NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

INDEX

2 - Science Fundamentals - T.T.1

5 - HEAT AND THERMODYNAMICS

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

2 - Science Fundamentals - T.T.1

5 - Heat and Thermodynamics

-1 - Carnot Cycle

0.0 INTRODUCTION

The Carnot cycle was devised and analyzed by a Frenchman named Carnot when he was 23 or 24 years old. It is the most efficient cycle conceivable. There are other ideal cycles as efficient as the Carnot cycle, but none more efficient. As will become apparent in the lesson, this cycle is an ideal cycle, and no practical engine has ever been built to operate on the Carnot cycle. However, it forms a basis of comparison for actual engines and actual cycles. Too great a difference between the actual and ideal thermal efficiencies suggests a search for means by which the actual efficiency can be raised.

A definition of symbols used in this lesson is located on the last page.

1.0 INFORMATION

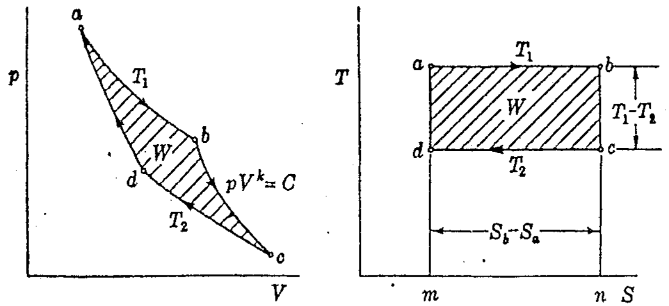
The Carnot cycle consists of two isothermal and two isentropic processes as shown on the pV and TS diagrams in Figure 1. It requires three main things to make it work:

1. a perfect gas confined in a cylinder and piston.
2. a source of energy at a high temperature.
3. a receiver of energy at a low temperature.

The processes that the substance goes through while completing a full cycle are as described below. Referring to Figure 1 and starting with the point 'a' we have:

1. An isothermal expansion a-b during which heat is added.
2. A reversible adiabatic (isentropic) expansion b-c; the power stroke.
3. An isothermal compression c-d during which heat is rejected.
4. An adiabatic (isentropic) compression d-a to the original condition.

It is apparent that to construct a practical engine to operate on this cycle, the following conditions would have to be met.



(a) For a Gas (b) For Any Substance

Let *a* be the initial point. The cycle is shown traversed in a clockwise direction, which, according to our convention of signs, is a power cycle. The net work *W* will be a positive number. Heat is added during the process *a–b*, rejected during *c–d*. It might be noted that for the state point to return to point *a*, thus completing the cycle, it *must* move toward the left in some manner after *b* on the *TS* plane is reached. Since a movement of the state point in the leftward direction on the *TS* plane indicates heat rejected, we may say now that *some heat must be rejected*, that all the heat supplied cannot be converted into work. The enclosed areas on each plane represent work. The unit of areas on the *pV* plane is ft-lb. The unit of areas on the *TS* plane is Btu. Since the Btu is 778 times larger than a ft-lb., it would be inconvenient to show these areas to scale.

Fig. 1 Carnot Cycle

1. The head of the cylinder in which expansions take place must be made alternately conducting and nonconducting, according to whether an isothermal or an adiabatic process is taking place within.

2. Provision must be made to apply the heat source, and the device for cooling alternately during the cycle.

Because of the difficulty in evolving a design to meet these conditions, no practical engine to operate on this cycle has ever been constructed. However, for comparison purposes this cycle is of considerable importance.

The Carnot cycle on the T-S plane is a rectangle and therefore its analysis is simple. The heat supplied is the area under curve a-b which is the same as the area of rectangle m-a-b-n. The mathematical expression for area m-a-b-n (which is the same as for heat added) is:

$$Q_A = T_1(s_b - s_a)$$

The heat rejected is the area under c-d on the TS diagram, m-d-c-n = Q_R

$$Q_R = T_2(s_c - s_d) = T_2(s_b - s_a)$$

Therefore the net work of the cycle is:

$$W = Q_A - Q_R = T_1(s_b - s_a) - T_2(s_b - s_a)$$

$$W = (T_1 - T_2)(s_b - s_a) \text{ .... } (1)$$

Observe that the cross-hatched area in Figure 1 (b) is equivalent to net work W. It then becomes apparent that for a cycle that is internally reversible, the area enclosed by the path of the state point on the TS plane represents the work in Btu, because the enclosed area will always be the heat added minus the heat rejected.

The thermal efficiency is:

$$e = \frac{Q_A - Q_R}{Q_A} = \frac{W}{Q_A} = \frac{(T_1 - T_2)(s_b - s_a)}{T_1(s_b - s_a)} =$$

$$e = \frac{T_1 - T_2}{T_1} \text{ .... } (2)$$

In discussing the Otto, Diesel and Brayton cycles you will have noticed that the efficiency was always dependent on the specific heat ratio k. The specific heat varies from substance to substance and therefore, the efficiency of these cycles is dependent on the substance used.

In the Carnot cycle, the efficiency is independent of the substance used, but depends on T_1 and T_2 only. The cycle is always a rectangle on the T-S diagram, no matter what substance is used. However, to draw the P-V diagram, some knowledge of the property of the substance has to be known.

We need to mention here that each process in the Carnot cycle is reversible internally and externally, and the cycle can perform either in the forward or reverse direction. This does not refer to direction of shaft rotation, but rather to the sequence in which the processes take place.

The following pages in this lesson will now deal with

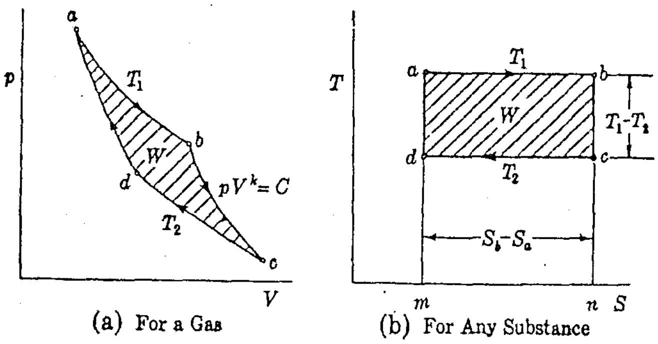
a) the Carnot cycle using a gas as the substance
b) the reversed Carnot cycle,
c) the Carnot cycle using vapor as a substance.

Carnot Cycle Using Gas

Work done in a Carnot cycle using a perfect gas can be calculated from a T-S diagram in a manner described below:

We have said in the lesson on Adiabatic and Isothermal Expansion and Compression T.T.2, that for an isothermal process:

$$W = (P_B'' \; V_B'')\ 2.3 \log_{10}\left(\frac{V_A}{V_B}\right) \text{ ft. lbs.}$$



(a) For a Gas (b) For Any Substance

Fig. 1 - Repeated

Notice that W in that case was not net work but rather total work and that if we divided the equation by J we would get the amount of heat transferred Q_A in Btu. Further, to apply it to this particular case for the isothermal expansion process, we would have to use the subscripts found in Figure 1 and get the equation:

$$Q_A = \frac{(P_aV_a)2.3}{J} \log_{10}\left(\frac{V_b}{V_a}\right) \text{ Btu.}$$

But from the characteristic gas equation, we know that:

$$P_aV_a = w\ R\ T_1$$

Substituting this into the equation for Q_A we get:

$$Q_A = \left(\frac{w\ R\ T_1}{J}\right) 2.3 \log_{10}\left(\frac{V_b}{V_a}\right) \quad \ldots\ldots\ldots\ldots(3)$$

This is the equation for heat added in the isothermal expansion process for the Carnot cycle. For the heat rejected Q_R, during the isothermal compression process c - d figure 1, we can similarly express the equation as follows:

$$Q_R = \left(\frac{w\ R\ T_2}{J}\right) 2.3 \log_{10}\left(\frac{V_d}{V_c}\right) \quad \ldots\ldots\ldots\ldots(4)$$

Then <u>net work</u> $W = \left(\frac{w\ R\ T_1}{J}\right) 2.3 \log_{10}\left(\frac{V_b}{V_a}\right) - \left(\frac{w\ R\ T_2}{J}\right) 2.3 \log_{10}\left(\frac{V_c}{V_d}\right)$

$$\ldots\ldots\ldots(5)$$

However, we know from the lesson on Heat Engine Cycles, T.T.2, that for the isentropic changes of state, b-c and d-a, in Figure 1 of this lesson we can write:

$$\frac{V_c}{V_b} = \left(\frac{T_b}{T_c}\right)^{\frac{1}{k-1}} = \left(\frac{T_1}{T_2}\right)^{\frac{1}{k-1}} \quad \text{and} \quad \frac{V_d}{V_a} = \left(\frac{T_a}{T_d}\right)^{\frac{1}{k-1}} = \left(\frac{T_1}{T_2}\right)^{\frac{1}{k-1}}$$

Since V_c/V_b and V_d/V_a are both equal to the same term, we must therefore conclude that:

$$\frac{V_c}{V_b} = \frac{V_d}{V_a}$$

or

$$\frac{V_c}{V_d} = \frac{V_b}{V_a}$$

Substituting this value of V_c/V_d in equation (5), we find:

$$W = (T_1 - T_2)\ \left(\frac{w\ R}{J}\right)\ 2.3\ \log_{10}\left(\frac{V_b}{V_a}\right)$$

which is represented by the enclosed area on the TS diagram. The thermal efficiency is:

$$e = \frac{W}{Q_A} = \frac{(T_1 - T_2)(w\ R/J)\ 2.3\ \log_{10}(V_b/V_a)}{(w\ R/J)(T_1)\ 2.3\ \log_{10}(V_b/V_a)}$$

Cancelling out equal terms in the numerator and denominator, we get:

$$e = \frac{T_1 - T_2}{T_1}$$

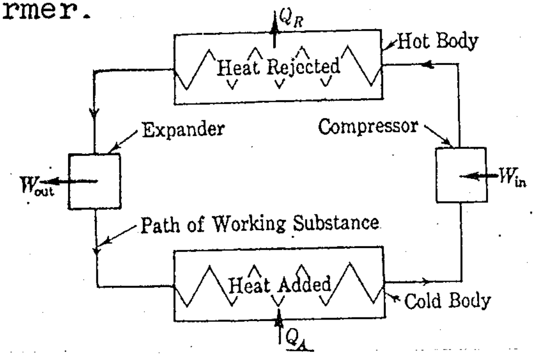
This is the expression for thermal efficiency of a Carnot cycle, using a perfect gas as the substance, which you will notice is the same as equation (2) of this lesson.

The Reversed Carnot Cycle

Normally, in describing the various changes and processes that a power cycle goes through, we move in a clockwise direction on the pV and TS diagrams. In a reversed cycle, we move in counter-clockwise direction.

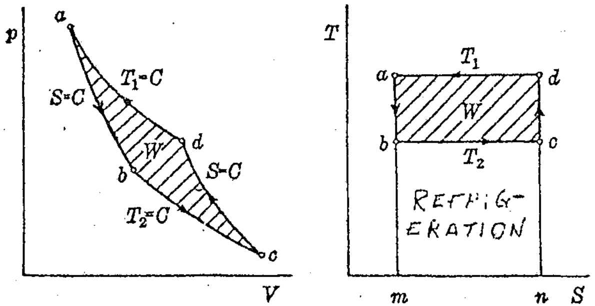
Reversed cycles are used to provide a refrigerating effect and sometimes a heating effect. In a reverse cycle, the net work is done on the substance, rather than by the substance. (See figure 2, and notice that it is the same as Figure 1, shown in the lesson on Heat Engine cycles, except that all the arrows are pointing in the reverse direction). Since each process of the cycle is reversible internally and externally, there is no reason why the cycle itself should not be performed in the opposite direction. Figure 3 shows the pV and TS diagrams for a reversed Carnot cycle, and an examination shows that for a given temperature range and a given isothermal curve a-d, the work must necessarily be the same as in the power cycle, the heat rejected by the reversed cycle, at the higher temperature must be equal to the heat added in the power cycle, and the heat added in the reversed cycle must be the same as the heat rejected in the power cycle.

A machine which operates on a reversed cycle is called a <u>heat pump</u>. It takes heat from a relatively cold body and discharges heat to a hotter body. The combined equipment in an ordinary household refrigerator can be considered a heat pump in the sense that it extracts heat from inside the refrigerator and discharges it into the room it is standing in - the refrigerator gets colder, but the room gets warmer.



This figure with certain devices not shown represents a reversed cycle. Note that the work is supplied *to* the system, that heat is rejected at the higher temperature, and that heat is added at the lower temperature. The *net* work is $W = W_{out} - W_{in} = \Sigma Q = Q_A - Q_R$, a negative number which indicates that work is done *on* the working substance.

<u>Fig. 2</u> Reversed Carnot Cycle



(a) For a Gas (b) For Any Substance

An isentropic expansion *a–b* lowers the temperature to a point where heat may be added to the substance reversibly along an isothermal *b–c*. A compressor isentropically compresses the substance along *c–d* to a temperature slightly higher than the hot body (say, $T_1 + \Delta T$), so that heat may be rejected to the hot body along *d–a*. The refrigeration is represented by the area *m–b–c–n*.

<u>Fig. 3</u> Reversed Carnot Cycle

The expression of efficiency of a reversed cycle is generally intended to mean the net work of the cycle divided by the heat transferred to the hot body, a number which is the same as that which would be obtained as the efficiency of the same cycle operating as a power cycle. However, in practice, a number called the <u>coefficient of performance</u> (C O P) is ordinarily applied to the reversed cycles to express the efficiency of operation. If the reversed cycle is used for refrigeration, the coefficient of performance is:

$$\text{C O P} = \frac{\text{Refrigeration}}{\text{Net Work}}$$

or for the reversed Carnot Cycle, Figures 2 and 3:

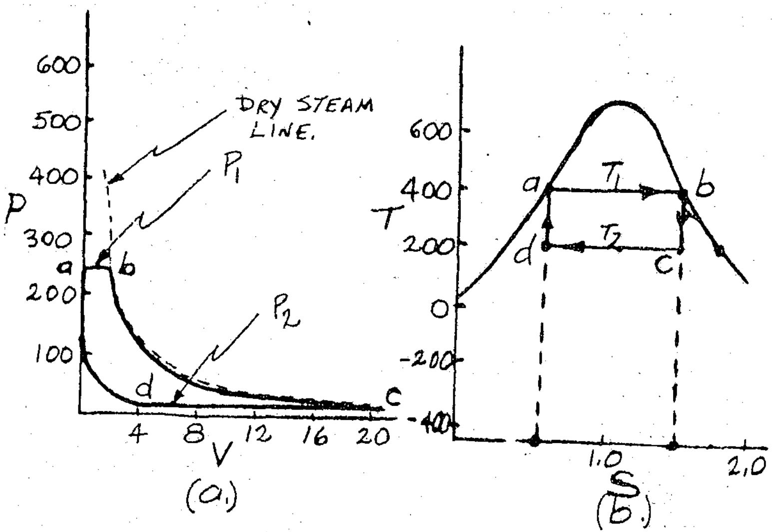
$$\text{C O P} = \frac{Q_A}{W} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (6)$$

We have previously said that the Carnot cycle is the most efficient cycle. This has been summarized in a statement called <u>Carnot's Principle</u>:

<u>Carnot's Principle</u> says that for given temperature limits, no engine can be more efficient than a reversible engine.

<u>Carnot Cycle Using Vapor</u>

As in the case for a gas, this cycle again consists of two isothermals connected by two isentropics, and its appearance on a TS diagram is the same as for a gas. The reader will note that vapor in this case refers to water vapor (or steam) and that the saturated liquid line and saturated steam line form a similar horseshoe shaped curve as appears on a temperature enthalpy diagram (lesson on Steam and Water, T.T.3).



<u>Fig. 4</u> Carnot Cycle for Vapors

If the cycle is carried out entirely within the area bordered by the saturated liquid and saturated vapor line, as in Fig. 4, then a constant temperature line (isothermal process a-b) is also a constant pressure line. A constant pressure line on the PV diagram means a horizontal line and thus for the Carnot vapor cycle the upper and lower lines on the PV diagram are horizontal lines. This changes the shape of the curve on a PV diagram as compared to a Carnot gas cycle.

The thermal efficiency remains unchanged and may still be expressed as:

$$e = \frac{T_1-T_2}{T_1}$$

Now let us consider what happens in the practical application of this cycle.

1. We must have an isothermal expansion during which heat is added. This is achieved by evaporation of water in a boiler. (Addition of latent heat of vaporization takes place at constant temperature).

2. We must have an isentropic expansion during which power is produced. This is achieved in a steam turbine. Since a turbine is well insulated, the process takes place adiabatically.

3. We must have an isothermal rejection of heat. This can be done in a condenser, since condensation takes place at constant temperature.

4. We must have an isentropic compression process. In Fig. 4(b) this would mean compressing wet steam from d to a. To do this we would have to put as much work in $W = (T_2-T_1)(s_b-s_a)$ as we originally got out by expansion from b to c $W = (T_1-T_2)(s_b-s_a)$. Therefore, we would have gained nothing as far as useful work is concerned.

In previous lessons where we have dealt with the various cycles using gas as the substance, the processes have all taken place within the engine itself and therefore, the thermal efficiency is the engine efficiency (neglecting friction and other losses).

In the Carnot vapor cycle, we have several pieces of equipment such as boiler, turbine, condenser and pump. Thus, if we spoke of ideal cycle efficiency regarding the turbine, the turbine manufacturer could justifiably complain that he was not responsible for inefficiencies due to the boiler, condenser and pump. Therefore, in the Carnot vapor cycle, we must consider separately the ideal cycle and the ideal engine.

The ideal cycle includes all the processes which occur such as in the boiler, turbine, condenser and pump.

The ideal engine includes only the processes associated directly with the turbine, those of delivering the substance to the turbine, of expansion in the turbine, of exhaust from the turbine and any heat transfer that takes place while the substance is in the turbine.

Thus in practice an actual steam power plant cycle may have an efficiency of around 35%, whereas the turbine itself may have an actual efficiency of around 85%.

In discussing the Carnot vapor cycle, we have been dealing with it as though all of the processes take place underneath the saturated liquid and saturated vapor lines on the TS diagram under ideal conditions. In actual practice vapor cycles do not follow the Carnot cycle and it would be fairer to devise a standard of comparison that more nearly accords with real conditions. This has been done by devising the Rankine Cycle which will be the last cycle to be covered in this course.

D. Dueck

Definition of Symbols Used in this Lesson

C O P = coefficient of performance

e = thermal efficiency

J = 1 Btu = 778 ft-lbs.

$P_{a,b,c,}$ etc. = pressure in lbs/sq.ft. at the various points

Q_A = heat added, Btu.

Q_R = heat rejected, Btu.

R = specific gas constant ($= pV/T$)

$S_{a,b,etc.}$ = specific entropy of a substance at the various points.

$T_{1,2,}$ $_{etc.}$ = absolute temperature °R at the various points

$V_{1,2,etc.}$ = volume of the substance cu.ft.

w = lb. of the substance

W = net work $Q_A - Q_R$.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

2 - Science Fundamentals - T.T.1
5 - Heat and Thermodynamics
-1 - Carnot Cycle
A - Assignment

1. What 3 main things are required to make a Carnot cycle work?
2. List the processes involved in a Carnot cycle. Draw the PV and TS diagram.
3. Give the formula for the efficiency of the Carnot cycle.
4. A Carnot cycle uses a perfect gas and works between a 2000°F source and an 80°F receiver. What will its thermal efficiency be?
5. If the reversible adiabatic compression process of a Carnot cycle requires a mechanical work input of 150 Btu/lb., what work does the reversible adiabatic expansion process produce?
6. Explain coefficient of performance in the reversed Carnot cycle and give the equation for it.
7. What is the difference on the PV diagram of a Carnot cycle when using vapor as the substance as compared to using a perfect gas?

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

2 - Science Fundamentals - T.T.1

5 - Heat & Thermodynamics

2 - Rankine Cycle

0.0 INTRODUCTION

The Rankine cycle, or complete expansion cycle was the first accepted standard of comparison for steam power plants. This is the cycle that will be described in this lesson. The latter part of this lesson dealing with improvements on the Rankine cycle uses terms that are more fully explained in the Turbine-generator course and it is assumed that the student has taken that course up to the T.T.2 level.

1.0 INFORMATION

Figure 1 shows a diagrammatic layout for a condensing power plant which would operate on a Rankine cycle. The main pieces of equipment involved are the boiler, turbine, condenser, boiler feed pump and cooling water pump. Figure 2 shows the TS diagram for this same Rankine cycle. The letters depicting the various state point conditions on the T S diagram correspond to the letters used in figure 1.

Notice that the area a-b-c-d is the same as the area in figure 4(b) in the lesson on the Carnot cycle. We said in that lesson that it is impractical to compress steam isentropically from d to a.

In the Rankine cycle instead of trying to compress from 'd' to 'a' we condense the steam right back to 'e.' The boiler feed pumps then compress the water adiabatically to boiler pressure along the line 'e' to 'f' where it enters the boiler. The boiler then heats the water to saturation temperature along 'f' to 'a', evaporates it isothermally along "a" to "b" to produce saturated steam; the steam then expands isentropically in the turbine from b to c; at the turbine exhaust the wet steam is then condensed isothermally to the saturated liquid line from c to e .

Note: In showing the line 'efa' we have taken into account that during compression in the boiler feed pump, the water gets hot to a certain extent. Usually the effect of boiler feed pumps is ignored when drawing the Rankine cycle and heating of liquid is assumed to proceed from e to a along the saturated liquid line.

