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PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

73

WARSZAWA—POZNAŃ 1976

PAŃSTWOWE WYDAWNICTWO NAUKOWE

PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

poświęcone są publikacjom naukowym z zakresu teorii i badań doświadczalnych w dziedzinie mechaniki i termodynamiki przepływów, ze szczególnym uwzględnieniem problematyki maszyn przepływowych

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THE TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

exist for the publication of theoretical and experimental investigations of all aspects of the mechanics and thermodynamics of fluid-flow with special reference to fluid-flow machinery

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80-952 Gdańsk, skr. pocztowa 621, ul. Gen. Józefa Fiszera 14, tel. 41-12-71

PAŃSTWOWE WYDAWNICTWO NAUKOWE – ODDZIAŁ W POZNANIU

Nakład: 350+90 egz. Ark. wyd. 13. Ark. druk. 10. Papier druk. sat. kl. V, 70g, 70×100. Oddano do składania 9 I 1976. Podpisano do druku 12 XI 1976. Druk ukończono w listopadzie 1976 r. Zam. nr 114/159. H-5/709. Cena zł 40,–

DRUKARNIA UNIWERSYTETU IM. A. MICKIEWICZA W POZNANIU

PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH
1976 Zeszyt 73

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Losses in Refrigeration and Heat Pump Cycles

The designer of refrigeration or heat pump plants must have a clear understanding of how his choice of design parameters for the plant will effect the power input needed to run the plant. Knowing in the first instance the combination of ideal processes which will give the desired refrigeration or heating effect he must then evaluate how the actual processes involved depart from the ideal. It is the function of this article to explain how these departures from the ideal in any given process contribute to the cycle work input and to state clearly how this work quantity may be calculated. The methods described can be universally applied. They can be used to evaluate losses due to the temperature differences needed for heat transfer in heat exchangers, losses due to heat transfer from or to the working fluid in compressors and losses due to friction effects such as those occurring in heat exchanger tubes, throttle valves and lines and in the ports of the compressor.

Notation

A_e – available energy,
P – pressure,
q – heat quantity,
s – entropy,
T – temperature,
u – internal energy,
V – volume,
WI – work input.

Suffixes

e – external,
f – friction,
0 – atmosphere,
r – rejected,
s – supplied,
si – sink,
s,so – source,
soc – relating to ideal fluid,
x' – denotes ideal value of x.

1. Introduction

The purpose of a refrigeration plant is to maintain a region at an artificially low temperature with respect to the prevailing atmospheric temperature, while that of a heat pump plant is to maintain a region at a temperature higher than atmospheric. Both plants contain similar elements which are needed to force the working fluid through a cycle. In each plant there are two principal heat exchangers, one for the removal of heat from the working fluid and the other for the addition of heat to the working fluid. In the

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popular vapour refrigeration plant these are the codenser and evaporator respectively. The other principal feature is a unit in which work energy is transferred to the working fluid.

Operating in conjunction with this unit are other auxiliary heat exchangers such as intercoolers and aftercoolers. There may also be, as in the less used air refrigerator plant, a unit in which some work is obtained by expanding the working fluid. This unit is not an essential to the operation of the plant and in vapour plants is replaced by a throttle. In the heat exchangers external heat is either supplied to or extracted from the working fluid in processes where there is a finite temperature difference across the heat exchanger tubes. Similarly there may also be external heat supplied to or rejected from the working fluid in the processes of the cycle where work is transferred and here again there will be finite temperature differences. The greater these temperature differences the greater are the losses in the cycle and these losses manifest themselves in more work input being required than in the ideal case where the temperature differences are minimal. Another and most important source of loss is due to fluid friction, one common example of which occurs in a throttling process where the kinetic energy generated due to the pressure drop is transformed into heat which is absorbed by the working fluid. Again friction losses necessitate more work input in the cycle. It is the function of this paper to analyse losses in a general manner applicable to any refrigeration or heat pump cycle so that relationships for losses may be determined and studied. The results may then be applied to any actual cycle so that the various sources of losses in the cycle are pinpointed.

2. Basic thermodynamic relationships

In the analysis use is made of the energy equation

$$q_e + q_f = u_2 - u_1 + \int_1^2 P\,dV, \tag{a}$$

which applies to any thermodynamic process taking place from state point *1* to state point *2*. In equation (a) q_e is the external heat crossing the system boundary into the working fluid, q_f – the internal heat of friction due to the dissipation of work energy or kinetic energy by friction forces (1), u – internal energy and P and V – pressure and specific volume respectively. Equation (a) may be expressed as

$$\int_1^2 P\,dV - q_f = q_e + u_1 - u_2, \tag{b}$$

which gives the net work done in the process. For the process between state points *1* and *2* the entropy change is given by

