Refrigeration and Air Conditioning
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Lecture No. # 10
Vapour Compression Refrigeration Systems

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Lecture No. # 10

Vapour Compression Refrigeration Systems

Welcome back in this lecture I shall introduce vapour compression refrigeration systems the specific objectives of this particular lesson are.

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10. VAPOUR COMPRESSION REFRIGERATION SYSTEMS (VCRS)

- The objectives of this lesson are to:

1. Introduce vapour refrigeration cycles
2. Discuss Carnot vapour compression refrigeration system and its practical limitations
3. Analyze standard Vapour Compression Refrigeration System (VCRS)
4. Compare Carnot system and VCRS
5. Present performance analysis of VCRS

To introduce vapour refrigeration cycles discuss Carnot vapour compression refrigeration system and its practical limitations. Analyze standard vapour compression refrigeration system abbreviated as VCRS compare Carnot system and VCRS and finally present performance analysis of VCRS.

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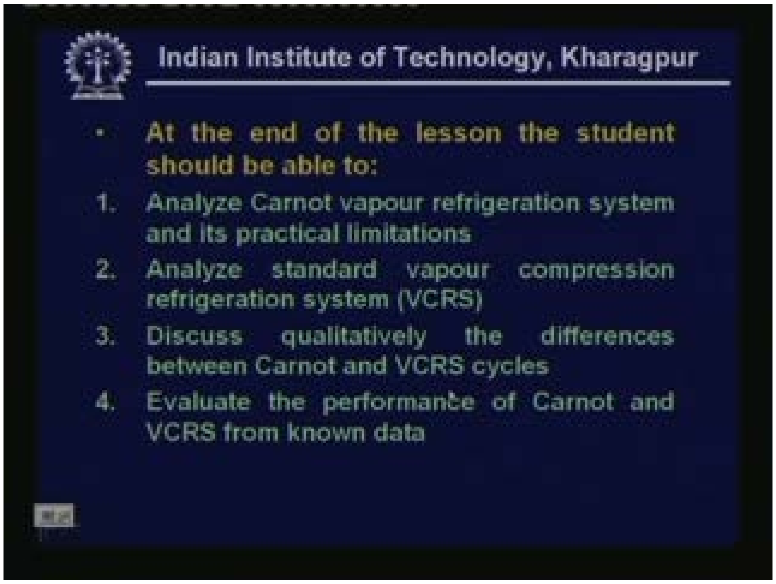

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- At the end of the lesson the student should be able to:

1. Analyze Carnot vapour refrigeration system and its practical limitations
2. Analyze standard vapour compression refrigeration system (VCRS)
3. Discuss qualitatively the differences between Carnot and VCRS cycles
4. Evaluate the performance of Carnot and VCRS from known data

At the end of the lesson you should be able to analyze Carnot vapour refrigeration system and its practical limitations. Analyze standard vapour compression refrigeration system discuss qualitatively the differences between Carnot and vapour compression refrigeration cycles evaluate the performance of Carnot and vapour compression refrigeration system from known data.

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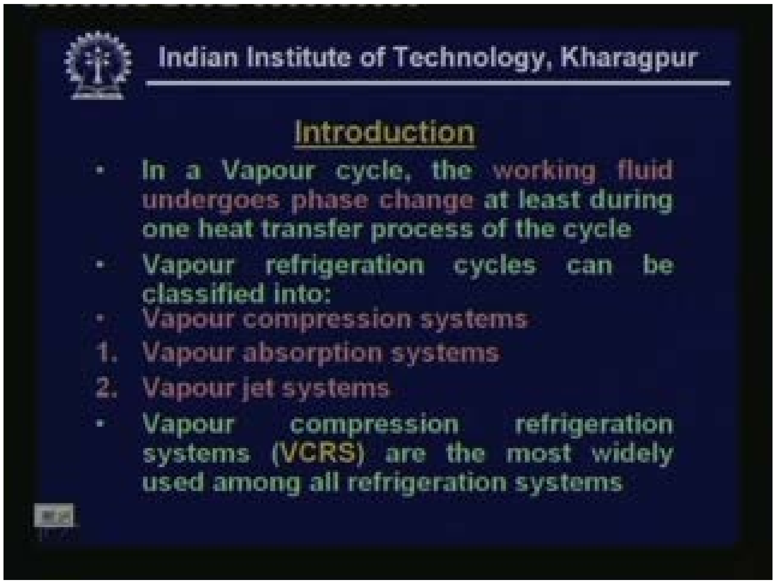

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Introduction

- In a Vapour cycle, the working fluid undergoes phase change at least during one heat transfer process of the cycle
- Vapour refrigeration cycles can be classified into:
- Vapour compression systems
1. Vapour absorption systems
2. Vapour jet systems
- Vapour compression refrigeration systems (VCRS) are the most widely used among all refrigeration systems

Let me give brief introduction first of all what is a vapour cycle. In a vapour cycle the working fluid undergoes phase change at least during one heat transfer process of the cycle this is against your gas cycle in which the working fluid does not undergo any phase change. So in a vapour cycle whether it is a power cycle or refrigeration cycle the working fluid undergoes phase change at least during one process okay. That is the definition of vapour cycle okay. So vapour refrigeration cycles can be classified into vapour compression systems vapour absorption systems and vapour jet systems. And among these three the vapour compression refrigeration systems are the most widely used among all the refrigeration systems in fact.

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- In a vapour compression refrigeration system:
- Refrigeration is obtained as the refrigerant evaporates at low temperatures
- The system input is in the form of mechanical energy required to run the compressor (mechanical refrigeration systems)
- Available to suit almost all applications, with refrigeration capacities ranging from few watts to few megawatts

In a vapour compression refrigeration system refrigeration is obtained as a refrigerant evaporates at low temperatures. The system input is in the form of mechanical energy required to run the compressor. Hence these systems are also known as mechanical refrigeration systems vapour compression refrigeration systems are available to suit almost all applications with refrigeration capacities ranging from few watts to few megawatts. That means this is one of the most versatile refrigeration systems.

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- Carnot vapor compression refrigeration cycle:
- Carnot refrigeration cycle is a completely reversible cycle
- It is used as a model of perfection for a refrigeration cycle operating between a constant temperature heat source and sink
- It is used as reference against which the real cycles are compared

First let us look at Carnot vapour compression refrigeration cycle. Let me give a brief introduction to Carnot cycle. The Carnot refrigeration cycle is a completely reversible cycle. That means it is internally as well as externally reversible it is used as a model of perfection for a refrigeration cycle operating between a constant temperature heat source and sink. This is one thing you must keep in mind this is model of perfection for constant temperature heat source and sink. It is used as a reference against which the real cycles are compared now let me explain Carnot vapour compression refrigeration cycle.

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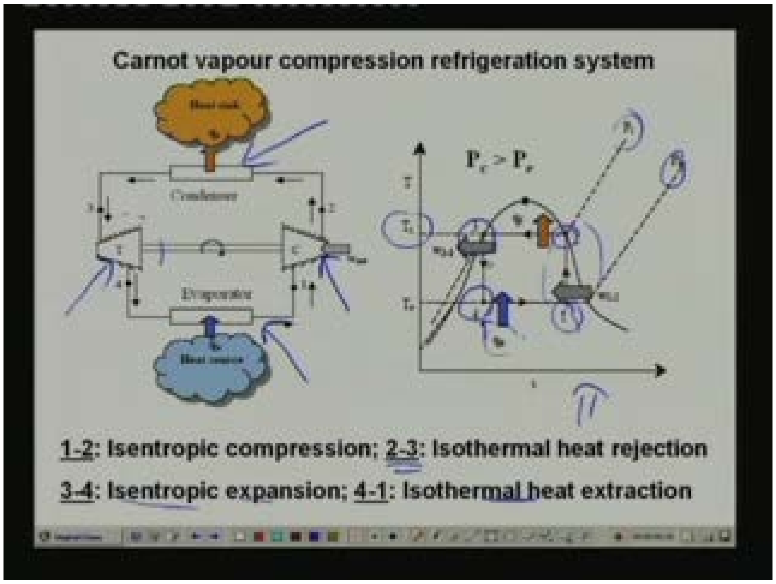

Carnot vapour compression refrigeration system

1-2: Isentropic compression; 2-3: Isothermal heat rejection

3-4: Isentropic expansion; 4-1: Isothermal heat extraction

As you can see here now the Carnot refrigeration cycle depends consists of four basic components the compressor condenser turbine and an evaporator and the cycle as shown on the T S diagram. Here consists of four basic processes, process one to two is isentropic compression process, two to three is isothermal heat rejection process, three to four is isentropic expansion in the turbine and finally process four to one is isothermal heat extraction. Now if you look at process one to two this is nothing but isentropic compression okay. So in this process you can see that the inlet at the inlet to the compressor you have the mixture of liquid and vapour. That means at point one you have the mixture of liquid and vapour.

This mixture of liquid and vapour is compressed isentropically from the evaporated pressure P e to the condenser pressure P c okay. For the exit of the compression process you have saturated vapour at condenser pressure okay. So that is the compression process. Next the isothermal heat rejection process during this process the working fluid rejects heat to the constant temperature heat sink at a temperature of T c that is the condensing temperature. And during this process the working fluid undergoes a phase change at the inlet to the condenser you have saturated vapour point two and at the exit of the condenser you have saturated liquid okay. So that is process two to three which is isothermal heat rejection process.

Next process three to four this is what happens in the turbine, this is an, as I have already said this is an isentropic expansion process. During this process saturated liquid, that means, that is at point three at condenser pressure expands isentropically in the turbine. And at the exit of the turbine you have a mixture of liquid and vapour at state four okay. And finally process four to one is an isothermal heat extraction process. During this process useful refrigeration effect q e is obtained okay. So this is the cycle of a simple Carnot vapour compression refrigeration system.

You can see that in this process heat transfer takes place during process four to one and during process two to three. Whereas work transfer takes place during process one to two and during process three to four now if you do a simple analysis of the system.

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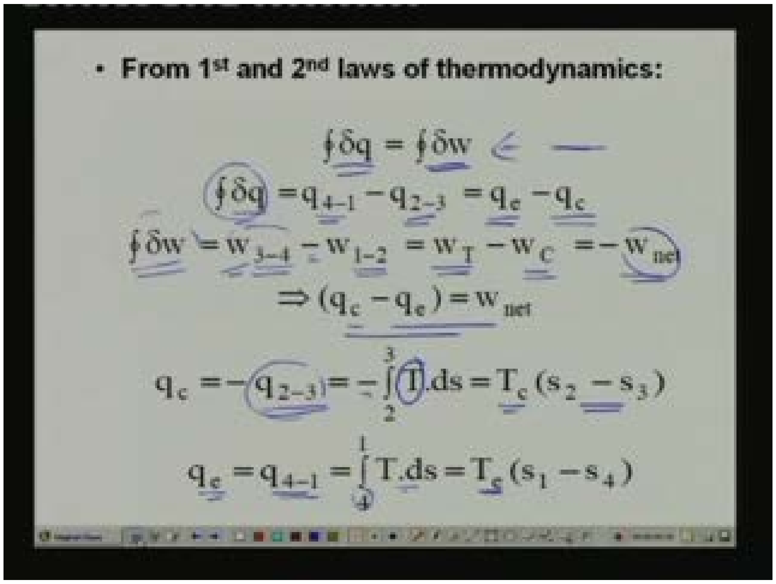

- From 1st and 2nd laws of thermodynamics:

$$\oint \delta q = \oint \delta w$$

$$\oint \delta q = q_{4-1} - q_{2-3} = q_e - q_c$$

$$\oint \delta w = w_{3-4} - w_{1-2} = w_T - w_C = -w_{net}$$

$$\Rightarrow (q_c - q_e) = w_{net}$$

$$q_c = -q_{2-3} = -\int_2^3 T.ds = T_c(s_2 - s_3)$$

$$q_e = q_{4-1} = \int_4^1 T.ds = T_e(s_1 - s_4)$$

From first and second law of thermodynamics you can write, for the cycle the first and the second laws are like this cyclic integral of Do q is equal to cyclic integral of Do w. And Do q is nothing but the heat transfer taking place. During the cycle heat transfer takes place during process four to one and this is positive. Because heat is added to the system and heat transfer also takes place during process two to three and this is negative because during this process heat is rejected from the system.

So cyclic integral of Do q is nothing but q four to one minus q two to three and q four to one is nothing but useful refrigeration effect q e and q two q two to three is nothing but heat rejected in the condenser q c. So cyclic integral of Do q is nothing but q e minus q c now cyclic integral of Do w. So work transfer takes place during process three to four and during process one to two.

During process three to four the system does the useful work in the turbine. So this is positive okay. And during process one to two works is done on the system. So this is negative okay, and if you look at the schematic w three to four is nothing but the work output from the turbine w T.

And w one to two is nothing but work input to the compressor w C. So cyclic integral of Do w is nothing but w T minus w C which is nothing but minus w net okay, where w net is the work input to the system. So if you substitute these in this first law for the cycle you find that q c minus q e is equal to W net. Now what is q c? q c is nothing but negative of heat transfer during process two to three. Since this is a reversible process you can write this as integral T d s from two to three with a negative sign and since this is isothermal process, temperature remains constant at condenser temperature. So T can be taken out so integral two to three integral T d s from two to three is nothing but T c into s two minus s three.

Similarly q e that is the refrigeration effect is nothing but the heat transfer during process four to one it is again, this is a reversible process. So you can write this as integral T d s from four to one temperature remains constant at evaporator temperature T e. So integral T d s is equal to T e into s one minus s four.