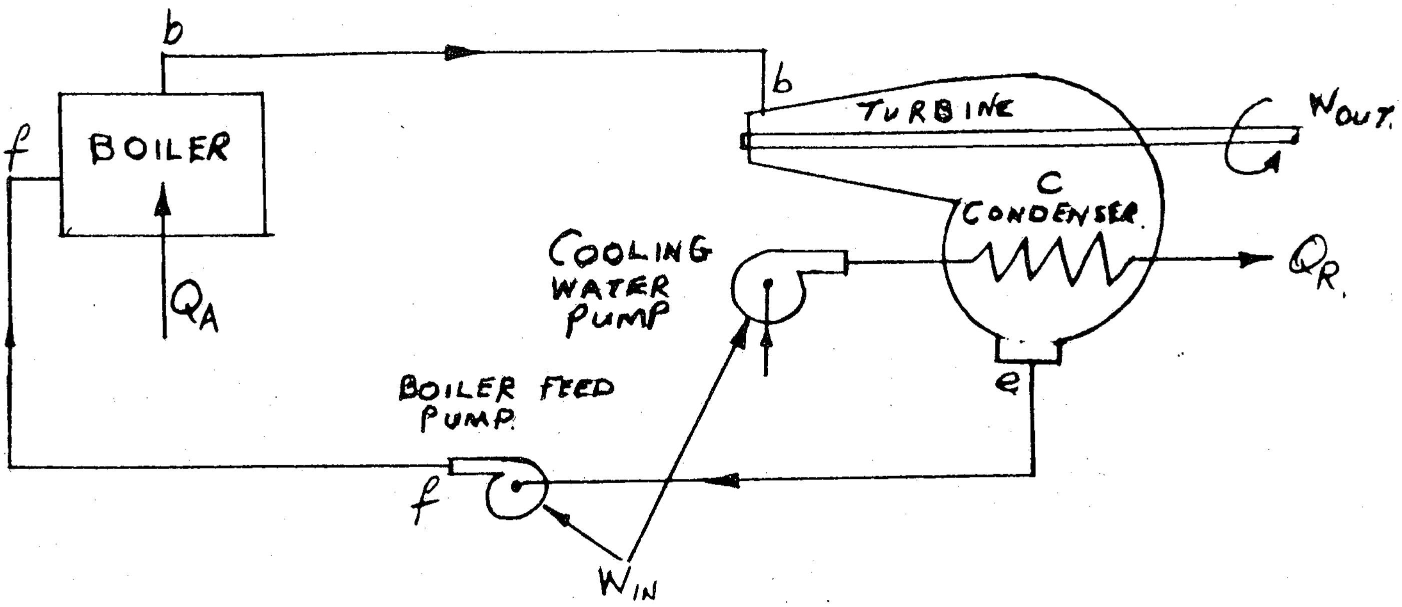

b
b
BOILER
f
Q_A
TURBINE
$W_{OUT.}$
C
CONDENSER
COOLING WATER PUMP
Q_R
BOILER FEED PUMP
e
f
W_{IN}

Fig. 1

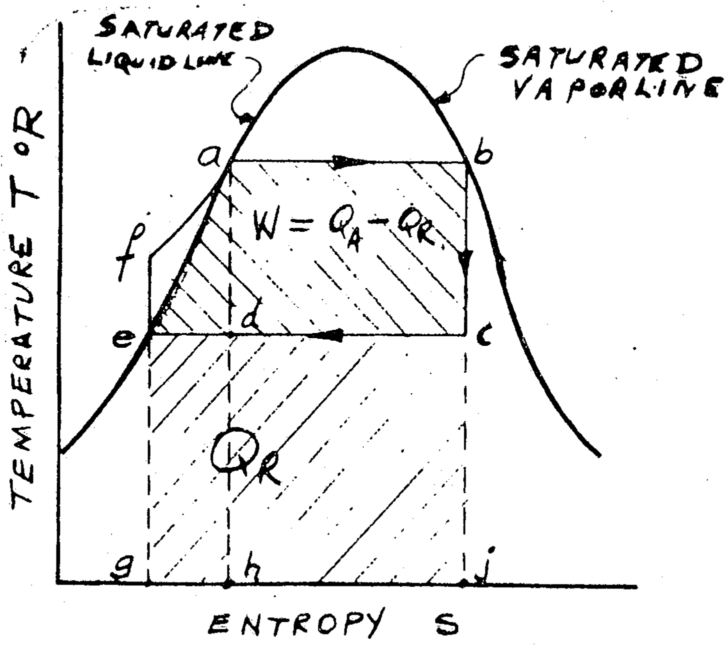

SATURATED LIQUID LINE
SATURATED VAPOR LINE
TEMPERATURE T °R
a
b
$W = Q_A - Q_R$
f
e
d
c
Q_R
g
h
j
ENTROPY S

Fig. 2

Notice that the extra heat added in a Rankine cycle as compared to a Carnot cycle is g-f-a-h. but the extra heat rejected is g-e-d-h. The work done is e-f-a-d and is a relatively small portion of the heat added. Therefore Rankine cycle is less efficient than the Carnot cycle.

Efficiency of the Rankine Cycle

Refer to figures 1 and 2 during this discussion. Since this is a closed cycle, all energy entering must equal all energy leaving. Thus from figure 1 we have:

$$Q_A + W_{in} = Q_R + W_{out}$$

Rearranging we get

$$W_{out} - W_{in} = Q_A - Q_R$$

Net Work $W = W_{out} - W_{in} = Q_A - Q_R$

The thermal efficiency of the Rankine cycle will then be:

$$e = \frac{W}{Q_A} = \frac{Q_A - Q_R}{Q_A} \qquad \text{- - - - - - - (1.)}$$

where:

Q_A = heat added, Btu, area g-e-a-b-j figure 2.

Q_R = heat rejected, Btu, area g-e-c-j figure 2.

W_{in} = work input to the cycle Btu

W_{out} = work output from the cycle Btu.

e = thermal efficiency.

W = net work, Btu, area a-b-c-e figure 2.

Referring to figure 2, let the enthalpy at b = h_b; at c = h_c; at e = h_e. Then the heat added:

$$Q_A = h_b - h_e$$

and heat rejected

$$Q_R = h_c - h_e$$

Substituting into equation 1 we get the thermal efficiency for the ideal Rankine cycle as:

$$e = \frac{(h_b - h_e) - (h_c - h_e)}{h_b - h_e} = \frac{h_b - h_e - h_c + h_e}{h_b - h_e}$$

$$e = \frac{h_b - h_c}{h_b - h_e}$$

Discussion

Notice that the particular Rankine cycle described produces saturated steam only and therefore as it expands in the turbine it will become wet steam. Wet steam erodes turbine blades and also makes the turbine less efficient. For this reason the simple Rankine cycle is very seldom used anymore.

Over the years metals have been developed that can withstand relatively high temperatures and along with this development improvements have been devised to the Rankine cycle. These improvements include 1.) Superheating 2.) Improving pressure conditions 3.) Reheating 4.) Regenerative feedheating.

All these improvements are made use of in modern conventional power stations. However, due to temperautre limitations of metals in the reactor, Nuclear Power station to date are not able to take advantage of the use of superheated steam; however the cycle does include reheating and feedwater heating.

Superheating

Since the thermal efficiency of the Carnot cycle is: $e = (T_1 - T_2) / T_1$ we can clearly see that if we could raise T_1 we could gain a greater efficiency. In a steam cycle one way this can be accomplished is by superheating the saturated steam leaving the boiler drum.

Referring to figure 3, and comparing the two illustrations it can be seen that as the superheat temperature is increased, the heat available for useful work increases more rapidly than the unavailable heat, and as a result the cycle efficiency improves.

Improving Pressure Conditions

a) Reducing the turbine exhaust pressure will lower the temperature at which steam will condense - i.e. the lower the pressure the lower the saturation temperature. As can be seen from figure 4 the lower the temperature, say, from 3-2 to k-d the less the

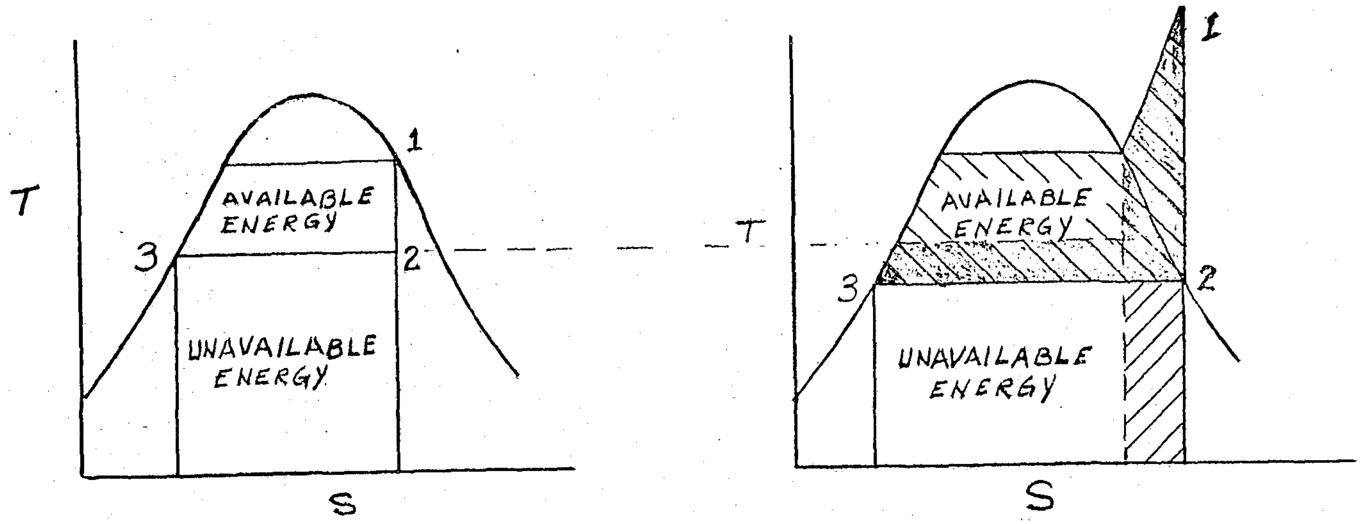


Figure 3
Effect of Superheating

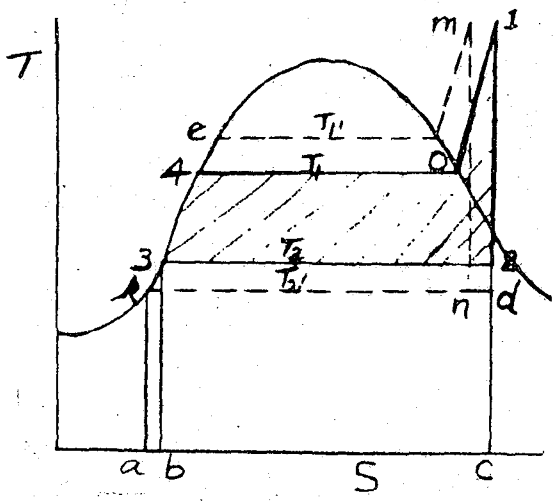


Fig. 4
Effect on Rankine Cycle of Improving Pressure Conditions

unavailable energy and the greater the available energy to do useful work. This also means greater efficiency as can be seen by examining the equation for thermal efficiency for the carnot cycle. $e = \frac{T_1 - T_2}{T_1}$ That is why modern steam turbines exhaust to almost a perfect vacuum. Incidentally the pressure that a turbine exhausts to is commonly known as its "back pressure". b) Raising the boiler pressure means raising the evaporation temperature say, from 4-1 to e-m figure 4, because the higher the pressure the higher the saturation temperature. More heat is transferred to the working substance at the higher pressure, but a greater proportion of the heat supplied is converted to work. Thus we find that the efficiency of plants using high pressures is greater than plants using low pressures. This accounts for the modern trend towards high pressures in steampower stations.

Reheating

Another method of improving the Rankine cycle is by reheating. In the last two methods of improving the Rankine cycle we have seen

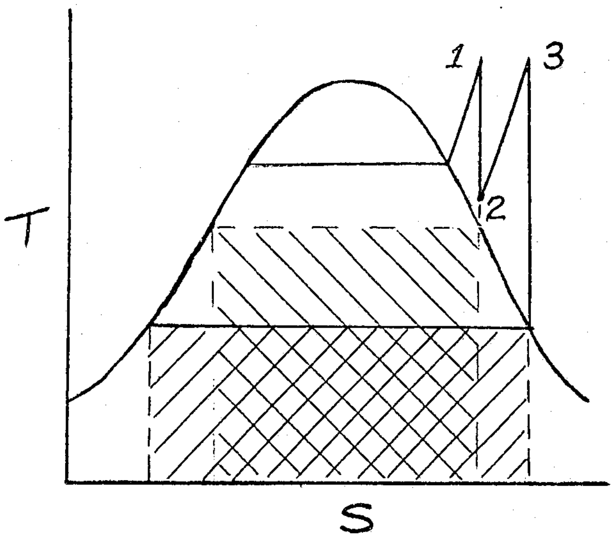


Figure 5
Effect of Reheating
on Rankine Cycle

that by increasing available work and decreasing unavailable work we can improve the cycle.

In the reheat cycle steam at point 1, figure 5 is passed through what is known as a high pressure turbine where it expands from 1 -2, the pressure being considerably reduced. From the high pressure turbine exhaust the steam is then led back to the boiler where it is again heated to its original temperature 2-3 after which it is completely expanded in lower pressure turbine cyclinders. As you can see this again increases the available energy and in greater proportion to the increase of unavailable energy. Thus there is a trend towards reheat cycles. Figure 5 shows a single reheat cycle. There are a few double reheat cycles in existence. However, more than double reheating is not economical at present.

Regenerative Feedheating

The Carnot cycle suggests another means of improving the Rankine cycle. In figure 6, if the heat represented by area 3-4-5-5'-4 could be used to supply the heat required to bring the feedwater to boiler saturation temperature at 2, represented by area 2-1'-6'-6-1 and if the heat represented by area 3-4-4' were identical to the heat represented by area 2-1'-1, then a cycle represented by area 1-2-3-4' would remain. This might be termed an ideal regenerative cycle. If areas 3-4-4' and 2-1'-1 are equal then the cycles represented by areas 1-2-3-4' and 1'-2-3-4 contain equal amounts of heat available for work and have equal amounts of unavailable heat. But the cycle represented by area 1'-2-3-4 is a Carnot cycle. Therefore the ideal regenerative cycle operating on saturated steam would have an efficiency equal to that of a Carnot cycle. The ideal regenerative cycle cannot be attained but it can be approximated as is shown in figure 7. If a quantity of heat represented by area a-b-c-5 is extracted from the turbine at condition "a" and mixed with feedwater at condition 1, the mixture will reach equilibrium at condition "d" and the heat represented by a-b-c-5 will have been used to supply the heat represented by 1-d-e-6 required to heat feedwater from condition 1 to condition d. Similarly if we increase the number of extractions for feedwater heating in this manner it brings the cycle closer and closer to the ideal regenerative cycle.

That is why regenerative feedheating is an integral part of the modern steam power station cycle. The ideal regenerative cycle is not attained but it certainly does make a big improvement on the efficiency of the overall cycle.

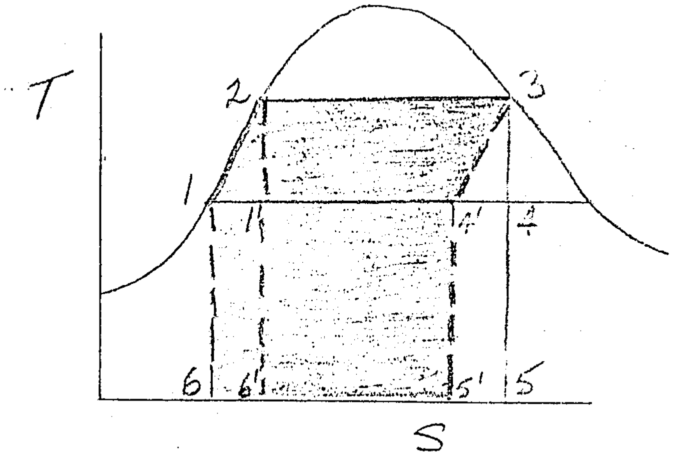


Fig. 6

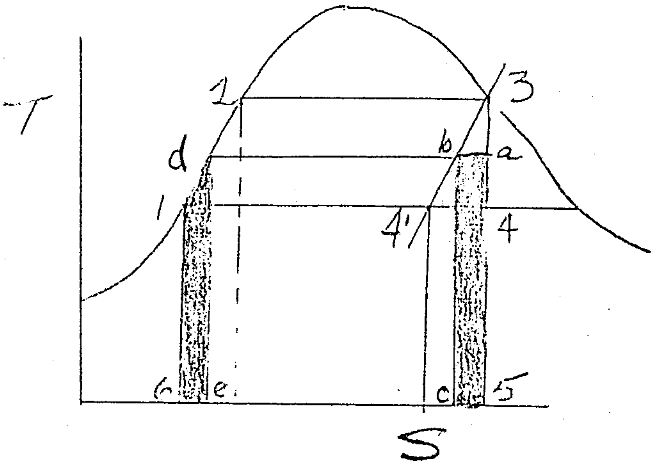


Fig. 7

Effect of Regenerative Feedheating

Summarizing then, superheating, improving pressure conditions, reheating and regenerative feedheating are methods whereby the Rankine cycle is being improved upon in modern steam power stations.

Expansion Line

As we have mentioned, in the Rankine cycle expansion takes place in the turbine and this can be represented as line 1-2 on a temperautre-entropy diagram as shown in figure 8. Therefore line 1-2 is referred to as the <u>expansion line</u>. However, this illustrates an ideal cycle condition. In an actual situation, as steam passes through a turbine there are friction losses, and since this is an irreversible adiabatic process, there is an increase in unavailable energy-that is, there is an increase in entropy.

Hence in the actual case the expansion line takes place from 1-2' resulting in more unavailable energy then under ideal condition.

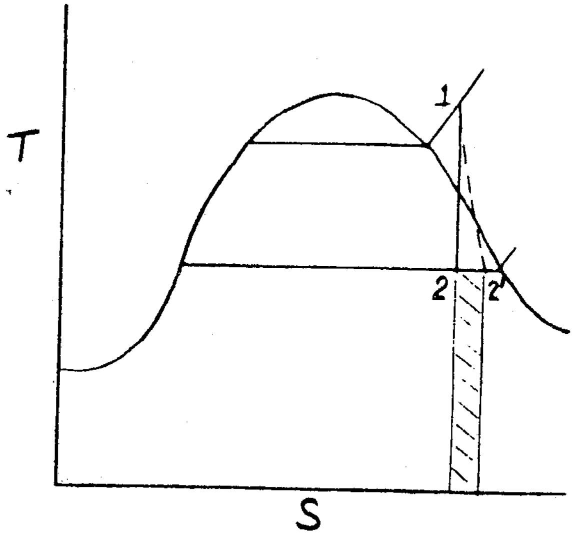


Figure 8

D. Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

2 - Science Fundamentals - T.T.1

5 - Heat & Thermodynamics

2 - Rankine Cycle

A - Assignment

1. With the aid of a TS diagram list the processes involved in a Rankine cycle using vapor as the substance
2. Which is more efficient-the Rankine cycle or Carnot cycle? Why?
3. What is the efficiency of a Rankine cycle power plant if steam is supplied to the prime mover saturated at 150 psia., and the exhaust pressure is 15 psia? The % moisture at exhaust is 12.5%. Use the steam tables supplied at the T.T. 3 level. What is the ratio of efficiency to that of the Carnot cycle?
4. What is the efficiency of a Rankine cycle steam power plant if steam is supplied to the prime mover saturated at 150 psia and the exhaust pressure is 4 psia? The % moisture at exhaust is 18%. What is the ratio of efficiency to that of the Carnot cycle?
5. State four ways in which the Rankine cycle can be improved on.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

2 - Science Fundamentals - T.T.1

5 - Heat and Thermodynamics

-3 - Mollier Diagram

0.0 INTRODUCTION

In the preceding lessons we have presented the properties of a vapor, especially those of steam, on pressure-volume diagrams, temperature-enthalpy diagrams and temperature-entropy diagrams. However, the properties of a vapor are often presented by still another method. If the total enthalpy of one pound of vapor as the ordinate is plotted against the total entropy of 1 lb. of vapor as the abscissa, the result is called a Mollier diagram, as shown in Figure 3. A chart like this is very convenient since the enthalpy at a given condition may be read directly at the edge of the chart and does not require the calculation of areas. The two most common processes for a vapor are those which take place at constant enthalpy and those for which the change is isentropic (reversible adiabatic). The first may be traced horizontally on the Mollier diagram, and the second is represented by a vertical line. Also on this diagram, changes of constant pressure, constant quality, constant superheat, or constant temperature may readily be traced.

1.0 INFORMATION

As you can see from Figure 3, the Mollier diagram contains a vast amount of information. The different properties of steam depicted on the diagram are listed below:

1. Enthalpy in Btu/lb as the ordinate.
2. Entropy as the abscissa.
3. Constant moisture % lines in the wet steam region.
4. Constant pressure lines in inches of mercury up to 2 inches, and from then on in psia.
5. Constant temperature lines °F.
6. Constant degrees of superheat, °F.

In order to clarify the Mollier diagram several lines of each of these variables have been traced from it, and are shown in the Figures that follow, just to give a general idea of the direction in which these lines run.

Figure 1 depicts the constant enthalpy and constant entropy lines. Notice that the diagram starts with enthalpy equal to 800 Btu/lb. rather than zero, and entropy starts from 1.0 and goes up to 2.2. The area depicted on the actual Mollier Diagram covers the conditions which are most commonly used in steam calculations and therefore, ther is no need to start from zero.