$$s_2 - s_1 = \int_1^2 \frac{dq_e + dq_f}{T}. \tag{c}$$

For a system in a given thermodynamic state where the surroundings of the system are at a state defined by pressure P_0 and temperature T_0, the available energy of the system is defined as the maximum work energy obtainable from the system when, in exchanging heat with the surroundings, the system finally attains state point 0. If the initial state is state point *1* it may be shown that the available energy A_{e_1} is given by

$$A_{e_1} = u_1 - u_0 + T_0(s_0 - s_1). \tag{d}$$

For a system which is initially at state point *1* and which is subjected to any process in which it finally attains state point *2*, the decrease in available energy is given by

$$A_{e_1} - A_{e_2} = u_1 - u_2 + T_0(s_2 - s_1). \tag{e}$$

The quantity of work given in equation (e) is the maximum work output obtainable from the system if the system exchanges heat only with the surroundings in attaining state point *2*. Once the system is at state point *2*, $A_{e_1} - A_{e_2}$ is then the minimum work input needed to get the system back to state point *1* in a process where the only heat interaction is with the surroundings.

3. The refrigeration cycle

The function of a refrigeration cycle is to maintain a region at a temperature or temperatures lower than atmospheric temperature. This region is then a heat source for a working fluid in a system and in the cycle to which the system is subjected heat is finally rejected to the atmosphere which is then the heat sink. Suppose we wish to maintain a

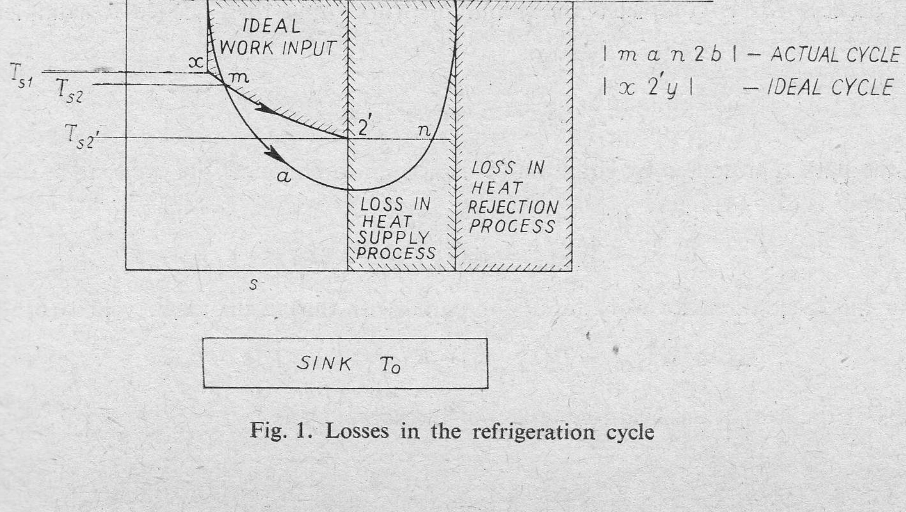

Fig. 1. Losses in the refrigeration cycle

variable temperature source (i. e. cold chambers) at temperatures T_{s_1} to T_{s_2}, all of which are below the atmospheric or sink temperature T_0. Consider Figure 1 in which the actual cycle is *1a2b1* so that in process *1a2* all the heat which is to be extracted from the source is transferred to the working fluid. It should be noted that parts *1m* and *n2* must be adiabatic expansion and compression processes respectively. The heat quantities involved in the small sub-processes are dq_{es_1} at T_{s_1}, dq_{es_2} at T_{s_2} etc., so that all the external heat supplied to the working fluid is $\Sigma dq_{es}=q_{es}$. Process *2b1* is then the heat rejection process of the cycle in which q_{er} units of heat are transferred from the working fluid to the sink. We wish to compare the cycle work input of this actual cycle with that of an ideal cycle with the same refrigeration effect or heat supplied q_{es}. The ideal heat supply process is $1\times 2'$ in which a fluid exchanges heat in a completely reversible manner with the source. This means that in the ideal heating process there is (*a*) no friction and (*b*) no temperature difference between the working fluid and the source. At state point *2'* at the end of the ideal heating process, the working fluid must then complete the ideal cycle while exchanging heat only with the sink which is at constant temperature T_0. The minimum work input to return the fluid to condition (1) is

$$A_{e_1}-A_{e_{2'}}=u_1-u_{2'}+T_0(s_{2'}-s_1). \qquad (1)$$

The work output of process $1\times 2'$ is

$$\int_1^{2'} P\,dV=q_{es}+u_1-u_{2'}. \qquad (2)$$

The ideal cycle work input is (1) - (2), i. e.

$$\mathrm{WI}_{\mathrm{ideal}}=T_0(s_{2'}-s_1)-q_{es}, \qquad (3)$$

where $T_0(s_{2'}-s_1)=q_{er'}$ is the heat rejected in the ideal cycle.

