The INSTITUTE of MARINE ENGINEERS
Transactions

1948, Vol. LX, No. 11

The Heat Pump with Some Notes on its Application to Air-Conditioning in Land and Marine Service

F. JODER

The principle of the heat pump has been known for a century, but its practical application has not been possible until comparatively recent times when, owing to scarcity and high cost of fuels, it has been used with success for space heating and industrial processes requiring heat at moderate temperatures.

The various types of heat pumps are briefly described and their application to different processes shown. Means for a fuller utilization of existing refrigerating installations are indicated, in particular how, by the addition of further compression stages, they can simultaneously serve as heat pumps in industrial undertakings.

The air cycle heat pump is of particular interest for ventilating and air conditioning problems, and the Lèbre pressure exchanger in the form of a cellular rotor of simple design may in future supplant the auxiliary vapour compression machine in the air conditioning field.

Comparative operating costs for the various types of heat pump installations are shown.

The heat pump, which until recently was considered as an impractical proposition, and unlikely to become competitive with conventional methods of raising heat, has slowly and laboriously emerged from semi-obscurity into a position of some prominence.

With the exception of some industries which make extensive use of distilling and evaporating processes in the concentration of lyes, fruit juices, etc., and similar installations, the heat pump was looked upon more as a scientific curiosity than a means of provided heat for buildings.

Nearly a century ago William Thomson, later Lord Kelvin, first drew attention to the fact that a refrigerating compressor could be used as a heating machine. He had realized that by suitably applying the thermodynamical cycle of the refrigerator it ought to be possible by extracting heat from the surroundings to obtain a multiple of the heat equivalent of the energy required to drive the compressor.

Apart from the application of this principle to distilling and evaporating processes it needed, at least in this country, a great fuel crisis to bring the heat pump to the notice of a wider circle of engineers. The savings in fuel it can effect in heating—and air conditioning—dwellings, office buildings, factories, ships, etc., may not be so spectacular as in industrial processes mentioned above in which for every kW. driving the heat pump as much as 20 times its equivalent in heat may be obtained. The more modest ratio, or coefficient of performance as it is commonly called, of 3 to 5 which obtains in space heating applications, will, as cost and scarcity of fuels increase, become a more attractive proposition for installations which require comparatively low temperatures for their efficient operation.

As already mentioned, the heat pump is not by any means a new idea. The first practical installation was put into operation more than sixty years ago in the Saltworks Bex in Switzerland. This plant is still in existence and in working condition. Since then a considerable number of heat pumps for a great variety of applications and with power inputs to the compressors ranging from a few horse-power to several thousand, have come into existence on the Continent and the U.S.A. Units of 4,750 h.p. were recently installed in a sugar refinery in Switzerland, and they are probably the largest heat pumps in Europe.

Most heat pumps are driven by electric motors, particularly in countries where electricity is derived from hydro-electric power plants and coal is not an indigenous product. There are, however, other prime movers which can come into consideration and amongst these the Diesel-engine is an attractive alternative.

Principles of the Heat Pump

The function of the heat pump is to raise the thermal value of the heat contained in the water of a river, warm waste water from a process or the ambient air to a higher temperature level where it can be used as a heating medium for space heating installations, drying stoves, air conditioning plants, the production of hot water for domestic or industrial purposes, etc.

The working cycle of the heat pump is similar to that of a refrigerating plant. The fundamental difference between the two lies in the fact that in the heat pump the heat removed from the refrigerant in the condenser, instead of being rejected to waste as in the refrigerating plant, is available for heating purposes.