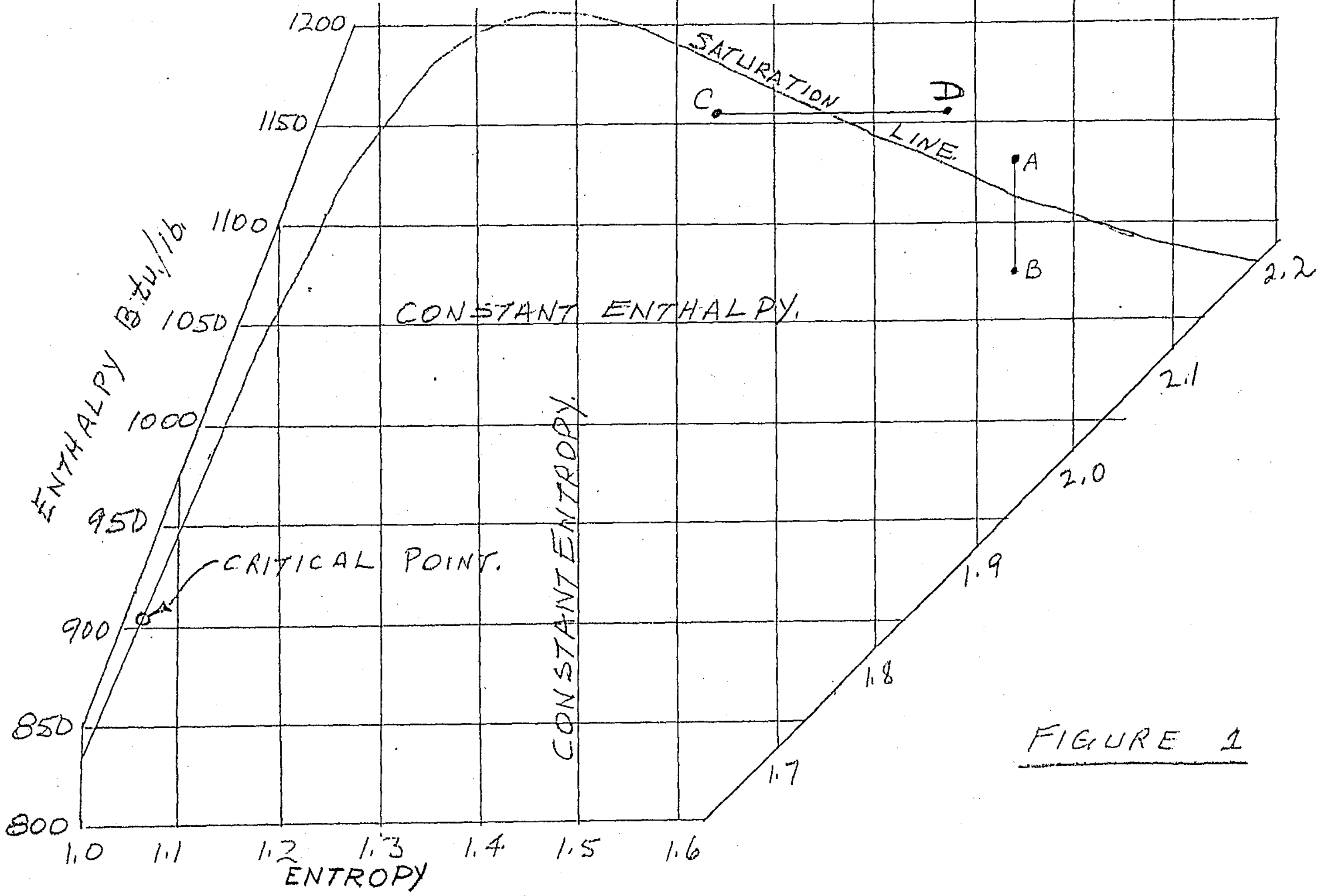


FIGURE 1

You will recall that an isentropic process is one that takes place at constant entropy. Constant entropy lines on the Mollier diagram are vertical. Therefore, an isentropic process would be shown as a vertical line, such as AB in Fig. 1.

There are also processes which take place at constant enthalpy This is known as a throttling process of which we will learn more later. Since constant enthalpy lines on the Mollier diagram are horizontal, therefore, a throttling process will always appear as a horizontal line such as line CD, figure 1, on a Mollier diagram.

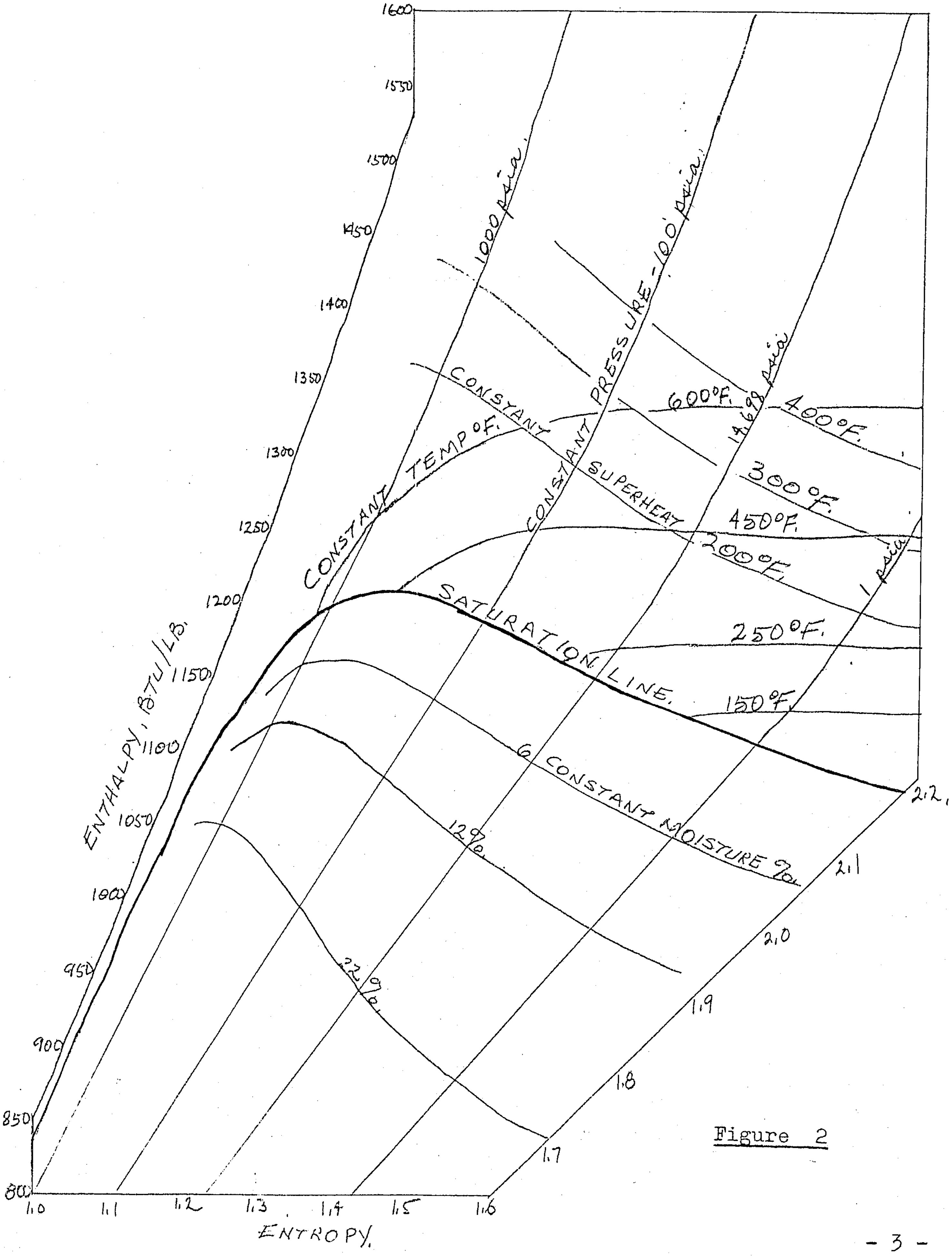

1600
1550
1500
1450
1400
1350
1300
1250
1200
1150
1100
1050
1000
950
900
850
800
ENTHALPY, BTU/LB.
1.0
1.1
1.2
1.3
1.4
1.5
1.6
1.7
1.8
1.9
2.0
2.1
2.2
ENTROPY.
1000 psia.
CONSTANT PRESSURE-100 psia.
14.698 psia
1 psia.
CONSTANT TEMP°F.
600°F.
400°F.
300°F.
450°F.
200°F.
250°F.
150°F.
CONSTANT SUPERHEAT
SATURATION LINE.
6 CONSTANT MOISTURE %
12%
22%

Figure 2

A MOLLIER CHART FOR STEAM

Abs. press., in. Hg.	Sat. temp., °F	Abs. press., lb. per sq. in.	Sat. temp., °F	Abs. press., lb. per sq. in.	Sat. temp., °F
0.20	34.60	1.0	101.74	120	341.25
0.25	40.23	2	126.08	140	353.02
0.30	44.96	3	141.48	160	363.53
0.35	49.06	4	152.97	180	373.06
0.40	52.64	5	162.24	200	381.79
0.45	55.87	6	170.06	220	389.86
0.50	58.80	7	176.85	240	397.37
0.55	61.48	8	182.86	260	404.42
0.60	63.95	9	188.28	280	411.05
0.65	66.26	10	193.21	300	417.33
0.70	68.40	12	201.96	400	444.59
0.75	70.43	14	209.56	500	467.01
0.80	72.32	14.696	212.00	600	486.21
0.85	74.13	16	216.32	700	503.10
0.90	75.84	18	222.41	800	518.23
0.95	77.47	20	227.96	1000	544.61
1.00	79.03	25	240.07	1200	567.22
1.10	81.95	30	250.33	1400	587.10
1.20	84.65	35	259.28	1600	604.90
1.30	87.17	40	267.25	1800	621.03
1.40	89.51	45	274.44	2000	635.82
1.50	91.72	50	281.01	2200	649.45
1.60	93.80	60	292.71	2400	662.12
1.70	95.77	70	302.92	2600	673.94
1.80	97.65	80	312.03	2800	684.99
1.90	99.43	90	320.27	3000	695.36
2.00	101.14	100	327.81	3206.2	705.40

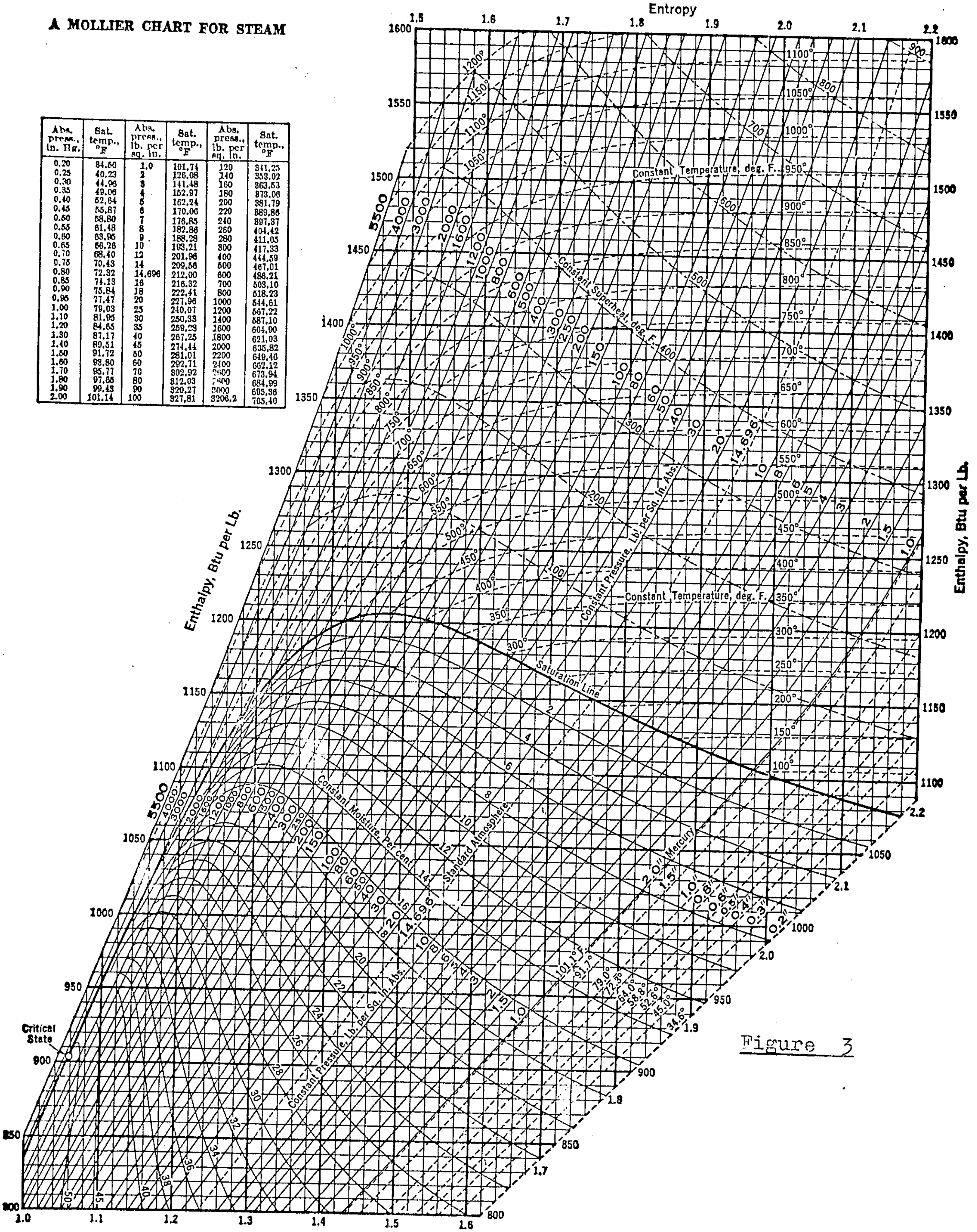


Figure 3

In Fig. 2, you can see that the saturation line has a shape similar to the saturation line on a TS diagram. Below this line is the wet steam region. This is shown by the constant moisture % lines which run approximately parallel to the saturation line.

Above the saturation line is the superheat region, in which are located two different types of temperature lines. One type shows constant superheat temperature which gives the number of degrees above saturation temperature. The other type shows constant temperature lines which is the actual temperature of steam in °F. Note that the latter temperature lines only run up to the saturation line and stop. In the wet steam region, constant temperature lines run parallel to lines of constant pressure. This is because water boils (evaporates) at constant temperature so one has to imagine the constant temperature lines extending from the saturated steam line down through the wet steam region parallel to the constant pressure lines.

The constant pressure lines already mentioned are straight lines in the wet steam region but curve slightly upwards in the superheat region. Also the lines diverge slightly while going in the upward direction.

Having all this information on one chart and knowing two variables one can, by determining at what point on the diagram the lines of these two variables intersect, find out what all the other unknowns are without doing any calculations. This is an enormous time-saver in calculations concerning turbine efficiencies, output heat balance, calculations, etc.

Notice also that the Mollier diagram is really a plot of all the values, except volume, that are listed in steam tables.

Let us now take an example to show how the Mollier diagram may be applied.

Sample Problem No. 1

Steam enters a steady flow machine (turbine) at a pressure of 100 psia and a temperature of 400°F, and then expands isentropically to a pressure of 5 psia. What work is done, if heat transferred $Q = 0$, and change in kinetic energy $\Delta K.E. = 0$?

Solution

In solving a problem of this type, draw a sketch as in Fig. 4 using properties of the entering steam, locate point 1 at the intersection of the curves $p_1 = 100$ psia and $t_1 = 400°F$. Now follow a constant entropy line (vertical line since the process is isentropic) until the 5 psia pressure line is reached and locate point 2. Move to the left ordinate from point 1 and read $h_1 = 1227$ Btu/lb., move to the left ordinate from point 2 and read $h_2 = 1011$ Btu/lb. Notice that point 2 is in the <u>wet steam</u> region.

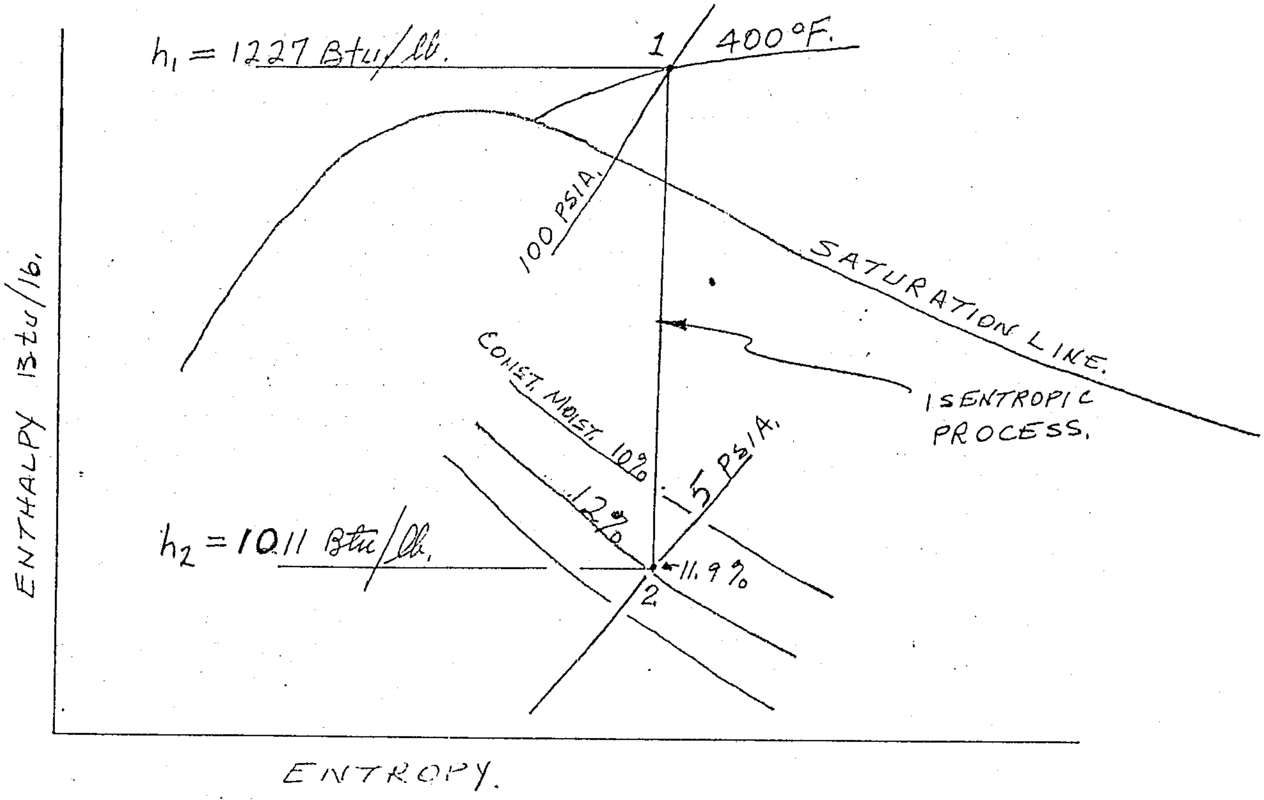


Fig. 4

The general energy equation as we learned in the T.T.2 level is:

$$\frac{v_{s_1}^2}{2gJ} + u_1 + \frac{p_1v_1}{J} + Q = \frac{v_{s2}^2}{2gJ} + v_2 + \frac{p_2v_2}{J} + W$$

Since there is no change in kinetic energy, the two terms on kinetic energy cancel each other out. The heat transferred Q = 0. Enthalpy is an energy term and by definition, h = internal energy u + flow work pv/J.
Thus we have:

$$u_1 + \frac{p_1v_1}{J} = u_2 + \frac{p_2v_2}{J} + W$$

$$h_1 = h_2 + W$$

$$W = h_1 - h_2$$

Thus net work done is: $W = 1227 - 1011 = \underline{\underline{216}}$ Btu/lb.

While we are at it we can read the quality of steam at point 2; % moisture at this point is 11.9%.

$$\% \text{ quality } x = 100 - \% \text{ moisture } y = 100 - 11.9 = \underline{\underline{88.1\%}}$$

That is, the substance in the wet steam region is 88.1% steam.

The reader should note that the Mollier diagram is recommended where great accuracy is not required. When great accuracy is required steam tables should be used. Also, remember that the Mollier diagram is for 1 lb. of steam only.

Throttling

In the throttling process, a working substance (such as steam) flows from a region of high pressure to a region of low pressure through a valve or some other constricted passage. What actually occurs is that enthalpy is converted into kinetic energy which is completely reconverted by friction into heat without doing external work. This process takes place adiabatically. Generally, we throttle a fluid primarily to control its rate of flow. The general energy equation applies in this case:

$$K.E._1 + H_1 + Q = K.E._2 + H_2 + W$$

where H has been substituted for $U + pV/J$.

Now since the throttling process is adiabatic, $Q = 0$; and since no work is done, $W = 0$. While the fluid is passing through the valve or restriction its velocity will increase considerably. However, after it has passed out of the nozzle it will soon slow down again if allowed a free path so that its initial velocity (v_{s_1}) is equal to its final velocity (v_{s_2}). Since $K.E. = wv_s{}^2/2g$ the kinetic energy terms $K.E._1$ and $K.E._2$ will cancel each other out. Thus, in the above equation, the enthalpy terms are the only ones that remain. That is:

$$H_1 = H_2 \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

Thus we can say that a throttling process is a constant enthalpy process.

On a Mollier Chart (or enthalpy-entropy diagram) the process will take place in a straight horizontal line. Consequently, if steam in a wet steam region is throttled to a low enough pressure, it can become superheated steam, as can be seen in Figure 5. This fact can be made use of in determining the wetness of steam. The device used is called a throttling calorimeter. It is attached to a pipeline, etc., and it throttles the steam down to atmospheric pressure. The measurements required to be made is the original steam pressure and the final pressure and temperature. It should be noted that in order for the throttling calorimeter to work, the

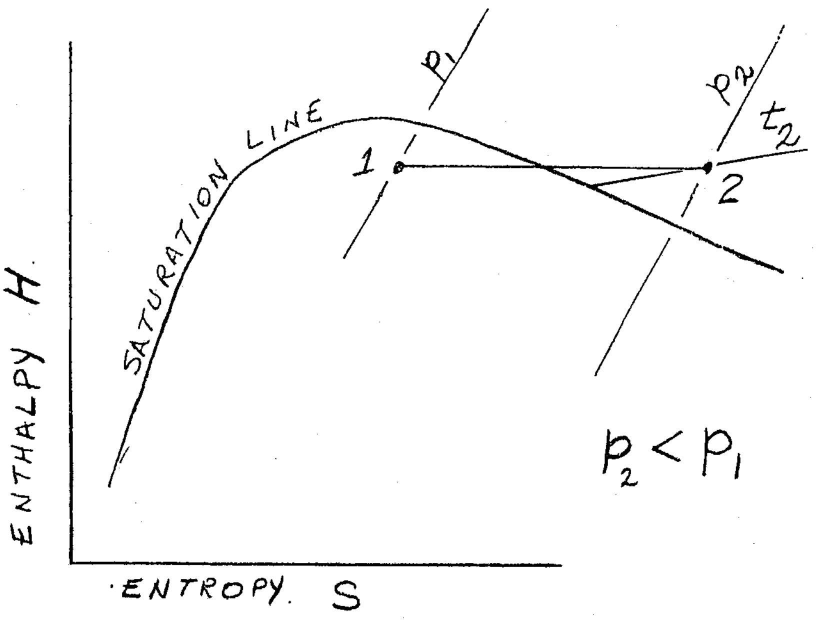


Fig. 5

outlet from the calorimeter must be in the superheat region. If wet steam comes out of the calorimeter then the values obtained are not applicable.

