum COP is 10.5, but actual compression system that as COP is 8.9, so this is how it is solved.

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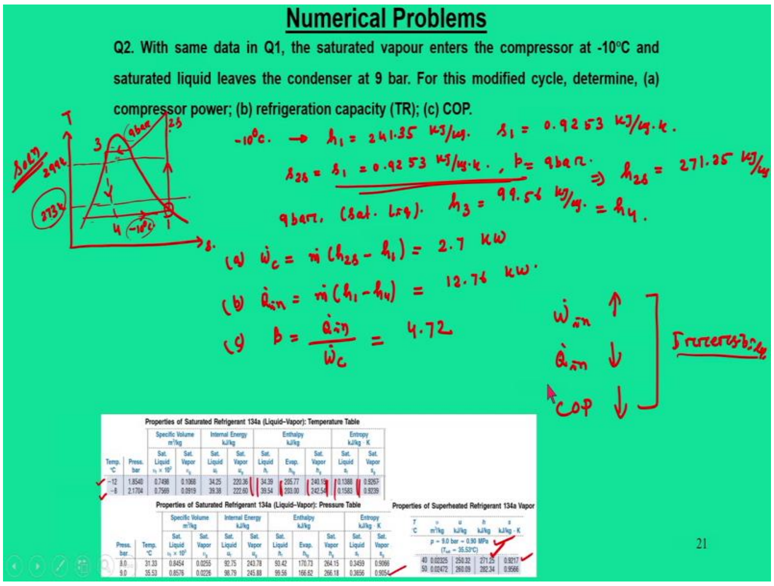

Numerical Problems

Q2. With same data in Q1, the saturated vapour enters the compressor at -10°C and saturated liquid leaves the condenser at 9 bar. For this modified cycle, determine, (a) compressor power; (b) refrigeration capacity (TR); (c) COP.

-10°C. → $h_1 = 241.35$ kJ/kg. $s_1 = 0.9253$ kJ/kg.K.

$s_{2s} = s_1 = 0.9253$ kJ/kg.K., $p = 9$ bar. ⇒ $h_{2s} = 271.25$ kJ/kg

9 bar, (Sat. Liq). $h_3 = 99.56$ kJ/kg. $= h_4$.

(a) $\dot{W}_c = \dot{m}(h_{2s} - h_1) = 2.7$ kW

(b) $\dot{Q}_{in} = \dot{m}(h_1 - h_4) = 12.76$ kW.

(c) $\beta = \frac{\dot{Q}_{in}}{\dot{W}_c} = 4.72$

$\dot{W}_{in}$ ↑
$\dot{Q}_{in}$ ↓
COP ↓
Irreversibility

Properties of Saturated Refrigerant 134a (Liquid–Vapor): Temperature Table

		Specific Volume m³/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy kJ/kg · K	
Temp. °C	Press. bar	Sat. Liquid $v_f \times 10^3$	Sat. Vapor v_g	Sat. Liquid u_f	Sat. Vapor u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Sat. Vapor s_g
−12	1.8540	0.7498	0.1068	34.25	220.36	34.39	205.77	240.15	0.1388	0.9267
−8	2.1704	0.7569	0.0919	39.38	222.60	39.54	203.00	242.54	0.1583	0.9239

Properties of Saturated Refrigerant 134a (Liquid–Vapor): Pressure Table

		Specific Volume m³/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy kJ/kg · K	
Press. bar	Temp. °C	Sat. Liquid $v_f \times 10^3$	Sat. Vapor v_g	Sat. Liquid u_f	Sat. Vapor u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Sat. Vapor s_g
8.0	31.33	0.8454	0.0255	92.75	243.78	93.42	170.73	264.15	0.3459	0.9066
9.0	35.53	0.8576	0.0226	98.79	245.88	99.56	166.62	266.18	0.3656	0.9054

Properties of Superheated Refrigerant 134a Vapor

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg · K
	p = 9.0 bar = 0.90 MPa (T_{sat} = 35.53°C)			
40	0.02325	250.32	271.25	0.9217
50	0.02472	260.09	282.34	0.9566

Now, the next question with same data in question 1, the question is little bit shifted to the fact that the saturated vapour enters the compressor -10°C. So, whereas, your cold region temperature is 0°C, but the refrigerant is now below this, so -10°C. And the saturated liquid leaves at the condensed pressure of 9 bar. So, the saturation pressure at that point is higher than the hot region temperatures. So, for this modified cycle you have to evaluate the same thing.