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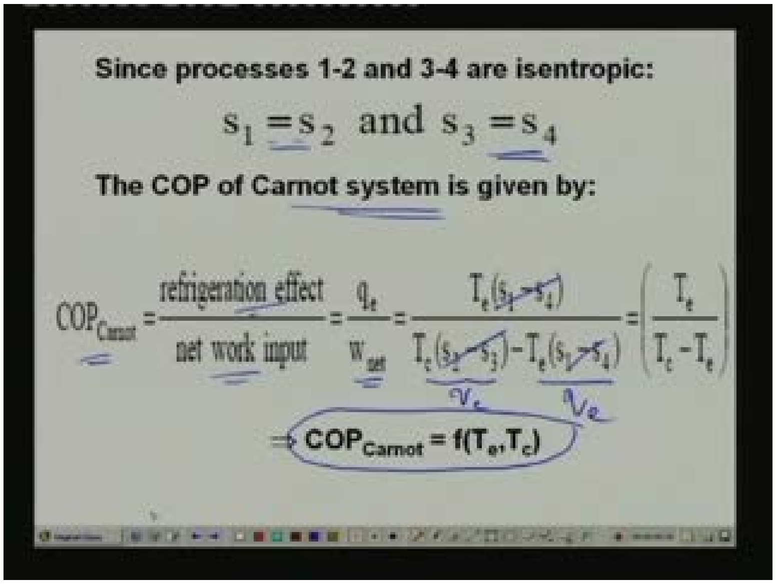

Since processes 1-2 and 3-4 are isentropic:

$$s_1 = s_2 \text{ and } s_3 = s_4$$

The COP of Carnot system is given by:

$$COP_{Carnot} = \frac{\text{refrigeration effect}}{\text{net work input}} = \frac{q_e}{w_{net}} = \frac{T_e(\cancel{s_1 - s_4})}{\underbrace{T_c(\cancel{s_2 - s_3})}_{q_c} - \underbrace{T_e(\cancel{s_1 - s_4})}_{q_e}} = \left(\frac{T_e}{T_c - T_e}\right)$$

$$\Rightarrow \boxed{COP_{Carnot} = f(T_e, T_c)}$$

Since processes one to two and three to four are isentropic s one is equal to s two and s three is equal to s four. Because the expansion and compression processes are isentropic processes. Now the COP of the Carnot system is given by COP of the Carnot is given as refrigeration effect divided by net work input that is q e divided by w net. W net is nothing but q c minus q e. This is q c and this is q e okay. And all q e and q c can be written in terms of temperatures and entropic changes that is q e is nothing but T e into s one minus s four similarly q c is equal to T c into s two minus s three.

Since s one is equal to s three and s three is equal to s four this terms get cancelled. So finally you find that the COP of the Carnot cycle is nothing but T e divided by T c minus T e where T e and T c are evaporator and condenser temperatures. That means the COP of the Carnot cycle is a function of operating temperatures only.

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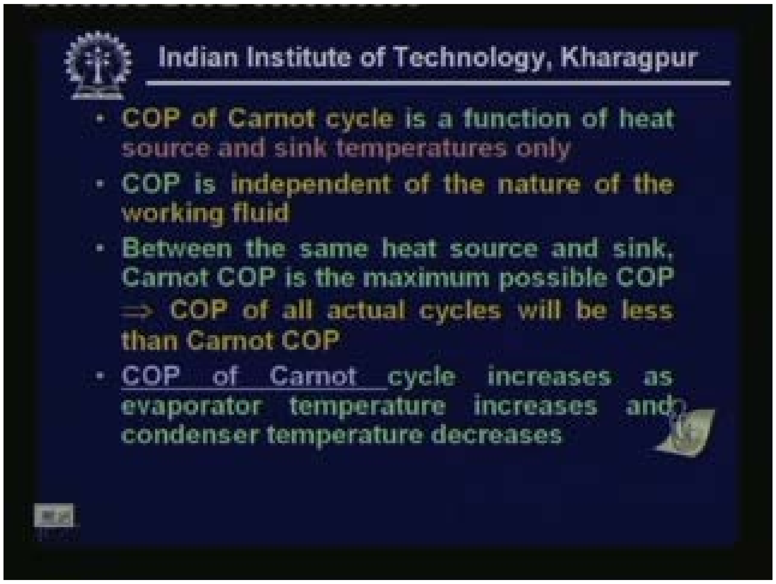

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- COP of Carnot cycle is a function of heat source and sink temperatures only
- COP is independent of the nature of the working fluid
- Between the same heat source and sink, Carnot COP is the maximum possible COP $\Rightarrow$ COP of all actual cycles will be less than Carnot COP
- COP of Carnot cycle increases as evaporator temperature increases and condenser temperature decreases

Now as I, as I have already explained COP of the Carnot cycle is a function of heat source and sinks temperatures. Only COP is independent of the nature of the working fluid. Whatever be the working fluid the expression for COP of the Carnot system remains same. Between the same heat source and sink Carnot COP is the maximum possible COP. That is why I said that this is a used as a standard for comparison okay, and COP of all actual cycles will be less than the Carnot COP. And COP of a Carnot cycle increases as evaporated temperature increases and as condenser temperature decreases. Let me explain this okay, so you can see here again I am plotting the Carnot cycle one two three four.

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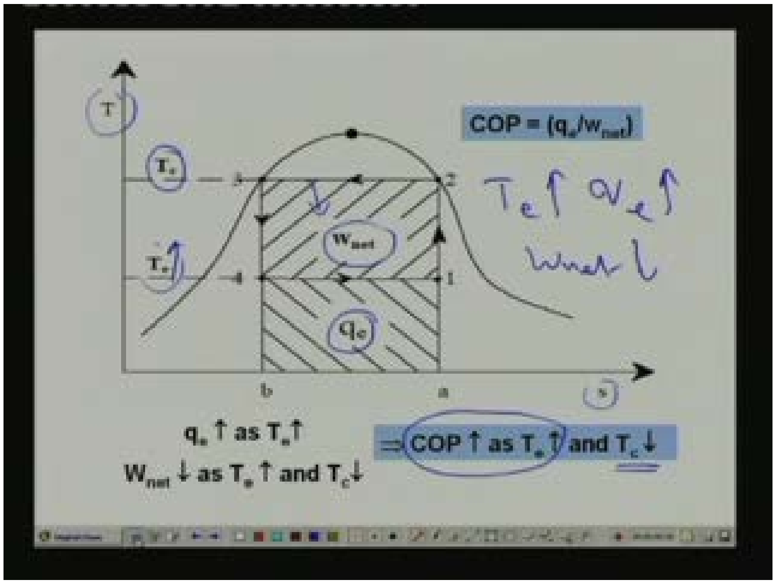

COP = (q_e/w_{net})

$q_e \uparrow$ as $T_e \uparrow$

$W_{net} \downarrow$ as $T_e \uparrow$ and $T_c \downarrow$

$\Rightarrow$ COP ↑ as $T_e \uparrow$ and $T_c \downarrow$

On temperature entropic coordinates okay. So for a given condenser temperature, that means I am fixing this condenser temperature and let us say that I am increasing T e. So what happens when you are increasing T e you find that q e increases and w net decreases okay, and since COP is equal to q e by w net as T e increases q e increases. That means the area under one to four increases and W net decreases. That means the area one two three four reduces as a result COP increases as T e increases. And similarly if you fix T e and reduce, let us say T c so when you are fixing evaporate temperature and reducing the condenser temperature q e does not change q e remains constant but w net reduces as a result again COP increases as condenser temperature reduces. The same thing is shown.

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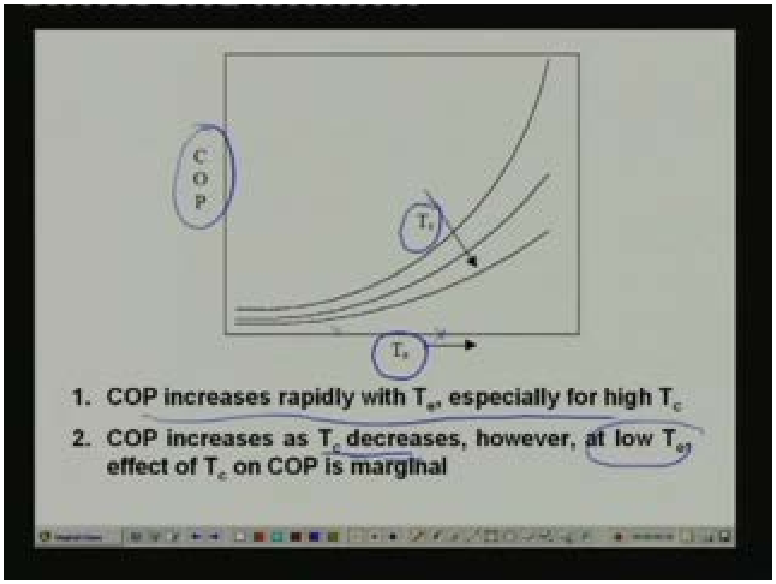

1. COP increases rapidly with T_e, especially for high T_c
2. COP increases as T_c decreases, however, at low T_e, effect of T_c on COP is marginal

On a graph here I have plotted COP of the Carnot system versus evaporated temperature for different condenser temperatures. You can see that for a given condenser temperature as evaporated temperature increases the COP increases. Similarly at a given evaporated temperature as a condenser temperature increases the COP reduces okay, that is what we have seen from the expression. And you can also see that the COP increases rapidly with T e especially for high T c. That means for and the condenser temperature is high the rate at which COP increases T e is high. Similarly COP increases as condenser temperature decreases however at low evaporated temperature. That means in this region the effect of T c on COP's marginal. So these are the observations we make from the expression for COP of a Carnot cycle.

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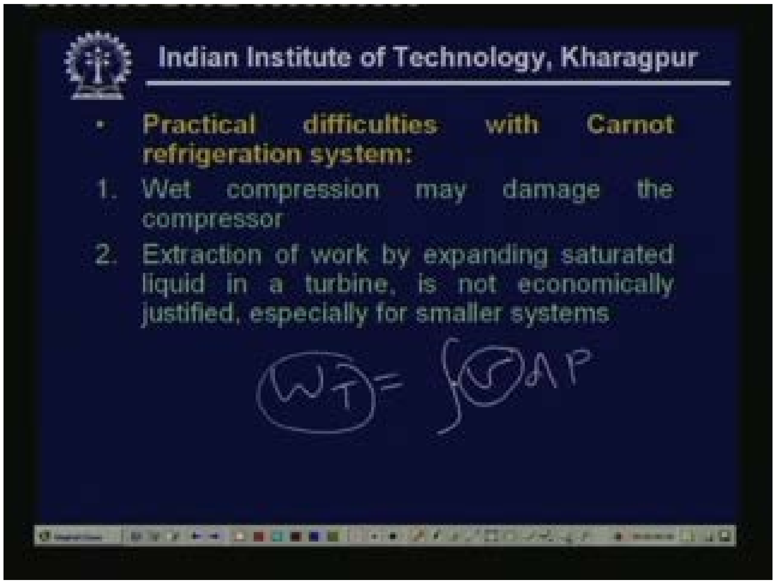

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- **Practical difficulties with Carnot refrigeration system:**

1. Wet compression may damage the compressor
2. Extraction of work by expanding saturated liquid in a turbine, is not economically justified, especially for smaller systems

$$\boxed{W_T} = \int \boxed{v}\, dP$$

Now what are the practical difficulties with Carnot refrigeration system we know that the Carnot refrigeration system gives the maximum COP then what is the problem in having Carnot refrigeration system. There are certain practical difficulties which will prevent us from constructing a cycle which operates on a Carnot cycle okay. These practical difficulties are like this. The first practical difficulty is with wet compression I have mentioned while explaining the Carnot refrigeration cycle that during the compression process the working fluid at the inlet to the compressor consists of liquid as well as vapour. That means you have to compress a mixture of liquid and vapour.

Most of the practical compressors are designed to compress vapour only. So if there is liquid this kind of a compression known as wet compression. That means whenever you have liquid at the inlet to the compressor you call that compression process as wet compression. And most of the compressors are not designed for wet compression. So when there is liquid at the inlet the compressor may get damaged okay. So this is one of the practical difficulties with Carnot refrigeration cycle okay. The second difficulty is extraction of work by expanding saturated liquid in a turbine is not economically justified especially for smaller systems. We have seen that in Carnot refrigeration cycle the expansion process is isentropic. That means we use a turbine and extract work that’s why you get an isentropic expansion process.

But what is the work output during this process if you assume that the process is steady flow process, then the work output of a turbine specific work output of the turbine is typically given by if you are neglecting kinetic and potential energy changes. Then you find that work input of a work output of a turbine is given by integral v d p and since you are expanding the liquid and liquids have typically very small specific volume. So you find that this is very small because of the small value of specific volume. As a result compare to the compressor work input the work output of the isentropic turbine is very small over. And above that when you consider actual turbine actual turbine will be difference from isentropic turbine because there will be some irreversibilities.

So the actual output of a turbine will be much less than the isentropic work output. So if you consider the cost involved in building a turbine and in incorporating a turbine and the work out output that you get out of it you will be find that in most of the cases especially for small capacity systems it is not economically feasible to use a turbine and to extract work okay. So the economic feasibility prevents the use of turbine okay. So this is second practical difficulty with a Carnot refrigeration system okay.

