main directed at the regulations prevailing in the country to which the contributor belonged. It was evident that there was a general acceptance of a distributed lane loading used in conjunction with a local concentration as the most suitable form for designers.

Some disappointment was expressed that contributions on the assessment of the strength of existing bridges was not as extensive as it might have been, and it was hoped that later written contributions might remedy this. The need to ascertain the maximum carrying capacity of old, or for that matter, new bridges, was one which must be a problem in most countries; here, it seemed, the sharing of individual experience could be of great value.

ASSESSING EXISTING BRIDGES

In those countries where more than one design load was used, there was a ready-made basis for classification; this might not, however, be entirely satisfactory, since the objects of design and assessment were to some extent different. In the former the aim was to provide a satisfactory bridge for general traffic with an adequate life, while the purpose of the latter was to ascertain the heaviest actual vehicles which could be permitted to use a bridge, in some cases at least, with the acceptance of a materially shortened life of the bridge. From the preliminary contributions it appeared that in some countries, higher stresses were acceptable for assessment than were permitted in design; that is to say, the load factor was reduced.

Old or complex structures presented considerable difficulty in assessment. The technique of testing to assist in reaching a conclusion seemed to be used most extensively in Sweden and Great Britain, and in the report on this theme attention was drawn to some of the difficulties inherent in this approach to the problem. In addition to those of interpretation and extrapolation of results, consideration was directed to the philosophic problem of the stresses acceptable in this technique; on the one hand it might be claimed that, as all relieving factors had been automatically taken into account in the test, somewhat lower stresses than those acceptable in design should be used, while on the other hand, since certain adverse unknown factors were similarly absorbed into the test results, higher acceptable stresses were justified. From the opinions expressed in the discussion, the latter conclusion seemed to be favoured.

From the Swedish and British contributions it appeared that the calculation of the strength of masonry arches continued to present major difficulties. A great deal of work had already been done on this intractable subject, particularly by the British Building Research Station, and one of the British contributions provided a description of a 110 ton testing vehicle specially designed by Ministry of Transport for the continuation of this research.

* * *

"QUEEN ANNE" BLAST FURNACE RELINED

After the completion of re-lining, the Queen Anne blast furnace at the works of the Appleby Frodingham Steel Company, Scunthorpe, Lincolnshire, a branch of the United Steel Companies, Limited, will have a hearth diameter of 31 ft. The furnace is one of the firm's famous four "Iron Queens," the others being Queen Mary, Queen Bess and Queen Victoria; it was commissioned in March, 1954, with a hearth diameter of 27 ft., a bosh diameter of 30 ft. 3 in., a throat diameter of 22 ft., and a height of 100 ft. The working volume was 42,372 cub. ft.

The re-lining operation began on July 21 and the furnace is expected to be back in operation on about September 1, when the bosh diameter will be 34 ft. 3 in. and the throat diameter 23 ft. 9 in. The effective volume of the furnace will be 51,615 cub. ft. and the height will remain unaltered at 100 ft.

During the first campaign, the Queen Anne furnace produced 800,000 tons of iron.

THE DOMESTIC HEAT PUMP

STANDARD EQUIPMENT OF THE FUTURE

By M. Komedera, B.SC. (ENG.)

The heat pump is by no means a new idea. The principle of it was first proposed to the Glasgow Philosophical Society by Lord Kelvin about 100 years ago, but it was nearly 80 years before Lord Kelvin's outlined procedure became more than an experiment.

The premises of the Southern California Edison Company in Los Angeles were heated and cooled by a heat pump as long ago as 1932. The Ohio Power Company had one plant in 1940. Similarly, in Switzerland, Zurich Town Hall utilised the heat of the nearby river in 1938, and following this, plants were installed in the Zurich Congress Building and Lanquart Paper Mills. In England, the Norwich Electricity Corporation installed a heat pump just before the war for heating and cooling its new buildings. There followed the Festival Hall heat pump, the story of which is well known.

During the past few years considerable attention has been given in the U.S.A., Switzerland, Germany and to a lesser extent in this country to the numerous possibilities offered by the system. One main advantage is of course, the low operating cost since 2 to 5 units of heat are given out for each heat unit of electricity consumed.

With regard to the domestic type of heat pump, very little progress was made before the war, as there was plenty of cheap coal available everywhere, and the development did not seem justified. However, times have changed and coal has become dearer—so have electricity, gas and oil. The problem of heating must still be solved. Furthermore, the Beaver Report has unveiled the truth about smog, and the general public realises now as never before something of the harm that is done to the whole nation by atmospheric pollution. It is thus obvious that smog-producing fuels should be eliminated from domestic heating and that the energy to be used in the future for that purpose will be electrical. Research into this field therefore must concentrate on improved methods of electrical heating.

Engineers know very well that the production of electricity is wasteful; in fact, the thermal efficiency of a good power station is only about 28 per cent. This means that only 28 per cent. of the total heat of combustion available in the coal is converted into electrical power, the remainder being rejected to cooling water, radiation and friction losses, etc. Allowing 2 per cent. for transmission and distribution losses, the final efficiency of 26 per cent. must be compared with an overall efficiency of 30 to 60 per cent. obtained by the direct burning of fuel. The difference in these efficiencies reflects clearly the competitive disadvantage of electrical heating. Yet from the point of view of the consumer, electric heating is ideal, since it eliminates dirt, soot and fumes, requires no stoking or ash removal, and lends itself to automatic control.