Sample Problem No. 2

The exhaust steam from a throttling calorimeter is measured as 280°F, and standard atmospheric pressure. Steam pressure in the pipeline to which the throttling calorimeter is connected is 120 psia. Calculate the % moisture in the steam in the pipe. (This can be represented as points 2 and 1 respectively as in Figure 5).

Solution

We know that $h = h_f + \frac{x}{100} \cdot h_{fg}$

$$\therefore \; x = \left(\frac{h - h_f}{h_{fg}}\right) 100$$

Since $h_1 = h_2$, we can look up the total enthalpy for one lb. at 14.7 psia and 280°F and get (by interpolation):

$h = 1182.9$

at 120 psia $h_f = 312.5$

and $h_{fg} = 878.1$

$$\therefore \; x = \left(\frac{1182.9 - 312.5}{878.1}\right) 100 = 99.1\%$$

% Moisture = 100 - 99.1 = 0.9%

An alternate solution would be by Mollier chart.

D. Dueck.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

2 - Science Fundamentals - T.T.1
5 - Heat and Thermodynamics
-3 - Mollier Diagram
A - Assignment

1. List the different properties of steam which are shown on a Mollier diagram.

2. When an isentropic process is depicted on a Mollier diagram in what direction does the line run?

3. When a throttling process is depicted on a Mollier diagram, in what direction does the line run?

4. Pressure in a main steam line is at $p_1 = 100$ psia. A throttling calorimeter used for measuring steam quality throttles the steam so that the conditions inside the calorimeter are 14.7 psia and 240°F. What is the quality x of the steam in the main steam line? Solve in two ways, by using equations and by using a Mollier Chart.

HEAT TRANSFER

In lessons in the course on "heat" so far we have repeatedly referred to heat transfer but have not mentioned how this transfer of heat is achieved. In level 4 we have defined three modes of heat transmission - heat conduction, heat convection and heat radiation. This lesson will deal mainly with heat transfer by conduction.

In the Second Law of Thermodynamics we have said that heat transfer always occurs from a body at a higher temperature to a body at a lower temperature. No doubt you are aware that that is what happens in heat exchangers.

We can define a heat exchanger as a device which affects a transfer of heat from one substance to another. These can be the type where a hot fluid mixes with a cold fluid and the two eventually reach equilibrium temperature. However, more often, engineers are required to estimate tha amount of heat transferred from one fluid on one side of a wall, to and through the wall, to another fluid on the other side of the separating wall. No doubt numerous examples will come to mind such as the exchange of heat in a closed feedwater heater, or oil cooler, or surface condenser, or the exchange of heat in a boiler; the flow of heat from the heated house to the outside air in winter, etc.

Conduction

We have said previously that heat is conducted from one part of a body to another part because faster moving molecules or atoms or electrons in a hotter part induce an activity of neighbouring molecules or atoms or electrons and thus heat "flows" from the hotter to colder parts of the body.

Conduction may occur under steady-state condition or unsteady-state conditions. The steady-state condition is one where the temperature of each point in the body remains constant; the unsteady-state condition is one where the temperature of various points in the body may change. Warming up period of a reactor is an example of an unsteady-state condition as far as conduction is concerned. We shall be dealing with the steady-state condition only.

Fourier's Law

At this stage we will introduce some equations used in estimating heat transfer, and it is appropriate to start off with a few definitions.

We know from experience that some materials conduct heat more readily than others. For example if we have a rod of copper and a rod of asbestos of equal length and we apply heat to one end of each rod, the copper rod will be hot on the unheated end, due to conduction, much more quickly than the asbestos rod. This characteristic is a property of the material and in thermodynamics is referred to as thermal conductivity of the material, we use the letter k to represent thermal conductivity which is directly proportional to the rate of heat transferred.

Note that this k is not to be confused with the specific heat ratio k = cp/cv.

If by experiment we know that we get a certain quantity of heat transferred through a material of area A_1 per unit time then we find that an area A_2 which is twice area A_1, will transfer twice as much heat per unit time. Thus the rate of heat transferred is directly proportional to the area A of the material.

The rate of heat transferred is dependent upon the thickness L of the material. The thicker the material the greater its resistance to conduction of heat. Therefore the rate of heat is inversely proportional to thickness L.

The rate of heat transferred is dependent on the difference in temperature Δt between the hot substance and the colder substance to which heat is transferred. The greater the temperature difference the greater the rate of heat transferred. Therefore this is a direct proportion.

Thus to summarize we can say that the rate of heat transferred Q is directly proportional to the conductivity k of the material, the surface area of the material, and the temperature difference Δt across the material and inversely proportional to the thickness L. This can be expressed in an equation as follows:

$$Q = k \frac{A}{L} \Delta t \qquad \ldots\ldots\ldots\ldots (1)$$

Where: Q = rate of heat transferred in Btu/hr
k = conductivity

A = surface area, sq. ft.
Δt = temperature difference $(t_1 - t_2)$°F
L = thickness of material, inches.

This equation is known as Fourier's Law.

Thermal Conductivity

If we solve for k from equation 1, in order to establish the units for k we get:

$$k = \frac{Q \times L}{A\Delta t} = \frac{\text{Btu}}{\text{hr.}} \times \frac{\text{in. thickness}}{\text{sq. ft. x °F}}$$

Thus by definition we have that thermal conductivity is the amount of heat transmitted in Btu/hr through 1 sq.ft. of homogeneous wall surface, per inch thickness of wall when the temperature changes 1°F in this thickness. In short thermal conductivity refers to how readily a material will conduct heat.

Or another way of expressing the units of conductivity k is Btu-in. per hr-sq.ft. °F.

Conduction in Solids

Table I is a list of conductivities for various materials and it is evident that conductivities vary widely for different materials. For example notice that per inch thickness, k for copper is 2616, whereas k for rock wool is 0.27. Hence copper is a good conductor of heat; rock wool is a good insulator.

The values in table I have been obtained experimentally and from experience there are several generalizations that can be made regarding conductivities of solids. These are:

1. The conductivities of all homogeneous solids are relatively high.
2. Materials that are poor conductors of heat are referred to as good insulating materials. Materials which are porous, cellular, fibrous, or laminated are good insulators. Air with a conductivity of 0.163 is one of the best insulators available.
3. In general conductivity increases with density and elasticity
4. With rare exceptions, conductivity of insulating materials increases very materially with temperature.
5. The absorption of moisture greatly impairs the insulating value of porous materials.

Table I - Conductivities

The units of *k* are Btu-in. per sq. ft-°F-hr.

Material	°F	*k*
Solids		
Aluminum piston alloy (a)	0–400	1209*
Asbestos, corrugated, 4 plies per in. (b)	300	0.828
Bearing metal, white (a)	68	164
Brickwork, low density (c)	68	5
Cast iron, grey (a)	68	350
	260	300
Copper, pure (b)	212	2616
Concrete, building construction (c)		12
Cork board (b)	86	0.3
Fir (c)		0.8
Glass, window (b)		6
Glass wool (c)		0.27
Gold (b)	64	2028
Gypsum plaster (c)		3.3
Monel (b)	68	242
Maple (c)		1.15
Oak (c)		1.15
Pine (c)		0.8
Plaster on wood lath, ¾ in. total thickness (c)		2.5
Rock wool (c)		0.27
Steel (S.A.E.1095) (a)	0–400	300*
Liquids		
Ammonia (a)	68	4.03
Petroleum oil (average)	68	1.00
Sulfur dioxide (a)	68	2.34
Water (a)	68	4.10
Gases		
Air (a)	32	0.163
Ammonia (a)	32	0.149
Carbon dioxide (a)	32	0.097
Carbon monoxide (a)	32	0.155
Hydrogen (a)	32	1.130
Nitrogen (a)	32	0.168
Oxygen (a)	32	0.170
Steam (a)	32	0.117

Conduction Through a Plane Wall

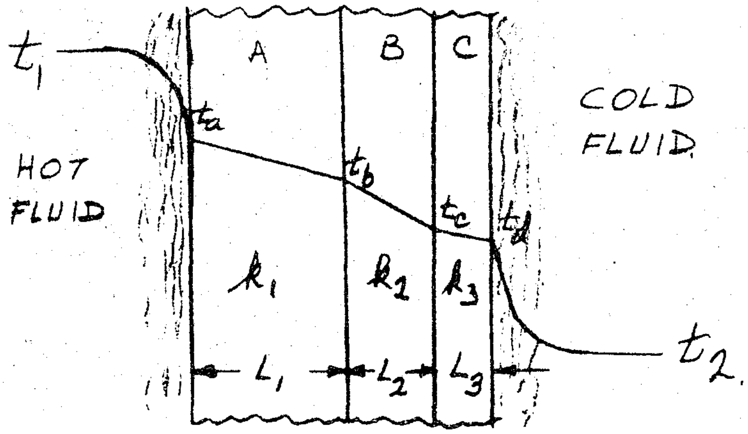


Figure 1 - Temperature gradients in a composite wall.

Figure 1 portrays a steady state condition where heat transfer is taking place from a hot fluid, through a composite wall (that is a wall composed of several different materials) to a cold fluid.

Notice that the temperature drops as heat flows from one material to the next. The rate of change of temperature with thickness is called temperature gradient.

The rate of heat transfer through material A, using equation 1 can be written as:

$$Q_A = \frac{k_1 A(t_a - t_b)}{L_1} \quad \text{Btu/hr.} \quad \ldots\ldots\ldots\ldots (2)$$

where t_a and t_b are the surface temperatures of partition A, figure 1. Similarly the rate of heat transfer through partitions B and C can be written:

$$Q_B = \frac{k_2 A(t_b - t_c)}{L_2} \quad \text{and} \quad Q_C = \frac{k_3 A(t_c - t_d)}{L_3} \quad \ldots\ldots (3)$$

Now note that after a steady state of heat transfer has been achieved there is no storage of heat and thus the heat that enters section A, leaves section A; the same amount of heat passes through section B and C. Consequently $Q_A = Q_B = Q_C = Q$. Solving for the temperature difference from each of equations (2) and (3) and dropping the subscripts for Q we get:

$$t_a - t_b = \frac{QL_1}{k_1 A} \quad t_b - t_c = \frac{QL_2}{k_2 A} \quad \text{and} \quad t_c - t_d = \frac{QL_3}{k_3 A}$$

But $(t_a - t_b) + (t_b - t_c) + (t_c - t_d) = t_a - t_d$

$$\therefore \quad t_a - t_d = \frac{QL_1}{k_1 A} + \frac{QL_2}{k_2 A} + \frac{QL_3}{k_3 A} = \frac{Q}{A}\left(\frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3}\right)$$

$$\text{and} \quad Q = \frac{A\ (t_a - t_d)}{\left(\frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3}\right)} \quad \text{Btu/hr} \quad \ldots\ldots\ldots (4)$$

Equation 4 represents the rate of heat transfer for the composite wall across which the temperature is $t_a - t_d$. It can be readily seen that if there were another section added to this wall the only change necessary in equation (4) would be the addition of another term L_4/k_4 in the denominator of the right side and the interpretation of t_d as being the temperature of the final surface. Thus one can adapt equation (4) to any wall of any number of homogeneous materials by the addition or deletion of L/k terms.

Surface Conductance

In figure 1 we have a hot and cold fluid respectively on either side of the wall. At each wall surface there is a thin film of fluid which offers a barrier to heat transfer through the wall. Thus there is a temperature drop across the film such that $t_1 > t_a$ and similarly $t_2 < t_d$. For a given temperature drop across such a film of fluid, there will be a flow of a quantity of heat. Let us represent this quentity of heat by the letter 'h' and its units will be Btu per hr - sq. ft. of surface for one degree of fahrenheit difference; or in equation form for rate of heat transfer across the films of fluid we have:

$$Q = h_1 A\ (t_1 - t_a) \quad \text{and} \quad Q = h_2 A\ (t_d - t_2) \quad \dots\dots \ (5)$$

This 'h' represents the surface coefficient of heat transfer for the fluid films adjacent to surfaces. The surface coefficient measures the transferred heat in the same manner as the thermal conductivity k, except that the surface coefficient includes the effect of the thickness of the fluid film.

The surface coefficient is dependent on specific heat of fluid, its density, its viscosity, its thermal conductivity, the velocity of fluid, and upon a characteristic dimension such as the diameter of a pipe in which fluid is flowing.

NOTE: The surface coefficient of heat transfer 'h' is not to be confused with the symbol 'h' for enthalpy.

Some typical values of surfaces coefficient are:
Inside building walls still air h = 1.65 Btu/sq.ft/°F/hr.
Outside building walls, 15 mph wind, h = 6.00 Btu/sq.ft/°F/hr.
Wet steam in a pipe from which heat flows h = 1000 Btu/sq.ft/°F/hr

Heat Transfer from Fluid to Fluid

Normally it is inconvenient to measure the temperature of surface of a wall and it is customary to use an equation which includes the surface coefficients and which permits the use of the temperatures of the fluids t_1 and t_2 as in figure 1.

To find the rate of heat transfer from fluid to fluid we again solve for the temperature differences in equations (2), (3) and (5) and proceed in a manner similar to the way we derived equation (4). Thus we get:

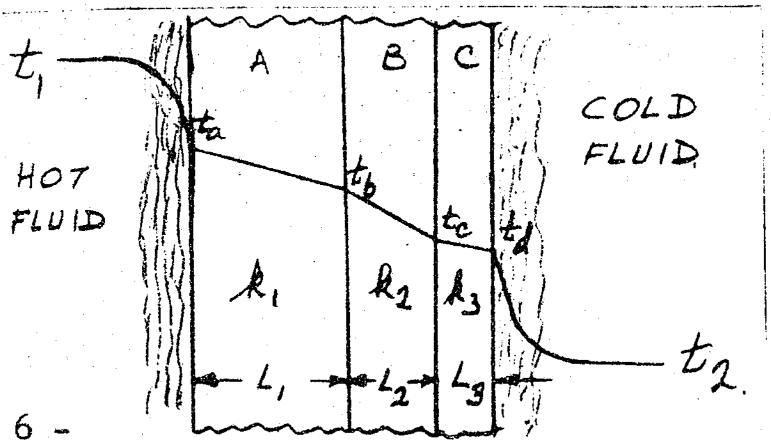


Fig. 1 (repeated)

$$t_1-t_a = \frac{Q}{h_1A} \;;\; t_a-t_b = \frac{QL_1}{k_1A} \;;\; t_b-t_c = \frac{QL_2}{k_2A} \;;\; t_c-t_d = \frac{QL_3}{k_3A}$$

and $t_d - t_2 = \frac{Q}{h_2A}$

again: $t_1-t_2 = (t_1-t_a)+(t_a-t_b)+(t_b-t_c)+(t_c-t_d)+(t_d-t_2)$

$$\therefore \quad t_1-t_2 = \frac{Q}{h_1A} + \frac{QL_1}{k_1A} + \frac{QL_2}{k_2A} + \frac{QL_3}{k_3A} + \frac{Q}{h_2A}$$

$$= \frac{Q}{A}\left(\frac{1}{h_1} + \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} + \frac{1}{h_2}\right)$$

and $$Q = \frac{A\,(t_1-t_2)}{\left(\frac{1}{h_1} + \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} + \frac{1}{h_2}\right)} \text{ Btu/hr.} \quad \ldots\ldots (6)$$

Equation (6) represents the rate at which heat is transferred from one fluid to another fluid through a composite wall as in figure 1.

Thus to estimate rate of heat transferred, all we have to do is measure the temperature of the two fluids, the rest of the data for the equation such as k, A, and L can be obtained from tables and design drawings.

The significance of the denominator on the right-hand side of equation (6) can be emphasized by an analogy with an electric circuit. Ohm's Law is expressed by the equation: $I = E/R$; i.e. the current I flowing in a given wire is directly proportional to the electromotive force E and inversely proportional to the resistance R. The electrical current is analogous to the heat, and the electromotive force, that is, the voltage drop, is analogous to the temperature difference. The greater the voltage drop the greater the current that will flow through a particular wire; the greater the temperature difference, the greater the heat that will flow through a given wall.

We have now established that Q is analogous to I and E is analogous to (t_1-t_2). Therefore the denominator in equation (6) must be analogous to resistance R.

The fluid films and the various sections of the wall are in a sense a series arrangement of resistances to the flow of heat, and the total resistance of a series connection is the sum of the resistances of the various parts.

We have defined k as the conductivity of a material and in accordance with the above discussion the reciprocal of k or 1/k expresses the resistance of a material to conduct heat. In heat and thermodynamics the resistance of a material to heat transfer 1/k is referred to as <u>thermal resistivity</u>.

To carry this one step further, the heat flow of the section A of the wall is L_1/k_1. Similarly 1/h is the resistance to heat flow through a film of fluid adjacent to the surface.

To repeat, the denominator in equation (6)

$$\frac{1}{h_1} + \frac{L_1}{k} + \frac{L_2}{k_2} + \frac{L_3}{k_3} + \frac{1}{h_2}$$ represents the resistance to heat transfer. The reciprocal of the sum of these resistances has the opposite effect to thermal resistivity - i.e. in equation (6) this would result in a high rate of heat transfer. (the material would transmit more heat.). Therefore the reciprocal of the sum of

$$\frac{1}{h_1} + \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} + \frac{1}{h_2}$$ is referred to as <u>transmittance</u>,U.

$$U = \frac{1}{\frac{1}{h_1} + \frac{L_1}{K_1} + \frac{L_2}{K_2} + \frac{L_3}{K_3} + \frac{1}{h_2}}$$

The equation is more convenient when expressed as:

$$Q = \frac{A\ (t_1-t_2)}{1/U}$$

$$\therefore\quad Q = UA\ (t_1-t_2)\ \text{Btu/hr.} \qquad \ldots\ldots\ (7)$$

From equation (7) it is apparent that a high transmittance means a higher rate of heat transfer, whereas a low transmittance would tend to mean a lower rate of heat transfer.

<u>NOTE</u>: Remember not to confuse transmittance U with internal energy U, defined in previous lessons.

Examining equation (6) still further we notice that increasing one or more of the terms in the denominator, heat will flow at a slower rate. This is the effect desired when providing insulation for steam lines, cold storage rooms, etc. On the other hand if a greater rate of heat flow is required we try to reduce the terms in the denominator of equation 6.

Speaking of heat exchangers the transmittance through heat exchanger tubes may be decreased considerably by accumulation of deposits on the tubes. Also the transmittance of insulated walls may increase due to deterioration of insulating materials.

Table II lists transmittances of a number of materials:

Table II - Transmittances

The units of U are Btu per sq. ft-°F-hr. These values are intended as typical and are not to be used in actual design unless it is known that they apply.

NATURE OF THE HEAT EXCHANGE		U
Ammonia condenser, double-pipe type, water inside, NH_3 in annular space, velocity of water 6 fps, mean temp. difference 3.5°F, inner tube 1¼ in. dia.	(a)	320
Brick wall, 8 in. thick, plaster inside	(b)	0.46
Brick veneer on 6-in. hollow tile, plaster inside	(b)	0.34
Feedwater heaters, closed; steam condensers, free convection	(a)	50–200
Ditto, forced convection	(a)	150–800
Hot-water radiators, air coolers, economizers, steam boilers, liquid to gas, gas to liquid, free convection	(a)	1–3
Ditto, forced convection	(a)	2–10
Steam, condensing, to air, free convection	(a)	1–2
Ditto, forced convection	(a)	2–10
Steam, condensing, to boiling water, free convection	(a)	300–800
Steam, condensing, to oil, free convection	(a)	10–30
Ditto, forced convection	(a)	20–60
Superheaters, steam, free convection	(a)	1.6–2
Ditto, forced convection	(a)	2–6
Water to water, free convection	(a)	25–60
Ditto, forced convection	(a)	150–300

Sample problem.

Calculate the rate of heat transfer through a wall 9' high by 12' long which is made of the following materials: 3/4" plaster, 2" glass wool, 2" air space, and 4" of low density brick. The temperature in the room, which this wall forms a part of is 70° F and the outside temperature is 0° F. For inside the room $h_1 = 1.65$ and for outside $h_2 = 6.00$

Solution: $Q = UA\,(t_1 - t_2)$

$$U = \frac{1}{\frac{1}{h_1} + \frac{L_1}{K_1} + \frac{L_2}{K_2} + \frac{L_3}{K_3} + \frac{1}{h_2}}$$

From Table I we get:

for plaster $k_1 = 2.5$ $L_1 = 3/4"$

for glass wool $k_2 = 0.27$ $L_2 = 2"$

for air space $k_3 = 0.16$ $L_3 = 2"$

for brick $k_4 = 5$ $L_4 = 4"$

$$\therefore\ U = \frac{1}{\frac{1}{1.65} + \frac{.75}{2.5} + \frac{2}{.27} + \frac{2}{.16} + \frac{4}{5} + \frac{1}{6}}$$

$$= \frac{1}{.606 + .300 + 7.41 + 12.5 + .8 + .17}$$

$$= \frac{1}{21.8} = .046 \text{ Btu/hr/sq ft/}^{\circ}\text{F}$$

$$A = 9 \times 12 = 108 \text{ sq ft}$$

$t_1 = 70$
$t_2 = 0^\circ F$

$$\therefore \quad Q = UA(t_1 - t_2)$$
$$= 0.046 \times 108 \times (70 - 0)$$
$$= 348 \text{ Btu/hr}$$

Transfer of Heat Through a Curved Wall

The equations studied so far apply to a plane surface only. You can readily see that for a curved wall, such as for a round pipe, the outside surface area is larger than the inside surface area and therefore we could not use the equation for a plane surface.