So, to solve this problem the we have to relook into your Ts diagram. So, we can redraw this dome again first thing we have to put a limit that what are the cold region temperatures. So, we have 0°C or 273 K and 299 kelvin, so this is cold region and hot region temperature.

But your cycle operates little bit different way the condenser temperature is higher than the hot region temperatures, evaporator temperature is lower than the cold region temperatures. So, that way we can draw this line, so it is now 1, 2s 3, 4. So, this point we say it is -10°C and this is 9 bar. So, we have to calculate the values at these points only.

So, in that sense, so we have to calculate the value at -10°C, we have data for -12 and -8. So, we can interpolate these values; that means, interpolate these values to find the data for -10°C.

$$h_1 = \frac{240.15 + 242.54}{2} = 241.35\text{kJ/kg};\ s_1 = \frac{0.9267 + 0.9239}{2} = 0.9253\text{kJ/kg-K}$$

Now, s_{2s} is equal to s_1 that is 0.9253 kJ/kg-K and p is equal to 9 bar. So, these two data gives the super heated regions. So, this will give h_{2s} as 271.25 kJ/kg.

Then again from 9 bar saturated liquid table, we can find out value as h_3 value as 99.56 kJ/kg. So, that is also equal to h_4. Now, we can find out

$$\dot{W}_c = \dot{m}\left(h_{2s} - h_1\right) = 2.7\text{kW}$$
$$\dot{Q}_{in} = \dot{m}\left(h_1 - h_4\right) = 13.76\text{kW}$$
$$\beta = \frac{\left(\dot{Q}_{in}\right)}{\left(\dot{W}_c\right)} = 4.72$$

So, we can see that due to this irreversibility, as compared to Q.1 the compressor load increases, Q_{in} drops, COP drops; this is due to irreversibility.

Now, what is this irreversibility here? Because the cold region temperature is higher than the evaporator temperature. And the hot region temperature is lower than the condenser temperatures. So, that is the irreversibility for which is there is a drop in COP.

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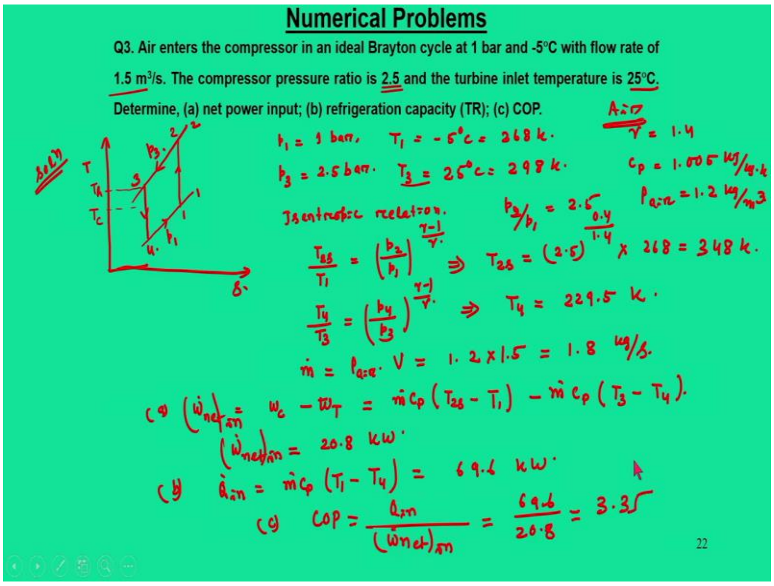

Numerical Problems

Q3. Air enters the compressor in an ideal Brayton cycle at 1 bar and -5°C with flow rate of 1.5 m³/s. The compressor pressure ratio is 2.5 and the turbine inlet temperature is 25°C. Determine, (a) net power input; (b) refrigeration capacity (TR); (c) COP.