For the heating process, *1a2*, of the actual cycle the work done is

$$\int_{1}^{2} {}^{a}\, P\,dV-q_{fa}=q_{es}-u_2+u_1. \qquad (4)$$

When process *1a2* is completed the minimum work input then needed to complete the cycle is

$$A_{e_1}-A_{e_2}=u_1-u_2+T_0(s_2-s_1). \qquad (5)$$

If the path represented by equation (5) is chosen to complete the cycle, then the cycle work input is (5) - (4) i. e.

$$\mathrm{WI}_{1a21}=T_0(s_2-s_1)-q_{es}. \qquad (6)$$

For this cycle the extra work input compared with that in the ideal cycle is (6) - (3) or

$$\text{Extra } \mathrm{WI}_{1a21}=T_0(s_2-s_1)-T_0(s_{2'}-s_1)=T_0(s_2-s_{2'}) \qquad (7)$$

and this is the loss in the heating process of the actual cycle.

The actual path chosen for the heat rejection process is, however, path *2b1* for which the work input is

$$-\int_2^1 {}_b P\,dV + q_{fb} = u_2 - u_1 + q_{er} \tag{8}$$

and this work quantity is greater than the minimum needed to complete the cycle, i.e. that given by equation (5). Thus due to losses in the heat rejection process of the actual cycle the extra work input required is (8) - (5) or

$$\text{Extra WI}_{2b1} = q_{er} - T_0(s_2 - s_1)\,. \tag{9}$$

Now q_{er} is rejected to an infinite sink at temperature T_0 and the entropy increase of the sink is given by

$$q_{er} = T_0(s_2 - s_1)_{si}\,, \tag{10}$$

where *si* denotes sink conditions. Thus equation (9) may be written as

$$\text{Extra WI}_{2b1} = T_0((s_2 - s_1)_{si} - (s_2 - s_1)) \tag{11}$$

and the total extra work input of the actual cycle compared with the ideal is (11)+(7), i.e.

$$\text{Extra cycle WI} = T_0(s_2 - s_{2'} + s_{2si} - s_2),$$

$$= T_0(s_{2si} - s_{2'})\,. \tag{12}$$

For any such cycle the external heat rejected less the external heat supplied is the cycle work input and hence when one compares an actual cycle with an ideal cycle where the same heat is supplied, the extra work input is also the extra heat rejected or

$$q_{er} - q_{er'} = T_0(s_{2si} - s_{2'})\,. \tag{13}$$

The same result may be obtained in a simpler manner by considering the irreversibilities in the processes of the actual cycle. If heat is transferred from a source to a working fluid at the same temperature as the source in a process where no friction occurs then there is no net gain in entropy of the system and source taken as an entity. Thus for an elemental part of the heating process *1a2* in which dq_{es} is removed from the source at T_s to a working fluid at temperature T in a process in which friction heat quantity dq_{fa} is generated, the sum of the external and internal „irreversible" changes in entropy are

$$\frac{dq_{es}}{T} - \frac{dq_{es}}{T_s} + \frac{dq_{fa}}{T}\,. \tag{14}$$

so that for the whole heating process the irreversible entropy increase is

$$\int_1^2 {}_a \left(\frac{dq_{es}}{T} - \frac{dq_{es}}{T_s} + \frac{dq_{fa}}{T}\right). \tag{15}$$

The corresponding expressions for the heat rejection process are

$$\frac{dq_{er}}{T_0}-\frac{dq_{er}}{T}+\frac{dq_{fb}}{T} \tag{16}$$

and

$$\int_2^1 b\left(\frac{dq_{er}}{T_0}-\frac{dq_{er}}{T}+\frac{dq_{fb}}{T}\right). \tag{17}$$

The entropy changes expressed in (15) will be seen to be identical with those in (7) so that the losses in the heating process are the increases in entropy therein times the sink temperature T_0. Similarly the entropy changes in (17) are identical with those in (11) and again the product of these and the sink temperature gives the loss. The total cycle loss or extra cycle work input, given by equation (12), may be obtained from $[(15)+(17)]\times T_0$. Now in (15)+(17) the terms containing T give the cycle change in entropy for the system which is zero and hence the cycle loss is

$$T_0\left[\int_2^1 b\,\frac{dq_{er}}{T_0}-\int_1^2 a\,\frac{dq_{es}}{T_s}\right]=q_{er}-T_0\int_1^2 a\,\frac{dq_{es}}{T_s}. \tag{18}$$

This equation indicates that to find the cycle loss we need only concern ourselves with the net entropy increase suffered by the source and the sink. If the cycle is completely reversible the loss is zero and from (18)

$$\frac{q_{er'}}{T_0}=\int_1^{2'} a\,\frac{dq_{es}}{T_s} \tag{19}$$

and if T_s is constant then

$$\frac{q_{er'}}{T_0}=\frac{q_{es}}{T_s} \tag{20}$$

and we are dealing with a Carnot cycle.