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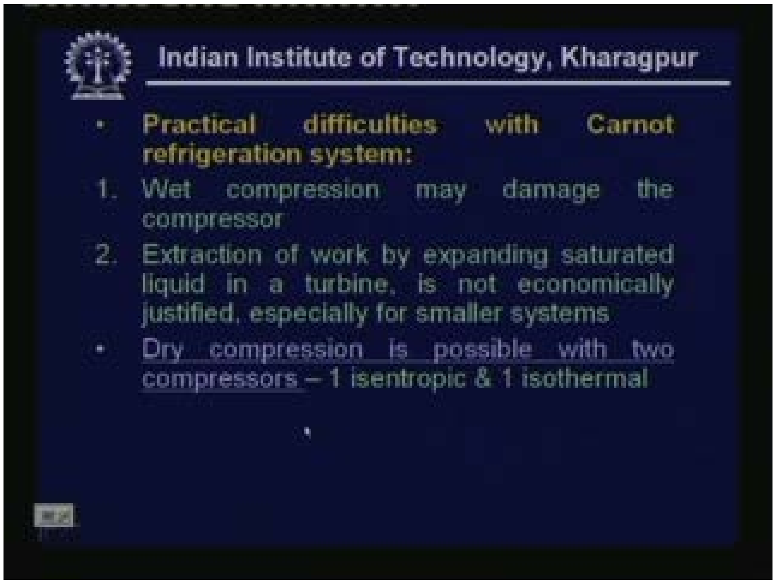

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- Practical difficulties with Carnot refrigeration system:

1. Wet compression may damage the compressor
2. Extraction of work by expanding saturated liquid in a turbine, is not economically justified, especially for smaller systems

- Dry compression is possible with two compressors – 1 isentropic & 1 isothermal

Of course the first difficulty that is the wet compression can be eliminated by using two compressors. Let me show what is the meaning of that.

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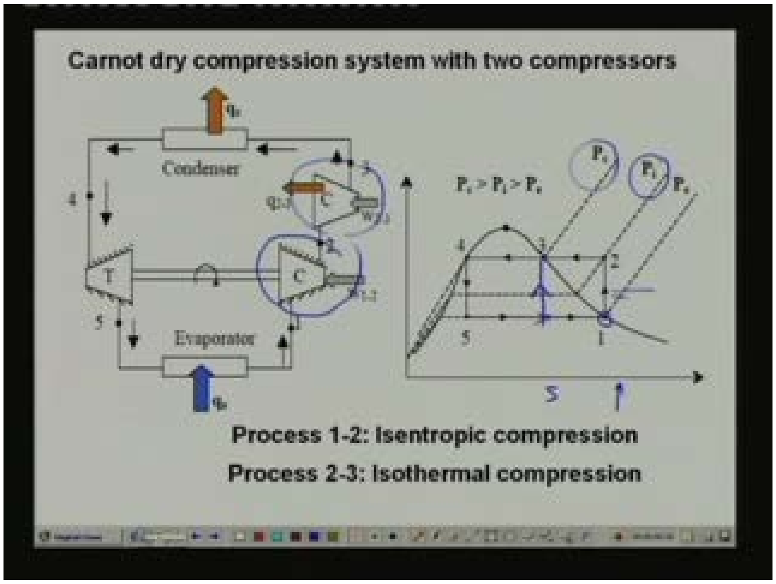

Carnot dry compression system with two compressors

Process 1-2: Isentropic compression

Process 2-3: Isothermal compression

For example you can look at the cycle here on T S okay. This is T S wet compression was something like this the wet compression can be eliminated if you move this point from this point to the saturated point one okay. Then the wet compression is avoided but you have to achieve the compression process now you have to add another compressor okay. That means the compression process is split into two parts part one to two is an isentropic compression okay. And from evaporator pressure P e to an intermediate pressure P I and process two to three is isothermal compression from intermediate pressure P i to condenser pressure P c okay. So the by using two compressors instead of one compressor you can eliminate the problem of wet compression okay.

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- Practical difficulties with Carnot refrigeration system:
1. Wet compression may damage the compressor
2. Extraction of work by expanding saturated liquid in a turbine, is not economically justified, especially for smaller systems
- Dry compression is possible with two compressors – 1 isentropic & 1 isothermal
  - Isothermal compression is difficult
  - Higher cost due to two compressors

However you find that isothermal compression is difficult in practice and the cost also will go up because you are using two compressors instead of one compressor okay. So these are the practical difficulties with Carnot refrigeration systems.

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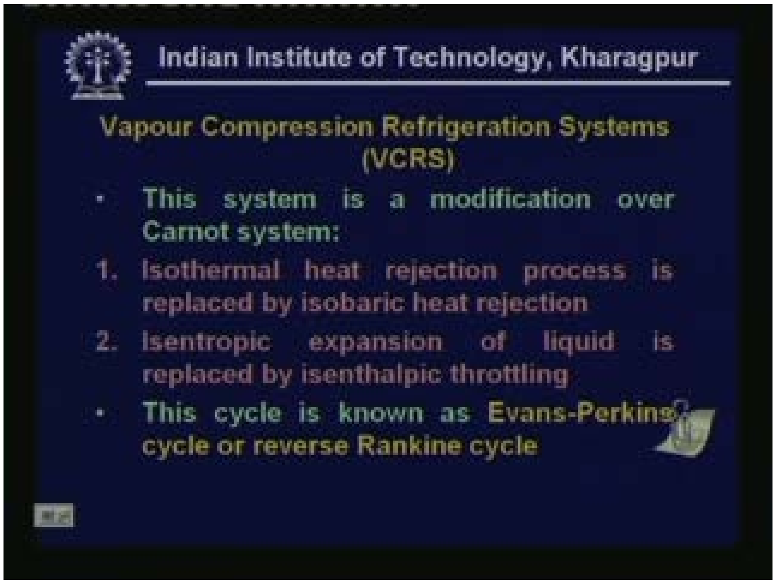

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Vapour Compression Refrigeration Systems (VCRS)

- This system is a modification over Carnot system:
1. Isothermal heat rejection process is replaced by isobaric heat rejection
2. Isentropic expansion of liquid is replaced by isenthalpic throttling
- This cycle is known as Evans-Perkins cycle or reverse Rankine cycle

Now let us look at vapour compression refrigeration systems which is the system used in practice. This system is the modification over Carnot system and what are the modifications?
The first modification is that the isothermal heat rejection process in the condenser is replaced by isobaric heat rejection this is the first deviation from the Carnot cycle.

Second deviation is that the isentropic expansion of liquid in the turbine is replaced by isenthalpic throttling in a throttling device okay. So these are the two major differences between the vapour standard vapour compression refrigeration system and the Carnot vapour compression refrigeration system okay. And this standard vapour compression refrigeration system is known as Evans Perkins cycle or sometimes it is also known as reverse Rankine cycle.

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- Standard, saturated single stage (SSS) VCRS:

1. Exit conditions of evaporator and condenser are saturated
2. The cycle consists of one low-side pressure and one high-side pressure
3. Cycle is internally reversible
4. Compression is isentropic
5. Expansion is isenthalpic

Now let me describe the standard saturated single stage cycle this is abbreviated as SSS this stands for standard saturated single stage vapour compression refrigeration system. First let me explain this a standard saturated single stage vapour compression refrigeration system the exit conditions of evaporator and condenser are saturated that is why you have this term saturated here and the cycle consists of one low side pressure and one high side pressure that is why you call it as a single stage cycle. And cycle is internally reversible and compression is isentropic and expansion is isenthalpic okay.

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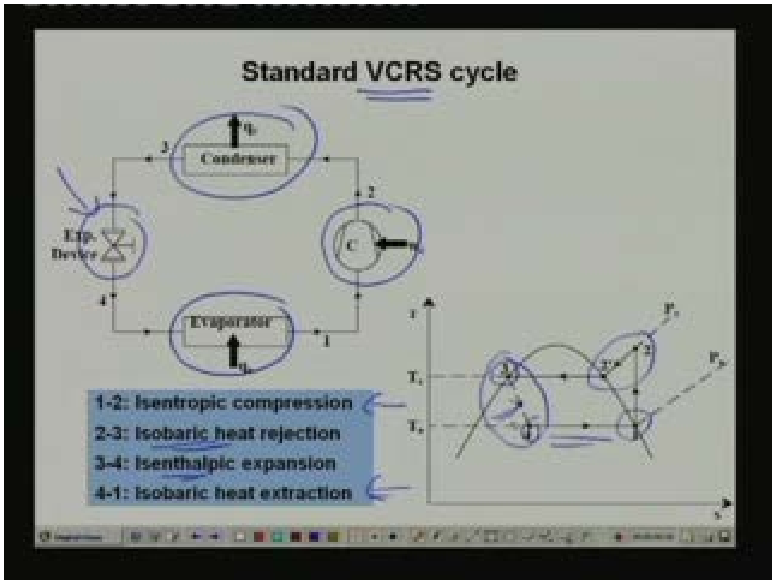

Standard VCRS cycle

1-2: Isentropic compression
2-3: Isobaric heat rejection
3-4: Isenthalpic expansion
4-1: Isobaric heat extraction

So now let me show the schematic of the cycle and the cycle on T S diagram. You can see here that this cycle also, that means the standard vapour compression cycle also consists of four basic components the compressor condenser expansion device or a throttling device and the evaporator okay. So if we compare this cycle with the Carnot cycle you will find that in terms of the components only difference is in the expansion device there we were using turbine here we use a throttling valve okay. And it can the cycle on T S diagram is shown here it consists of four processes. Process one to two is isentropic compression process two to three is isobaric heat rejection process three to four is isenthalpic expansion and four to one is isobaric and isothermal heat extraction okay.

Now again if you see compare this cycle with the Carnot cycle you find that here the compression is dry compression because you are compressing only vapour okay. So the problem of wet compression is eliminated. But as a result of this you find that the condensation process or the heat rejection process is isobaric but not isothermal. Because during the process two to two dash the temperature is varying where as during the process two dash to three the temperature remains constant. That means initial heat rejection is an isobaric process but non isothermal process where as the rest of the process is isobaric as well as isothermal. Now it coming to the expansion process three to four is isenthalpic process and this is shown by a dash line because isenthalpic throttling process is highly irreversible. So we really do not know the path we only know the end states three and four okay. And process four to one as I said is isobaric heat extraction and if we are using a pure fluid this is also isothermal.

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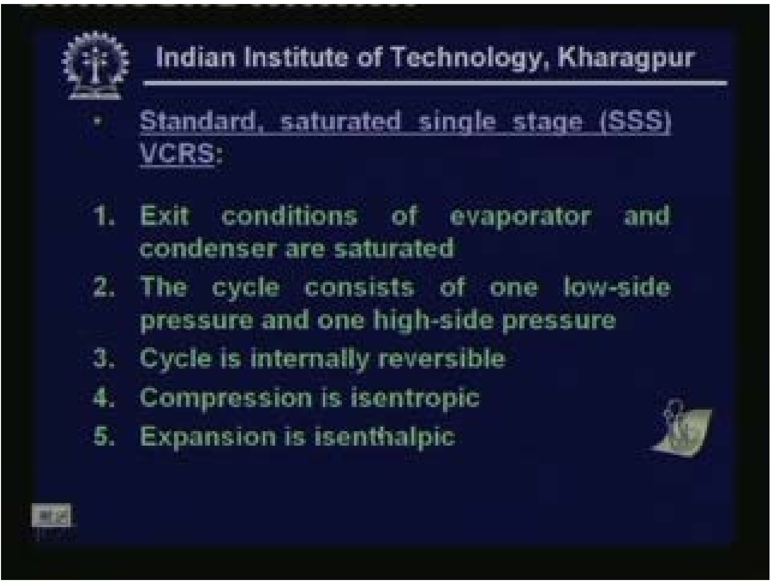

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- Standard, saturated single stage (SSS) VCRS:

1. Exit conditions of evaporator and condenser are saturated
2. The cycle consists of one low-side pressure and one high-side pressure
3. Cycle is internally reversible
4. Compression is isentropic
5. Expansion is isenthalpic

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Okay this is the standard saturated single stage vapour compression refrigeration cycle.

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Comparison between Carnot and SSS cycle

For the same heat source and sink temperatures, compared to Carnot cycle:

1. Refrigeration effect of SSS cycle decreases
2. Heat rejection increases

Now how does this cycle compare with Carnot cycle, what are the differences? For the same heat source and sink temperature that means for the same evaporator and condenser temperatures compare to Carnot cycle we find that the refrigeration effect of the standard cycle decreases. So how do you show that this can be shown with the help of the T s diagram?