No big improvement in the overall efficiency of power stations can be envisaged during the next few years. Power consumption is rising very steeply and it is estimated that within 20 years the consumption will be trebled. Nuclear power will not mean unlimited supplies of electricity for heating purposes and, although this extra power will make a substantial contribution to the generation programme, it will not solve fuel crises, unless wastage of energy is stopped and the efficiency of heating appliances increased.

A possible method of using electrical energy in a more efficient manner is by employing the heat-pump cycle. The high coefficient of performance (C.O.P.), which usually lies between 2 and 5, raises the overall efficiency for electrical heating from 28 per cent., to over 100 per cent., and at the same time all the advantages of electrical heating are retained. Since the heat pump also gives cooling with the same equipment that is used for heating, it offers a substantial advantage over other methods. It is easy to see, therefore, why the electrical industry is taking more and more interest in the developments which are carried out in this field. The heat pump seems to be the only logical answer to the present fuel economy problem, since nearly all the heat losses at the power station can be regained by the consumer.

HEAT PUMP CYCLE

The principles of the heat pump are similar to those of the refrigerator and require only a knowledge of the behaviour of vapours under varying conditions of temperature and pressure. The vapour or refrigerant is the working fluid used in the system. In the case of a domestic refrigerator and small plants of automatic type, a refrigerant known as "Freon 12" (or Arcton 6) is almost invariably used. The full name of Freon 12 is dichlorodifluoromethane, and its chemical formula is $C\,Cl_2\,F_2$. Fig. 1 shows the heat pump circuit; the operation differs from that of a refrigerator only in that the heat extracted from the low temperature source is utilised instead of being dissipated.

Consider the heat pump cycle, and assume that Freon 12 is used as a refrigerant. Starting

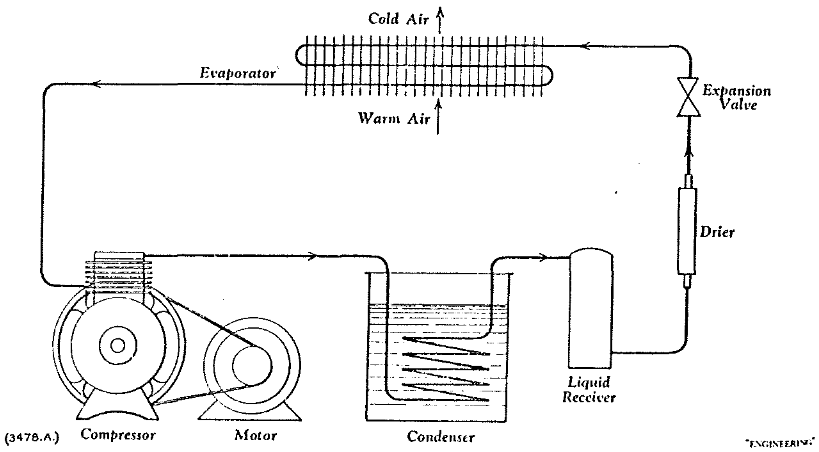

Fig. 1 The elements of a heat pump circuit are the same as those of a refrigerator, and consist basically of a compressor, condenser, expansion valve and an evaporator.

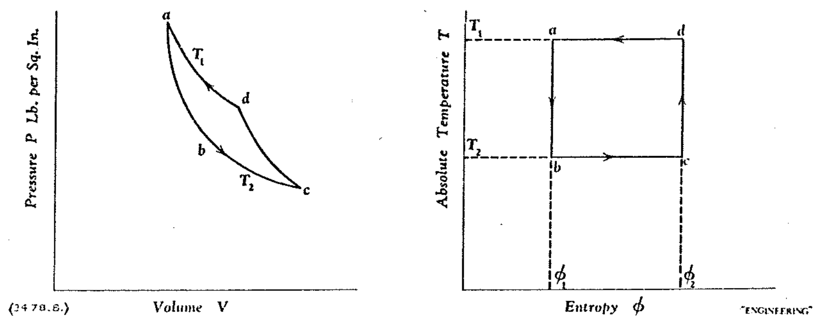

Fig. 2 The Carnot cycle for a refrigerator shown on (a) P-V and (b) T-ϕ diagrams.

with the compressor, Freon gas is induced into the cylinder and compressed to a pressure of, say, 206 lb. per sq. in. gauge, which corresponds to a saturation temperature of Freon of 140 deg. F. Thus, if this hot gas is passed through a coil immersed in water (the condenser) which is at a temperature of, say, 60 deg. F., the gas will be cooled down and liquified. The liquid is collected in a receiver which serves as a storage bottle and from there it passes through the expansion valve, which both controls the rate at which the refrigerant is admitted to the evaporator and lowers the pressure of the vapour, and therefore the saturation temperature. At, say, 10 lb. per sq. in. gauge, the saturation temperature is 2 deg. F. Now this cold vapour at low pressure is passed through the evaporator coil and extracts the heat from the surrounding air. Even when the air temperature is at 32 deg. F. it is still warmer by 30 deg. than the Freon vapour at 10 lb. per sq. in., and extraction of heat necessary for the evaporation of the Freon is still possible. Thus it can be seen that the equipment, as a heater, has a better performance than as a cooler, because the total heat output is equal to the energy input plus the heat absorbed from the heat source.

Before an investigation of the economy which is possible with a heat pump can be carried out, a theoretical analysis must be made. The object of this is to show the theoretical possibilities, limitations and difficulties with which engineers are confronted when using this particular system.