The derivation for the equation for transfer of heat through a curved surface such as a pipe requires a knowledge of calculus and therefore we will state, without proof, that it can be shown that the following equation is true:

$$Q = \frac{2\pi kZ\,(t_a - t_b)}{2.3 \log_{10} (r_o/r_1)} \qquad \ldots\ldots\ldots\ldots \quad (8)$$

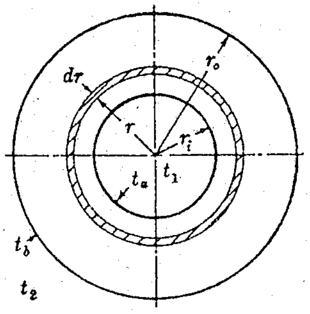


Figure 2 Curved Wall

Referring to figure 2 the definition of these symbols are:

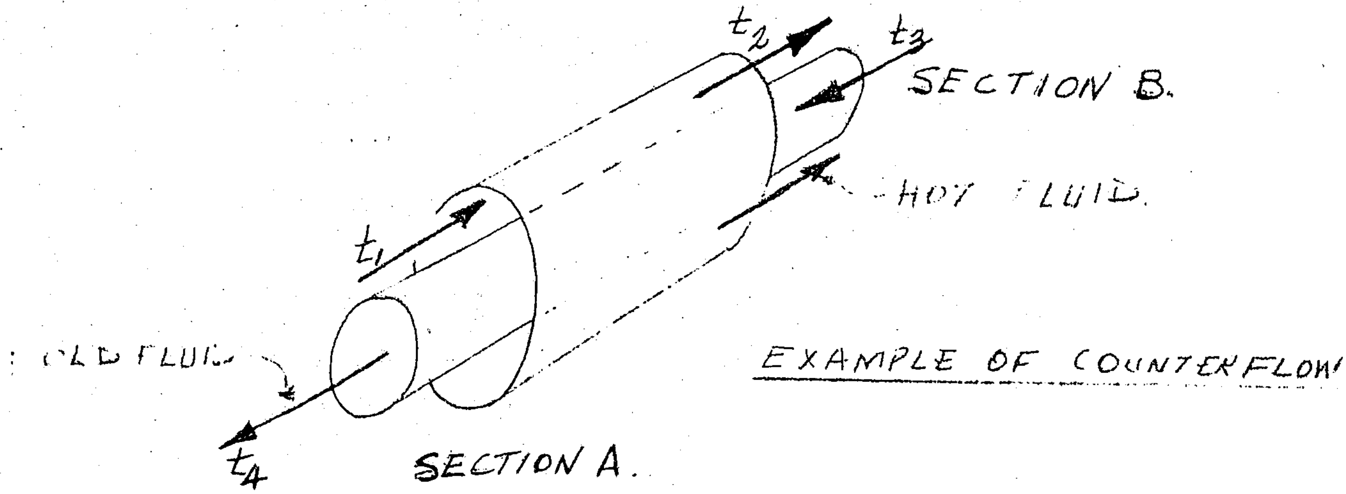
Q = rate of heat transfer, Btu/hr.
k = conductivity, Btu-in per hr. sq.ft-°F.
Z = length of pipe, ft.
t_a = temperature inside pipe, °F
t_b = temperature outside pipe, °F
r_o = outside diameter of pipe, inche
r_i = inside diameter of pipe, inches.

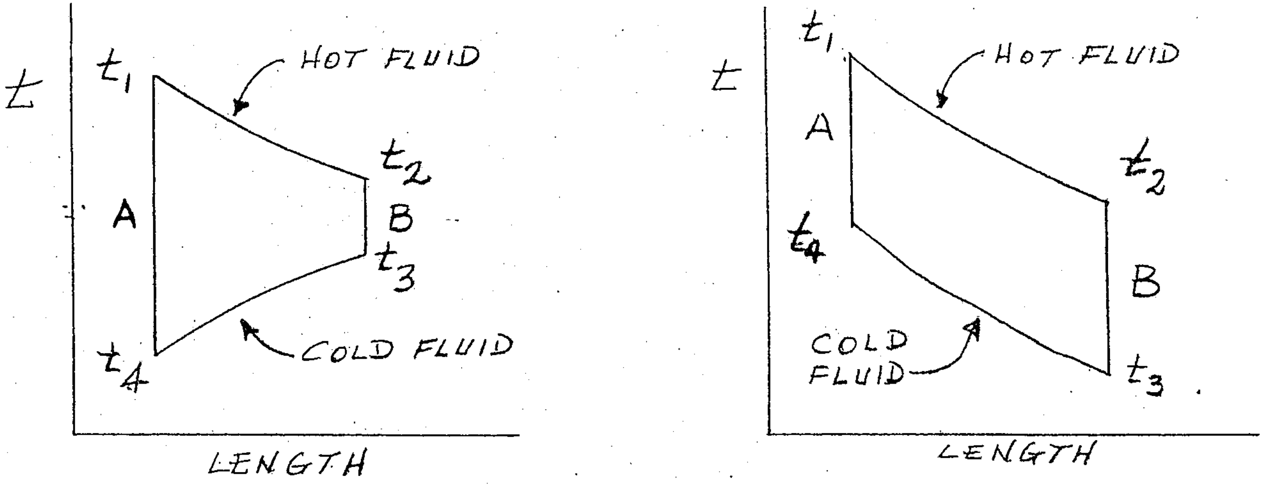
Flow in Heat Exchangers

In heat exchangers we can have parallel flow in which case the fluids flow in the same direction through the heat exchanger, and counter flow, where the fluids flow in opposite directions.

In figure 3 we have a heat exchanger represented by an oute tube and an inner tube. The hot fluid flows in the former while the cold fluid flows in the latter.

Figure 3(a) portrays graphically what happens in parallel flow. As the hot fluid flows from A to B it will give up its heat, by conduction to the cold fluid, and will drop in temperature from t_1 to t_2. The cold fluid which is flowing in the same direction will pick up heat, quite rapidly starting at section A, because the temperature difference is great, but as it approaches B, the hot liquid has a lower and lower temperature, while the cold liquid will be having a higher and higher temperature.





a) Parallel Flow

b) Counterflow

Figure 3 Temperature variations, parallel and counterflow. In each case, the temperature difference at section 'A' is t_1-t_4 and at section 'B' it is t_2-t_3.

Thus, as the two fluids approach section B, their temperature difference will be less and less; but we have learned in this lesson that the higher the temperature difference between the two fluids the greater the rate of heat transfer will be. Thus the heat exchanger will not be operating very efficiently as it approached section B. Temperature t_3 can never be equal to t_2 because of the thermal resistivity of the film and metals surfaces.

Figure 3(b) portrays graphically what happens in counterflow. In this case the hot liquid, say, flows from section A to B, and the cold liquid will now flow from sections B to A. The cold liquid thus enters at the section where the hot liquid is at its lowest temperature and flows towards the hot end of the heat exchanger. Thus the temperature difference between fluids remains fairly constant all the way along as they pass each other from one section to the other, and the temperature difference can be maintained relatively high during the whole process.

This then obviously is a much more efficient type of arrangement. In a counterflow heat exchanger it is even possible to have the cold fluid leave the heat exchanger at a higher temperature than the temperature at which the hot fluid leaves the heat exchanger - i.e. t_4 can be higher than t_2. This is often achieved in high pressure feedwater heaters in steam power stations.

Before concluding this lesson we want to introduce another term. You will notice from figure 3(a) and (b) that the temperature of the hot and cold fluids are changing as they pass through the heat exchanger and hence the temperature differences will be changing as well. Therefore to calculate a mean temperature difference the way we calculate a mean temperature normally would be inaccurate. However, in many calculations on heat exchangers we need to know the average temperature difference between the hot and cold fluid. To get around this problem we introduce what is called the "<u>logarithmic mean temperature difference</u>" which is not the same as the mean temperature difference between the hot and cold fluids. Using the subscripts for temperature as shown in figure 3, its meaning can best be expressed in equation form, thus:

$$\Delta t_m = \frac{(t_1-t_4) - (t_2-t_3)}{2.3 \log_{10}\left(\frac{t_1-t_4}{t_2-t_3}\right)} \quad \ldots\ldots\ldots \quad (9)$$

where:

Δt_m = logarithmic mean temperature difference, °F
t_1 = temperature of the hot fluid at section A, °F
t_4 = temperature of cold fluid at section A, °F
t_2 = temperature of hot fluid at section B, °F
t_3 = temperature of cold fluid at section B, °F

This equation applies whether there is parallel or counterflow in the heat exchanger.

The logarithmic mean temperature difference (Δt_m) is used in calculating the amount of heat transferred in a heat exchanger. We modify equation (7) by substituting Δt_m for $(t_1 - t_2)$ to get:

$$Q = UA\Delta t_m \quad \ldots\ldots\ldots\ldots\ldots (10)$$

where:

Q = Heat transferred, Btu/hr
U = Transmittance, Btu/hr/sq ft/$^{\circ}$F
A = surface area of tube through which heat is being conducted, sq ft
Δt_m = logarithmic mean temperature difference, $^{\circ}$F

If the heat exchanger is the tube type then the transmittance U must be calculated for a round pipe.

In conclusion note that you should be familiar with the equations given in this lesson and be able to use them for solving a problem, but they need not be memorized. In an exam any equations required from this lesson will be given.

ASSIGNMENT

1. Define the term "heat exchanger".
2. Give Fourier's Law in equation form and define the symbols used.
3. Compare the merits of air and copper as conductors of heat.
4. A wall 20 ft long x 1 ft high has steam @ 400°F on one side and water @ 40°F on the other side. The wall consists of ¼" sheet steel. Calculate the amount of heat transferred in Btu/hr.

   For steam h_1 = 1000 Btu/sq ft/$^{\circ}$F/hr; assume for water h_2 = 10 Btu/sq ft/$^{\circ}$F/hr.
5. In transferring heat in a heat exchanger is it better to have parallel flow or counterflow? Explain your answer.
6. In an oil cooler arranged for counterflow the inlet and outlet temperatures of the oil are 140°F and 92°F respectively; the inlet and outlet temperatures of the process water are 62°F and 90°F respectively. Calculate the logarithmic mean temperature difference between the oil and water. What would be the mean temperature difference between oil and water? Compare your answers.

D. Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

2 - Science Fundamentals - T.T.1

5 - Heat and Thermodynamics

-5 - Heat Loss - Insulation

0.0 INTRODUCTION

We have learned in the lesson on Heat Transfer that the rate of heat transfer takes place in accordance with the following equation.

$$Q = \frac{A\,(t_1 - t_2)}{\frac{1}{h_1} + \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} + \frac{1}{h_2}}$$

We said further that increasing one or more of the items in the denominator would cause heat to flow at a slower rate. This effect is desired when we have a piece of equipment from which flow of heat or loss of heat is undesirable. In order to slow down the rate of heat transfer, we provide insulation. Looking at the above equation, if we want to increase one or more of the terms in the denominator, we would have to increase the thickness of L, of insulation, or provide a different insulating material (that is one with a lower conductivity k,) or perhaps a combination of both.

Taking the above discussion into account, this lesson will discuss heat losses when no insulation is provided, and heat losses when insulation is provided.

1.0 INFORMATION

The value of using insulation to conserve heat becomes readily apparent when comparing heat losses from say a bare pipe, with heat losses when the pipe is properly insulated.

Losses from Bare Surfaces

The two curves in Figure 1, show a comparison between losses from a bare pipe and heat losses from a pipe that is insulated with laminated asbestos felt for various temperature differences between pipe and room.

It is quite evident that an uninsulated high temperature pipe means high heat losses. This may be due to conduction, convection and/or radiation. On the other hand, great savings can be made by providing insulation as the lower curve would indicate. Take for example at a temperature difference of 400°F between pipe and room, the bare pipe will lose 4 Btu/hr/sq ft/°F, whereas the insulated pipe will lose only 0.46 Btu/hr/sq.ft/°F.

Thus we can say that insulation conserves heat, which means a saving in fuel. Other functions performed by insulation are that it minimizes temperature drop in superheated steam lines, hot-air ducts, etc. It also ensures more comfortable working conditions in the vicinity of heated surfaces, because it prevents transfer of heat from a pipe or header into the room. Keeping heat away from where it is not wanted is often as important as keeping it where it is useful.

Figure 2 shows heat losses from bare surfaces in Btu, occurring in still air at various temperatures up to 900°F. It is estimated that the savings involved in using proper insulation will result in the insulation almost paying for itself in the first year of application.

Effect of Air Velocity

The rate of heat loss from a bare surface maintained at constant temperature is greatly increased by air circulation over the surface. Figure 3 shows this effect.

Take for example a bare surface of which the temperature is 500°F higher than surrounding temperature. For still air at 0 mph., the heat loss is 2300 Btu/sq.ft./hr. while with a wind velocity of 18 mph, the heat loss is 4900 Btu/sq.ft./hr - more than twice as much.

In the case of well-insulated surfaces, the losses are very small compared with increases shown in Figure 3. The maximum increase in heat loss due to air velocity is about 30% in the case of 1 inch thick insulation, to about 10% in the case of 3 inch thick insulation.

Heat Losses from Bare Fittings

It is often the practice to leave flanges and other fittings uninsulated because it is thought that there isn't much heat loss from them anyway. However, let us take the example of a flanged joint for a 10" pipe suitable for 125 lb. pressure. The surface area of this flange is 3.43 sq.ft. With a steam pressure of 100 psi, the heat loss from this joint over a period of a year would amount to the equivalent of more than 1 ton of coal. One can thus see the need for insulating fittings for there are hundreds of them in a steam power station.

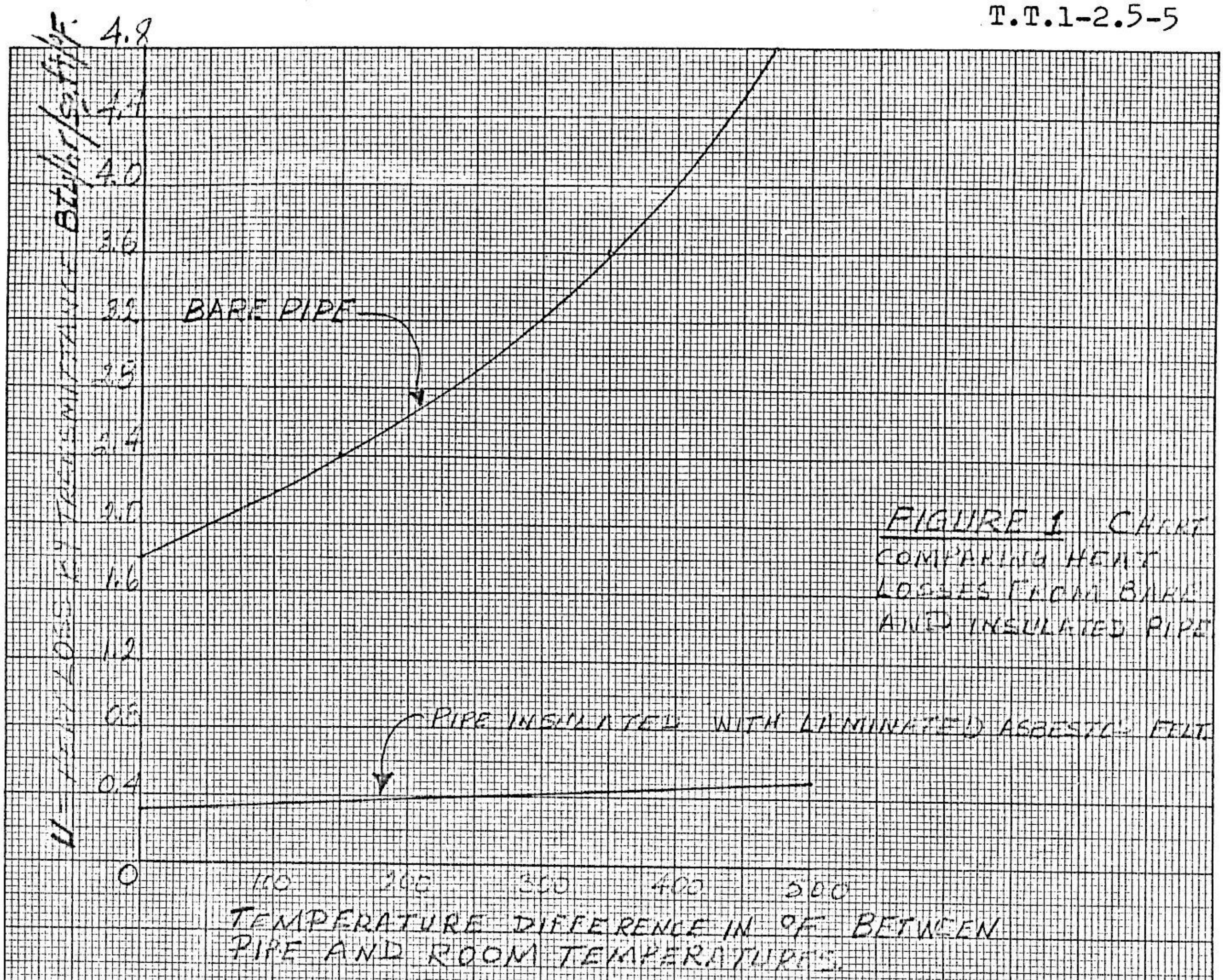


FIGURE 1 CHART COMPARING HEAT LOSSES FROM BARE AND INSULATED PIPE

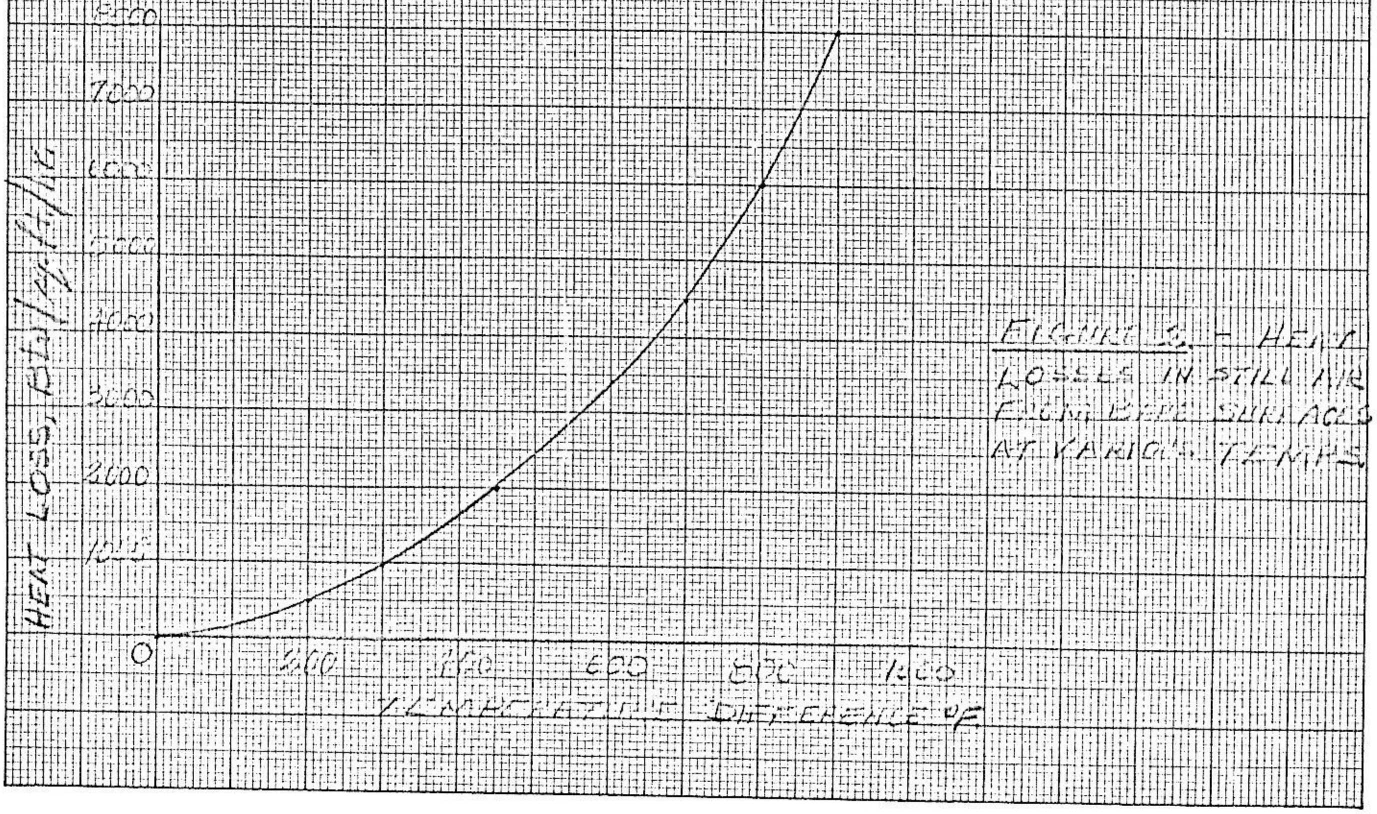


FIGURE 2 – HEAT LOSSES IN STILL AIR FROM BARE SURFACES AT VARIOUS TEMPS.