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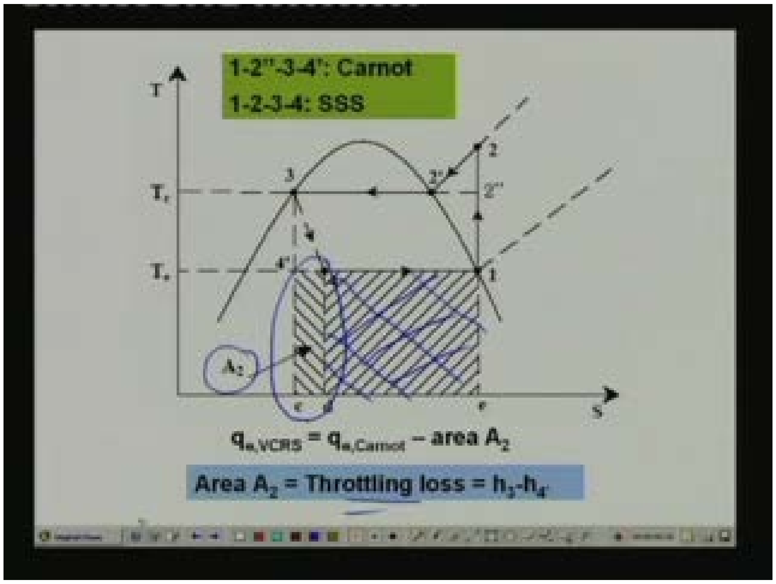

1-2''-3-4': Carnot

1-2-3-4: SSS

$q_{e,VCRS} = q_{e,Carnot} - \text{area } A_2$

Area A_2 = Throttling loss = $h_3 - h_{4'}$

You can see here that one two dash three four is the Carnot cycle that means this cycle is the Carnot cycle okay, whereas one two three four that means this one is your standard vapour compression refrigeration cycle okay. Now what is the refrigeration effect of Carnot cycle and what is the refrigeration effect of the standard cycle and what is the difference between these two, okay, let us write the expressions for this. Now if we look at the Carnot cycle one two double dash three four dash the refrigeration effect is nothing but the area under the curve four dash to one. That means this entire area okay, this entire area right, where as the refrigeration effect of the standard cycle is nothing but area under process four to one that means this area okay.

So you find that the refrigeration difference between the refrigeration effect of the Carnot cycle and vapour compression refrigeration cycle is nothing but this area four dash c d four that is area A two and this area A two is because of the throttling and this area A two is also known as throttling loss okay. So in terms of you can write derive the expressions.

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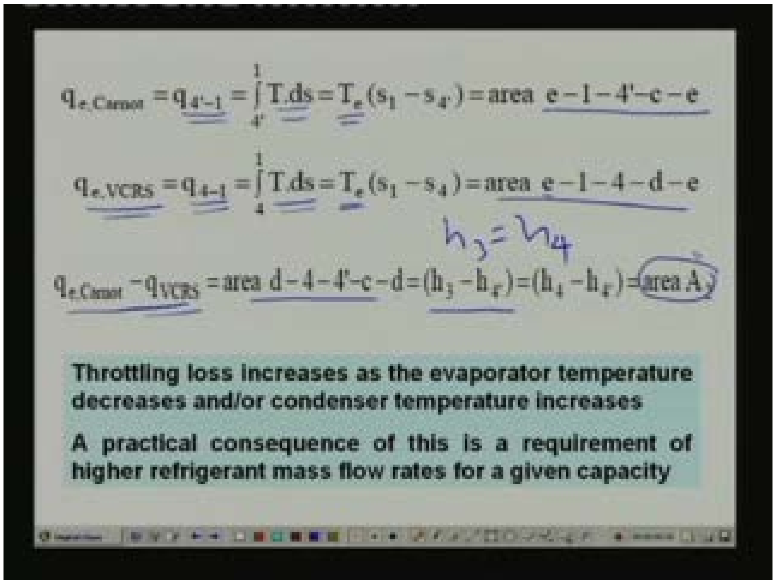

$$q_{e,Carnot} = \underline{\underline{q_{4'-1}}} = \int_{4'}^{1} \underline{\underline{T.ds}} = \underline{\underline{T_e}}(s_1 - s_{4'}) = \text{area } \underline{e-1-4'-c-e}$$

$$\underline{\underline{q_{e,VCRS}}} = \underline{\underline{q_{4-1}}} = \int_{4}^{1} \underline{\underline{T.ds}} = \underline{\underline{T_e}}(s_1 - s_4) = \text{area } \underline{e-1-4-d-e}$$

$$h_3 = h_4$$

$$\underline{q_{e,Carnot} - q_{VCRS}} = \text{area } \underline{d-4-4'-c-d} = \underline{(h_3 - h_{4'})} = (h_4 - h_{4'}) = \boxed{\text{area } A_2}$$

Throttling loss increases as the evaporator temperature decreases and/or condenser temperature increases

A practical consequence of this is a requirement of higher refrigerant mass flow rates for a given capacity

For this, so for example the refrigeration effect of the Carnot cycle as I have already told you in nothing but heat transfer during process fourth four dash to one so that is nothing but integral T d s from four dash two one temperature remains constant. So this is nothing but T e into s one minus s four dash which is equal to area e one four dash c e okay. And refrigeration effect of the vapour compression refrigeration cycle is nothing but heat transfer during process q four to one that is nothing but integral T d s from four to one that is equal to T e into s one minus s four that is equal to area e one four d e.

So the difference between the refrigeration effect that q e Carnot minus q e vapour compression refrigeration cycle is nothing but the area d four four dash c d which is equal to h three minus h four dash or this is also equal to h four minus h four dash because h three is equal to h four because this is an isenthalpic expansion process. This is nothing but area A two which we call as throttling loss area okay. And if you look at the T S diagram you find that the throttling loss increases as evaporated temperature decreases and or the condenser temperature increases and a practical consequence of this is a requirement of higher refrigerant mass flow rate for a given capacity okay.

Because you are loosing refrigeration effect if the capacity is fixed to get the same capacity more amount of refrigerant has to circulate okay. So this is the a consequence of changing the expansion process from isentropic expansion to isenthalpic expansion process okay.

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Comparison between Carnot and SSS cycle

For the same heat source and sink temperatures, compared to Carnot cycle:

1. Refrigeration effect of SSS cycle decreases
2. Heat rejection increases

Now let us look at the heat rejection you if you find the same heat source and sink temperatures compare to Carnot cycle the heat rejection increases in a standard a vapour compression refrigeration cycle again with the help of the T s diagram. This can be shown very easily.

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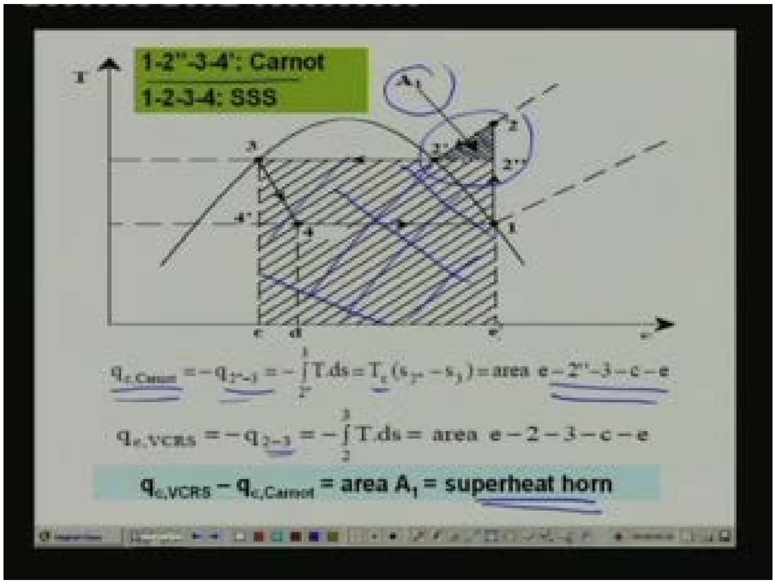

$q_{c,Carnot} = -q_{2''-3} = -\int_{2''}^{3} T.ds = T_c(s_{2''} - s_3) = \text{area } e-2''-3-c-e$

$q_{c,VCRS} = -q_{2-3} = -\int_{2}^{3} T.ds = \text{area } e-2-3-c-e$

$q_{c,VCRS} - q_{c,Carnot} = \text{area } A_1 = \text{superheat horn}$

For example if you look at the T s diagram again one two dash three four dash is your Carnot cycle and one two three four is your standard vapour compressor refrigeration cycle. And what is the heat rejection in the Carnot cycle? Heat rejection in the Carnot cycle is nothing but heat rejected during process two dash to three, that is nothing but area under two dash to three which is equal to T c into s two dash minus s three that means this entire area okay, that is area e two dash e two dash three c e. Now what is the heat rejection in standard vapour compression refrigeration cycle that is nothing but heat rejected during process two to three that is nothing but the area under curve two to three okay that means this area plus this area okay. So this is equal to area e two three c e that is area e two three c e right the entire area. So if you find the difference between the heat rejection of vapour compression cycle and the Carnot cycle is nothing but this area two two dash two double dash or area A one okay, and this area is known as superheat horn okay. And this is coming because of the fact that we are replacing the isothermal heat rejection of the Carnot cycle by an isobaric heat rejection process.

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Comparison between Carnot and SSS cycle

For the same heat source and sink temperatures, compared to Carnot cycle:

1. Refrigeration effect of SSS cycle decreases
2. Heat rejection increases
3. Compressor work input increases

$\Rightarrow COP_{SSS} < COP_{Carnot}$

Now because of the differences in refrigeration effect and heat rejection rate we find that the compressor work input increases when you change over from Carnot cycle to standard vapour compression cycle again with the help of T s diagram, we can show this one.

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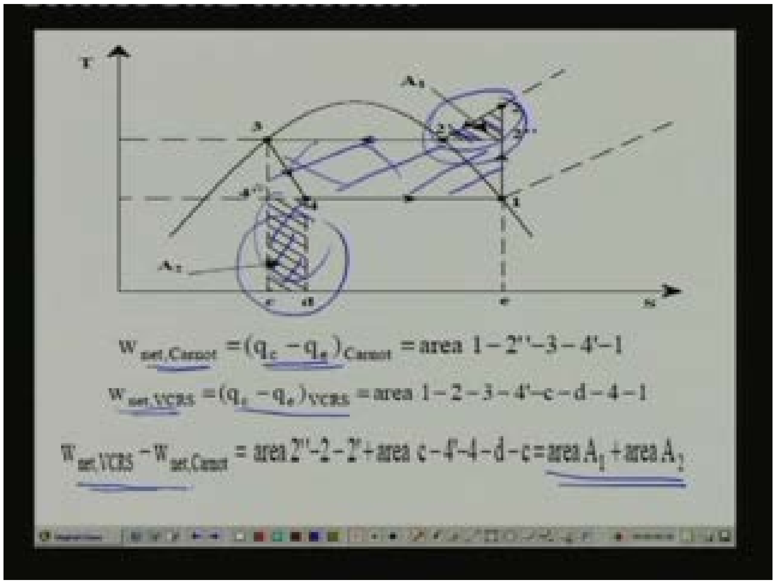

$$W_{net,Carnot} = (q_c - q_e)_{Carnot} = \text{area } 1-2''-3-4'-1$$

$$W_{net,VCRS} = (q_c - q_e)_{VCRS} = \text{area } 1-2-3-4'-c-d-4-1$$

$$W_{net,VCRS} - W_{net,Carnot} = \text{area } 2''-2-2' + \text{area } c-4'-4-d-c = \text{area } A_1 + \text{area } A_2$$

If you look at again the T s diagram you find that the net work input of Carnot cycle is nothing but q c minus q f, Carnot cycle that is nothing but area one two double dash three four dash one that means this area okay. Whereas the net work input of a vapour compression refrigeration cycle in nothing but q c minus q e f vapour compression refrigeration cycle that is nothing but this area. That means area one two three four dash c d four okay. That means this area plus this area plus this area okay. So you find the difference between the net work input of a standard cycle minus Carnot cycle is nothing but this area that is the area rectangular area one two dash three four dash plus super heat horn area A one plus throttling area A two okay. That means the difference is nothing but area A one plus area A two okay in this area and this area right.

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Comparison between Carnot and SSS cycle

For the same heat source and sink temperatures, compared to Carnot cycle:

1. Refrigeration effect of SSS cycle decreases
2. Heat rejection increases
3. Compressor work input increases

$\Rightarrow COP_{SSS} < COP_{Carnot}$

So as a result of this so you find that when you have moved away from the Carnot cycle you lost refrigeration effect and you also have to spend more amount of a work input okay. So as a result obviously the COP of the system decreases. So you find that the COP of the standard cycle is always less than the COP of the Carnot cycle okay. For you can write this expression.

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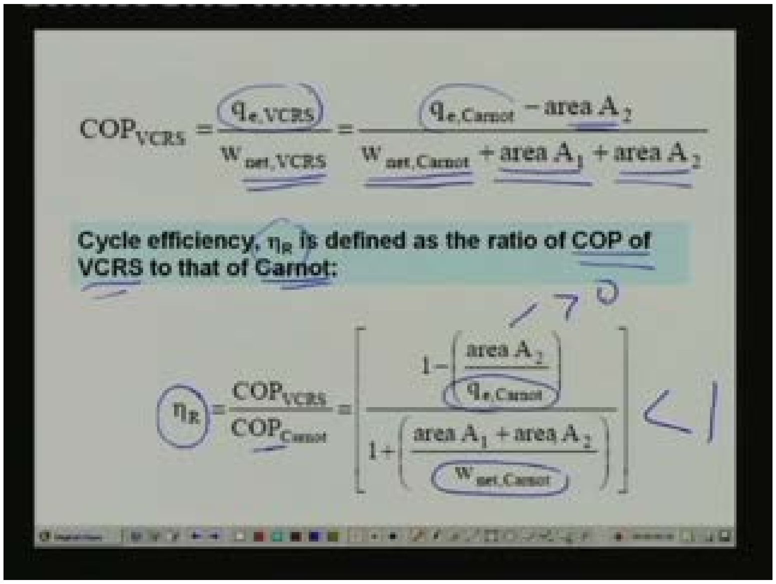

$$COP_{VCRS} = \frac{q_{e,VCRS}}{w_{net,VCRS}} = \frac{q_{e,Carnot} - \text{area } A_2}{w_{net,Carnot} + \text{area } A_1 + \text{area } A_2}$$

Cycle efficiency, η_R is defined as the ratio of COP of VCRS to that of Carnot:

$$\eta_R = \frac{COP_{VCRS}}{COP_{Carnot}} = \left[\frac{1 - \overset{> 0}{\left(\frac{\text{area } A_2}{q_{e,Carnot}}\right)}}{1 + \left(\frac{\text{area } A_1 + \text{area } A_2}{w_{net,Carnot}}\right)}\right] < 1$$

For example COP of the vapour compression refrigeration cycle is defined as the refrigeration effect of the vapour compression refrigeration cycle divided by net work input in the vapour compression refrigeration cycle. And the q e vapour compression refrigeration cycle is written as q e Carnot minus throttling area A two and work input of the standard cycle is given in a it can be written as work input to the Carnot cycle plus super heat horn area A one plus throttling area A two okay. Now we can design what you known as a cycle efficiency eta subscript R this is defined as a ratio of COP of standard vapour compression cycle to that of Carnot COP right. That means cycle efficiency eta R is given by COP of standard vapour compression cycle divided by Carnot COP.