CARNOT CYCLE

The theoretically perfect refrigerator, using air as a working fluid, may be regarded as a reversed Carnot engine in which heat is absorbed at a temperature T_2 and dissipated at a temperature of T_1, where T_2 and T_1 are measured on the absolute scale. The operating cycle of this perfect refrigerator is shown on the pressure-volume (P-V) diagram and also on the temperature-entropy diagram (T-ϕ) (Fig. 2). The same lettering has been used for both diagrams to facilitate comparison.

The cycle commences at point *a*, and the clearance volume of air is expanded adiabatically to *b*. During this expansion the temperature changes from T_1 to T_2. The air expands further isothermally to *c* during which process the heat necessary to keep the temperature constant is absorbed from a cold body (evaporation). Next, adiabatic compression takes place along *c-d*, which causes the temperature to rise to T_1. To complete the cycle, further compression takes place at constant temperature, and consequently, heat is rejected to a hot body (condensation).

The ideal efficiency of such a cycle (i.e., Carnot cycle efficiency) can easily be determined from the T-ϕ diagram. The total heat supplied is represented by the area under isothermal *a-d* which is equal to $T_1(\phi_2 - \phi_1)$. The heat absorbed is given by the area under the isothermal *c-b* and is equal to $T_2(\phi_2 - \phi_1)$.

Now work done

$$= \text{total heat supplied} - \text{heat absorbed}$$
$$= (T_1 - T_2)(\phi_2 - \phi_1)$$

Thus Carnot cycle efficiency

$$= \frac{\text{work done}}{\text{heat supplied}}$$
$$= \frac{(T_1 - T_2)(\phi_2 - \phi_1)}{T_1(\phi_2 - \phi_1)}$$
$$= \frac{T_1 - T_2}{T_1} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

In refrigeration practice the efficiency as such is seldom used, since a refrigeration engineer is only interested in the net refrigerating effect, and a quantity known as coefficient of performance (C.O.P.) is used which is defined as the ratio of $\frac{\text{net refrigeration effect}}{\text{work done}}$, and this is equal to $\frac{\text{heat absorbed from a cold body}}{\text{work done.}}$

This will become, therefore:

$$\frac{T_2}{T_1 - T_2} \quad . \quad . \quad . \quad . \quad (2)$$

In the case of a heat pump the C.O.P. is defined as a ratio of $\frac{\text{heat available at the condenser}}{\text{work done}}$ and this can be expressed as:

$$\frac{T_1}{T_1 - T_2} \quad . \quad . \quad . \quad . \quad (3)$$

Re-arranging this expression gives:

$$\frac{T_1}{T_1 - T_2} = 1 + \frac{T_2}{T_1 - T_2} \quad . \quad (4)$$

which shows that the C.O.P. of a heat pump is greater by unity than the C.O.P. of a refrigerator operating between the same limits of temperature.

PRACTICAL HEAT PUMP CYCLE

In practice, some modifications are necessary. The diagram (Fig. 3) shows the modified T-ϕ and P-H cycles of a heat pump using liquid and vapour as a working fluid.

At this stage it must be made clear that the reason for using a liquid and vapour instead of a perfect gas is that at predetermined temperatures latent heat allows considerable transfer of heat to be made, and also evaporation can take place at a low pressure and condensation at a higher pressure.

Considering the modified cycle, as in Fig. 3, it will be observed that the diagrams are similar to the Rankine cycle using superheated steam, and in the writer's opinion, the basis of comparison should be the Rankine cycle rather than the Carnot cycle.

The pressure-enthalpy (P-H) diagram shown in Fig. 3 is the most usual and convenient form of diagram for refrigeration work, because the evaporation and condensation occur at constant pressure. Furthermore, the changes in enthalpy of the refrigerant during one complete cycle are easily determined if the conditions of the vapour during the various processes are known.

The compression *c-d* is shown to be adiabatic. In practice, however, using vapour as a refrigerant, the compression follows a law $PV^n = C$ and not $PV^\gamma = C$, and in the case of small compressors using Freon 12, the adiabatic index *n* is sometimes taken as 1·15, which is nearer the isothermal operation. Furthermore, the expansion is not an exact constant-total-heat operation (throttling), and a small divergence can be observed, especially when the liquid is undercooled. For these reasons the efficiency of a practical heat pump cycle can only be about 0·75 to 0·8 of the ideal cycle (Carnot).

What, then, is the coefficient of performance that can be expected in practice? Using equation (3) the theoretical coefficient of performance can be found for various evaporation temperatures (heat source) and condensation temperatures (discharge). Fig. 4 shows the variation of the C.O.P. with evaporation temperature, assuming condensing temperatures of 120 deg. F., 140 deg. F. and 160 deg. F. An analysis of these graphs shows that a high C.O.P. can only be obtained with high evaporative temperatures.

It will, of course, be noticed that with a higher temperature of condensation the coefficient of performance becomes correspondingly lower.