The basic arrangement of a heat pump using an auxiliary

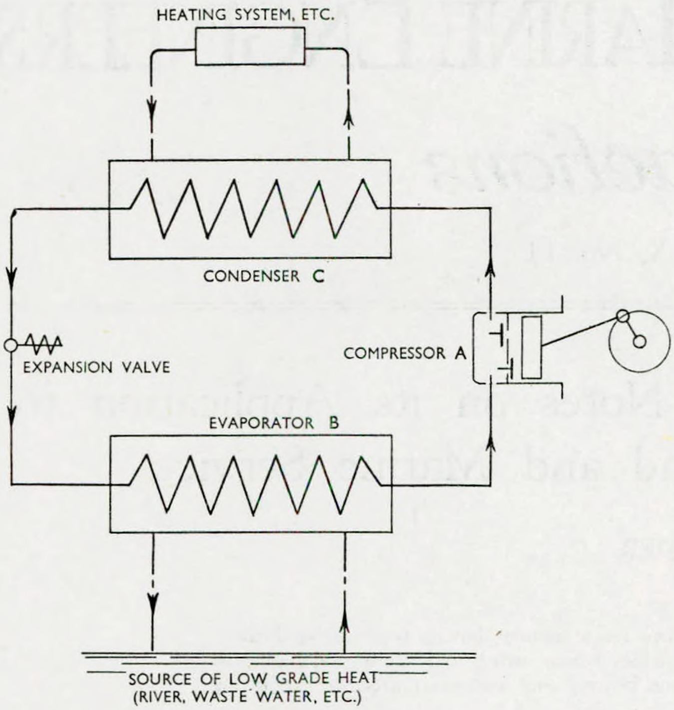


FIG. 1.—*The basic arrangement of a heat pump using an auxiliary vapour*

vapour is shown in Fig. 1 and consists essentially of a compressor, A, which compresses a highly volatile liquid such as ammonia, Freon, sulphur dioxide or methyl chloride; an evaporator, B, in which the refrigerant is evaporated at low pressure, the necessary heat for this process being supplied by the surrounding air, the water of a river, etc., which is at a slightly higher temperature than the refrigerant; and a condenser, C, in which the hot vapours leaving the compressor are liquefied by a cooler secondary medium which is usually the water of a heating system or the air of a ventilating—or air conditioning plant.

The theoretical cycle of the heat pump is most conveniently shown with the help of the pressure-enthalpy diagram. In the following example (Fig. 2) it is assumed that the refrigerating medium is Freon 12 evaporating and condensing at a temperature of 50 deg. F. and 140 deg. F. respectively. The corresponding temperature of the source of heat and that of the heating medium leaving the condenser would be 60 deg. F. and 130 deg. F. respectively.

To the left of the liquid boundary curve lies the liquid region of the refrigerant, to the right of the saturated vapour boundary curve is the superheated vapour region. The space between the two curves is that of the wet condensing or evaporating vapours.

Dry saturated vapour at the temperature t_o=50 deg. F. with the heat content h_o'' (1) is adiabatically compressed to the pressure of the condensing refrigerant (2). In doing so the power absorbed by the compressor raises the heat content of the vapour to h_2 superheating it at the same time. Liquefaction takes place in the condenser at the temperature t_c=140 deg. F. when the heat absorbed in the evaporator together with the heat equivalent of the power consumed by the compressor is given up to the cooling water of the condenser, which, in the case of the heat pump, is the heating medium of the heating plant. The liquid refrigerant then returns to the evaporator (4) via the expansion valve expanding to the evaporator pressure. Part of the liquid refrigerant evaporates during expansion, the necessary heat for the evaporation being supplied by the liquid itself. The product of the difference in heat content between point (1) and (4) and the circulating weight of refrigerant is the net refrigerating effect.

To make the heat pump an economical producer of heat the problem to be solved is to extract from the surroundings the greatest possible amount of heat with the least possible expenditure of shaftwork at the compressor. The ratio

$$\frac{\text{Heat given up in condenser}}{\text{Heat equivalent of compressor shaftwork}}$$

is called the coefficient of performance.

Put in thermodynamical language, the coefficient of performance or c.o.p. for short is the ratio of the absolute condensing temperature to the temperature difference between the condensing and evaporating refrigerant.

$$\text{Thus coefficient of performance} = \frac{T \text{ condensing}}{T \text{ condensing} - T \text{ evaporating.}}$$

Fig. 3 shows graphically the theoretical coefficients of performance for various evaporating and condensing temperatures.