So that can be shown to be equal to here if you take q e Carnot by w Carnot is common then this can be written as one minus area A two divided by q e Carnot divided by one plus area A one plus area A two divided by w net of Carnot. You find that this is always less than one because area A two is always is greater than zero and area A one and area A two will all also be greater than zero okay. So as a result cycle efficiency will always be less than one.

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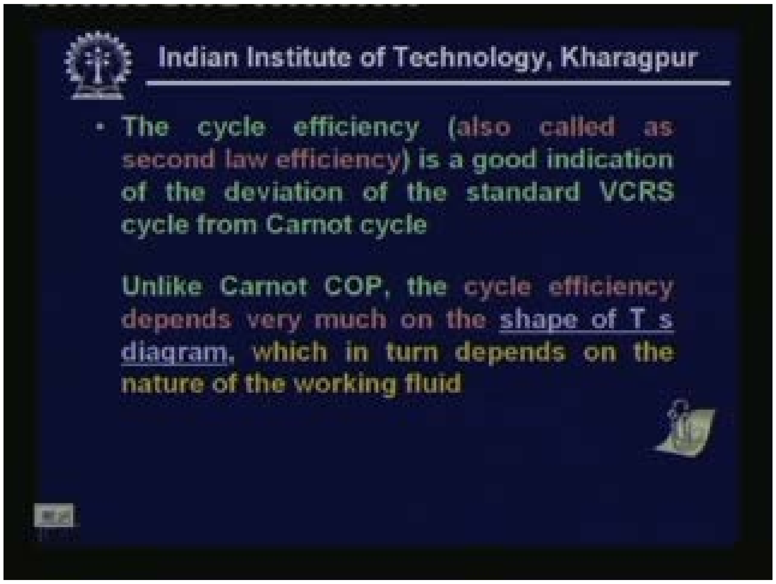

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- The cycle efficiency (also called as second law efficiency) is a good indication of the deviation of the standard VCRS cycle from Carnot cycle

  Unlike Carnot COP, the cycle efficiency depends very much on the shape of T s diagram, which in turn depends on the nature of the working fluid

So the cycle efficiency or sometimes it is also called as second law efficiency. This is a good indication of the deviation of the standard vapour compression refrigeration system from Carnot cycle unlike Carnot COP the cycle efficiency depends very much on the shape of temperature entropy diagram which in turn depends on the nature of the working fluid. We have seen that the COP of the Carnot system is a function of the heat source and sink temperatures only and it is independent of the nature of the working fluid okay. But you find that the COP of a standard vapour compression refrigeration cycle also depends upon the shape of the vapour dome on T s diagram.

That means which in turn depends upon the nature of the working fluid okay, how is it let me explain that with the help of the T s diagram.

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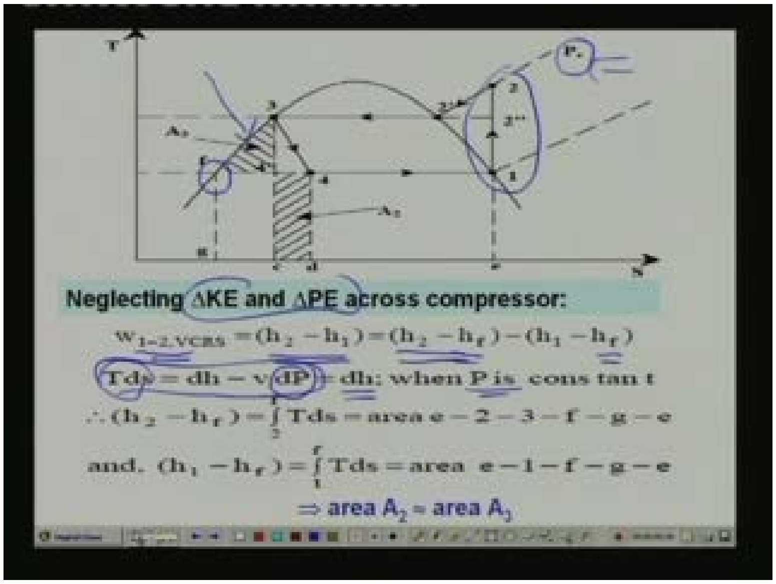

Neglecting ΔKE and ΔPE across compressor:

$$w_{1-2,VCRS} = (h_2 - h_1) = (h_2 - h_f) - (h_1 - h_f)$$

$$Tds = dh - v\,dP = dh;\ \text{when } P \text{ is constant}$$

$$\therefore (h_2 - h_f) = \int_2^f Tds = \text{area } e-2-3-f-g-e$$

$$\text{and. } (h_1 - h_f) = \int_1^f Tds = \text{area } e-1-f-g-e$$

$$\Rightarrow \text{area } A_2 \approx \text{area } A_3$$

Okay, so as I have already explained to you here the standard vapour compression refrigeration cycle is given by one two one two three four okay. And neglecting kinetic and potential energy changes across the compressor compression process is one to two okay. So if you apply energy balance for the compression process you find that the work input to the compressor w one to two of the vapour compression refrigeration cycle is nothing but h two minus h one. That means the enthalpy different across the compressor which is nothing but work of compression okay. So what it do here is I subtract and add h f where h f is nothing but the saturated liquid enthalpy at evaporated temperature okay. So I am writing h two minus h one as h two minus h f minus h one minus h f okay.

Now I make one small assumption here I assume that the saturated liquid line okay. This is the saturated liquid line this coincides with the isobar P c okay. That means if I extend the isobar in the liquid region I assume that this isobar coincides with the saturated liquid line. This is the reasonably good assumption because if you look at the actual T s diagram you find that the isobars are very close to the saturated liquid lines okay. That means the process two to f is an isobaric process at pressure P c okay. And using the thermodynamic relations T d s is equal to d h minus v d P and if you apply this thermodynamic relation this two the process two to f you will find that this is equal toT d d h because d P is equal to zero because this is an isobaric process. So T d s is equal to d h right then P is equal to constant so what is d h here d h is nothing but h two minus h f right so h two minus h f that means this quantity is equal to integral T d s okay, integral T d s is nothing but area under.

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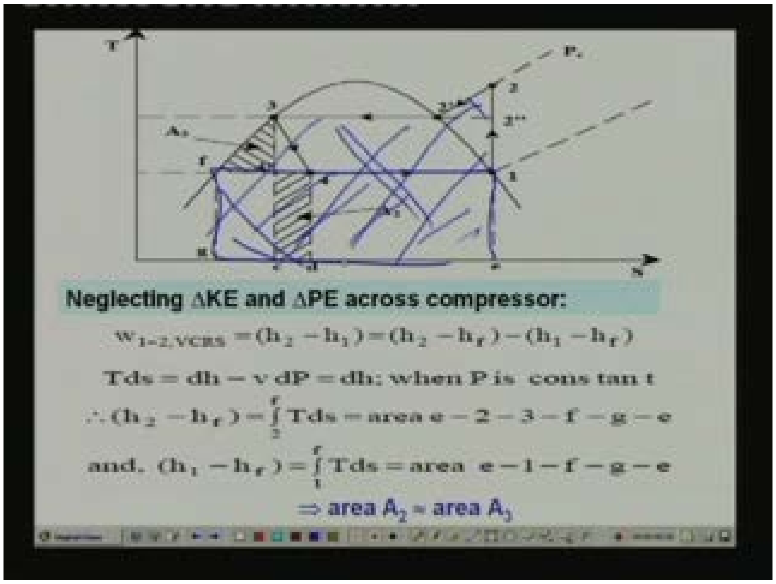

Neglecting ΔKE and ΔPE across compressor:

$$w_{1-2,VCRS} = (h_2 - h_1) = (h_2 - h_f) - (h_1 - h_f)$$

$$Tds = dh - v\,dP = dh;\ \text{when } P \text{ is constant}$$

$$\therefore (h_2 - h_f) = \int_2^f Tds = \text{area } e-2-3-f-g-e$$

$$\text{and. } (h_1 - h_f) = \int_1^f Tds = \text{area } e-1-f-g-e$$

$$\Rightarrow \text{area } A_2 = \text{area } A_3$$

The process two to f that means this entire area okay this entire area right that is equal to h two minus h f. Now what is h one minus h f h one minus h f is nothing but area under the curve one to f okay one to f is an isobaric process. So this is nothing but again h one minus h f is equal to integral T d s temperature remains constant. So h one minus h f is nothing but area under curve one to f on T s diagram which is equal to this rectangular area okay, this rectangular area right. So you find that under this assumption h two minus h one which is equal to h two minus h f minus h one minus h f is nothing but this area okay, right.

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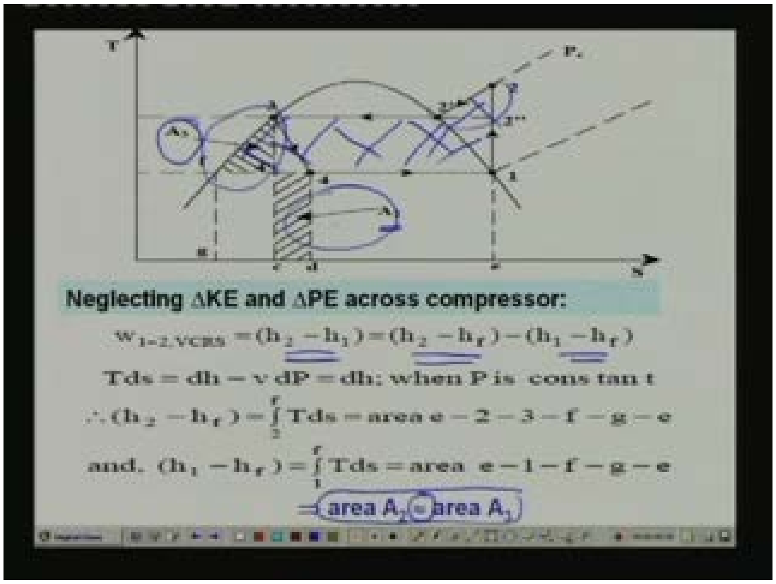

Neglecting ΔKE and ΔPE across compressor:

$$w_{1-2,VCRS} = (h_2 - h_1) = (h_2 - h_f) - (h_1 - h_f)$$

$$Tds = dh - v\,dP = dh;\ \text{when P is constant}$$

$$\therefore (h_2 - h_f) = \int_2^f Tds = \text{area } e-2-3-f-g-e$$

$$\text{and, } (h_1 - h_f) = \int_1^f Tds = \text{area } e-1-f-g-e$$

$$\Rightarrow \text{area } A_2 \approx \text{area } A_3$$

So this area is nothing but the rectangular area plus the superheat horn area plus this area A three okay. And in the previous slides we have seen that for vapour compression refrigeration cycle the net work input is equal to this rectangular area plus this superheat horn area plus the throttling area A two. So if you compare that one with this area which both of them have got to be equal this will show that area A two is equal to area A three okay. The area A two is approximately equal to Area A three approximately equal to area A three. Because here I am making an assumption that the isobar coincides with the saturated liquid line okay, now this area A three depends very much on the shape of the T s diagram okay.

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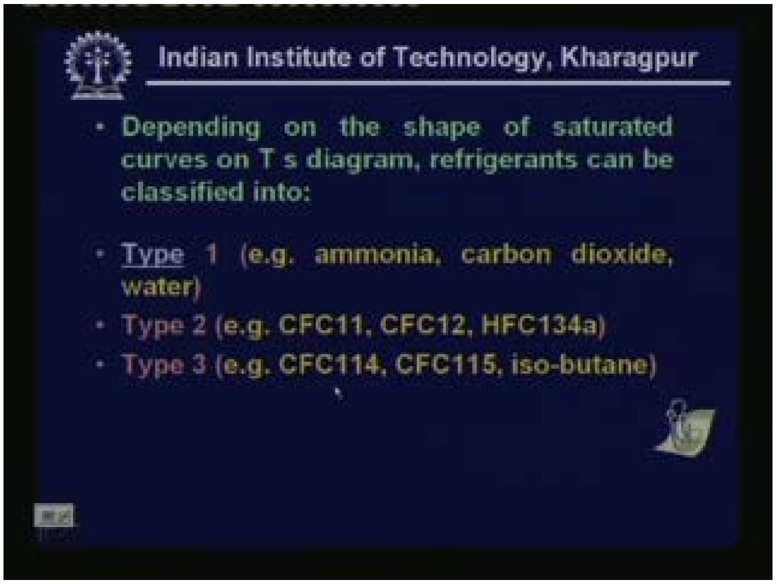

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- Depending on the shape of saturated curves on T s diagram, refrigerants can be classified into:
- Type 1 (e.g. ammonia, carbon dioxide, water)
- Type 2 (e.g. CFC11, CFC12, HFC134a)
- Type 3 (e.g. CFC114, CFC115, iso-butane)

Now depending upon the shape of saturated curves in T s diagram refrigerants can be classified into type one refrigerants. That means ammonia carbon dioxide and water type two refrigerants. For example CFC eleven CFC twelve HFC one thirty-four and type three refrigerants that is CFC one one four CFC one one five isobutane okay. So let me show these types on T s diagram and what is the meaning of that.