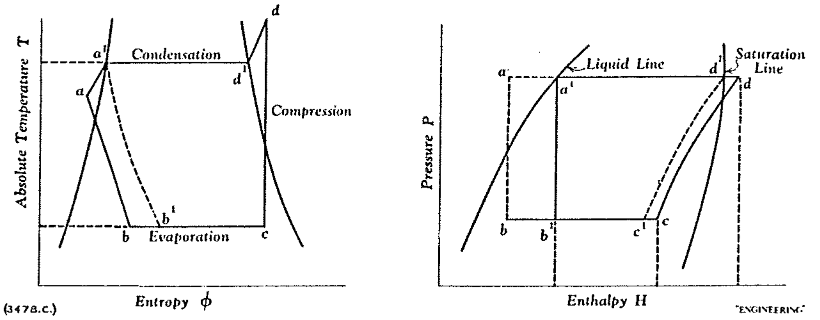

Fig. 3 The practical heat pump diagram approaches the Rankine cycle. The T-ϕ diagram is on the left, and the P-H diagram is on the right. cd = compression, da = condensation, ab = expansion, bc = evaporation, dd' = superheating, ac' = undercooling.

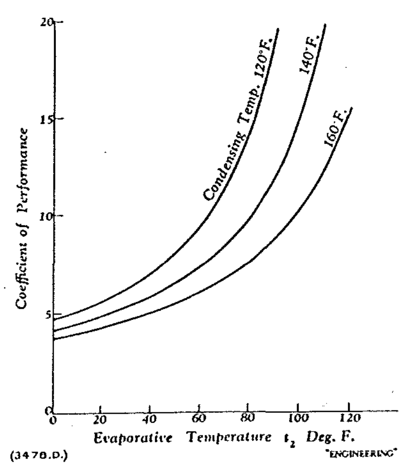

Fig. 4 The coefficient of performance varies with both the input and output temperatures.

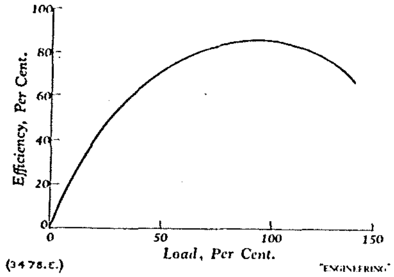

Fig. 5 Load-efficiency curve for ½ h.p. induction motor.

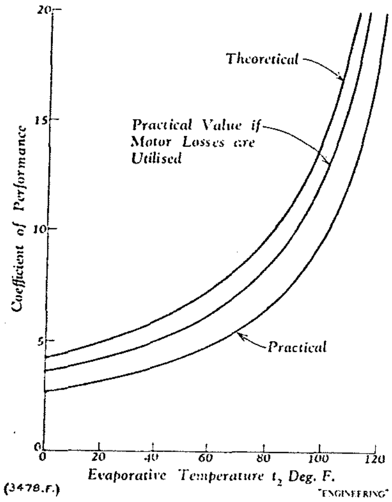

Fig. 6 Comparison of theoretical and actual C.O.Ps. The practical value is improved if the heat losses of the motor and compressor can be usefully absorbed in the system.

Thus it is seen that the actual (as distinct from the theoretical) C.O.P. of a heat pump is dependent upon suction and discharge pressure.

This, of course, is the first complication in the design, since if water is required to be heated to say 140 deg. F., pressures in the region of 250 lb. per sq. in. are to be expected (using Freon 12). Furthermore, the heat source available is usually of low temperature, such as air, and suction pressures of 10 to 20 lb. per sq. in. can be expected (which correspond to evaporative temperature of 2 deg. F. to 18·5 deg. F.) This will apply particularly in winter. However, even under these adverse conditions a reasonably high theoretical C.O.P. can be expected.

In order to determine practical C.O.P. a further analysis is necessary and the mechanical and electrical efficiencies of a heat pump system must be investigated.

PRACTICAL EFFICIENCIES

It is very difficult to estimate the overall efficiency of a small compressor, since this quantity varies considerably with the size, type and make. Such information is not generally available, but in many cases 3 h.p., 1 h.p. or ½ h.p. compressors may have an efficiency of only 60 to 70 per cent. Insufficient area in the suction valves will throttle the vapour in its return to the compressor. Too large a clearance is another source of trouble and this is particularly important in the case of the heat pump, where condensation is expected to take place at abnormally high temperatures. Furthermore, any restriction in the delivery valve will require additional power and frictional losses may be high.

A more definite picture can be obtained with regard to the electrical efficiency. The motor efficiency depends on the size, make of motor and its load, and Fig. 5 shows a typical efficiency curve for a 1 h.p. single-phase induction motor. In order to obtain maximum efficiency the motor must be fully loaded.

For small powers, the electrical efficiency of the motors should be between 78 and 82 per cent. and for larger installations it should be slightly higher. In the case of the sealed type of motor compressor, which is nowadays used by the majority of refrigerator manufacturers, the overall efficiency of the drive should be used in the design.

It is difficult to estimate the actual efficiency of a heat exchanger since many factors influence its performance. However, in general, an efficiency of 90 to 96 per cent. may be assumed for a well designed coil. The type of exchanger used depends on the basic design selected but good performance depends on the rate of heat transfer, and this in turn depends on the temperature difference between the heating and cooling medium. The greater the temperature gradient, the better will be the heat transmission. However, it would be wrong to make the refrigerant evaporate at lower temperatures, since, as mentioned previously, by lowering the suction pressure the C.O.P. will be correspondingly lowered. It is also essential that the pressure drop along the evaporator coil be kept to a minimum, otherwise the compressor suction will be lowered, again reducing the C.O.P. Also with high frictional losses the mean temperature difference will be decreased, reducing proportionally the heat transferred by a given coil. In the case of air-to-air heat pumps careful consideration must also be given to the atmospheric conditions, especially humidity, and to a method of coil defrosting in the winter time.