In practice the coefficient of performance is of course much lower owing to the unavoidable losses in the compressor and its driving motor, and the irreversibility of the heat pump cycle. The actual overall coefficient of performance would thus be:

$$\text{Coefficient of performance} = \eta_{co}\eta_m\eta_c \frac{T\text{cond.}}{T\text{cond.} - T\text{evap.}}$$

where η_{co} = mechanical efficiency of compressor

η_m = mechanical efficiency of motor

η_c = efficiency of cycle (=1 for ideal Carnot cycle)

For heat pump plants using turbo-compressors the overall efficiency is approximately as follows:

Output of unit, B.Th.U. per hr.	Overall efficiency
1,000,000 to 4,000,000	0·45 to 0·55
4,000,000 to 12,000,000	0·55 to 0·60
over 12,000,000	0·60 to 0·65

Types of Heat Pumps

There are several types of heat pumps and they can be roughly divided into three groups. The classical heat pump is of course the refrigerating machine, but it does not concern this paper. Moreover, the refrigerating machine is a necessity for technological reasons rather than considerations of fuel economy.

In the first group are the heat pumps for evaporating and boiling processes in the chemical and certain food industries, e.g. for the concentration of lyes and other chemical solutions, distillation of water, manufacture of common salt, concentration of milk, fruit juices and other organic solutions. Owing to the relatively small increase of temperature which some of these processes require the coefficient of performance can reach the very high figure of 15 to 20, which is one of the reasons why heat pumps are used here to a much greater extent than for space heating purposes, and were employed in these industries long before they were applied for the latter purpose. About 90 per cent of all the heat pumps installed in the last twenty-five years belong to this group and today their total capacity exceeds $2{,}000 \times 10^6$ B.Th.U. per hr.

Fig. 4 shows schematically the arrangement of a heat pump for an evaporating plant.

The weak solution E to be concentrated enters the evaporator A via the heat exchanger D, where it is preheated by the condensate leaving the evaporator heating coil. The vapour from the boiling solution is evacuated by the compressor B, which raises its pressure and temperature to a slightly higher degree than that of the boiling solution. The whole of the latent heat contained in the compressed vapour is, therefore, available for the concentration process.

This is the simplest type of heat pump since it does not require an auxiliary vapour such as ammonia, Freon, etc., for its operation. Heat pumps for boiling processes in the food industry are very similar to the one just described except that the temperatures are in most cases kept lower to avoid the destruction of valuable ingredients or vitamins. Owing to the low pressures at which these plants are operated the volume of vapour that has to