In the actual cycle the source was in effect considered to be infinite, so that when heat quantity dq_{es} is extracted from an individual cell of the cold chamber, the cell temperature remains constant at T_s. In fact in any actual refrigerator the low temperature region remains at constant temperature only because the heat supplied to the system is replaced by the same quantity of heat which leaks into the region from the atmosphere. Thus in fact the heat quantity q_{es} is obtained originally from the atmosphere or sink at temperature T_0. Hence in the cycle q_{es} is extracted from the sink and q_{er} is rejected to the sink and we have a cycle operating in conjunction with a single heat reservoir, i. e. the atmosphere. For the cycle

$$q_{er}-q_{es}=\text{cycle WI} \tag{21}$$

and the net result of a cycle is that the sink suffers an increase in entropy given by

$$\frac{q_{er}-q_{es}}{T_0}=\frac{\text{cycle WI}}{T_0}. \tag{22}$$

Thus all the work input is degenerated to heat causing an irreversible entropy increase given by equation (22). Since in fact there is no other sink at a lower temperature than T_0, there is no way of recovering any of this heat as work energy. These latter remarks would apply even if the cycle were ideal.

4. The heat pump cycle

In the heat pump cycle a region is maintained at a temperature or temperatures above that of the atmosphere. In the cycle heat is supplied to the working fluid from the atmosphere which is then the heat source for the heating process of the cycle and the region is maintained at an artificially high temperature by rejecting heat from the working fluid to the region in the heat rejection process of the cycle. This heat supplied to the region finally leaks back to the atmosphere. Comparison of different heat pump cycles should be made by considering that in each cycle the same heat is rejected.

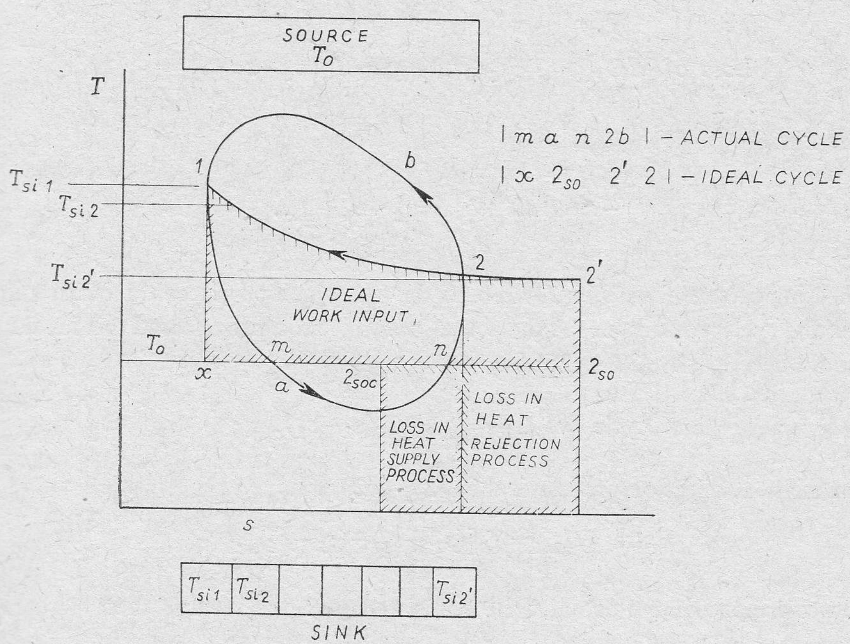

Fig. 2. Losses in the heat pump cycle

Considering Figure 2, suppose that we wish to maintain a sink at temperatures T_{si_1}, T_{si_2}, $T_{si2'}$ by supplying to the sink heat quantities dq_{er_1}, dq_{er_2} etc. so that $q_{er} = \int dq_{er}$ is the heat rejected by the system to the sink. Since the atmosphere is the source for the cycle, the ideal cycle which will effect this heat rejection process is $1 \times 2_{s0}\, 2'1$, where the area under the line $1 \times 2_{s0}$ is the heat supplied and the area under $2'1$ is the heat rejected q_{er}.

The ideal cycle work input is then

$$\mathrm{WI}_{\mathrm{ideal}} = q_{er} - T_0(s_{2'} - s_1), \tag{23}$$

where $q_{es'} = T_0(s_{2'} - s_1)$ is the ideal heat supplied. Suppose that the actual cycle is *1a2b1* in which the heat rejected in *2b1* is again q_{er}. For process *1a2* where heat is supplied parts *1m* and *n2* are adiabatics and for *1a2* the work done is

$$\int_1^2 {}^a P\,dV - q_{fa} = q_{es} - u_2 + u_1 . \tag{24}$$

The work quantity given by equation (24) is less than that which could have been obtained by taking a reversible path between *1* and *2* while the system interacts with the source. This maximum work output is given by

$$A_{e_1} - A_{e_2} = u_1 - u_2 + T_0(s_2 - s_1). \tag{25}$$

Hence in process *1a2* the loss of work output is (25)–(24) or

$$\text{Loss of } \mathrm{WD}_{1a2} = T_0(s_2 - s_1) - q_{es} = T_0\left(s_2 - s_1 - \frac{q_{es}}{T_0}\right). \tag{26}$$