PRACTICAL COEFFICIENT OF PERFORMANCE

Having investigated the various factors influencing C.O.P., the expression for practical C.O.P. for the heat pump can be written as follows:—

$$\text{C.O.P.} = \text{C.O.P. (Carnot)} \times k \times \eta_M \times \eta_E \times \eta_H \quad . \quad (5)$$

where k = diagram factor (taking into account deviation from adiabatic and throttling operations); η_M = mechanical efficiency (motor and compressor); η_E = electrical efficiency; η_H = heat transfer efficiency.

If, therefore, the various values for the given individual efficiencies are substituted, the C.O.P. at a chosen evaporation temperature could be obtained, and a curve can be plotted showing the variation of the practical C.O.P. for the heat pump. Such a curve is shown in Fig. 6 for a heat pump using a ½ h.p. sealed-type motor compressor and operating at 140 deg. F. condensing temperature.

Some writers regard as incorrect the expression of the overall efficiency of a refrigerator or a heat pump as a product of the individual efficiencies, since the factors influencing the efficiency cannot be clearly separated. This is true for a generalised expression, but if the procedure is applied to a particular size of heat pump and the various factors are determined separately, the overall efficiency can then be correctly expressed as a product of the partial efficiencies. The C.O.P. expression must be modified if the motor heat losses and the compressor frictional losses (appearing as heat) are utilised in the system. Such a motor can be regarded as 100 per cent. efficient, and the expression, therefore, would become:

$$\text{C.O.P.} = \text{Carnot C.O.P.} \times k \times \eta_M \times \eta_H \quad . \quad . \quad . \quad (6)$$

The curves in Fig. 6 show the variation of practical C.O.P. for a heat pump using a ½ h.p. motor-compressor and utilising the motor heat losses.

In the case of heating water from cold, say from 60 deg. to 140 deg. F., an average C.O.P. must be used, and this can be found by intergrating equation (3), i.e., C.O.P. $= \frac{T_1}{T_1 - T_2}$ in the limits of $T x_1 = 140$ deg. F. and $T x_2 = 60$ deg. F., and finding the average ordinate, T_1 being the variable quantity.

Average C.O.P.

$$= \frac{1}{T x_1 - T x_2} \int_{T x_2}^{T x_1} \frac{T x}{T x - T_2}\, d\,Tx;$$

$$= 1 + \left[\frac{T_2}{T x_1 - T x_2}\right] \log_e \frac{T x_1 - T_2}{T x_2 - T_2} \quad . \quad . \quad . \quad (7)$$

If the ambient temperature is assumed to be 40 deg. F., then the average theoretical C.O.P. = 11·1.

The average practical C.O.P. for these conditions will be approximately 4. This is quite good, and a heat pump utilising low-grade heat from the atmosphere seems to be an economical proposition. However, from a thermodynamic point of view, to ensure good C.O.P., consideration must also be given to superheating and undercooling of the refrigerant.

SUPERHEATING AND UNDERCOOLING

The diagram in Fig. 7 shows a practical cycle for a heat pump which is used for water heating.

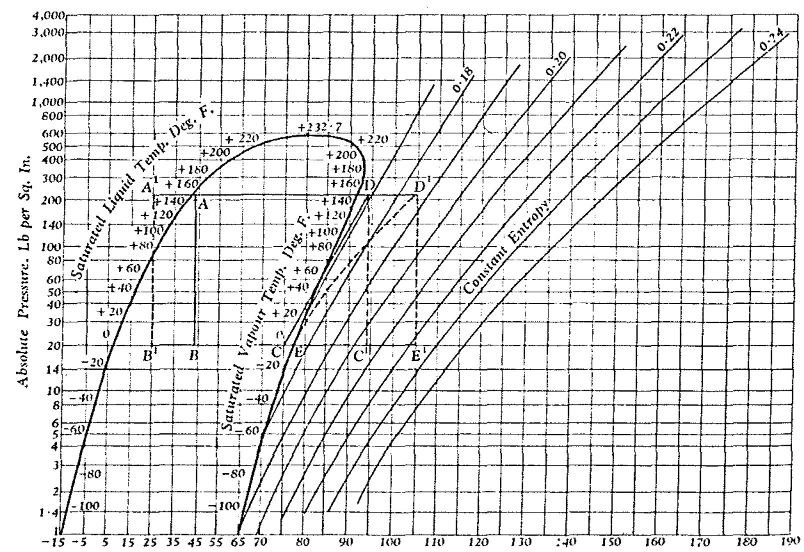

Fig. 7 Complete P-H diagram for a heat pump using Freon 12.

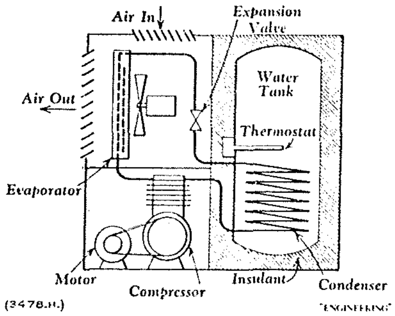

Fig. 8 Combined water heating and space cooling system for a fairly large installation.