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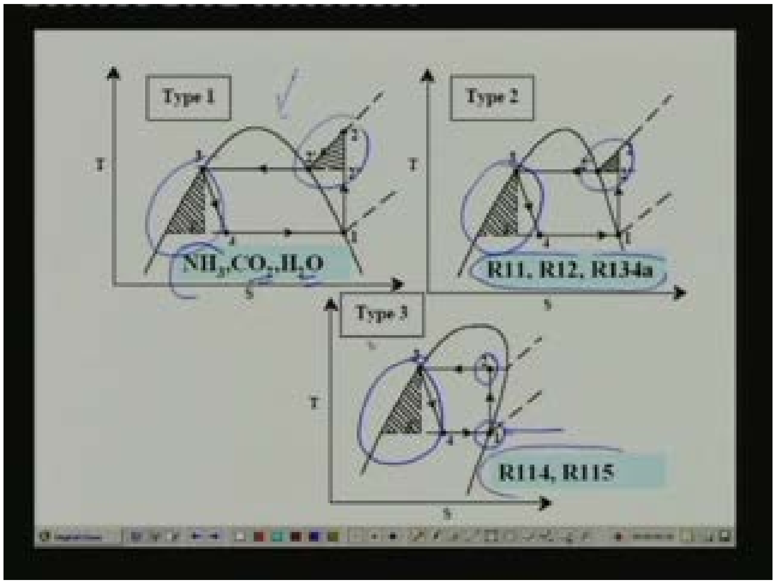

You find that type one refrigerant has a T s vapour dome like this even you can see that the vapour dome is almost symmetrical okay. And the type one refrigerants are ammonia carbon dioxide and water etcetera. Since the vapour dome is symmetric you find that the losses due to superheat horn and losses due to throttling both are of the almost same magnitude that means both are important right for type one refrigerants. Now type two refrigerants that means R eleven R twelve and R one thirty-four a you find that the loss due to superheat horn is much less compare to the loss due to throttling okay, because this area is much less compared to this area. Now type three that means refrigerants high molecular weight refrigerants such as R one one four and R one one five have a peculiar vapour dome and here you find that there is no superheat horn at all because the slope of the vapour saturated vapour curve is such that when you start compression with a saturated vapour the exit of the compression lies in the two phase region okay.

So they, that means there is no superheat horn that means there is superheat loss due to superheat horn is non existence for this type of refrigerants whereas the loss due to throttling is quite considerable okay. So depending upon the type of the refrigerants and depending upon the vapour dome shape on T s diagram you can classify the refrigerants and what is the use of this classification you know where you are losing okay, and if you want to do some modifications you know where you have to concentrate okay.

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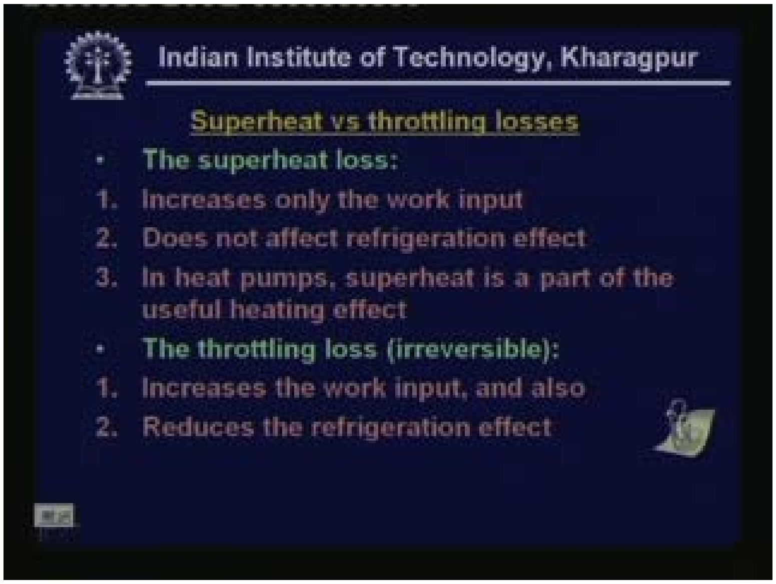

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Superheat vs throttling losses

- The superheat loss:
  1. Increases only the work input
  2. Does not affect refrigeration effect
  3. In heat pumps, superheat is a part of the useful heating effect
- The throttling loss (irreversible):
  1. Increases the work input, and also
  2. Reduces the refrigeration effect

Now let us quickly look at the differences between superheat and throttling losses. I have mentioned that because you have changed over from Carnot cycle to vapour compression refrigeration system because of some practical difficulties two additional losses have been introduced. And for the first loss is because of the superheat and the second loss is because of the throttling loss. Now let us look at these losses okay if you look at the superheat loss the superheat loss increases only the work input it does not effect the refrigeration effect. And as I said it does not effect refrigeration effect and in heat pumps superheat is a part of the useful heating effect.

That means if you are using the refrigeration system for heat pumping. That means if you are interested in the heat rejected in the condenser then you find that the heat loss in the superheat due to superheat horn is not a loss at all because that can be recovered okay. Now if you look at the throttling loss the throttling process is highly irreversible and you find that in the because of the throttling process the work input increases and also the refrigeration effect is reduced okay. So if you look at the expression for COP the numerator reduces and the denominator increases okay, so you loose on both sides. So in this aspect throttling loss is more serious compare to or it has more significant effect compare to the superheat loss okay.

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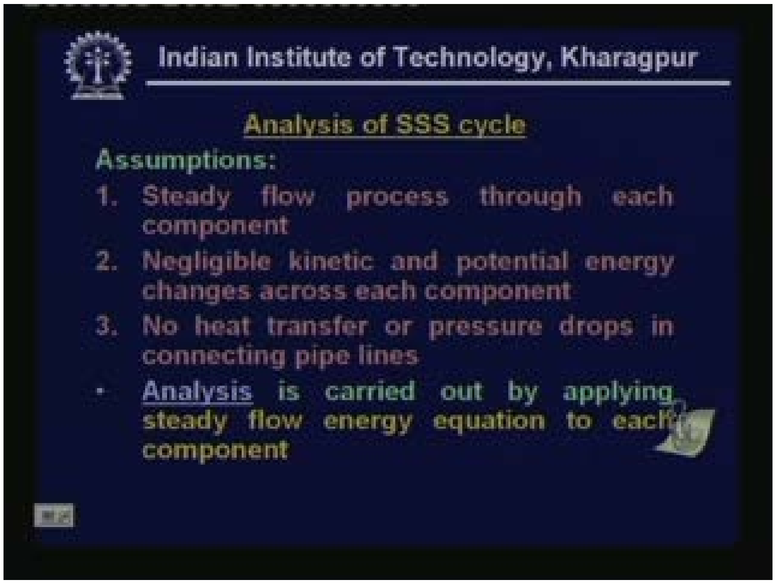

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Analysis of SSS cycle

Assumptions:

1. Steady flow process through each component
2. Negligible kinetic and potential energy changes across each component
3. No heat transfer or pressure drops in connecting pipe lines

- Analysis is carried out by applying steady flow energy equation to each component

Now let us look at a simple analysis of standard vapour compression refrigeration cycle. And this analysis carried out under the following assumptions we assume that the all the processes taking place in each component is a steady flow process and we neglect kinetic and potential energy changes across each component. And there are no heat transfer or pressure drops in connecting pipe lines and the analysis carried out by applying steady flow energy equation to each component. So let me show the analysis for each component.

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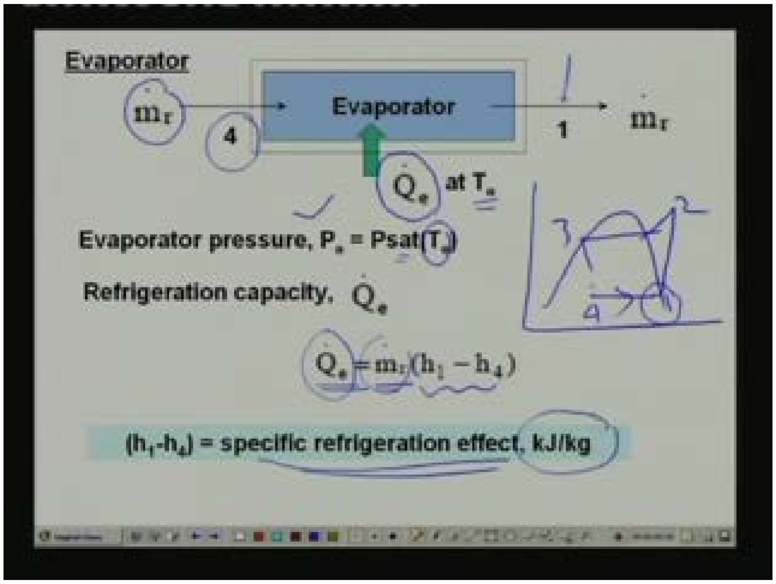

Evaporator

Evaporator pressure, $P_e = Psat(T_e)$

Refrigeration capacity, $\dot{Q}_e$

$$\dot{Q}_e = \dot{m}_r(h_1 - h_4)$$

$(h_1\text{-}h_4)$ = specific refrigeration effect, kJ/kg

First let us take the evaporator remember that the evaporator process is given by process four to one okay. So if you look at the T s diagram okay, so this is one two three four so process four to one so at the inlet to the evaporator you have a mixture of liquid and vapour and at the exit of the evaporator you have saturated vapour okay. And m dot r is mass flow rate of the refrigerant and q dot e is the refrigeration capacity of the evaporator. So if you take the control volume across the evaporator and if you apply the steady flow energy equation since we are neglecting the kinetic and potential energy changes you find that the refrigeration capacity q dot e is nothing but mass flow rate of the refrigerant multiplied by the enthalpy difference h one minus h four okay. And the pressure at which the evaporator operates is nothing but the saturated pressure corresponding to the evaporator temperature T e okay.

So if you know the saturation pressure temperature characteristics and if you know the operated temperature you can calculate what is the evaporator pressure okay. And you can if you know mass flow rate and enthalpy difference you can also calculate what is the refrigeration capacity. Now this parameter h one minus h four is known as specific refrigeration effect and the units are kilo joules per kg okay.

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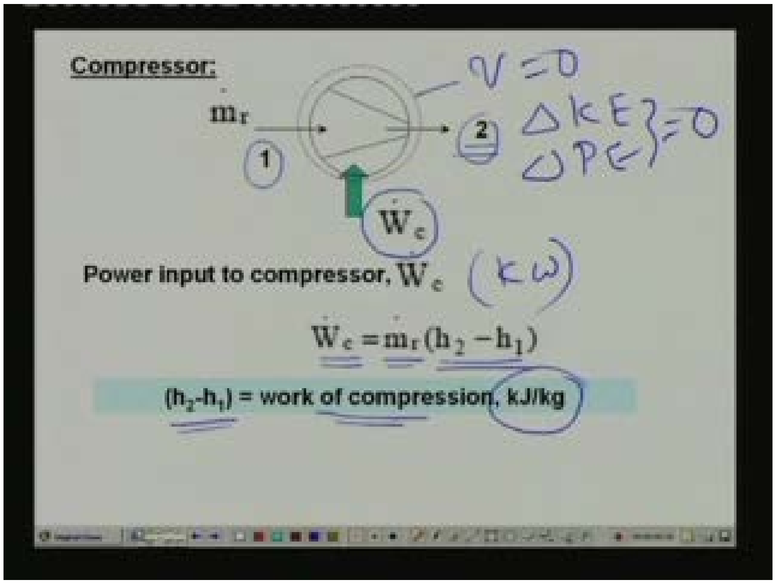

Compressor:

Power input to compressor, $\dot{W}_c$ (kW)

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

$(h_2\text{-}h_1)$ = work of compression, kJ/kg

Now if you apply the steady flow energy equation to the compressor remember that we are assuming the compression process to be isentropic so there is no heat transfer q is zero okay. Again we are neglecting delta K E and delta P E both are negligible okay. And refrigerant enters at state one at evaporated pressure and temperature and it exists at state two which is at condenser pressure and at a discharge temperature T two okay and W dot c is the power input to the compressor. So if you again apply the steady flow energy equation you find that the power input to the compressor W dot c is nothing but m dot r into h two minus h one where h two minus h one is the enthalpy rise across the compressor. And this h two minus h one is known as work of compression and it has units kilo joules per k g where is the power input as you know is in the units of kilo watts okay.