For process *2b1* the work input is

$$-\int_2^1 {}^b P\,dV + q_{fb} = u_2 - u_1 + q_{er}. \tag{27}$$

If the heating process from *1* to *2* had been the one which produced the maximum work output then the cycle work input for the actual heat rejection process and this ideal heating process would be given by (27)–(25) or

$$\text{cycle } \mathrm{WI}_{1-2b1} = q_{er} - T_0(s_2 - s_1). \tag{28}$$

The loss in the heat rejection process is then (28)–(23) or

$$\text{Extra } \mathrm{WI}_{2b1} = T_0[(s_{2'} - s_1) - (s_2 - s_1)]. \tag{29}$$

The extra work input of the actual cycle compared with the ideal is then

$$\text{Extra } \mathrm{WI}_{2b1} + \text{Loss of } \mathrm{WD}_{1a2} = T_0(s_{2'} - s_1) - q_{es} = q_{es'} - q_{es} = T_0\left[(s_{2'} - s_1) - \frac{q_{es}}{T_0}\right] \tag{30}$$

which is the total cycle loss.

It will be seen from equation (30) that, as the total loss increases, the heat supplied reduces below the ideal value. The entropy q_{es}/T_0 in equations (26) or (30) may be written as

$$\frac{q_{es}}{T_0} = (s_2 - s_1)_{soc}, \tag{31}$$

where $(s_2-s_1)_{soc}$ is the entropy increase suffered by a fluid which takes part in a completely reversible interaction with the source, where the heat supplied to the fluid is q_{es}, i.e. the heat supplied in the actual process *1a2*.

From equations (26) and (31) the loss in the heating process is then

$$\text{Loss of work}_{1a2}=T_0[(s_2-s_1)-(s_2-s_1)_{soc}] \tag{32}$$

and from equations (29) and (31) the loss in the heat rejection process is

$$\text{Loss of work}_{2b1}=T_0[(s_{2'}-s_1)-(s_2-s_1)]. \tag{33}$$

The total loss, from equations (30) and (31) is

$$\text{Total cycle loss}=T_0[(s_{2'}-s_1)-(s_2-s_1)_{soc}]. \tag{34}$$

These losses are shown in Figure 2 and may also be obtained from the irreversibilities in the various processes of the cycle. Thus (32) for the heating process may be written as

$$\text{Loss of work}_{1a2}=T_0\int_1^2 {}_a\left[\frac{dq_{es}}{T}+\frac{dq_{fa}}{T}-\frac{dq_{es}}{T_0}\right] \tag{35}$$

and for the heat rejection process equation (33) may be written as

$$\text{Loss of work}_{2b1}=T_0\int_2^1 {}_b\left[\frac{dq_{er}}{T_{si}}-\left(\frac{dq_{er}}{T}-\frac{dq_{fb}}{T}\right)\right]. \tag{36}$$

For the total cycle loss (35)+(36) gives

$$T_0\left[\int_2^1 {}_b\frac{dq_{er}}{T_{si}}-\int_1^2 {}_a\frac{dq_{es}}{T_0}\right] \tag{37}$$

which is the same as (34).

Since in the actual cycle the heat rejected leaks back to the atmosphere, the heat pump cycle in effect operates in conjunction with one heat reservoir which is the atmosphere and for any cycle

$$q_{er}-q_{es}=\text{cycle work input} \tag{38}$$

and the net effect in the cycle is to produce an entropy increase in the atmosphere of

$$\frac{q_{er}-q_{es}}{T_0}=\frac{\text{cycle work input}}{T_0} \tag{39}$$

which is an irreversible increase in entropy and is a minimum for the case of the ideal cycle.

5. Concluding remarks

Inherent in this method of analysis is the separation of friction losses from those due to temperature differences. The method may therefore be used at the design stage to assess the effect on overall loss in the cycle of the choice of diameter in heat exchanger tubes, since on this choice depends the pressure drop due to friction and the temperature difference needed for heat transfer. The method may also be used to compare, on a basis of the losses which are likely to occur, different possible plants for the same duty and finally may be used in a given plant to pinpoint the individual processes which are major contributors to the overall loss in the cycle.

Short illustrative examples on the analysis of losses in refrigeration and heat pump cycles are contained in questions 1 and 2 and in Appendix.

Received by Editor, May, 1974.

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Appendix

1. A refrigeration circuit consists of two compressors, a condenser, two throttles and a cold chamber and uses ammonia as the working fluid. The intermediate pressures between the throttles and the compressors are such that the temperature of the ammonia at these points is the atmospheric temperature, 60°F, which is also of course the sink temperature of heat rejected in the cycle. The cold chamber is maintained at a constant temperature of 30°F and in each cycle 458 Btu/lb ammonia is removed from the cold chamber to the ammonia. There is no undercooling in the condenser and the heat rejected from the ammonia goes directly to the atmosphere. During evaporation in the cold chamber and condensation in the condenser 20 deg F temperature difference is allowed for heat transfer.