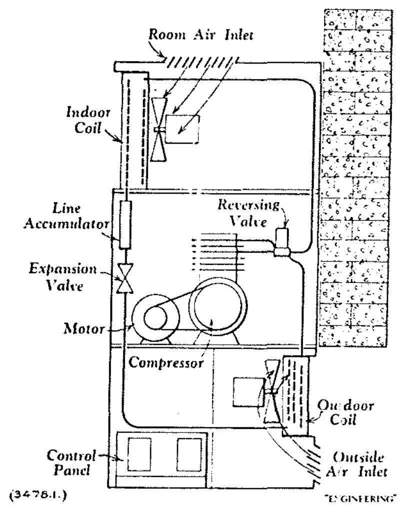

Fig. 9 Space heating and air conditioning plant for a building.

The heat removed from the air by an evaporator is shown by distance BC, the heat available at the condenser is denoted by DA, and CC′ is work done by the machine. The ratio of $\frac{DA}{CC'}$ is the C.O.P. of the unit.

It must, of course, be realised that the above cycle represents one set of working conditions only. (In this case the condenser pressure was 230 lb. per sq. in.). The same diagram shows that if point D is moved to the right, i.e., if the refrigerant leaves the compressor in a superheated condition, then the heat at the condenser and the work of compression will both be increased, but their ratio, i.e., $\frac{\text{Heat to condenser}}{\text{Heat to compression}}$, which is the C.O.P., is lowered.

The effect of superheating can be explained in the following way. Due to temperature rise in the cylinder head, condenser pressure will increase, and, as shown previously, the increase of head pressure lowers the C.O.P. In practice, 20 deg. F. superheat will reduce a C.O.P. of say, 3·5 by approximately 1·5 per cent.

Superheating is very common in refrigeration practice, and in the case of an hermetically sealed motor compressor it is almost always present, since the refrigerant vapour passes over the hot motor windings. It must also be mentioned that if an installation uses a thermostatically-operated expansion valve, at least 8 to 10 degrees of superheat are required in the evaporator for the valve to operate satisfactorily.

If point A in Fig. 7 moves to the left and passes across the liquid line, the refrigerant will be undercooled. In this case the heat at the condenser will be increased without any change in the work of compression and consequently the C.O.P. will be increased.

In the case of any heat pump, provision for undercooling must always be made, not only because of increased efficiency but because with a high-temperature liquid throttling is not satisfactory, and defrosting can be observed. In practice there are two ways of undercooling: by using an external fluid; or by using the refrigerant vapour.

The first method is the more efficient but requires the installation of a special heat exchanger between the condenser and the expansion valve. The second method is less efficient but more practical, and the usual way is to precool the liquid using cold gas at the outlet of the evaporator. This precooling of the liquid serves automatically to preheat the low pressure vapour, and the heat lost by the precooled liquid is equal to the heat gained by the superheated vapour.

PRACTICAL DESIGN

After this theoretical analysis consideration must be given to the possibilities of adopting the heat pump in the home for space and water heating. Water heating seems to be the more difficult case and it will be considered first.

It was shown by the curves and also by the pressure-enthalpy diagram that the heating up of the water will vary the condenser conditions and the evaporator conditions will vary correspondingly but not proportionally.

However, the problem is more complicated than it looks. In order to obtain a water temperature of 140 deg. F., high pressures must be used. As already mentioned, a pressure of about 250 lb. per sq. in. can be expected when using Freon 12. Furthermore, serious thought must be given to the other problems closely related to the design of a heat pump, such as:

(*a*) Source of heat, whether air, water or ground; (*b*) Climatic conditions, which are far the most important in design problems, since they will determine the heating and cooling load required, and to a large extent the practicability of any given heat source; (*c*) Whether dual purpose, i.e., cooling and heating is required in the same installation; and (*d*) Whether heating is required continuously or only occasionally.

Some of these problems require very careful consideration. If economy is essential, the dual-purpose heat pump would be best. There are, however, quite a few other problems which present themselves when the requirements of a dual-purpose domestic unit are considered; one is the effect of the seasons of the year upon the refrigerating and heating requirements of the installation. During cold weather the heat produced will be of more importance than the refrigeration, and the opposite will be the case on hot summer days. To make matters worse, consider the case of a family vacating a house for the weekend during hot weather. No heat or hot water whatsoever would be used, yet full refrigeration would be required; consequently some form of " heat-leak " would be necessary. The designer is, therefore, confronted with a number of problems which are new to refrigerating and heating practice. That there is a solution is certain and already two firms are mass producing small water-heating units in this country and very shortly more will enter this field. In the case of small heat pumps, air is used as the source of heat or, more precisely, the heat pump is placed in a larder and the whole space is changed into one large refrigerator. The heat collected is utilised to heat the water and in each case heat leaks are incorporated. The Brentford Electric Duo-Therm heat pump is fully automatic and a special valve controls the radiator; in the case of the Ferranti Fridge-Heater, excess heat is dissipated by removing a portion of lagging from its auxiliary water cylinder. The diagram in Fig. 8 shows a larger heat pump installation used for water heating and space cooling.

In the case of sets used for space heating, considerable progress has been made during the past few years, and various firms are working on prototypes. One or two units are already on the market. One such air-to-air unit is shown in Fig. 9. Cooling in the summer is obtained by simply reversing the refrigerant flow and Figs. 10 and 11 show the schematic operation of such a system using hand operated or automatic valves.