Soln

$p_1 = 1$ bar, $T_1 = -5°C = 268$ K.

$p_3 = 2.5$ bar. $T_3 = 25°C = 298$ K.

Air

$\gamma = 1.4$

$c_p = 1.005$ kJ/kg.K

$\rho_{air} = 1.2$ kg/m³

Isentropic relation. $p_3/p_1 = 2.5$

$$\frac{T_{2s}}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_{2s} = (2.5)^{\frac{0.4}{1.4}} \times 268 = 348\text{ K}.$$

$$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4 = 229.5\text{ K}.$$

$$\dot{m} = \rho_{air} \cdot V = 1.2 \times 1.5 = 1.8\text{ kg/s}.$$

(a) $(\dot{W}_{net})_{in} = W_c - W_T = \dot{m} c_p (T_{2s} - T_1) - \dot{m} c_p (T_3 - T_4).$

$(\dot{W}_{net})_{in} = 20.8$ kW.

(b) $\dot{Q}_{in} = \dot{m} c_p (T_1 - T_4) = 69.6$ kW.

(c) $COP = \frac{\dot{Q}_{in}}{(\dot{W}_{net})_{in}} = \frac{69.6}{20.8} = 3.35$

Now, next problem is for a gas refrigeration cycle that operates in a ideal Brayton cycle. So, here we have to first draw the temperature entropy diagram for a reversed Brayton cycle. So, we can draw in this diagram you can draw the processes as 1, 2, 3, 4.

So, there are two pressures p_1 and p_3, this pressure ratio is 2.5. p_1 is 1 bar, T_1 is -5°C that is 268 K, p_3 is 2.5 bar and T_3 is 25°C, that is 298 K. For air we can take gamma as 1.4, C_P as 1.005 kJ/kg-K, ρ as 1.2 kg/m^3.

$$\frac{T_{2s}}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_{2s} = 348K \qquad \frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_4 = 229.5K$$

$$\dot{m} = \rho V = 1.2 \times 1.5 = 1.8\text{kg/s}$$

$$\dot{w}_{net} = w_c - w_T = \dot{m}c_p\left(T_{2s} - T_1\right) - \dot{m}c_p\left(T_3 - T_4\right) = 20.8\text{kW}$$

$$\dot{Q}_{in} = \dot{m}c_p\left(T_1 - T_4\right) = 69.6\text{kW}$$

$$\text{COP} = \frac{\dot{Q}_{in}}{\dot{w}_{net}} = 3.35$$

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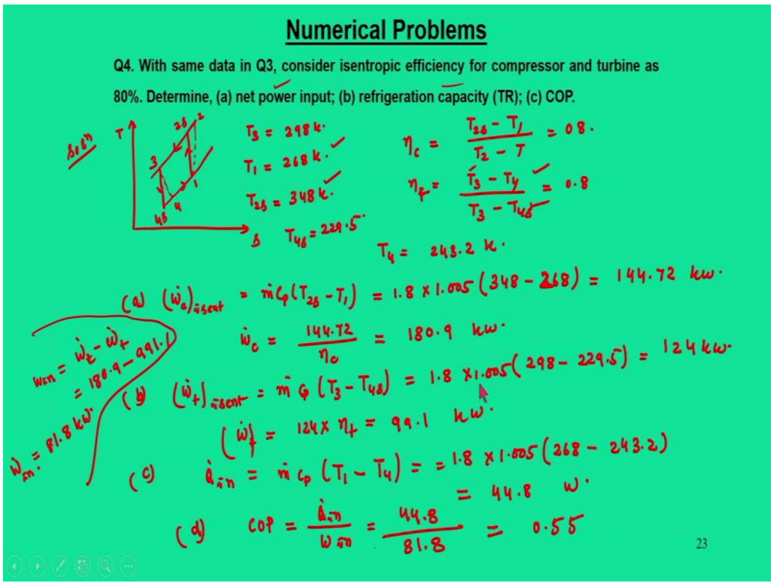