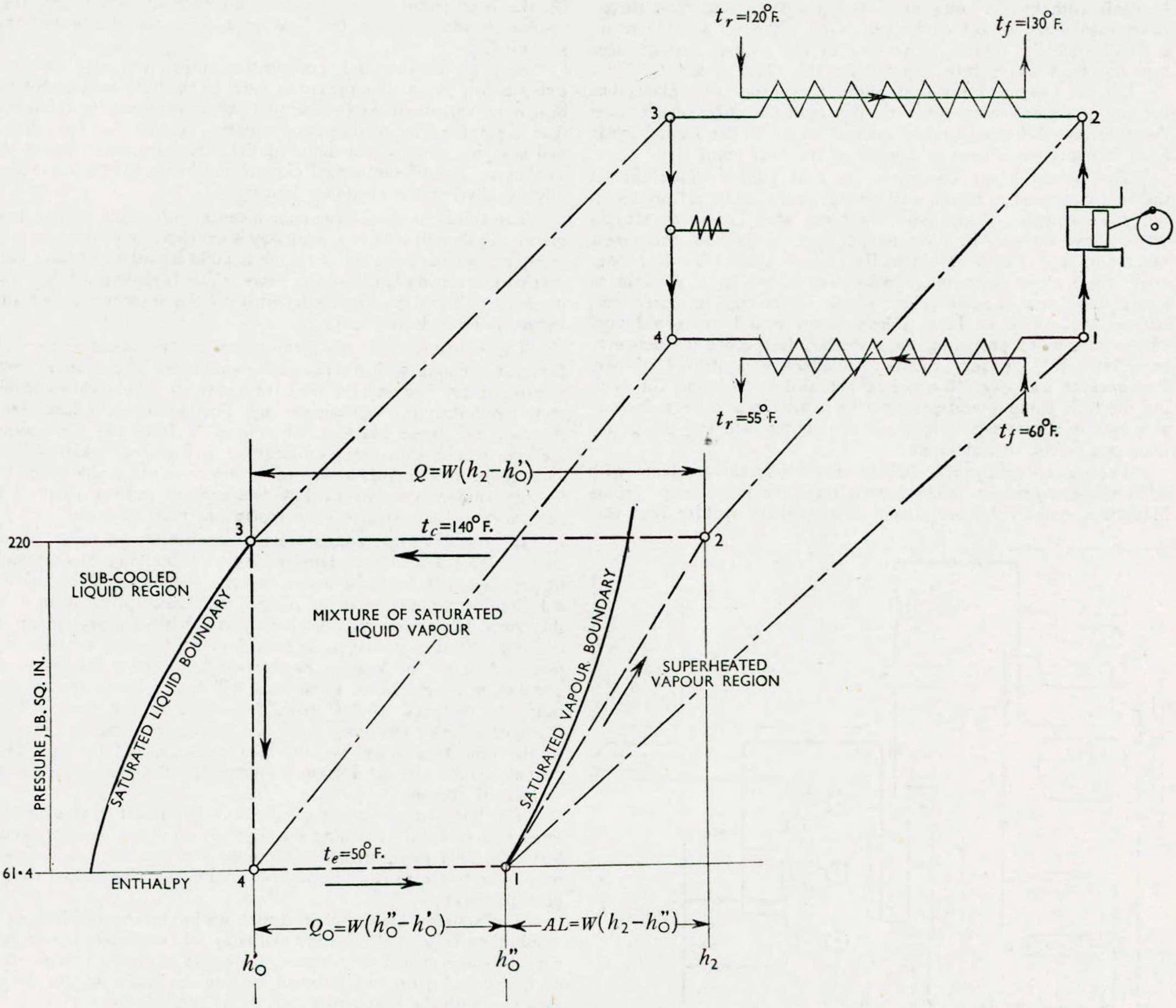


FIG. 2.—*Pressure-enthalpy diagram*

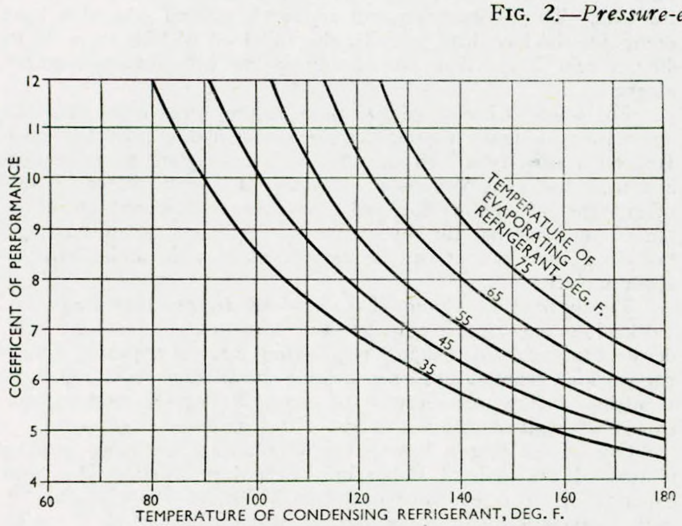


FIG. 3.—*Coefficients of performance for various evaporating and condensing temperatures*