The compression in the LP compressor is adiabatic, the friction heat generated being 10 Btu/lb ammonia and the compression line is assumed to be straight on the T/s diagram. Compression in the HP compressor is cooled so that 10 Btu/lb ammonia is transferred directly to the sink. This rejection of heat from the ammonia coupled with the internal heat generated due to friction in the compression process is such that the state line during compression follows an isentropic path.

Determine the actual cycle work input needed to maintain the cold chamber at 30°F under these circumstances. Also find the minimum cycle work input required and give a detailed account of the losses that occur in the individual parts of the actual cycle.

Solution. In Figure 3 a circuit diagram for the refrigerator plant is shown together with a T/s diagram for the ammonia in the cycle.

Determination of entropy and enthalpy values of the ammonia. All work and heat quantities are in Btu/lb ammonia and entropy values in Btu/lb °R.

With 20 deg F temperature difference for heat transfer the evaporation temperature is 10°F (470°R) and the condensing temperature is 80°F (540°R).

At state point *6*

$$T_6 = 80°\text{F} \quad \therefore \quad P_6 = 153 \text{ lbf/in}^2,$$

$$h_6 = 132 \quad \text{and} \quad s_6 = 0.2749.$$

At state point *1*

$$T_1 = 60°\text{F} \quad \therefore \quad P_1 = 107.6 \text{ lbf/in}^2,$$

$$h_1 = 132 = 109.2 + x_1(627.3 - 109.2),$$

$$\therefore \; x_1 = 0.0441,$$

$$\therefore \; s_1 = 0.2322 + 0.0441(1.2294 - 0.2322)$$

$$= 0.2762.$$

At state point *2*

$$T_2 = 10°\text{F} \quad \therefore \; P_2 = 38.51 \text{ lbf/in}^2,$$

$$h_2 = 132 = 53.8 + x_2(614.9 - 53.8),$$

$$\therefore \; x_2 = 0.1412,$$

$$\therefore \; s_2 = 0.1208 + 0.1412(1.3157 - 0.1208)$$

$$= 0.2898.$$

At state point *3*

$$h_3 = 132 + 458 = 590,$$

$$\text{and } s_3 = 0.2898 + \frac{458}{470} = 1.2638.$$

LP compression *3–4*, $T_4 = 520°\text{R}$

$$\therefore \text{ internal heat} = 10 = \int_3^4 T ds = \frac{T_3 + T_4}{2}(s_4 - s_3),$$

$$\therefore \; 10 = \frac{470 + 520}{2}(s_4 - 1.2638),$$

$$\therefore \; s_4 = 1.2840.$$

These values of T_4 and s_4 suggest that point *4* is in the superheat field and at a pressure of 73.32 lbf/in² (where the saturation temperature is 40°F and the saturation entropy and enthalpy 1.2618 and 623) a temperature of 60°F (20 deg F superheat) gives an entropy of 1.2838 and an enthalpy of 634.95. The entropy value of 1.2838 will be taken to be identical with $s_4 = 1.2840$ so that $h_4 = 634.95$.

At state point *5*

$$\text{here} \quad P_5 = P_6 = 153 \text{ lbf/in}^2$$

$$\text{and} \quad s_5 = s_4 = 1.2840.$$

These figures give 75.4 deg F superheat and $h_5 = 680.2$.

The various enthalpy and entropy values are shown in Figure 3.

Cycle work input:

The cycle work input is the work input in the two compressors.

For the LP cylinder, $WI = h_4 - h_3 = 634.95 - 590 = 44.95$.

For the HP cylinder, $WI = h_5 - h_4 + q_{er\ 4-5} = 680.2 - 634.95 + 10 = 55.25$.

Total actual cycle work input $= 44.95 + 55.25 = 100.2$.

The minimum cycle work input is that for a reversed Carnot cycle between 490°R and 520°R with 458 Btu/lb as the heat supplied in the cycle.