SPACE-HEATING PERFORMANCE

The C.O.P. of space-heating units is dependent upon the source of heat available and the method of heat transfer employed. If hot air is used a higher C.O.P. will be obtained than with hot-water radiator systems. Also, proper air

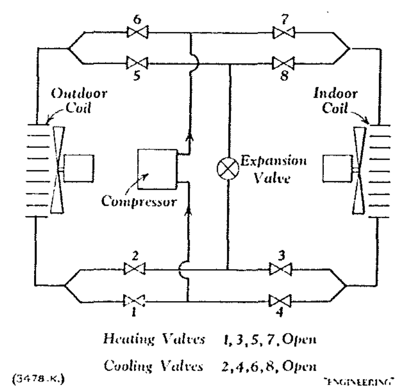

Fig. 10 Valve system using hand operated valves for giving either heating or cooling. For heating, valves 1, 3, 5 and 7 are opened and for cooling, 2, 4, 6 and 8.

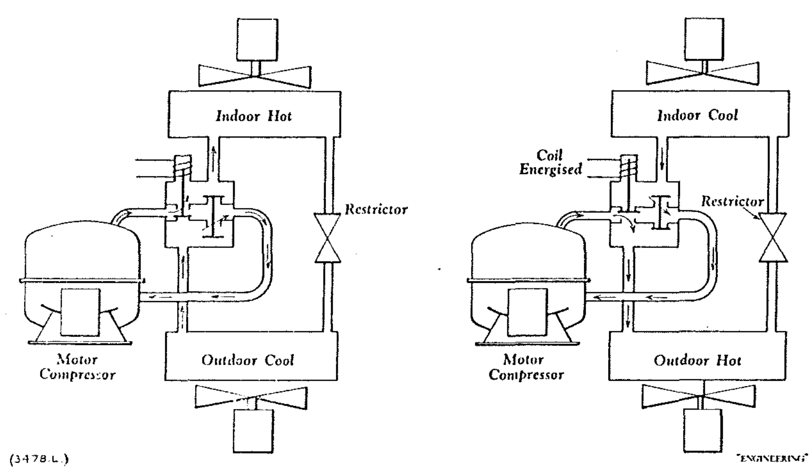

Fig. 11 Alternative heating and cooling system employing an automatic magnetic valve.

TABLE I. COMPARATIVE COSTS OF REFRIGERATION AND WATER HEATING

Type of appliance	Calorific value of fuel	Cost of fuel	Assumed efficiency, per cent.	Running costs per day		Total running costs (per day)
				Refrigerator	Water heater	
					s. d.	s. d.
Heat pump	—	1*d.* per kWh	C.O.P. 2·8	—	8	8
Immersion heater	—	1*d.* per kWh	100	1½*d.*	2 4	2 5½
Anthracite boiler	12,500 B.Th.U. per lb.	180*s.* per ton	70	1½*d.*	11	1 0½
Gas geyser	500 B.Th.U. per cub. ft.	1*s.* 8*d.* per therm	75	1½*d.*	2 3	2 4½

TABLE II. COMPARATIVE COSTS OF SPACE HEATING

Type of appliance	Calorific value of fuel	Cost of fuel	Assumed efficiency	Approximate running costs (per heating season)
				£ s. d.
Heat pump	—	1*d.* per kWh	C.O.P. = 4	20 16 8
Electric resistance heaters	—	1*d.* per kWh	100	83 6 4
Gas boiler	500 B.Th.U. per cub. ft.	1*s.* 8*d.* per therm	78	67 15 0
Oil unit	18,000 B.Th.U. per lb.	1*s.* 3*d.* per gallon	75	31 16 0
Anthracite boiler	12,500 B.Th.U. per lb.	180*s.* per ton	70	26 16 6
Coke boiler	11,800 B.Th.U. per lb.	140*s.* per ton	70	23 19 6

conditioning will be possible only in the case of air-to-air units.

If we assume that the outside temperature will decrease uniformly during the night, from $Tx_1 = 40$ deg. F., to $Tx_2 = 20$ deg. F., and hot air will be discharged at say 80 deg. F., then the average C.O.P. will be given by:

$$\frac{1}{T\,x_1 - T\,x_2}\int_{T\,x_2}^{T\,x_1} \frac{T_1}{T_1 - T\,x}\,d\,T\,x$$

the solution of which is

$$\frac{T_1}{T\,x_1 - T\,x_2}\log_e\frac{T_1 - T\,x_1}{T_1 - T\,x_2} \quad . \quad (8)$$

For the above conditions its value will be approximately 11, and a practical average C.O.P. for a 3 h.p. unit could be as high as 4 to 5. Assuming that a 3 h.p. unit consumes 1 kW per h.p. then the output of such a unit would be over 12 kW/hr., sufficient to heat a house of some 1,200 sq. ft.

ECONOMY

In order to appreciate the economy which is possible with a heat pump installation, a particular case has to be considered, and a house having 1,200 sq. ft. floor area has been chosen for this purpose. This particular house has a water-heater cum larder-cooler heat pump and an air-to-air space heating unit.

For a water-heater heat pump a reasonable average C.O.P. would be 2·8 for a ⅓ h.p. unit. The power consumed by such a unit would be between 300 and 350 watts per hour, i.e., 8*d.* per day—assuming continuous running and electricity at 1*d.* per unit. To give an overall picture of the efficiency and economy of the heat pump as compared with other appliances giving the two basic services, i.e., hot water and refrigeration, Table I has been computed.

A comparison of the operating costs of a heat pump used for space heating with a conventional system is very difficult as the standard of heating varies with different types of heating appliances. However, Table II depicts the cost of heating a house requiring 20,000 kW heating load per one heating season. The C.O.P. of a heat pump has been assumed to be 4.