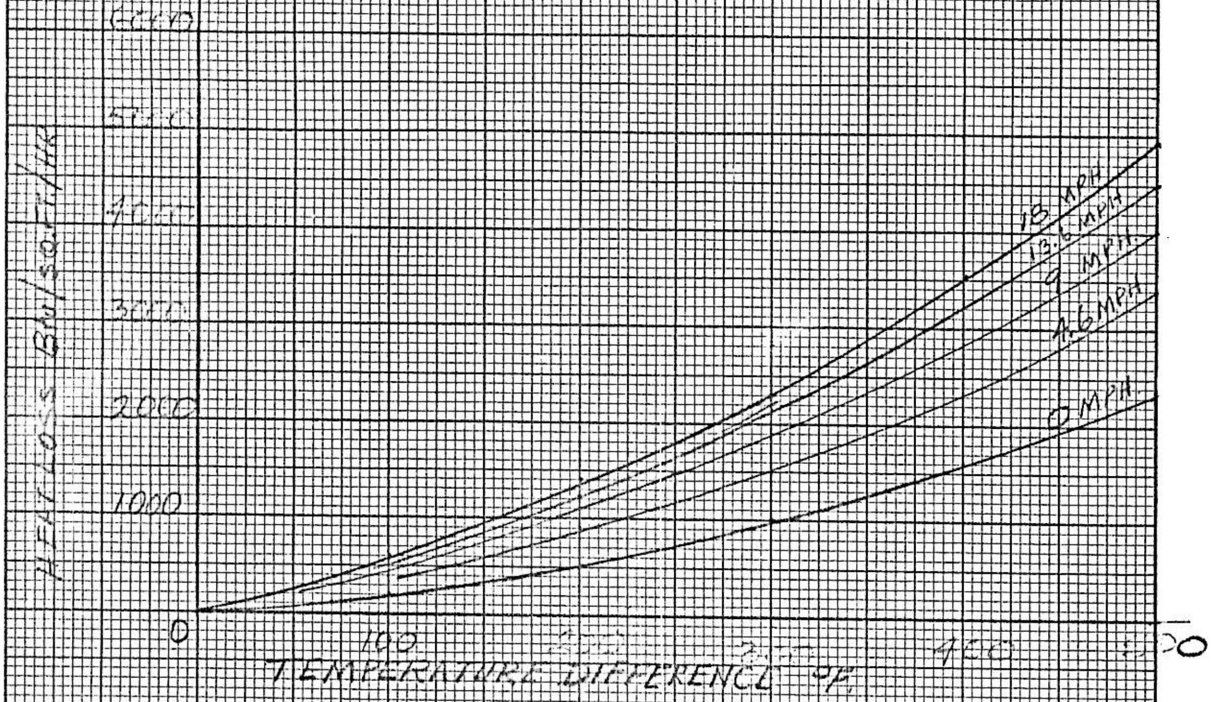


FIGURE 3 EFFECT OF AIR VELOCITY ON HEAT TRANSFER FROM A BARE SURFACE

Insulating Cold Surfaces

Surfaces such as pipes, tanks, etc., are often at temperatures below the dew point temperatures of the surrounding air. These surfaces, if uninsulated, will thus become wet due to condensation of water vapor. This condensation in turn drips on floors, equipment, etc., causing corrosion, dampness and other undesirable affects. Insulating these cold surfaces will stop this type of condensation. For this reason, circulating water or process water pipes and equipment in a steam power station are almost always insulated.

When applying this insulation, care must be taken to seal it against the penetration of moist air; moisture damages insulation causing its deterioration and reducing its insulating characteristics.

Heat Insulating Materials

The following briefly describes some of the common types of insulating materials in use to prevent heat transfer from (or to) pipes, tanks, etc.

1. Diatomaceous Silica is a mineral of low density consisting of the fossilized cellular skeletons of microscopic organisms called diatoms. It consists of almost pure silica and has a high heat resistance. It is recommended as a first layer insulation on steam lines 600 to 1200°F. It generally comes in split sections, 3 ft. long, 1 to 3 in. thick for pipe sizes up to 12" diameter.

2. 85% Magnesia is a rigid molded insulation consisting of approximately 85% by weight of basic carbonate of Magnesia and 10 to 15% of asbestos fibre with additions of clay or other cementing materials. It is recommended for temperatures up to 600°F. It is furnished in split sections 3 ft. long for piping up to 8" diameter.

3. Mineral Wool is made from clayey limestone or furnace slag. It is made into small fibres by blowing steam through the fused limestone or furnace slag to form a soft woolly type of material. It is usually used as loose fill in house insulation and for insulation in metal jackets around heaters and furnaces. It can also be made in molded form when mixed with asphaltic compound. It is recommended for use up to 1000 to 1200°F when used as loose fill bats and 1600°F to 1800°F when used as molded blocks.

4. Glass Wool is made by blowing steam through streams of molten glass to fiberize it. The composition of the glass batch and the fibre diameters determine the type of service and use - temperature limit for which various types of glass wool are recommended. It is apparently unaffected by water and other

destructive materials. Pads made of glass cloth covering loose fill are often used for insulating steam turbines. It is recommended up to 600°F.

5. Asbestos is a heat-resisting fibrous mineral, the most important deposits of which are found in Canada. Because of its incombustibility, its low heat and electrical conductivity, and its resistance to most active chemical agents, has a wide variety of uses. Examples of its use are as electrical insulation, packings and gaskets for high temperature service, cloth for theatre curtains, and heat protective clothing, roofings and building materials. It is spun and woven into yarn, rope and cloth, is formed into asbestos felt, paper, roll board,mill board for insulation, is mixed with portland cement to make fireproof shingles and asbestos wood. It can be used for insulation in the range of temperatures from 150 to 1900°F, depending on its composition.

6. Amosite Asbestos insulation consists of long amosite asbestos fibre laminated and mixed with sodium silicate binder. It can be used for hot surface temperatures up to 1050°F.

7. Calcium Silicate insulation consists of a chemical compound of lime and silica with asbestos fibre added. It can be used for hot surface temperatures up to 1050°F. also.

8. Cork-mastic - Cork is the outer bark of the cork oak. It is a good nonconductor of heat, is waterproof, is unaffected by moisture, and is slow burning. Cork-mastic is a quick drying pasty cement which is applied to the outside surfaces of pipes and other equipment which will be at low temperatures. It is used as anti-sweat insulation.

This list of insulating materials is by no means complete. There are other types of insulating materials such as expanded mica, hair felt, wool felt, cork, balsa wood, etc. The temperatures stated in the above examples are representative only and may vary quite a bit from manufacturer to manufacturer.

Uses

The following gives a few examples of the type of insulation used on various pieces of equipment found in a Nuclear Power Station such as Douglas Point.

Amorite Asbestos or Calcium Silicate - used on main steam lines, feedwater heaters and associated piping, boiler feed pumps and associated piping, station heating steam, etc.

Anti-sweat Insulation

Pre-shrunk insulating felts - used on process water system, vacuum priming system, domestic water system, circulating water system, air-conditioning system, etc.

Cork mastic - used on condenser water boxes, gravity filters, coagulation tank, activated carbon filters, cation unit tank, etc.

D. Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

2 - Science Fundamentals - T.T.1
5 - Heat and Thermodynamics
-5 - Heat Loss - Insulation
A - Assignment

1. Compare heat losses from bare surfaces with heat losses from properly insulated surfaces.
2. What effect does air velocity have on heat loss from bare surfaces?
3. Why should pipe fittings in a steam power station be insulated?
4. Why should cold surfaces in a steam power station be insulated?

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

2 - Science Fundamentals - T.T.1

5 - Heat & Thermodynamics

6 - Nucleate & Film Boiling

0.0 INTRODUCTION

Before we proceed any further let us review some of the things we have learned about heat transfer. We have said that whenever there is transfer of heat through a wall there is a temperature drop as heat progresses from the high temperature side to the low temperature side. We defined the rate of change of temperature drop with wall thickness as temperature gradient. The temperature drop depends on the conductivity k of the wall and on the wall thickness.

We said further that transmittance was a property of a material which depended not only on the type of material, but also on the thickness of material in the following manner:

$$\frac{1}{U} = \frac{1}{h_1} + \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{1}{h_2}$$

and that the rate of heat transfer in a heat exchanger takes place in accordance with the relationship

$$Q = UA\,\Delta tm$$

where

U = transmittance, Btu/hr./sq.ft./°F
h_1 & h_2=surface coefficient of heat transfer, Btu/hr./sq/ft/°F
k_1 & k_2=thermal conductivity of a material, Btu.inch/hr/sq.ft/°F
L_1 & L_2=thickness of material, inches.
Q = rate of heat transfer Btu/hr.
A = surface area across which heat is transferred, sq.ft.
Δtm = logarithmic mean temperature difference, °F.

In a mculear power station the reactor and boiler are heat exchangers. In order to get the best possible performance out of the reactor & boiler it is necessary to have the transmittance U, surface area A, and logarithmic mean temperature difference, tm as high as possible within allowable limits so that the rate of heat transfer Q is as large as possible.

You will notice from the above equation for the transmittance U that the larger the surface coefficients of heat transfer, (h_1 and h_2) are, the larger U will be. A film at the wall of a fuel rod or boiler tube tends to reduce the values of h_1 and h_2 and hence reduce the value of U.

This lesson on nucleate and film boiling deals with how water behaves when it evaporates to form steam and how the film formed by water in contact with tubes affects the rate of heat transfer.

1.0 INFORMATION

Boiling of water takes place in two ways: 1) nucleate boiling 2) film boiling.

Referring to ordinary boiling, it is well known that the vapor formed on the free surface is at first invisible, but becomes visible when heating is increased. With further increase of heat input boiling starts somewhere below the water surface; steam bubbles are formed, rise to the top surface and break through it to join the vapor above the water surface. The bubbles formed are attached to some solid particles or sorts of nuclei. Thus this is known as nucleate boiling.

In explaining film boiling we can use the example of spilling water on a hot stove. You will have noticed that when this occurs water globules dance on the hot surface for quite some time without sensible evaporation. In fact a film of superheated steam exists below each water globule and serves as a thermal insulation between surface and the water, and thus prevents the bulk of the water from fast evaporating. This is called film boiling.

However, film boiling is, only a special case of nucleate boiling. Consider the case of quenching a glowing steel bar by throwing it into water. On the metal surface an insulating vapor layer is formed through which only a little heat penetrates to the water. In this case the vapor formed cannot escape sidewards as it can in the case of the droplets on a hot stove. Therefore the vapor formed increases the thickness of the vapor film which surrounds the steel bar. When the vapor layer has grown to such an extent that its bouyancy can overcome the capillary forces which hold it in place and together, it blows up. This vapor then forms a bubble different only by its larger size from the bubbles formed in nucleate boiling. After the blowing up of the vapor film on the steel bar the film grows again and the process repeats itself until the steel bar is cool enough so that nucleate boiling takes place.

It should be noted here that most liquids can boil by either the nucleate or film process. These depend on both the heat flux Q/A and the temperature difference.

In general, boiling depends on the following conditions:

1. Heat flux entering the fluid -- i.e.

   $$\frac{\text{Rate of transfer } Q_1 \text{ Btu/hr}}{\text{surface area, } A\text{, sq. ft.}}$$

2. Surface tension of the fluid being boiled. Surface tension measures the ability of the fluid to wet the heated surface. Wettability depends on the adhesive forces between the fluid molecules and heated surface as compared to the cohesive forces between the fluid molescules.

3. Smoothness and cleanliness of the heat transfer surface. A rough clean surface will produce boiling more easily than a smooth unclean surface.

4. Temperature difference between the heated surface and saturation temperature of the boiling liquid.

5. Position of heat transfer surface - horizontal vertical or inclined.

6. Thermal conductivity of liquid & vapor.

Figure 1 shows the effect of low medium and high surface tension of the boiling liquid in forming vapor bubbles. A low surface-tension liquid readily wets the hot surface and easily surrounds the vapor bubble so that it floats off the surface to rise through the liquid.

In a medium surface-tension liquid the cohesive forces between liquid and adhesive forces between liquid and surface are about equal. The vapor must form a larger bubble before it can break away from the surface & float up through the liquid.

In high surface-tension liquids the liquid-molecule forces are so large that the liquid has trouble trying to surround the vapor and lift it off the hot surface. This causes vapor blanketing of the surface, or film boiling.

Nucleate Boiling

In recent years Heat Transfer Engineers have drawn attention to the fact that there are two separate regions in nucleate boiling. In the first region, which corresponds to low heat flux (Q/A) densities, bubbles do not interfere with one another so they

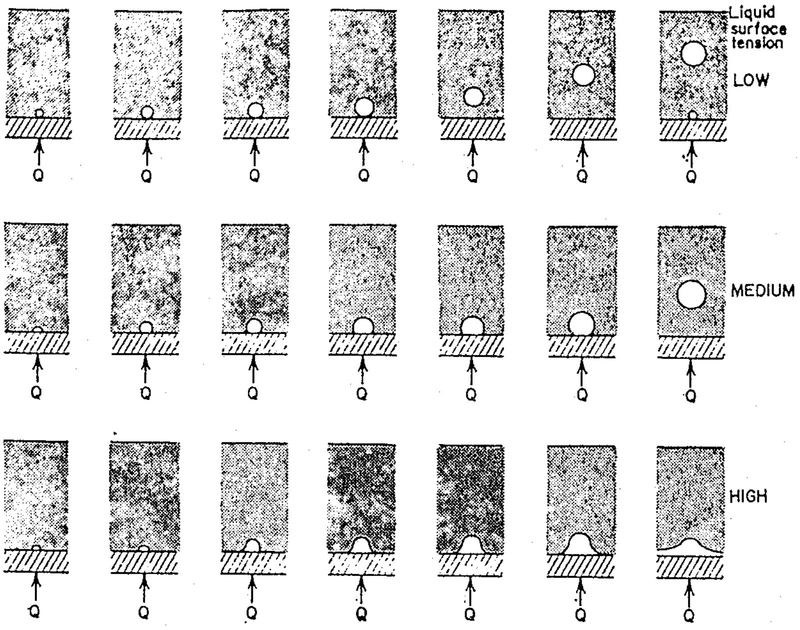


Figure 1 Steam-bubble formation and size depend on surface tension of the liquid, among other factors. As surface tension increases, the bubbles stick more tightly to the surface.

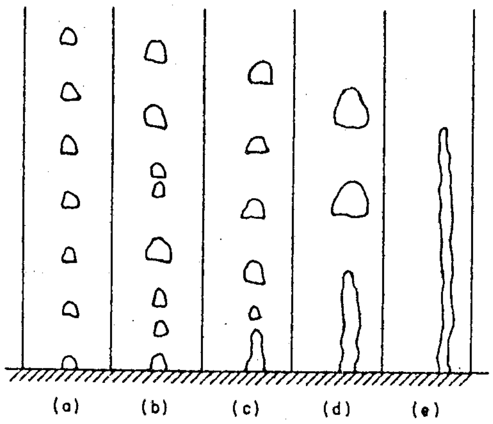


Figure 2 Stages in the transition from the region of isolated bubbles to the region of continuous columns (vapor flow increasing from a to e)

can be thought of as isolated. As the heat transfer rate is increased, the process of vapor removal from the heating surface changes from an intermittent to a continuous one as the isolated bubbles change to continuous vapor columns.

The stages of the transition process from isolated bubbles to continuous vapor columns are illustrated diagramatically in Fig. 2. In figure 2(a) the bubles are spaced sufficiently far apart so that there is no interaction between them. However, if the heat flux is increased the bubbles will be formed more rapidly and the distance between bubbles will become less and less. When the distance between two consecutive bubbles becomes smaller than some critical value, the trailing bubble influenced by the wake of the leading one rises faster, until, eventually the two agglomerate.

The stages of this development are shown in fig. 2(b to e) until at (e) a continuous vapor column is shown.

Figure 3 shows actual photographs taken of water boiling at atmospheric pressure. Figure 3 a) shows nucleate boiling in the isolated bubble region with a heat flux of 38,400 Btu/sq. ft./hr. while Figure 3 b) shows nucleate boiling in the continous column region with a heat flux of 116,000 Btu/sq. ft./hr.

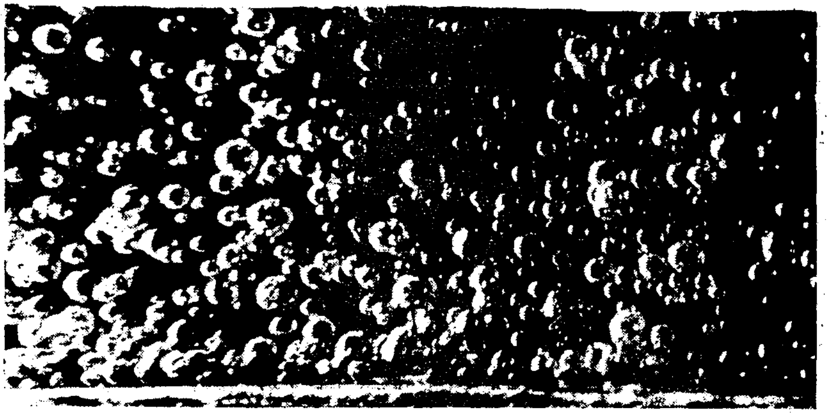


Fig. 3 a) Water at atmospheric pressure Q/A = 38,400 Btu/hr/sq.ft. isolated bubble region.

Fig. 3 b) Water at atmospheric pressure Q/A = 116,000 Btu/hr/sq.ft. continuous column region.

Burnout Heat Flux

Nucleate boiling in the continuous column region which we have discussed so far has taken place in such a way that they move parallel to each other and in a direction perpendicular to the heated surface.

If there is a still further increase of heat flux there will tend to be lateral interaction of the continuous vapor columns and the inflowing liquid in a direction parallel to the heated surface. This then limits the amount of heat which can be transfered from the heated surface. When this happens, then the upper limit of the heat transfer rate due to nucleate boiling has been reached. This is known as burnout heat flux. This is the transition point between nucleate and film boiling. Film boiling slows down the heat transfer rate because the film has an insulating effect and therefore film boiling is an undesirable state in a heat exchanger.

The foregoing iscussion on nucleate and film boiling is summarized in the graph shown in fig. 4. It shows the relation for water at atmospheric pressure (14.7 psia) and at 2400 psia. The process of heat transfer takes place in four distinct regions which are designated by a, b, c, and d and marked at the top of the graph.

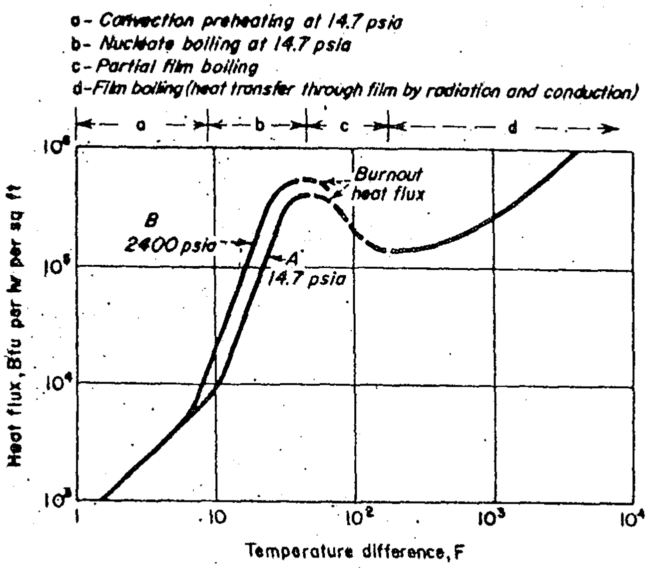


Fig. 4 Heat transferred to a boiling liquid depends on the temperature difference and whether the boiling takes place by the nucleate or the film process.

From the graph you will notice that as we raise the temperature difference from 1 to 10°F(region "a") the water is being preheated at increasing heat fluxes through the heat-transfer surface. At this stage the heat distributes through the water by conduction aided by natural convection currents.

Between 10 and 50°F temperature difference (region "b") nucleate boiling takes place. At the lower end of this range we have both nucleate boiling and superheating of the liquid with lower heat fluxes. At the higher end we have boiling only, and the heat flux reaches a maximum. This is the burnout heat flux mentioned earlier.

If the heat-transfer-surface material can stand the higher temperature, we can continue the process andincrease the temperature difference between heated surface and water above 50°F (region "c"). Notice that when we do this the heat flux, i.e. the rate of heat transfer per sq.ft. drops in spite of the fact that the temperature difference increases.
This is because we are now entering a region where we're getting nucleate and film boiling. Above 100°F temperature difference the heat flux reaches a minimum where film boiling takes over completely. In this range the heat flux drops because of lower conductivity of the steam film covering the heating surface, (h_1 is lower and therefore U is lower.)

As we continue increasing the temperature difference radiation from the heated surface will be adding to the heat flux and, then even though we're still in the region of film boiling the heat flux will again rise. Eventually the heated surface becomes so hot that it melts and fails.

Now you might ask, how does all this affect us? The type of reactors we have in Canadian Nuclear Power Plants at present are not designed for boiling liquids at all. However, there is a possibility that fuel channels can get partially plugged. This means that the flow of heavy water is not rapid enough to carry away the heat produced by the fuel rods. When this happens the fuel would heat up very rapidly and if the temperature difference between fuel rods and cooling water were allowed to get high enough then a vapor film would develop around the fuel rods, causing film boiling. But in film boiling there is a decrease in rate of heat transfer and a decrease in heat flux even though the temperature of the fuel rods (and hence the temperature difference between fuel rods and cooling liquid) is rising quite rapidly. Thus in film boiling the inlet and outlet temperatures of the cooling liquid could be quite deceiving as far as giving an indication of the temperature of the fueld rods is concerned. As the fuel rods approached film boiling temperature the outlet temperature of the cooling water would at first rise but then would decrease again. To the inexperienced this might indicate that the danger was past, where as, if the fuel rod temperature continued to rise the cladding and possibly the fuel might actually melt. For this reason there is an alarm when the outlet temperature of the cooling water is too much above the inlet temperature so that corrective action can be taken long before film boiling would occur.

In ordinary type of heat exchangers such as oil coolers, or process water coolers film boiling and subsequent melting of cooling tubes is not likely to occur because the temperatures involved are not high enough. However in heat exchangers where high temperatures are involved such that film boiling could occur, then the deceptive situation described in the preceding paragraph is a distinct possibility and should be watched for. That is, in a heat exchanger where the outlet temperature of the hot fluid is continuing to rise but the outlet temperature of the cold fluid is going down there is a possibility that film boiling is taking place.

Circulation in Boiling Water Reactor

In the case of boiling water reactors with water and steam flowing together in tubes parallel to each other, circulation problems do occur which are similar to problems encountered in coal-fired boilers. Figure 5 shows three tubes receiving different heat fluxes but connected across common manifolds with a pressure difference ΔP. The tube A receiving the greatest amount of heat produces steam more quickly, so it has the least water and most steam in its length. But because of the greater specific volume of steam this tube has the least lb. per hr. flow.

The poorer conductance of steam film allows the upper tube-wall temperature to rise. At the upper end of the tube we eventually will have a burnout. Tube C, with the lowest heat flux, will run the coolest and produce the least steam - - in fact it might even carry water up into the discharge manifold.