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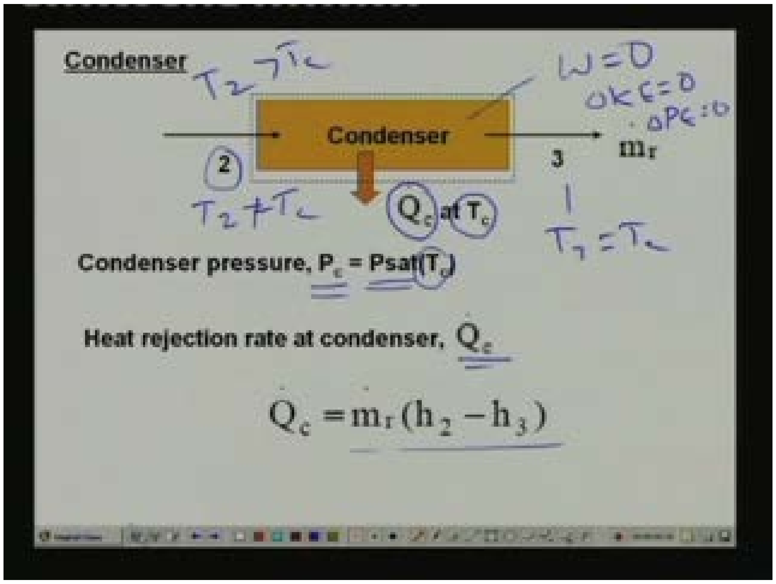

Condenser

Condenser pressure, $P_c = \mathrm{Psat}(T_c)$

Heat rejection rate at condenser, $\dot{Q}_c$

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

Similar to evaporator you can also take control volume across the condenser again there is no work transfer here W is zero and delta K E and delta P E are negligible delta K E is zero and delta P E is zero okay. And Q dot c is the heat rejection rate from the condenser and heat rejection is taking place at a condensing temperature of T c okay. And the condenser pressure P c is nothing but the saturation pressure P sat corresponding to this temperature T c. One thing you must keep in mind is that temperature T two is not equal to T c where as temperature T three is equal to T c okay so this temperature T two is always greater than T c. Because of the desuper heating in the initial portion okay. So you can find out the condenser pressure if you know the saturation pressure temperature characteristics and the heat rejection rate Q dot c can be obtained by this equation this is from the steady flow energy equation Q dot c is nothing but m dot r into h two minus h three.

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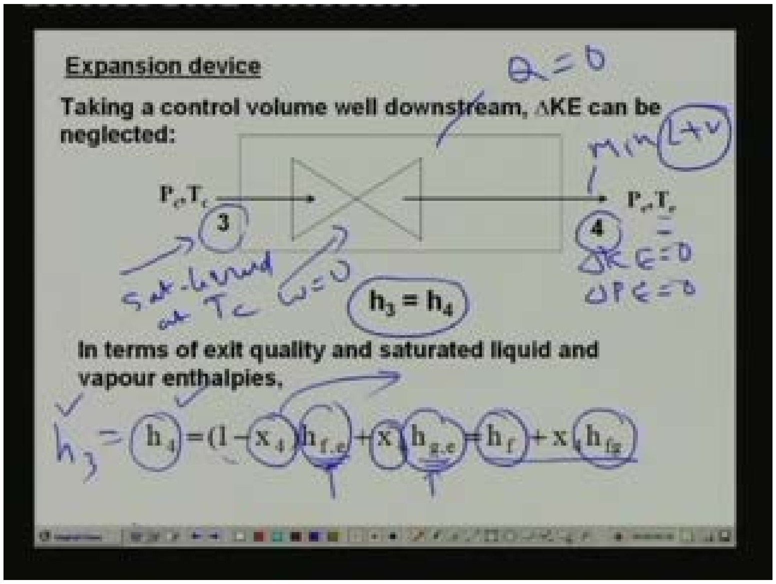

Expansion device

Taking a control volume well downstream, ΔKE can be neglected:

$h_3 = h_4$

In terms of exit quality and saturated liquid and vapour enthalpies,

$$h_3 = h_4 = (1 - x_4)h_{f,e} + x_4 h_{g,e} = h_f + x_4 h_{fg}$$

Now if you take the expansion device expansion process is throttling process. That means an isenthalpic process and if you take the control volume for downstream of the expansion device then delta K E will be zero and delta P E also will be zero. And there is no work interaction W is zero and there is no heat interaction Q is zero so you find that from steady flow energy equation h three is equal to h four or this process is isenthalpic process. Now what is the state of the refrigerant at the inlet to the throttling device this is saturated liquid right saturated liquid at condenser temperature whereas state four if you look at the T s diagram is a mixture of liquid and vapour okay.

So this is a mixture of liquid and mixture of liquid plus vapour at condenser temperature at evaporated temperature and pressure okay. So you can write the enthalpy at h four in terms of the saturated liquid enthalpy at evaporated temperature and saturated vapour enthalpy at evaporated temperature and the quality or dryness fraction x four okay. So h four is nothing but one minus x four into h f e plus x four into h e this can also be written as h f plus x four h f g where h f is nothing but h f e that is the saturated liquid enthalpy and h f g is nothing but latent heat of vaporization at evaporated temperature okay. And h four is equal to h three right. So h three is known because this is nothing but the saturated liquid enthalpy. So h four is also known because h three is equal to h four and h f e and h g e are nothing but saturated liquid and vapour enthalpies at evaporated temperatures so they can be obtained from the properties. So using this expression we can find out what is the quality of the refrigerant at the exit of the expansion device okay, so that is the use of this equation.

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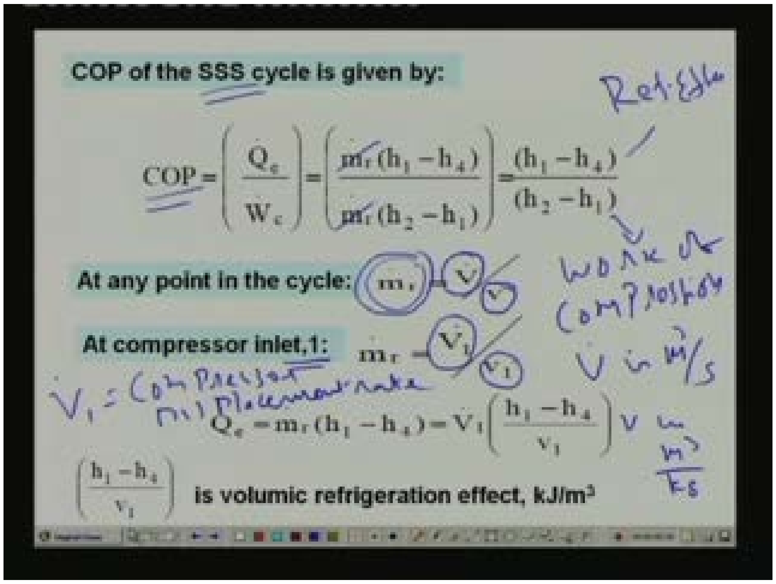

COP of the SSS cycle is given by:

Ref. Effe

$$COP = \left(\frac{\dot{Q}_e}{\dot{W}_c}\right) = \left(\frac{\cancel{\dot{m}_r}(h_1 - h_4)}{\cancel{\dot{m}_r}(h_2 - h_1)}\right) = \frac{(h_1 - h_4)}{(h_2 - h_1)}$$

work of compression

At any point in the cycle: $\dot{m}_r = \dot{V}/v$

$\dot{V}$ in m³/s

At compressor inlet,1: $\dot{m}_r = \dot{V}_1/v_1$

$\dot{V}_1$ = compressor displacement rate

$$\dot{Q}_e = \dot{m}_r(h_1 - h_4) = \dot{V}_1\left(\frac{h_1 - h_4}{v_1}\right)$$

v in m³/kg

$\left(\frac{h_1 - h_4}{v_1}\right)$ **is volumic refrigeration effect, kJ/m³**

So finally the COP of the SSS cycle is given by Q dot e divided by W dot c right. So Q dot e is nothing but the refrigeration capacity which is equal to m dot r into h one minus h four divided by m dot r into h two minus h one. So this m dot r get cancelled so finally you find that COP of the standard saturated cycle is given by h one minus h four divided by h two minus h one where h one minus h four is nothing but your refrigeration effect and this is nothing but your work of compression okay. So finally COP is expressed in terms of enthalpies only right and at any point in cycle the mass flow rate m dot r can be written in terms of volumetric flow rate V dot V dot is volumetric flow rate in meter cube per second okay. And v small v is a specific volume in meter cube per kg okay so V dot divided by v becomes kg per second which is nothing but the mass flow rate of the refrigerant.

Now if you apply this equation to compressor inlet one then m dot r is equal to V one dot divided by v one where V one dot is nothing but the compressor displacement rate. That means V one dot is compressor displacement rate right this is again meter cube per second and v one is nothing but the specific volume of the refrigerant at the compressor inlet okay.

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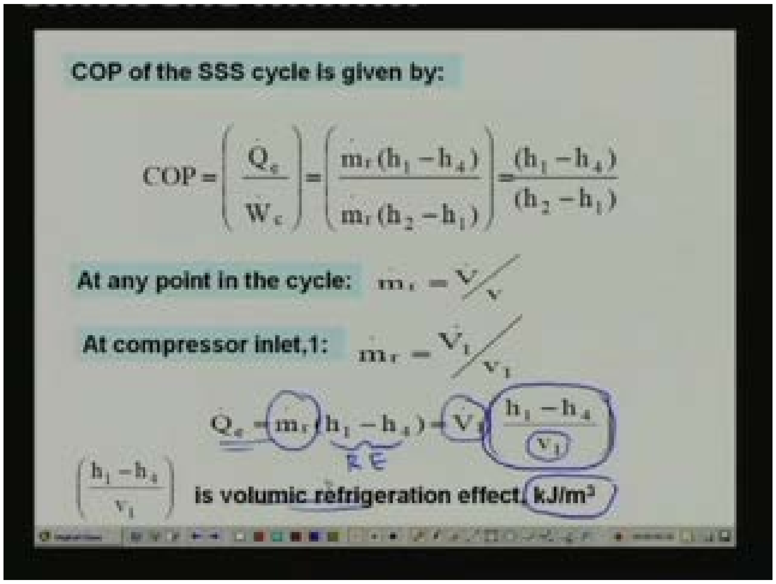

COP of the SSS cycle is given by:

$$COP = \left(\frac{\dot{Q}_e}{\dot{W}_c}\right) = \left(\frac{\dot{m}_r(h_1-h_4)}{\dot{m}_r(h_2-h_1)}\right) = \frac{(h_1-h_4)}{(h_2-h_1)}$$

At any point in the cycle: $\dot{m}_r = \dot{V}/v$

At compressor inlet,1: $\dot{m}_r = \dot{V}_1/v_1$

$$\underline{\dot{Q}_e} = \dot{m}_r \underbrace{(h_1-h_4)}_{RE} = \dot{V}_1 \boxed{\frac{h_1-h_4}{v_1}}$$

$\left(\frac{h_1-h_4}{v_1}\right)$ is volumic refrigeration effect, kJ/m^3

Now what we do is we write the refrigeration capacity which is equal to mass flow rate into h one minus h four which is nothing but your refrigeration effect right. And the mass flow rate is written in terms of the volumetric flow rate at the compressor inlet V dot one and the specific volume of the refrigerant at the compressor inlet v one okay so Q dot e is written as V V dot one into h one minus h four divided by v one. Now this parameter h one minus h four divided by v one has the units of kilo joules per meter cube okay. And this parameter is called as volumic refrigeration effect okay and the units are kilo joules per meter cube so what is the practical significance of this volumic refrigeration effect.

This is an indication of the size of the compressor okay, so the higher volumetric refrigeration effect smaller will be for a given capacity higher the volumetric refrigeration effect smaller will be the required displacement rate of the compressor okay. So this is an important performance parameter. Similarly the refrigeration effect which has unit of kilo joules per kg is an indication of the required mass flow rate okay, because for a given capacity mass flow rate is nothing but Q dot e divided by refrigeration effect so higher the refrigeration effect smaller will be the mass flow rate of the refrigerant okay.

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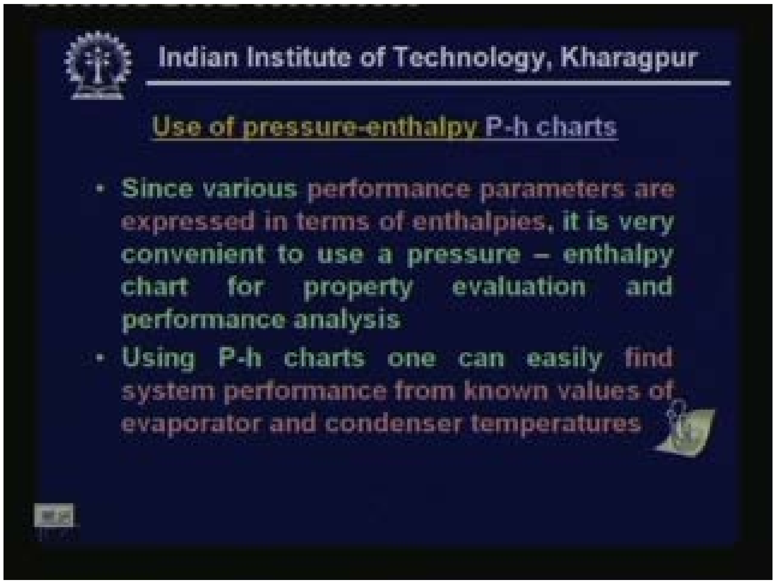

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Use of pressure-enthalpy P-h charts

- Since various performance parameters are expressed in terms of enthalpies, it is very convenient to use a pressure – enthalpy chart for property evaluation and performance analysis
- Using P-h charts one can easily find system performance from known values of evaporator and condenser temperatures

Now we have seen that of most of the properties that means most of the performance parameters like refrigeration effect work of compression COP volumic refrigeration effect there all expressed in terms of enthalpies. So it is really useful to use a pressure enthalpy chart in place of a temperature entropy chart okay. This P h diagram is sometimes known as mollier diagram so this is very widely used refrigeration cycle analysis because using this cycle you can straight away calculate the required performance parameters okay. So that is what is mentioned here since various performance parameters are expressed in terms of enthalpies it is very convenient to use in pressure enthalpy chart for property evaluation and performance analysis using P h chart one can easily find system performance from known values of evaporator and condenser temperatures. Let me show this P h chart.