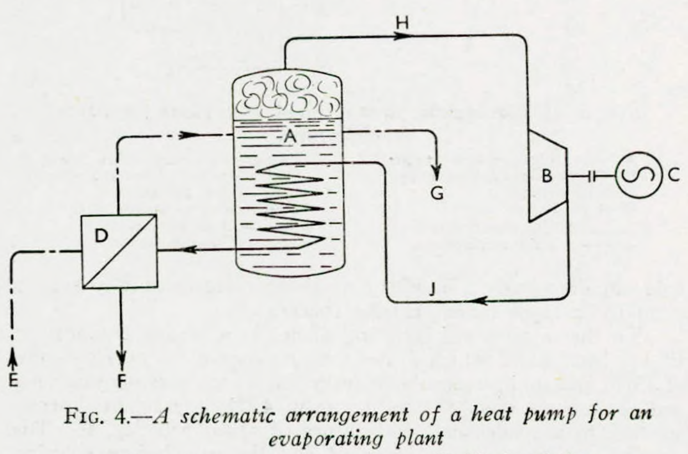


FIG. 4.—*A schematic arrangement of a heat pump for an evaporating plant*

A—Evaporator.
B—Compressor.
C—Motor.
D—Heat exchanger.
E—Weak solution.
F—Condensate.
G—Concentrate.
H—Vapour.
I—Compressed vapour

be dealt with by the compressor is often very large and turbo-compressors are almost exclusively used, although for plants on a small scale the rotary compressor of the positive displacement type has been successfully adapted for this class of work.

On the Continent the heat pump for evaporating plants has become a serious competitor to the multiple effect evaporator. Apart from the considerable savings in fuel, the capital costs have in many cases been in favour of the heat pump.

The second group comprises the heat pumps which extract the heat contained in warm waste water from industrial processes, the cooling water from power stations, etc., i.e. from sources where heat due to its low temperature is useless for direct utilization, and, therefore, normally thrown away. To this category also belong swimming baths and where it is possible to locate them near a power plant whose warm cooling water constitutes the source of heat, a heat pump would prove a highly efficient means of providing the necessary heat since the temperature lift would be very small. An effective coefficient of performance of well over 10 could be obtained and current for driving the heat pump would probably be available at a very favourable rate in a case where both the swimming bath and the power plant are public undertakings.

The second group also includes the refrigerating plants with additional compression stages which form the heat pump. Some industries, notably breweries and many others require heat and

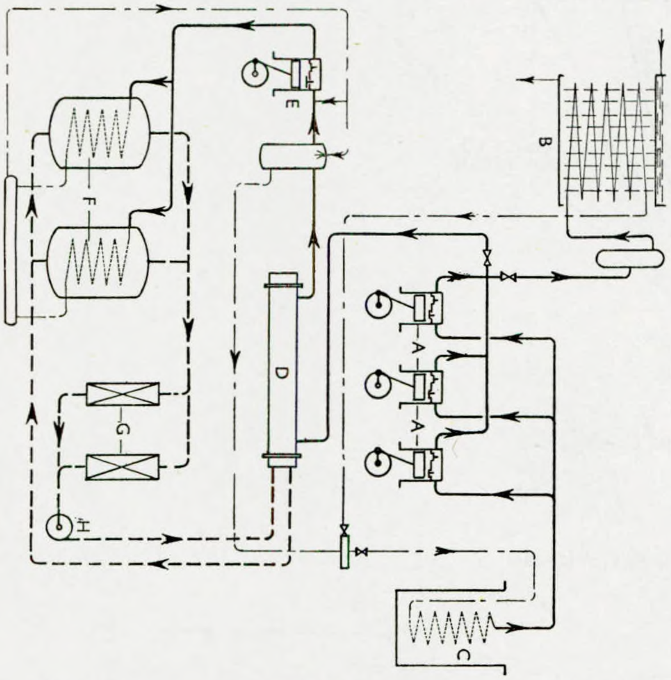


Fig. 5.—*Diagrammatic arrangement of a plant for rayon manufacture*

A—First compression stage.	E—Second compression stage.
B—Condenser of first stage.	F—Condenser of second stage.
C—Evaporator.	G—Heating system.
H—Gas cooler.	H—Circulating pump of G.
—— Refrigerant vapour.	——— Heating circuit.
—·— Liquid refrigerant.	– – – Cooling water.

cold simultaneously. In Fig. 5 is shown diagramatically such a plant for a rayon manufacturing concern.