Numerical Problems

Q4. With same data in Q3, consider isentropic efficiency for compressor and turbine as 80%. Determine, (a) net power input; (b) refrigeration capacity (TR); (c) COP.

Soln

$T_3 = 298$ K.

$T_1 = 268$ K.

$T_{2s} = 348$ K.

$T_{4s} = 229.5$

$\eta_c = \frac{T_{2s} - T_1}{T_2 - T} = 0.8$

$\eta_t = \frac{T_3 - T_4}{T_3 - T_{4s}} = 0.8$

$T_4 = 243.2$ K.

(a) $(\dot{W}_c)_{isent} = \dot{m} C_p (T_{2s} - T_1) = 1.8 \times 1.005 (348 - 268) = 144.72$ kW.

$\dot{W}_c = \frac{144.72}{\eta_c} = 180.9$ kW.

$W_{en} = \dot{W}_c - \dot{W}_t = 180.9 - 99.1$

$\dot{W}_{in} = 81.8$ kW.

(b) $(\dot{W}_t)_{isent} = \dot{m} C_p (T_3 - T_{4s}) = 1.8 \times 1.005 (298 - 229.5) = 124$ kW.

$(\dot{W}_t) = 124 \times \eta_t = 99.1$ kW.

(c) $\dot{Q}_{in} = \dot{m} c_p (T_1 - T_4) = 1.8 \times 1.005 (268 - 243.2) = 44.8$ W.

(d) $COP = \frac{\dot{Q}_{in}}{W_{in}} = \frac{44.8}{81.8} = 0.55$

Now, we will move to the last problem this is in connection with the data from question 3. But here we have introduced the term isentropic efficiency. So, when we are introducing isentropic efficiency then we have to recall our data based on the expression of isentropic efficiency.

So, for this we can introduce isentropic efficiency for compressor, isentropic efficiency for turbine. So, there we can say processes are 1, 2s, 3, 4 that is isentropic, processes 1, 2, 3, 4 this is real process.

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{4s}} = 0.8; \ \eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} = 0.8$$

Other fixed datas are from the question 3 we have T_3 is 298 K, T_1 is 268 K, then T_{2s} is equal to 348 K and T_{4s} is equal to 229.5 K. Now using this equations we can calculate T_4 as 243.2 kelvin. Now we have to calculate power input, refrigeration capacity and COP.

$$\left(\dot{W}_c\right)_{isent} = \dot{m}\left(h_{2s} - h_1\right) = 1.08 \times 1.005\left(348 - 268\right) = 144.72\text{kW}$$

$$\dot{W}_c = \frac{144.72}{\eta_c} = 180.9\text{kW}$$

$$\left(\dot{W}_t\right)_{isent} = \dot{m}\left(h_3 - h_{4s}\right) = 1.08 \times 1.005\left(298 - 229.5\right) = 124\text{kW}$$

$$\dot{W}_c = 124 \times \eta_t = 99.1\text{kW}$$

$$\dot{Q}_{in} = \dot{m}\left(h_1 - h_4\right) = 1.08 \times 1.005\left(268 - 243.2\right) = 44.8\text{kW}$$

$$\beta = \frac{\left(\dot{Q}_{in}\right)}{\left(\dot{W}_{net}\right)} = \frac{44.8}{180.9 - 99.1} = 0.55$$

So, you can see that by introducing the efficiency of turbine the COP has drastically reduced. So, this means irreversibilities has a very detrimental effect in gas refrigeration systems. So, with this I conclude this lecture for today.

Thank you for your attention.