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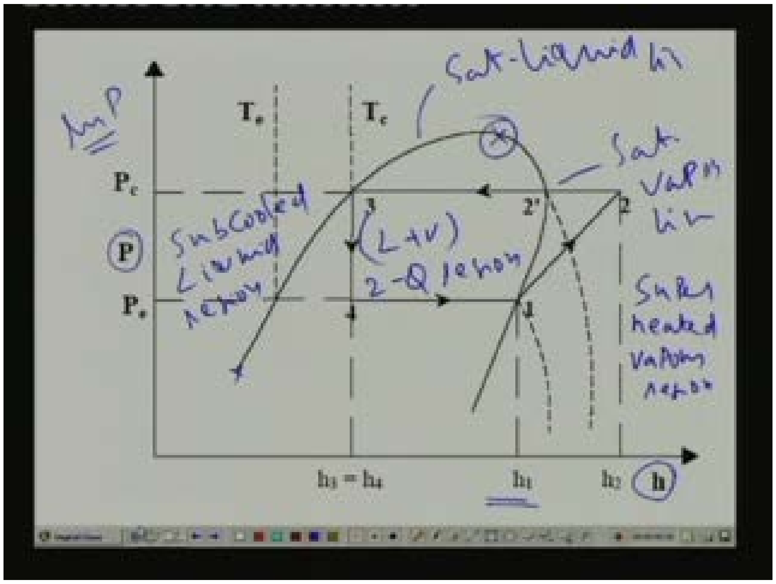

Okay this is a typical P h chart okay. So here the x axis is enthalpy and y axis is pressure okay. Normally the pressure is drawn on a log semi log plot this is drawn on semi log plot so that the, at the lower temperatures this plot opens up okay. So generally this is drawn on log scale whereas the enthalpy is on a linear scale okay. So this is normally l n P versus this thing and this is the saturation this is a vapour dome okay. You have the critical point here and from this point to this point is your saturation liquid line okay and from this point to this point is your saturation vapour line okay. So this is your sub cooled liquid region okay and this is your liquid plus vapour two phase region okay, and this is your superheated vapour region okay.

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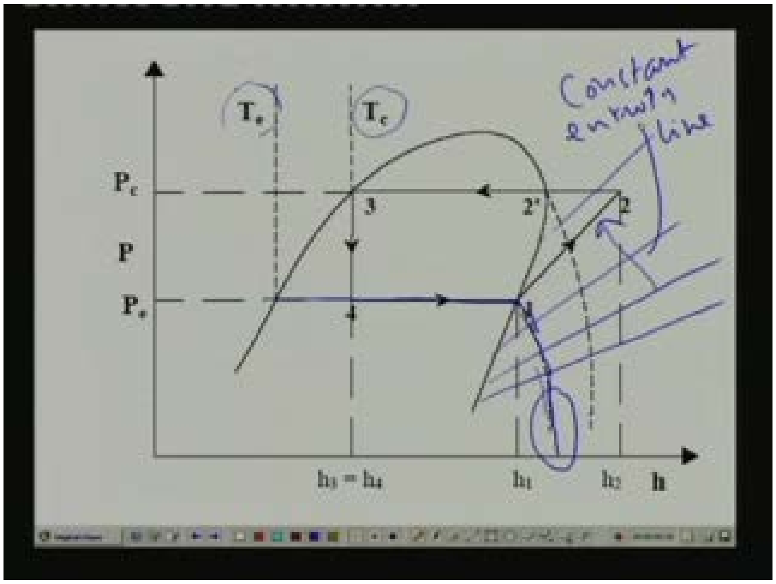

Now on the P h diagram obviously the enthalpy constant enthalpy lines are vertical lines and constant pressure lines are horizontal lines okay. And the isotherms for example isotherm at evaporated temperature it is almost vertical in the sub cooled liquid region and it becomes horizontal in the two phase region and again it becomes inclined okay and it goes like this and it again becomes almost vertical at low pressure region because the pressure is reducing in this direction when the pressure is low you find that the isotherm shape of the isotherm in the super heated region becomes almost vertical. Why is it so?

Because when the pressure becomes very small the vapour starts behaving as a, an ideal gas and we know that for an ideal gas the enthalpy is the function of temperature only okay. So if you are fixing the temperature enthalpy also gets fixed okay. So if it is a constant temperature line then it also has got to be constant enthalpy line as a result at low pressure region the isotherm becomes almost vertical okay. So this is isotherm for evaporator temperature and this is the isotherm corresponding to the condenser temperature okay. And this is an isentrope, that means this is a constant entropy line okay in fact you will find that the constant entropy lines will be varying like this okay. These varying lines they will be diverging right.

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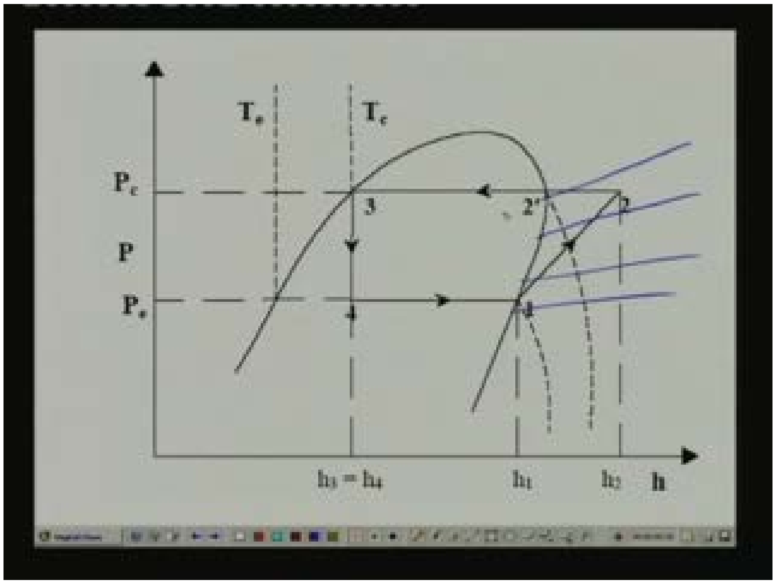

You can also have constant specific volume lines on this constant specific volume lines will be something like this okay. If you look at an actual P h diagram constant specific volume lines and constant entropy lines in the super heated region are also shown in addition to the isotherms in the liquid two phase and super heated regions okay.

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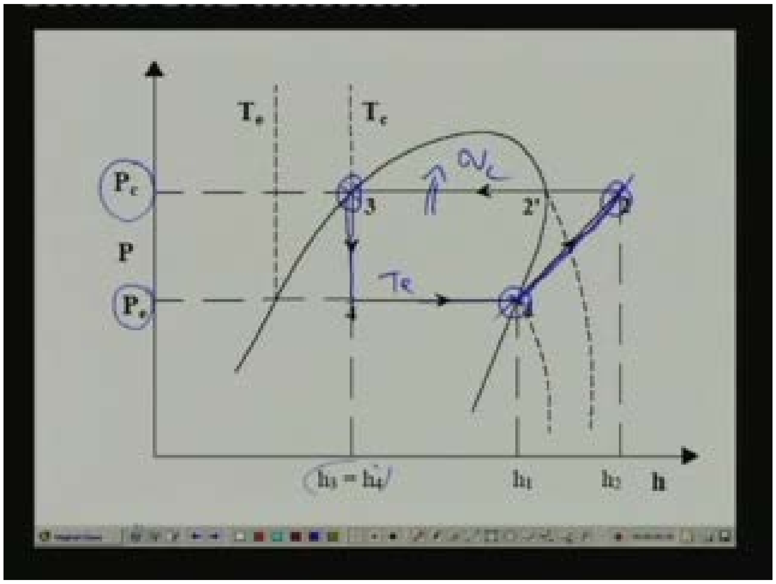

Now how do we represent the standard vapour compression cycle on P h diagram we know that for example if you look at all the four processes beginning with the compression process. During the compression process saturated vapour is compressed from evaporator pressure to condenser pressure and this process is isentropic process okay. So if you know the, this temperature then you can find out this pressure okay and you can locate this point right. And you can also find the entropy at this point since compression process is isentropic you have to go along the constant isentropic, isentrope right. Along the constant isentrope and where the constant isentropic line intersects the constant condenser pressure line that is your exit of the compressor okay. So you can draw this process right.

Now process two to three is nothing but heat rejection in the condenser and this is we know that is an isobaric process. So all that you have to do is you draw a horizontal line start beginning with point two okay. Where it intersects the saturated liquid line that is the exit of your condenser, okay. So the process two to three is nothing but a horizontal line on P h diagram and process three to four is nothing but isenthalpic throttling process during this process pressure drops from P c to P e and the enthalpy remains constant. So this process is nothing but a vertical line on P h diagram okay. And finally the evaporation process the heat extraction process is isobaric. So it is a constant, it is a horizontal line on P h diagram so the process four to one is given by the horizontal line okay.

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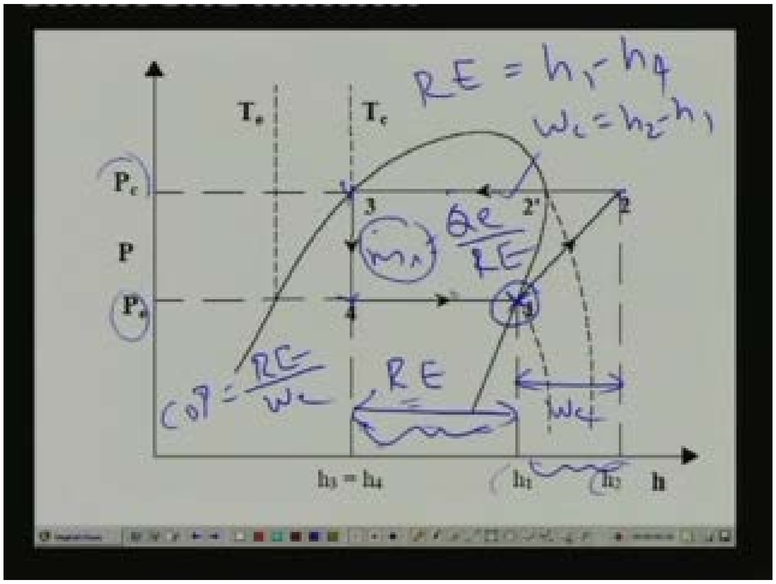

So finally you find that the vapour compression refrigeration cycle takes the shape one two three four okay. So you can the once you know the condenser pressure and condenser evaporator pressure or condenser temperature and evaporator temperature you can easily locate this points right and you can draw the cycle on P h diagram. Once you have drawn the cycle on the P h diagram the calculation become very simple. For example I want to calculate I want to find out what is the refrigeration effect refrigeration effect as you know R E is the refrigeration effect in nothing but h one minus h four okay. This is nothing but this enthalpy minus this enthalpy or this is nothing but this okay this is your refrigeration effect okay. And what is the work of compression work of compression is nothing but h two minus h one that is nothing but this enthalpy minus this enthalpy that is nothing but this so this is your work of compression okay.

So the movement you locate all the points you can straight away read the enthalpy values and you can get the refrigeration effect and work of compression.

So the COP as you know is nothing but R E divided by W c that means this length divided by this length okay. So straight away you get the value of COP and you can also find out required mass flow rate etcetera. If you know the capacity because mass flow rate is nothing but refrigeration capacity divided by R E and R E is nothing but this length refrigeration capacity is given so you can find out what is mass flow rate. And if you read the specific volume from the P h chart then you can also find out what is the required volumetric displacement at this point okay.

So like that using the P h diagram you can find all the required performance parameters okay. So that is the reasons why we use this P h chart vary widely in refrigeration cycle analysis okay.

So in next lectures we will be using this to evaluate the performance of standard vapour compression refrigeration cycles okay. And as I have already mentioned this P h chart it is sometimes known as mollier diagram okay.

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Conclusions

In this lesson:

1. Carnot refrigeration cycle and its practical limitations are discussed
2. The standard vapour compression refrigeration system (VCRS) is introduced
3. VCRS is compared with Carnot system
4. Performance analysis of VCRS is presented

- The performance of standard VCRS and modified VCRS will be discussed in next lessons

Okay, so let me conclude this lesson what is that we have learn learned in this lecture in this lesson the Carnot refrigeration cycle and its practical limitations are discussed. The standard vapour compression refrigeration system is introduced and its working principles explained and we have compared the vapour compression refrigeration system with Carnot system and we have seen what the deviations are and what the consequences of these deviations are. And we have presented a simple performance analysis of a standard vapour compression refrigeration system by applying steady flow energy equation to each component and for we have also presented the cycle on a P h diagram. And I have also explained how to evaluate the required performance parameters using the P h diagrams from the known values of evaporator and condenser temperatures okay. So performance of standard vapour compression refrigeration system and modified vapour compression system will be discussed in next lectures okay.

Thank you.