To the original refrigerating plant, A, a second compressor, E, has been added which draws the ammonia gases at a pressure of 137lb. per sq. in. from the delivery side of the existing machines and compresses them further to about 450lb. per sq. in., corresponding to a condensing temperature of about 150 deg. F. The original condenser, B, is bypassed and the gas, before entering the suction branch of the second compressor, has its temperature reduced in a gas cooler D. The latter can be the first stage of the heating system or provide heat for an independent process circuit. The gas from the delivery side of the fourth compressor, i.e. the heat pump is condensed in an enclosed submerged type condenser which forms the heat exchanger for the heating installation.

Since the heating and refrigerating output may vary quantitatively at any given time provision must be made to supplement the heat requirements of the heating plant when the need for refrigeration is small. The refrigeration capacity is regulated by starting and stopping compression units of the first compression stage the suction volume of the second compressor being varied automatically by altering the clearance spaces.

The third group of heat pumps deals with space heating problems. Although this is a relatively small field to which the heat pump has so far been applied some notable installations have been carried out during the last ten years. The high cost of fuel and its scarcity have greatly contributed to the introduction of this relatively novel heat raiser.

The conditions for the heat pump are the least favourable for space heating with direct heating surfaces. The annual operation hours are low and the heat they have to supply varies within wide limits during a heating season. Furthermore, the large temperature difference between the source of heat and the heating medium results in a low coefficient of performance which seldom exceeds 3·5. A slightly better performance is obtainable when the heating surface consists of low temperature radiant panels. In such a case the coefficient of performance may reach 4·5.

It is well known among heating engineers that over 80 per cent of the seasonal heat requirement of a building can normally be supplied with half the boiler capacity installed. This fact has led Continental engineers to design their heat pump plants for this basic load only, and install in series with the heat pump an auxiliary heating device, usually a fuel fired boiler to boost the temperature of the heating medium during coldest weather. An installation conceived on these lines will (a) be lower in first cost than one designed to deal with the whole of the heating load, (b) have a better efficiency, since the smaller heat pump can most of the time be run at its full rated capacity, and (c) provide a partial standby during forced shutdowns of the heat pump owing to lack of current.

The last named feature is especially important in this country where the habit of "shedding the load" in very cold weather would leave the heat pump owner without heat altogether. A company may offer lower rates if current is voluntarily interrupted during peak periods.

The standby boiler plant would preferably be oil-fired as it would then be possible to start and stop it immediately the current supply is interrupted or restored again. It might be thought that an installation with two different heating devices would be dearer than one with the heat pump capable of supplying the whole heating load. This is, however, not so, as the capital cost of a heat pump for the base load, i.e. half the full load will be about 30 to 40 per cent lower than one supplying the full heating requirements.

For small and average size space heating installations the reciprocating auxiliary vapour compressor is undoubtedly the most frequently used type. It has attained a high state of perfection and from the point of view of reliability it is unsurpassed by any other type of compressor. By providing additional clearance spaces and means for bypassing of cylinders in multi-stage machines, their output can be varied within wide limits without appreciable losses.

The normal heat pump has only one temperature stage, i.e. works into one circuit only but the provision of a second condenser can enable it to supply two heating circuits requiring different working temperatures, e.g. heating installations with radiators requiring a flow temperature of say 160 deg. F. and radiant panels where it should not exceed 120 deg. F.

One of the largest heat pump installations for space heating is that of the Federal Polytechnic School in Zurich. The total capacity of its three compressors is 24,000,000 B.Th.U. per hr. with a temperature of the source of heat of 36 deg. F. Two of the machines are turbo-compressors and the other is a Sulzer 3-stage ammonia compressor with wide range load regulation shown in Fig. 6. The coefficient of performance is about 3, and