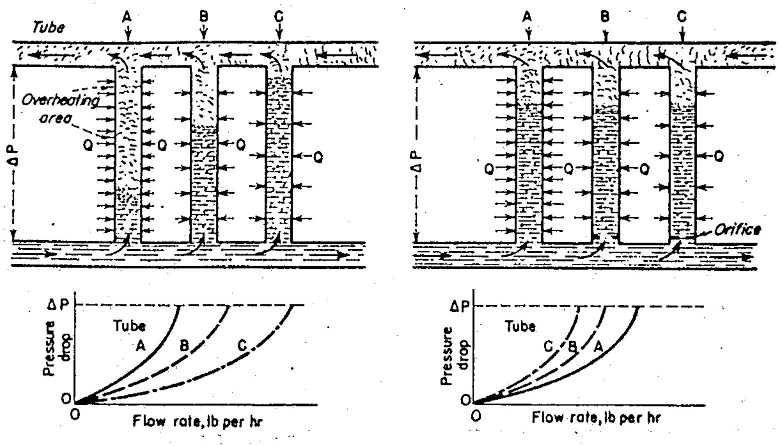


Fig. 5 Circulation in parallel tubes of the reactor or steam generator must be proportioned according to the heat input to avoid a flow unbalance and possible tube burnout, left. Orifices in the tube inlets, right, help to control pressure drops and flows.

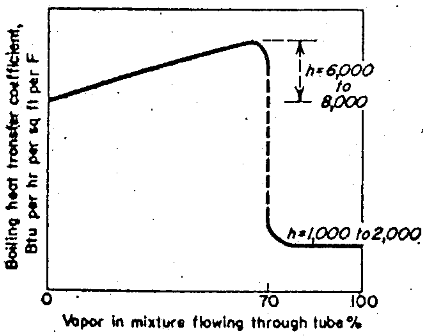


Fig. 6 Boiling heat transfer coefficient depends on the proportion of vapor in the boiling steam-water mixture.

This is an unbalanced condition causing burnout and possibly excessively wet steam. To correct this condition we must first recognize that ΔP stays constant, and we must force more flow into tubes receiving the greater amount of heat. This can be done by inserting orifices into tubes B and C having excessive flows, as in figure 5, right.

Fig. 6 shows the boiling heat-transfer coefficient "h" for various proportions of vapor in a water-vapor mixture flowing through a tube. For vapor content less than 70%, "h" varies from about 6000 to 8000 Btu/hr/sq.ft/°F. Above 70% it drops to a range of 1000 to 2000. Under laboratory conditions of a clean heat transfer surface "h" might be as high as 10,000 to 20,000 Btu/hr/sq.ft./°F.

Scale deposits on a surface may drop the heat transfer coefficient considerably. While the exact coefficient depends on the chemical condition of the water, it can drop the heat transfer coefficient for a tube roughly by about 2000 Btu/hr/sq.ft/°F. For this reason it is always desirable to keep scale away from heat exchanger tubes.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

2 - Science Fundamentals - T.T.1

5 - Heat & Thermodynamcis

6 - Nucleate & Film Boiling

A - Assignement

1. What is meant by nucleate boiling?
2. What is meant by film boiling?
3. List 6 general conditions upon which boiling depends.
4. How does surface tension affect the boiling of a liquid?
5. What two regions of nucleate boiling are there?
6. Explain burnout heat flux.
7. Why is film boiling undesirable?

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

2 - Science Fundamentals - T.T.1

5 - Heat and Thermodynamics

-7 - Psychrometry

0.0 INTRODUCTION

Psychrometry is the science which deals with water vapor in air. At the T.T.3 level in Heat and Thermodynamics in the lesson on Air and Steam and the lesson on the Dew Point Hygrometer, this subject has already been introduced. Also the lessons "Gas Laws and Gas Flow", "Gas Mixtures" and "Psychrometry" in Fluid Mechanics at the T.T.3 level have dealt with this subject.

The reader should note that when we are talking about steam produced by a boiler, then we are talking about water only in the gaseous state. Steam used in a Nuclear Power Plant is not a mixture of steam and air; the air being present in such minute quantities, if at all, that it may be entirely disregarded. So when we talk about psychrometry we are generally referring to conditions which exist in the atmosphere or air-conditioning systems, etc.

This lesson will review a number of definitions given in the lessons mentioned above as well as give several examples. Use will be made of the psychromatic chart. This chart has been introduced previously. It should be noted that the psychromatic chart is plotted for a barometric pressure of 29.92 in. of mercury. It can be used with reasonable accuracy if the barometric reading is within 29.92 ± 1 in. mercury. However, beyond this limit the accuracy of the chart begins to fall off and in this case it is more accurate to use appropriate equations to calculate psychromatic values.

1.0 INFORMATION

Generally when we speak of air, we mean moist air. However, in air conditioning we also use a term called "dry air" which is used to indicate the water-free part of moist air. Terms used to describe the condition of mixtures of air and water are defined in the paragraphs below. Some are a review from lessons you have taken previously; some are new.

Saturation - saturated air at a particular temperature and pressure holds as much water vapor as it can possibly hold. If the temperature is lowered, condensation will take place. If the temperature is raised, the air will contain superheated vapor.

Specific Humidity (w_v) is the weight of water vapor per unit weight of dry air in a vapor-air mixture. This is expressed in lbs. water vapor/lb. dry air, or sometimes also as grains water vapor/lb. dry air. (7000 grains = 1 lb). Note that this definition is sometimes also referred to as humidity ratio. This value can be obtained from the following equation:

$$w_v = \frac{0.622\, p_v}{p_m - p_v}$$

where w = specific humidity, lb/lb dry air
p_v = actual vapor pressure
pm = pressure of mixture or barometric pressure.

Dew Point Temperature is the saturation temperature corresponding to the existing humidity ratio and barometric pressure. (In practise, dew point temperature is the temperature at which condensation will just begin, when the moist air mixture under consideration is cooled at constant pressure).

Mol One mol of any substance is the number of pounds of that substance equal to its molecular weight. For example, the molecular weight of water is 18.016 and one mol of water is 18.016 lbs. of water. The molecular weight of air is 28.966 and one mole of air is 28.966 lbs.

Mol Fraction of a substance in a given amount of a homogeneous mixture is the ratio of the number of mols of the substance in the given amount of mixture to the total number of mols of all substances present in the given amount of mixture. For example, if a particular amount of moist air contains n_a mols of air and n_w mols of water, the mol fraction of air in the mixture is $\frac{n_a}{n_a + n_w}$. If we pursue this example further, and let us say that we have .01 lbs. of vapor /1 lb. of air then:

$$\text{no. of mols of air } n_a = \frac{1}{28.966} = .0345$$

$$\text{no. of mols of water } n_w = \frac{.01}{18.016} = .0005$$

Then the mol fraction of air is: $\frac{n_a}{n_a + n_w} = \frac{.0345}{.0345 + .0005} = \underline{\underline{0.985}}$

The mol fraction of water vapor is: $\frac{n_w}{n_a + n_w} = \frac{.0005}{.0345 + .0005} = 0.015$

Partial Pressure of any substance in a given vapor phase mixture is defined as the product of the mol fraction of that substance and the total pressure of the mixture. Using the values from the previous example and at 14.7 psia, the partial pressure due to water vapor for that particular case is:

Partial pressure = mol fraction x total pressure of mixture

Partial pressure = 0.015 x 14.7 = 0.22 psia.

Or another way of calculating partial pressure is by using the equation:

$$p_v = p_{s_w} - \frac{(p - p_{s_w})(t_d - t_w)}{(2831 - 1.43\ t_w)}$$

where

p = barometric pressure
P_v = actual vapor pressure
p_{s_w} = saturation pressure at wet-bulb temperature. (this can be obtained from steam tables.)
t_d = dry-bulb temperature °F
t_w = wet-bulb temperature, °F

Some pychrometric charts have vapor pressure values as well, in which case the partial pressure due to vapor can be obtained directly.

Relative Humidity has been defined in two different ways in lessons at the T.T.3 level. They are as follows:

1. Relative Humidity is the density of water vapor actually contained in the air at a certain temperature, divided by the density of water vapor, if the air were saturated at that temperature. This is express in equation form as:

$$\phi = \frac{\rho_v}{\rho_s} \times 100$$

where ϕ = relative humidity %

ρ_v = density of water vapor actually contained lb/ft^3

ρ_s = density of vapor for saturation conditions, lb/ft^3

2. Relative Humidity is the actual partial pressure of water vapor of the air-vapor mixture to the partial pressure normal for saturated water vapor at the existent dry-bulb temperature of the mixture. In equation form this is:

$$\phi = \frac{p_v}{p_{s_d}} \times 100$$

where

ϕ = relative humidity

p_v = actual vapor pressure of the mixture

p_{s_d} = saturated water vapor pressure at the dry-bulb temperature

The units in this case can be either psi or inches mercury (in Hg.)

$$\therefore \phi = \frac{p_v}{p_{s_d}} \times 100 = \frac{\rho_v}{\rho_s} \times 100$$

Relative humidity can be obtained directly from a psychrometric chart if the wet and dry bulb temperatures are known. The definition of these two temperatures are reviewed below.

<u>Dry Bulb Temperature</u> is the temperature of the mixture of air and water vapor at rest.

<u>Wet Bulb Temperature</u> is the temperature obtained from a wet bulb thermometer on a psychrometer. This is explained further, under the heading Sling Psychrometer below.

<u>Air Density</u> (ρ_a) is the pounds of air in one cubic foot. In equation form this can be expressed as:

$$\rho_a = \frac{(p - p_v)}{R\ T_d}$$

where:

ρ_a = lbs. of air/ft^3.

p = barometric pressure in either psi or in. Hg.

p_v = vapor pressure in psi or in. Hg.

R = Gas constant = 0.3704 for pressures in psi.
= 0.7541 in in. Hg.

(Note that the gas constant for air is 53.3 but since we are using pressure in psi rather than lbs/sq.ft., we have to divide R by 144. Similarly when we use pressure in in. Hg. we have to divide R by the appropriate constant to get 0.7541).

T_d = absolute dry bulb temperature = $(460 + t_d)$

<u>Vapor Density</u> (ρ_v) is the pounds of vapor in one cubic foot. In equation form:

$$\rho_v = \rho_a \times w_v$$

Mixture Density ($_m$) is the sum of lbs. of air plus lbs of vapor in one cubic foot.

$$m = a + v$$

Sling Psychrometer

The sling psychrometer has also been covered in previous courses. It is used to give an indication of dry bulb temperature and wet bulb temperature of the atmosphere. As can be seen in Figure 1, it consists of two ordinary mercury thermometers. The one on the right measures the dry bulb temperature; the one on the left has the bulb covered with a piece of light fabric gauze. This gauze is wetted with water and is therefore known as the wet bulb thermometer.

Fig.1

Prior to taking a wet and dry bulb temperature reading the sling psychrometer is whirled around by means of the handle on the upper end. As the air rushes past the wet bulb thermometer, some of the water on the gauze will evaporate. The latent heat of vaporization required for evaporation will cool the wet bulb thermometer resulting in a lower wet bulb temperature reading than the dry bulb temperature reading. The amount of difference between the wet and dry bulb temperature readings will depend on how much water vapor exists in the air. If the air is very dry - i.e. the temperature is much above the saturation temperature, then there will be rapid evaporation and the difference between the two thermometers readings will be large. If the air temperature is almost the same as the saturation temperature - that is the air is holding almost as much water vapor as it can for that particular temperature, then, there will be little evaporation as the psychrometer is whirled around. Consequently the difference between the two thermometers readings will be quite small. For saturated air there will be no evaporation and the wet and dry buld readings will be the same. The wet and dry bulb temperature readings thus obtained can be used to calculate the relative humidity, dew point, etc. of the air by using appropriate equations or by using the psychrometric chart.

For good results on the sling psychrometer, the air should be rushing past the wet bulb at about 1000 ft/min. How many revolutions a minute does this mean for the sling psychrometer? If the psychrometer is 1 ft. long from handle to wet bulb, as is the case for the one shown in Figure 1, then the circumference of the circle circumscribed by the wet bulb as it is whirled around is:

Circumference = $2\pi r$ = 2.x 3.14 x 1 ft. = 6.28 ft.

For a velocity of 1000 fpm, for the wet bulb, the number of rpm required for the psychrometer would be:

Circumference x rpm = ft/min.

$$\text{rpm} = \frac{1000}{6.28} = 160.$$

$$\text{or} \quad \text{revs/sec} = \frac{160}{60} = 2.7$$

Thus in a 10 second period the sling psychrometer should be rotated approximately 27 times to give a velocity of 1000 ft/min. For a psychrometer with a different length the number of revolutions required per 10 second period would differ. Of course, if you are measuring wet and dry bulb temperatures in a duct in which the velocity of air is already 1000 ft/min, or more, then all you have to do is stick the psychrometer into the duct and hold it stationary.

When taking psychrometric readings, wet the gauze on the wet bulb thermometer and whirl the psychrometer. Take readings of thermometers approximately every 10 seconds. When two consecutive readings on the wet bulb thermometer are the same, then this is the true wet bulb reading.

We are now ready to do some examples in psychrometry. For this purpose a psychrometric chart has been included in this lesson.

Sample Problem No. 1 - Heating & Cooling Above Dew Point

This entails no condensation of vapor. When barometric pressure and the composition remain unaltered, partial pressures remain constant. The process is represented in Figure 2 from A to B.

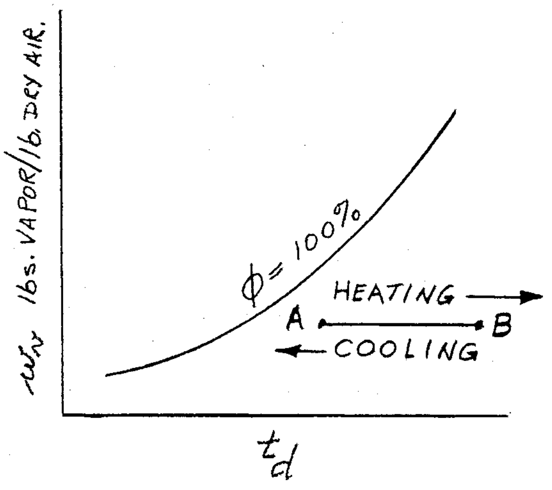


Figure 2

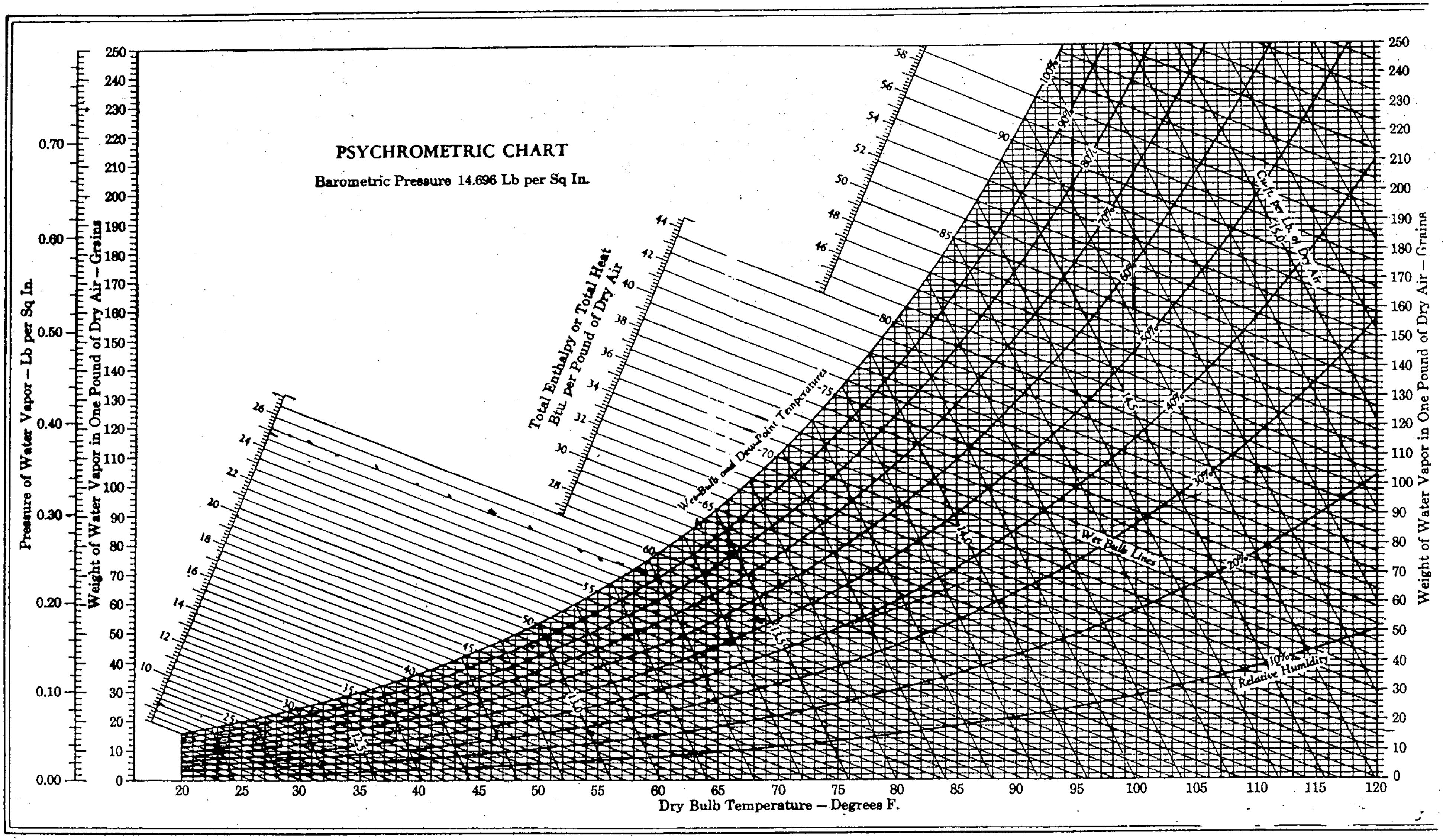
Initial Condition A

Barometric pressure p = 28" Hg.
Dry bulb temp. t_d = 60°F
Wet bulb temp. t_w = 50°F
Total Volume of air V = 1200 cu.ft.

Final Condition B

Dry bulb temp. t_d = 82°F.

Calculate vapor pressure p_v, relative humidity ϕ, specific humidity w_v, for each condition



Psychrometric chart.

plus the volume for the final condition.

Initial computed Values (at A)

Vapor pressure $p_v = p_{s_w} - \frac{(p-p_{s_w})(t_d-t_w)}{(2831 - 1.43\ t_w)}$

p_{s_w} = 0.3624 "Hg. at 50°F as obtained from steam tables for saturation conditions -
p = 28"Hg.
t_d = 60°F.
t_w = 50°F.

$$\therefore\ p_v = 0.3624 - \frac{(28-0.3624)(60-50)}{2831 - (1.43 \times 50)} = 0.26\text{"Hg.}$$

Relative Humidity: $\phi = \frac{p_v}{p_s} \times 100$

p_{s_d} = 0.52"Hg at 60°F as obtained from steam tables for saturation conditions. Note that to calculate vapor pressure we use the saturation pressure p_{s_w} at the wet bulb temperature, whereas for calculating relative humidity we use the saturation pressure p_{s_d} at the dry bulb temperature.

$$\therefore\ \phi = \frac{0.26}{0.52} \times 100 = \underline{\underline{50\%}}$$

Specific Humidity: $w_v = \frac{0.622\ p_v}{p_m - p_v} = \frac{0.622 \times 0.26}{28 - 0.26}$ = .0058 lb. vapor per lb. dry air.

Density of air: $\rho_a = \frac{(p - p_v)}{R\ T_d} = \frac{28 - 0.26}{0.7541(460+60)}$ = 0.0707 lbs air/ft^3

Total weight of air involved = $\rho_a \times V = 0.0707 \times 1200 = \underline{\underline{84.9}}$ lbs

Final computed values (at B)

When heating to a dry bulb temperature t_d = 82°F, the vapor pressure p_v, specific humidity w_v, and total weight of air are unaltered since the barometric pressure and composition remain unaltered.

Relative humidity $\phi = \frac{p_v}{p_{s_d}} \times 100$

p_{s_d} at 82°F from steam tables = 1.101"Hg.

$$\therefore\ \phi = \frac{0.26}{1.101} \times 100 = 23.6\% \text{ or } \underline{\underline{24\%}}$$

Air density at new temp. $\rho_a = \frac{(p - p_v)}{R\ T_d} = \frac{28 - 0.26}{0.7541(460+82)} = 0.0679$ lb.air/ft^3

Hence the total Volume at the new temp.

$$V = \frac{\text{Total weight of air}}{\rho_a} = \frac{84.9}{0.0679} = 1250 \text{ ft}^3$$

The above method took into account the actual barometric pressures. Now let us compare this with the values obtained when solving this same problem using the psychrometric chart which has been plotted for a barometric pressure of 29.92"Hg. (Our problem states that the conditions take place at 28"Hg barometric pressure.)

Initial Psychrometric Chart values (at A)

From the junction of the lines for dry bulb temp $t_d = 60°F$ and wet bulb temp. $t_w = 50°F$ we get:

vapor pressure from the chart in this lesson is:

$p_v = 0.125$ psi $= .125 \times \frac{30}{14.7} = 0.255$ "Hg as compared with 0.26 "Hg by means of calculations.

Relative humidity $\phi = 49\%$ as compared with 50% from calculations

Specific humidity $w_v = 39$ grains/lb.dry air $= \frac{39}{7000} = 0.0056$ lbs vapor/lb. dry air

as compared with 0.0058 from calculations.

Density of air: volume of air = 13.21 cu.ft/lb.

$\rho_a = \frac{1}{13.21} = 0.0757$ lb. vapor/cu.ft.dry air.

as compared with 0.0707 from calculations.

Total weight involved $= \rho_a \times V = .0751 \times 1200 = 90.8$ lb.

Final Psychromatic Chart values (at B)

Relative humidity $= \phi = 23\%$ as compared with 24% from calculations.

Specific volume of air = 13.77 cu.ft/lb.

Total volume of air at 82°F = specific volume x total weight
$= 13.77 \times 90.8 = 1250$ cu.ft.

Comparing the two methods, the weight of air is in error by about 7% using the psychrometric chart, i.e. almost in proportion to the barometric pressure; the others agree reasonably closely.

Sample Problem No. 2 Cooling Below The Dew Point (Dehumidification)

This entails condensation of vapor; the final atmosphere will be saturated, and some liquid will appear in the process. Refer to figure 3.