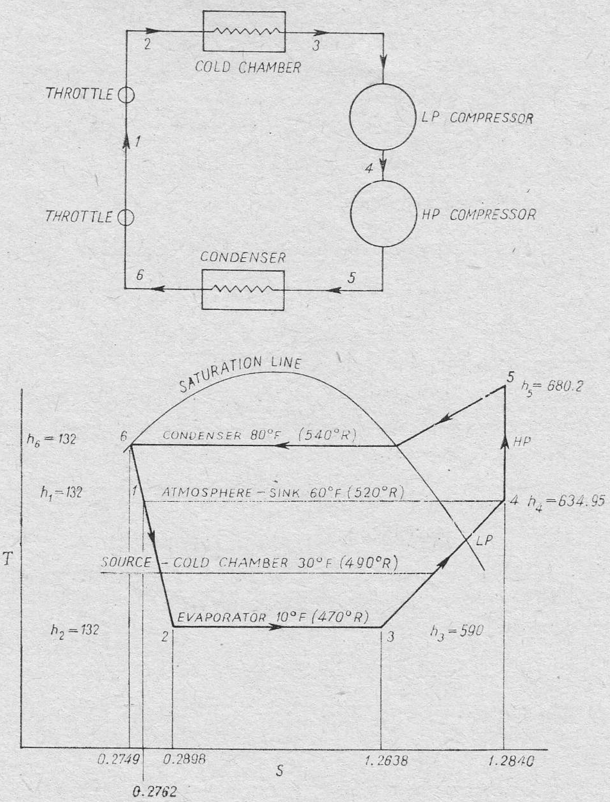

Fig. 3

Thus minimum cycle work input

$$= \frac{458}{490}(520 - 490) = 28.0.$$

Hence the total loss of cycle work input in the actual cycle is $100.2 - 28.0 = 72.2$.

Losses in the various parts of the cycle:

1 - 2 throttle loss $= T_0(s_2 - s_1) = 520\ (0.2898 - 0.2762) = 7.07$,

2 - 3 loss in evaporator

$$= T_0\left[(s_3 - s_2) - \left(\frac{q_{es}}{T_{so}}\right)\right]$$

$$= 520\left[(1.2638 - 0.2898) - \frac{458}{490}\right] = 20.8,$$

3 - 4 loss in LP compressor $= T_0(s_4 - s_3) = 520\ (1.2840 - 1.2638) = 10.5$,
4 - 5 loss in HP compressor

$$= T_0\left[\left(\frac{q_{er}}{T_0}\right) - (s_5 - s_4)\right]$$

$$= 520\left(\frac{10.0}{520} - 0\right) = 10.0,$$

5 - 6 condenser loss

$$= T_0\left[\left(\frac{q_{er}}{T_0}\right) - (s_5 - s_6)\right]$$

$$= 520\left[\frac{680.2 - 132}{520} - (1.2840 - 0.2749)\right] = 23.3,$$

6 - 1 throttle loss $= T_0(s_1 - s_6) = 520\ (0.2762 - 0.2749) = 0.676$.
The total loss in the actual cycle is thus

$$7.07 + 20.8 + 10.5 + 10.0 + 23.3 + 0.676 = 72.346.$$

and the discrepancy in this figure compared with 72.2 may be attributed to inaccuracies involved when interpolating in tabulated values of ammonia properties.

Since the heat removed from the cold chamber originally came from the atmosphere, it follows that in one actual cycle the increase in entropy of the atmosphere is

$$\frac{\text{actual cycle work input}}{T_0} = \frac{100.2}{520} = 0.1921$$

and this figure should be compared with the minimum possible increase in the entropy of the atmosphere namely

$$\frac{\text{ideal cycle work input}}{T_0} = \frac{28}{520} = 0.0538.$$

2. A heat pump to provide water at 150°F from cold water at 60°F uses Freon 11. Heat input is from the atmosphere at 60°F and 20 deg F temperature difference is allowed for heat transfer in the evaporator. The Freon is dry saturated at compressor suction, the compression is adiabatic with a measured enthalpy increase of 22 Btu/lb, the condensing temperature is 160°F and the Freon is undercooled to 100°F before entering the throttle valve.

Compare the actual work input required with the minimum which would be required with a reversible cycle using an ideal fluid, then find the losses as they occur in the actual plant.

Solution. *T/s* diagrams for the Freon 11 and for an ideal fluid are shown in Figure 4. The properties of Freon 11 are shown in the table below. The pressures are in lbf/in^2, enthalpies in Btu/lb Freon and entropies in Btu/lb °R.

Saturation properties of Freon 11

°F	P	h_f	h_g	s_f	s_g
40	7.032	15.89	97.11	0.0346	0.1972
100		28.27		0.0580	
160	61.04	41.23	111.12	0.0798	0.1926

Properties of superheated Freon 11 vapour at 61.04 lbf/in^2

°F	h	S
180	114.10	0.1973
200	117.09	0.2019
220	120.12	0.2065

Evaluation of the enthalpy and entropy at the various points in the Freon 11 circuit – see Figure 4. All enthalpies in Btu/lb freon and entropies in Btu/lb °R.
At state point *1*

$$h_1 = 97.11 \qquad s_1 = 0.1972.$$

At state point *2*

$$h_2 = 97.11 + 22 = 119.11,$$

$$T_2 = 200 + \frac{119.11 - 117.09}{120.12 - 117.09}(20),$$

$$= 200 + \tfrac{2}{3}(20) = 213.3°\text{F}$$

and

$$s_2 = 0.2019 + \tfrac{2}{3}(0.2065 - 0.2019) = 0.2050$$

At state point *3*

$$h_3 = 28.27, \qquad s_3 = 0.0580$$

At state point *4*

$$h_4 = 28.27,$$

$$\therefore\ 28.27 = 15.89 + x_4(97.11 - 15.89),$$

$$\therefore\ x_4 = 0.152,$$

$$\therefore\ s_4 = 0.0346 + 0.152(0.1972 - 0.0346)$$

$$= 0.0592.$$

Heat supplied from the Freon 11 to the water

$$= h_2 - h_3 = 119.11 - 28.27 = 90.84$$

and the work input required

$$= h_2 - h_1 = 22.$$

The ideal fluid circuit:

The water is heated from 60°F ($h_f = 28.1$ Btu/lb water) to 150°F ($h_f = 117.9$ Btu/lb water) and hence the heat taken in by the water is

$$117.9 - 28.1 = 89.9 \text{ Btu/lb water}.$$

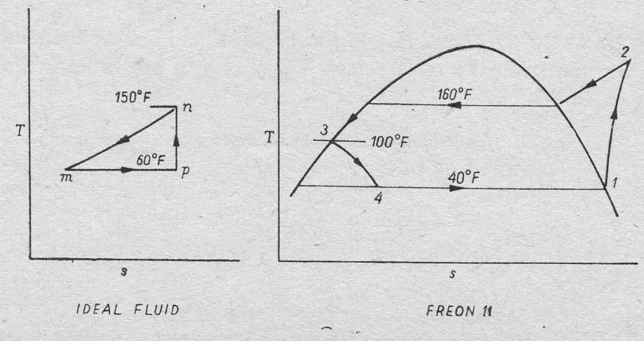

Fig. 4

The ideal cycle which will affect this heating of the water is *mpnm* for the ideal fluid in Figure 4. Here the ideal fluid absorbs heat from the atmosphere in process *mp* at 60°F, is then compressed isentropically (p - n) to 150°F. Thereafter in process *nm* heat is rejected by the ideal fluid in a completely reversible manner to the water. The minimum work input in this cycle is

$$89.9-520(0.2149-0.0555)=6.9 \text{ Btu/lb water},$$

where 0.2149 and 0.0555 are the entropies of water at 150°F and 60°F respectively.

The circulation of Freon 11 per pound of water is

$$\frac{89.8}{90.84}=0.989$$

and hence the work input in the actual cycle is

$$0.989\times 22=21.8 \text{ Btu/lb water}.$$

The total loss of work is thus

$$21.8-6.9=14.9 \text{ Btu/lb water}.$$

Breakdown of the losses in the actual cycle:

All values are in Btu/lb water.

1 - 2, loss = 520 (0.2050 − 0.1972) 0.989 = 4.025,

2 - 3, loss = 520 [(0.2149 − 0.0555) − (0.2050 − 0.0580) 0.989] = 7.38,

3 - 4, loss = 520 (0.0592 − 0.0580) 0.989 = 0.625,

$$4\text{ - }1,\ \text{loss}=520\left[(0.1972-0.0592)-\frac{(97.11-28.27)}{520}\right]0.989=2.93.$$

Hence the total loss in the actual cycle is

$$4.025+7.38+0.625+2.93=14.96 \text{ Btu/lb water}.$$

Straty w obiegach chłodniczych i grzejnych

Streszczenie

Pobór mocy dla zrealizowania obiegów chłodniczych lub grzejnych określony jest przez parametry projektowe instalacji pompowej. Podstawą wyboru parametrów projektowych jest znajomość idealnych przemian termodynamicznych, które mają zapewnić pożądany efekt chłodniczy lub grzejny. Projektant musi jednak ocenić, w jakim stopniu procesy rzeczywiste odbiegają od procesów idealnych. Celem niniejszego artykułu jest wyjaśnienie wpływu takich odstępstw na pracę włożoną do układu oraz przedstawienie sposobu obliczania wartości tej pracy. Opisane metody postępowania mogą być stosowane dla dowolnych obiegów. Można ich użyć do oceny strat spowodowanych różnicami temperatur podczas przekazywania ciepła w wymiennikach ciepła i podczas wymiany ciepła w sprężarkach oraz strat spowodowanych tarciem w rurach wymienników ciepła, zaworach dławiących, przewodach i przepustach sprężarek.

Потери в циклах холодильников и тепловых насосов

Резюме

Мощность для осуществления циклов холодильников и тепловых насосов определяется их проектными параметрами. Основой для подбора проектных параметров является знание идеальных термодинамических превращений, которые должны обеспечить необходимый холодильный или отопительный эффект. Однако проектировщик должен оценить, в какой степени действительные

процессы отличаются от идеальных процессов. В настоящей статье поставлена цель выяснить влияние таких отклонений на работу затраченную в системе, а также представить способ расчета значения этой работы. Представленные методы производства могут быть применены при любом цикле. Их также можно применить при оценке потерь, вызванных разницей температур во время отдачи тепла в теплообменниках и во время теплообмена в компрессорах, а также потерь, вызванных трением в трубах теплообменников, в дроссельных клапанах, в трубопроводах и в пропусках компрессоров.