The two Tables are self-explanatory and show clearly the running economy which is possible with heat pump installations. It must not be forgotten that heat pumps have other advantages and, as already mentioned, not only can they provide heating and cooling, but can filter the air, provide ventilation and humidity control, and achieve the ideal in cleanliness, compactness and quick response to changes in temperature requirements. However, the basic disadvantage of a heat pump is its high initial cost but if there is taken into account the cost of building chimneys, providing boilers, stoves, fire-places, coal bunkers, hot-water cylinders and lagging, and also the refrigerator, it can be shown that the unit is able to be compared very favourably with the more conventional heating system.

It does seem to be generally believed that heat pumps will soon become extremely popular and that some 50,000 units will be in operation within the next six years. In the U.S.A. for example, during 1955, the sale of heat pumps jumped to double the number of units which had been in operation until then. By their use not only is there a saving in running costs and domestic use of high grade coal, but there will also be a cleaner atmosphere and improved comfort in the home.

CONCLUSION

Many developments remain to be made in this fascinating method of heating and the time will come when the old draughty house with its traditional fire place will vanish, and there will be no more roasted shins and frozen backs. Economic circumstances alone make the heat pump a "must" and there is no doubt that the traditional method of heating is already on its way out. Heat pumps will initially become the conventional means of water heating; later they will provide central heating and air conditioning. The new appliance which a few years ago was not even imagined, will be accepted as part of modern life.

The Beaver Report (Report of the Committee on Air Pollution) has indicated the urgent task of ending smog, which is so costly to the nation. Further, at least one-third of our coal, roughly 63,000,000 tons per year, is used for domestic heating. There is, therefore, a clear justification for the industry to develop heat pumps. It is in the national interest that such heat appliances should be developed, since with them, the fuel situation could be vastly improved, the atmosphere cleared and the standard of comfort in the home increased.

(Among earlier articles dealing with heat pumps which have been published in ENGINEERING are:—

"The Norwich Heat Pump," page 116, vol. 164, 1947.

"Fuel and Power Economy with Special Reference to Heat Pumps" pages 285 and 309, vol. 166, 1948.

"Heat Pump for Small Office," page 95, vol. 169, 1950.

"Gas Driven Heat Pump at the South Bank Exhibition," page 741, vol. 171, 1951.

"Small Heat Pumps," page 750, vol. 178, 1954.

"Central Heating by Heat Pump," page 439, vol. 179, 1955.)

PRODUCER GAS FOR ROAD TRANSPORT

RESULTS OBTAINED WITH COMPRESSION-IGNITION ENGINES

During the last war, one of the alternatives to petrol for the propulsion of road vehicles was producer gas and before the war ended, over 3,000 vehicles in this country were equipped for the use of this fuel. In reports published by the Ministry of Fuel and Power* and the Ministry of Transport and Civil Aviation, details are given of a series of tests carried out on four vehicles, with particular reference to those fitted with compression-ignition engines. The aim of the reports is to consolidate present knowledge and stimulate interest in the use of fuels such as low-grade coals, lignites, peat, wood, sawdust, charcoal, bagasse, coconut shells and groundnut, rice and cotton-seed husks—all of which can be satisfactorily burnt in the producer. The work was carried out jointly by the two Ministries and the Fuel Research Station.

PRODUCER PLANT

The vehicles used were (1) a 14 h.p. Vauxhall private car; (2) a 5 ton Bedford lorry; (3) a 10 ton A.E.C. Mammoth Minor lorry; and (4) a double-decked bus. The first two had petrol engines with standard spark ignition, and the others were each fitted with 7·7 litre A.E.C. Diesel engines. The first stage was obviously to develop a producer plant, and the final form is shown on the opposite page. This shows the diagrammatic layout of a plant for Diesel engines. It is basically similar to that developed for agricultural tractors. The main components are a hopper for the fuel, a pre-heater or heat exchanger, gas-cooling tubes, a water washer or irrigator and a slag-wool filter. Operation is by the pressure drop in the induction manifold when the engine is running, which draws air through a water carburettor. The water taken up is vaporised in the heat exchanger and air/steam mixture passes into and through the fire bed. A cross up-draught flow is used. The resulting producer gas (largely a mixture of carbon monoxide, hydrogen, methane, carbon dioxide and nitrogen) gives up most of its heat in the heat exchanger, and then is further cooled, washed and filtered before entering the mixing valve or chamber. In this a sufficient quantity of air is added to form a combustible mixture which is then fed to the engine. The equipment is mounted on a two wheel trailer and its weight is about half a ton.

* *Producer Gas for Road Transport*, Ministry of Fuel and Power, Information Branch, Thames House South, Millbank, London S.W.1.

ENGINE CONVERSIONS

To compensate partially for the reduced volumetric efficiency and the lower calorific value of producer gas, conversion of a spark-ignition engine should include an increase in compression ratio. This is hardly practicable owing to the cost and also to the fact that the engine must initially be started on petrol. Consequently only about 50 per cent. of the normal engine power can be developed.

On the other hand, with the compression-ignition engine the alterations are less complicated, the automatic gas/air mixing valve makes driving easier and effects a saving of more than half the fuel oil, though some is still required to give proper combustion. Under these conditions it is quite possible to obtain full power from the engine.

As producer gas burns comparatively slowly, the injection timing of the fuel pump has to be