Initial conditions (Point 1)

Barometric pressure $p=29''$Hg.
Dry bulb temp. $t_d=75°F$.
Wet bulb temp. $t_w=65°F$.
Vapor pressure $p_v=0.52''$Hg
Volume of air $V=1500$ cu.ft.

Final Conditions (Point 2)

Dry bulb temp. $t_d = 45°F$

Calculate relative humidity ϕ, specific humidity w_v, density of air involved for both conditions and the vapor pressure, volume and amount of liquid formed for the final condition. Also find the dew point.

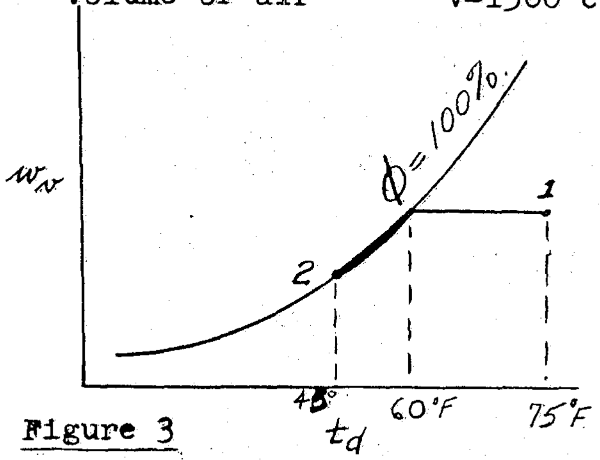


Figure 3

Initial Computed Values (Point 1)

$$\text{Relative humidity } \phi = \frac{p_v}{p_{s_d}} \times 100$$

From steam tables @ 75°F: $p_s = 0.875''$Hg.

$$\therefore \quad \phi = \frac{0.52}{0.875} \times 100 = 59.4\%$$

$$\text{Specific humidity } w_v = \frac{0.622\ p_v}{p_m - p_v} = \frac{0.672 \times 0.52}{29 - 0.52} = 0.0113 \text{ lb. vapor/ lb. dry air}$$

$$\text{Density of air} = \rho_a = \frac{(p - p_v)}{R\ T_d} = \frac{29 - 0.52}{0.7541(460+75)} = 0.0706 \text{ lb. air/ cu.ft.}$$

Total Weight of air = $0.0706 \times 1500 = 106.0$ lb. of air.

Dew point temperature is the temperature at which the vapor contained in the air would begin to condense. The dew point temperature can be looked up in steam tables. We said in the previous example that if no more vapor is allowed to enter or leave the air, then as we raise or lower the dry bulb temperature the vapor pressure p_v will remain unchanged. Thus for this example the p_v will be 0.52 at 75°F as well as at the saturation conditions or dew point. But temperature steam tables give the saturation pressure opposite the temperature column. Therefore at saturation pressure of 0.52"Hg. we read 60°F. Thus the dew point temp. = 60°F.

Final Computed Values (Point 2)

$$\text{Vapor pressure } p_v = p_{s_w} - \frac{(p - p_{s_w})(t_d - t_w)}{(2831 - 1.43\ t_w)}$$

p_{s_w} = 0.300"Hg at 45°F. Note that this pressure is to be for wet bulb temperature, which we are not given for point 2. However, since point 2 is at a temp. below the dew point we must be at saturation, then wet and dry bulb temperature are the same and therefore we can use the given dry bulb temp. $t_d = 45°F.$ here.

$t_d = 45°F.$
$t_w = 45°F$ as per the above discussion.

Note that since $t_d - t_w = 45 - 45 = 0$ the second term in the equation for vapor pressure will be 0.

$\therefore\ p_v = p_s = 0.300"Hg.$

Relative Humidity $= \phi = \frac{p_v}{p_{s_d}} \times 100 = \frac{0.300}{0.300} \times 100 = 100\%$

Specific humidity $w_v = \frac{0.622\ p_v}{p_m - p_v} = \frac{0.622 \times 0.300}{29 - 0.300} = 0.0065$ lb.vapor per lb. dry air

Density of air $\rho_a = \frac{p - p_v}{R\ T_d} = \frac{29 - 0.300}{0.7541(460+45)} = 0.0754$ lb.air/ft^3.

Total volume at point 2 $= \frac{106 \text{ lb air}}{0.0754 \text{ lb/ft}^3} = 1408$ cu.ft.

Water condensed per lb. air = w_v at point 1 - w_v at point 2 =
= 0.0113 - 0.0065 = 0.0048 lb vapor/lb. air.

Total water condensed = 106 x .0048 = 0.58 lbs.

Solution of the same problem by psychrometric chart yields:

Initial condition (Point 1)

Relative humidity = ϕ = 59%.

Specific humidity w_{v_1} = 76 grains/lb. $= \frac{76}{7000} = 0.0109$ lb.vapor/lb.air

Density of air $= \frac{1}{\text{volume/lb.}} = \frac{1}{13.7 \text{ cu.ft/lb.}} = 0.073$ lb.air/cu.ft.

Total weight of air = 0.073 x 1500 = 109.3 lbs.

Dew Point: to find the dew point on the psychrometric chart start at the junction of the lines for $t_d = 75°F$ and $t_w = 65°F$. From this point move on a horizontal line to the left till you arrive at the saturation line (100% relative humidity). At this point read the dew point temperature as 59.5°F.

Final Condition (Point 2)

From the above description on finding the dew point we arrived at the saturation line in the process of cooling. Condensation then begins as cooling continues and the cooling process will follow the saturation line until a dry bulb temperature of 45°F is reached. This means that point 2 must be on the saturation line at a point where the vertical dry bulb temperature line intersects the saturation line. At this point we find that:

Vapor pressure p_v = 0.288"Hg.

Relative humidity = 100%
Specific humidity $w_{v_2} = 44$ grains/lb. of dry air $= \frac{44}{7000} = 0.00628$ lbs/lb. of dry air.

Density of air $= \frac{1}{\text{volume/lb.}} = \frac{1}{12.84} = 0.0779$ lbs air/cu.ft of air.

Total Volume of air at point 2 = total weight of air x cu.ft./lb.
= 109.3 x 12.84 = 1405 cu.ft.

Water condensed/lb.air $= w_{v_1} - w_{v_2} = .0109 - .00623 = 0.0046$ lbs/lb.air

Total water condensed = 0.0046 x 109.3 = 0.53 lbs.

Comparing the two methods in this case indicates that at a barometric pressure = 29"Hg. the error in using the psychrometric chart is within reasonable limits.

Mixing Two Atmospheres

Mixing of two different types of atmospheres is a situation which can arise in recirculating ventilation systems, or in Nuclear Power stations where atmospheres from two different rooms are mixed, etc.

In this particular example two atmospheres (1 and 2) are mixed to form a third (3). Refer to figures 4 and 5. The state of the final atmosphere is readily found graphically on the psychrometric chart as shown in figure A. First locate points 1 and 2 on the chart which represent the states of the initial atmospheres. Then connect these two points with a straight line.

The state of the final mixture (3) will be somewhere along this line. Also the state of the final

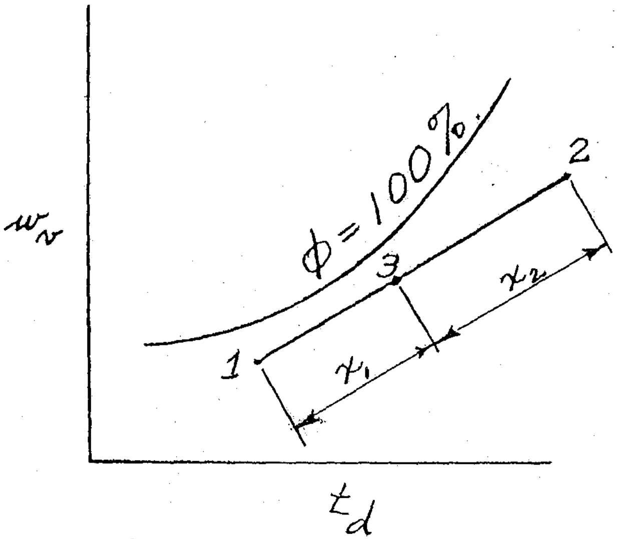


Figure 4

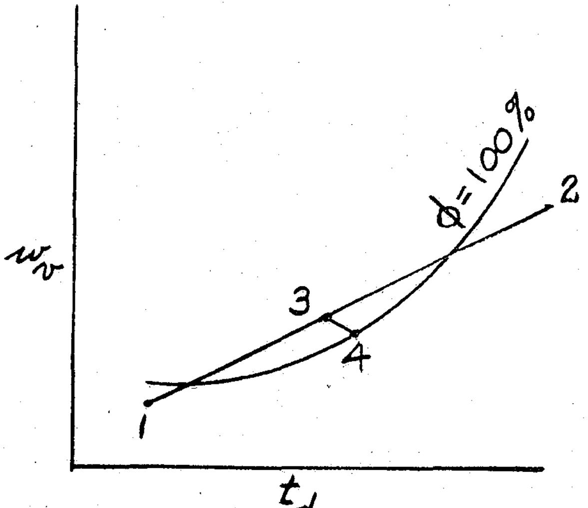


Figure 5

mixture (3) will be inversely proportional to the weights of air in the respective atmospheres. Thus all we have to do in locating point 3 on the psychrometric chart is to divide line 1-2 into segments inversely proportional to the weights of air in the respective atmospheres. This division point represents the state of the final mixture, so long as it falls below the saturation curve $(\phi = 100\%)$. If the final points (3) falls above the saturation curve as is the case in figure 5, condensation will occur and the true final point (4) is found by drawing a line from the apparent point 3, parallel to the lines of constant wet-bulb temperature to its intersection of the saturation curve.

The specific humidity may be read at all the points involved, including point 3 when it falls above the saturation line, and in this case the difference between specific humidity at point 3 and specific humidity at point 4 $(w_{v_3} - w_{v_4})$ will be the weight of water condensed in lbs/lb. of dry air.

For state point 3, the specific humidity and dry bulb temperature are the weighted averages of the specific humidity and dry bulb temperatures for the original atmospheres. The weighing factors used are the respective weights of air. This can be expressed mathematically thus:

$$W_{a_3} = W_{a_1} + W_{a_2} \quad \ldots\ldots\ldots\ldots \quad (1)$$

$$W_{a_3} w_{v_3} = W_{a_1} w_{v_1} + W_{a_2} w_{v_2} \quad \ldots\ldots\ldots \quad (2)$$

$$W_{a_3} t_{d_3} = W_{a_1} t_{d_1} + W_{a_2} t_{d_2} \quad \ldots\ldots\ldots\ldots \quad (3)$$

where:

$W_{a_1}, W_{a_2}, W_{a_3}$ represent the total weights of air at points 1, 2 & 3 respectively.

$w_{v_1}, w_{v_2}, w_{v_3}$ represent the specific humidity at points 1, 2 & 3 respectively.

$t_{d_1}, t_{d_2}, t_{d_3}$ represent the dry bulb temp. at points 1, 2 &3 respectively.

This method is not exact; the result may be in error 1 or 2 degrees F.

Sample Problem No. 3

The following two atmospheres are to be mixed (Refer to figure 5):

	Atmosphere 1 (Point 1)	Atmosphere 2 (Point 2)
Dry bulb temp. t_d	34°F	100°F
Relative humidity ϕ	90%	80%
Total volume V	1.5 cu.ft.	1.0 cu.ft.
Barometric pressure p	30"Hg.	

Using the psychrometric chart the following results can be obtained (these values can also be obtained using equations):

Specific humidity: $w_{v_1} = 0.0037$ lb.vapor/lb.air

$w_{v_2} = 0.034$ lb. vapor/lb.air

Density of air: $\rho_{a_1} = 0.080$ lb.air/cu.ft.

$\rho_{a_2} = 0.067$ lb.air/ft^3

Since total weight W = total volume V x density ρ.

at point 1 $W_{a_1} = 1.5 \times 0.080 = 0.120$ lbs. air

at point 2 $W_{a_2} = 1.0 \times 0.67 = 0.067$ lbs. air

Weight of mixed air $= W_{a_3} = W_{a_1} + W_{a_2} = 0.120 + 0.067 = 0.187$ lbs. air

From equation 2:

$$w_{v_3} = \frac{W_{a_1} w_{v_1} + W_{a_2} w_{v_2}}{W_{a_3}}$$

$$= \frac{(0.120 \times .0037) + (0.067 \times 0.034)}{0.187}$$

Specific humidity of mixture: $w_{v_3} = 0.0146$ lb.vapor/lb.air
$= 0.0146 \times 7000 = 102.2$ grains vapor/lb.air

From equation 3:

$$t_{d_3} = \frac{W_{a_1} t_{d_1} + W_{a_2} t_{d_2}}{W_{a_3}}$$

$$= \frac{(0.120 \times 34) + (0.067 \times 100)}{0.187}$$

Dry bulb temp. of mixture: $t_{d_3} = 57.6$°F.

Join points 1 & 2 with a straight line. Then Point 3 should be located where the constant specific humidity line $w_{v_3} = 102.2$ grains/lb. intersects the straight line 1-2. The vertical dry bulb temp. line $t_d = 57.6$ should intersect line 1-2 at the same place. This is a check as to the accuracy of the calculations.

Notice that point 3 falls far above the saturation curve. We said previously that it was impossible for the relative humidity to be greater than 100%. The state point giving the actual condition of the mixture must be somewhere along the saturation line. This point which we'll call point 4 can be found by drawing a line from point 3 to the saturation line, which is parallel to the wet bulb temp. lines. The intersection of this line with the saturation line is the location of point 4. A vertical line from point 4 projected downward to the dry bulb temperature scale is a reading of dry bulb temperature $t_{d_4} = 64.7°F$.

Specific humidity at 4; $w_{v_4} =$ 92 grains vapor/lb. dry air.

$= 92/7000 = 0.013$ lb. vapor/lb. dry air.

$\therefore$ Amount of Condensation $= w_{v_3} - w_{v_4} = 0.0146 - 0.013 = 0.0016$ lb/lb.dry air.

If point 3 had fallen below the saturation line then of course the actual condition of the mixture of the two atmospheres would be as at point 3 on the psychrometric chart.

Note that the student should be familiar with the equations given in this lesson and know how to use them, but they need not be memorized. Any equations required in a check-out will be given.

D. Dueck.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

2 - Science Fundamentals - T.T.1

5 - Heat and Thermodynamics

-7 - Psychrometry

A - Assignment

1. Find the specific humidity of a saturated air-steam mixture at 85°F dry-bulb temperature when atmospheric pressure is 30.5"Hg. absolute. Solve by using appropriate equations and by using the psychrometric chart.

2. Find the vapor density in question No.1. Use both methods.

3. An air vapor mixture has a dry-bulb temperature $t_d = 90°F$ and a wet bulb temperature $t_w = 64°F$. Find the dew point using appropriate equations and also using the psychrometric chart. Barometric pressure p = 28"Hg.

4. An air-vapor mixture with 72% relative humidity has a temperature of 92°F at a total pressure of 27.5"Hg. absolute. Find the dew point using appropriate equations and compare it with psychrometric chart results.

5. An air-vapor mixture at 15.2 psia total pressure and 120°F dry-bulb temperature has a wet-bulb temperature of 70°F. Find its a) specific humidity, b) relative humidity, c) dewpoint, d) specific volume. Use the psychrometric chart.

6. Two streams of air-vapor mixture merge into a common stream. One with 1 lb/sec. flow has a dry bulb temp. $t_d = 60°F$ and a wet bulb temp. $t_w = 55°F$. The other with 2 lb/sec.flow has a dry-bulb temp. $t_d = 98°F$. and wet-bulb temp. $t_w = 81°F$. Find the specific humidity and dry bulb temp. of the merged streams when the total pressure of all streams is 29.92"Hg. absolute.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

2 - Science Fundamentals - T.T.1
5 - Heat and Thermodynamics
-8 - Summary of Equations Applicable to T.T.1 level

Carnot Cycle:

$$Q_A = T_1 (s_b - s_a) \text{ Btu/lb.}$$

$$Q_R = T_2 (s_c - s_d) = T_2 (s_b - s_a) \text{ Btu/lb.}$$

$$W = (T_1 - T_2)(s_b - s_a) \text{ Btu/lb.}$$

$$e = \frac{T_1 - T_2}{T_1}$$

$$\text{C.O.P.} = \frac{Q_A}{W} = \frac{T_1}{T_1 - T_2}$$

Rankine Cycle:

$$Q_A = h_b - h_e \text{ Btu/lb.}$$

$$Q_R = h_c - h_e \text{ Btu/lb.}$$

$$W = h_b - h_c \text{ Btu/lb.}$$

$$e = \frac{h_b - h_c}{h_b - h_e}$$

Steam Quality:

$$h = h_f + \frac{x}{100} \cdot h_{f_g}$$

where:

C.O.P. = coefficient of performance in refrigeration
e = thermal efficiency %
h_b = enthalpy of steam entering steam engine, Btu/lb.
h_c = enthalpy of exhaust steam, Btu/lb.
h_e = enthalpy of condensed liquid, Btu/lb.
h = enthalpy of mixture of vapor & steam, Btu/lb.
h_f = enthalpy of saturated liquid, Btu/lb.
h_{f_g} = latent heat of vaporization, Btu/lb.
Q_A = heat added to the process, Btu.
Q_R = heat rejected from the process, Btu.

Summary of Equation Applicable to T.T.1 (cont'd)

s_a = minimum entropy in the cycle
s_b = maximum entropy in the cycle
T_1 = maximum absolute temperature °R
T_2 = minimum absolute temperature °R
W = net work done, Btu/lb. $(Q_A - Q_R)$

Heat Transfer $Q = \frac{kA\Delta t}{L}$ Btu/hr.

Heat Transfer through a Composite Wall (fluid to fluid)

$$Q = \frac{A\,(t_1 - t_2)}{\left(\frac{1}{h_1} + \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} + \frac{1}{h_2}\right)} \quad \text{Btu/hr}$$

Surface Conductance $Q = h_1\,A\,(t_1 - t_a)$ Btu/hr.

Transmittance $\frac{1}{U} = \frac{1}{h_1} + \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} + \frac{1}{h_2}$

Heat Transfer through a Curved Wall

$$Q = \frac{2\pi kz\,(t_a - t_b)}{2.3\,\log_{10}\left(r_o/r_i\right)}$$

Logarithmic Mean Temperature Difference (Heat Exchangers)

$$\Delta t_m = \frac{(t_1 - t_4) - (t_2 - t_3)}{2.3\,\log_{10}\left(\frac{t_1 - t_4}{t_2 - t_3}\right)}$$

Heat Transfer in a Heat Exchanger

$$Q = U.A.\ \Delta t_m$$

where:

A = surface area sq. ft.
h_1, h_2 = surface coefficients of heat transfer for fluid films, Btu/sq.ft/°F/hr.
k, k_1, k_2, k_3 = thermal conductivities of various material, Btu-inches/hr/sq.ft/°F.

L, L_1, L_2, L_3 = thicknesses of wall or insulating material, inches.

Q = rate of heat transfer, Btu/hr.

r_o = outside diameter of pipe, inches.

r_i = inside diameter of pipe, inches.

t_a = temp. of inside surface, °F.

t_b = temp. of outside surface, °F.

t_1 = temp. of hot fluid, °F.
= inlet temp. of hot fluid for heat exchangers, °F.

t_2 = temp. of cold fluid, °F.
= outlet temp. of hot fluid for heat exchangers, °F.

t_3 = inlet temp. of cold fluid for counterflow heat exchanger, °F.

t_4 = outlet temp. of cold fluid for counterflow heat exchanger, °F.

Δt = temperature difference between two fluids $(t_1 - t_2)$°F.

Δt_m = logarithmic mean temp. difference, °F.

U = transmittance Btu/hr./sq.ft./°F.

z = length of curved wall or pipe, ft.

Psychrometry

Specific Humidity: $$w_v = \frac{0.622\, p_v}{p_m - p_v}$$

Partial Pressure: $$p_v = p_{s_w} - \frac{(p - p_{s_w})(t_d - t_w)}{(2831 - 1.43\, t_w)}$$

Relative Humidity: $$\phi = \frac{\rho_v}{\rho_s} \times 100 = \frac{p_v}{p_{s_d}} \times 100$$

Air Density $$\rho_a = \frac{(p - p_v)}{R\, T_d}$$

Vapor Density $$\rho_v = \rho_a \times w_v$$

Mixture Density $$\rho_m = \rho_a + \rho_v$$

Mixing Two Atmospheres: $W_{a_3} = W_{a_1} + W_{a_2}$

$$W_{a_3} \cdot w_{v_3} = W_{a_1} w_{v_1} + W_{a_2} w_{v_2}$$

$$W_{a_3} \cdot t_{d_3} = W_{a_1} \cdot t_{d_1} + W_{a_2} \cdot t_{d_2}$$

where:

p = barometric pressure, either psia, or "Hg.

p_m = total pressure of mixture (or barometric pressure) psia or "Hg.

p_{s_w} = saturation vapor pressure at wet-bulb temp. psia or "Hg.

p_{s_d} = saturation vapor pressure at dry-bulb temp. psia or "Hg.

p_v = actual vapor pressure, psia, or "Hg.

R = gas constant = 0.3704 for pressure in psi.
= 0.7541 for pressure in "Hg.

t_d = dry bulb temp. °F.

t_w = wet bulb temp. °F.

t_{d_1}, t_{d_2}, t_{d_3} = dry bulb temps. for atmosphere 1 and 2 and mixture 3, °F.

T_d = absolute dry bulb temp. $(460 + t_d)$ °R.

w_v = specific humidity lbs water vapor/lb. dry air.

w_{v_1}, w_{v_2}, w_{v_3} = specific humidities of atmospheres 1 & 2 and mixture 3 respectively lbs/lb.

W_{a_1}, W_{a_2}, W_{a_3} = total weights of atmospheres 1 & 2, and mixture 3 respectively. lbs.

$\emptyset$ = relative humidity

ρ_a = density of air, lbs/ft^3

ρ_m = density of vapor plus density of air, lbs/ft^3

ρ_s = density of vapor for saturation conditions lbs/ft^3

ρ_v = density of vapor actually contained in air lb/ft^3.

D. Dueck.