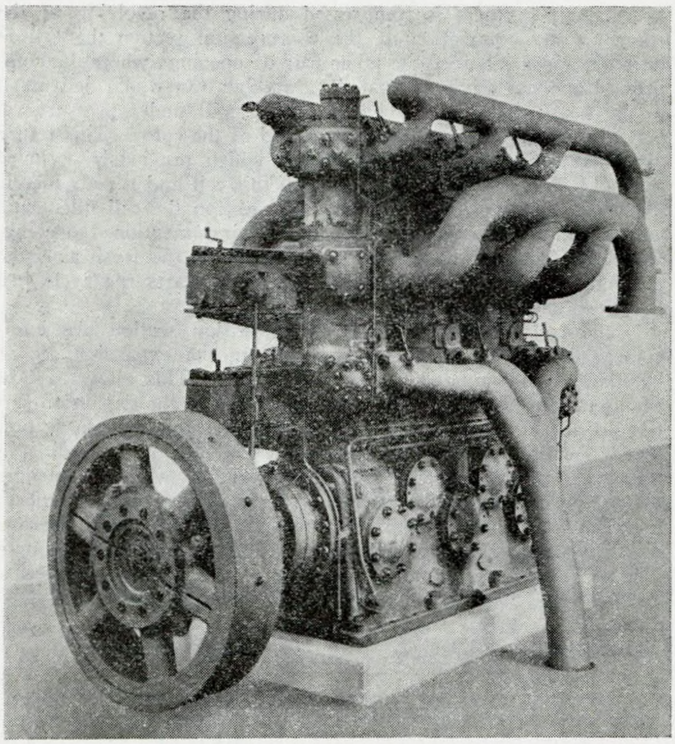


FIG. 6.—*Sulzer 3-stage ammonia compressor with wide range load regulations*

the maximum temperature of the heating medium is 160 deg. F. The source of heat is the river Limmat from which about 13,000 gals. per min. are taken. In its passage through the evaporators the temperature of the water drops about 2 deg. F. after which it is returned to the river.

Having dealt with the three main groups of heat pumps, another machine, which is a comparative newcomer amongst heat pumps, should be mentioned. This is the air cycle type which promises to supplant to some extent the auxiliary vapour compression machine in the field of air conditioning and drying.

The principle of the cold air machine which it resembles has of course been known for a long time, having first been propounded by the American, Dr. John Gorrie, in 1845, and put into practice by Bell-Coleman in 1877.

The air cycle heat pump has the great advantage that the refrigerating medium required by the vapour compression machine is replaced by atmospheric air which is at all times available in abundance free of cost and non-toxic.

Fig. 7 shows the cold air machine in principle.

The compressor and expansion machine are arranged on the same shaft and coupled to an electric motor. The compressor draws in primary air from the source of heat and compresses it, raising its temperature thereby. After passing through the heat exchanger where part of the heat content is given up to the secondary air stream the air enters the expansion machine expanding to the original atmospheric pressure whereby a certain amount of the work expended in compression is recovered. Owing to the extraction of heat in the heat-exchanger and the work done by the expansion process, the primary air will have a lower temperature than when it entered the compressor. Heat has, therefore, been abstracted from the source of heat, and, together with the heat equivalent of the difference in power absorbed by the compressor, and regained by the expansion machine, transmitted to the secondary air stream. The lower part of the figure shows the changes of state in the process just described. The contraction of the air volume due to heat given up to the secondary air stream can be clearly seen, and it is obvious that the power regained in the expansion machine must be smaller than the power absorbed by the compressor.

Fig. 8 shows the same arrangement somewhat modified by the addition of an auxiliary compressor which handles a quantity of air equivalent to that of the volume contraction in the heat exchanger. The main compressor, therefore, delivers the same amount of air as the expansion machine. In a perfect machine the power regained in the expansion machine would just equal the power absorbed by the compressor, and no driving motor would, therefore, be needed. The consumption of power would of course be the same as in Fig. 7, and would be absorbed by the auxiliary compressor.

The practical realization of the two processes just described with reciprocating compressor and expansion machine would have disappointing results in small installations. Owing to the unavoidable losses in converting mechanical work into compression work and expansion back into mechanical work the useful heat output would be very low indeed. For larger air quantities and by employing turbo-machinery the performance could be considerably bettered but might still be uneconomical.

The air cycle heat pump invented by Lèbre constitutes a great advance over the cold air machines just described. It is in principle as shown in Fig. 9, using an auxiliary compressor to make up for the loss in volume of the primary air which occurs in the heat exchanger due to the temperature drop. The essential difference, however, is that the reciprocating compressor and expansion machine with their double conversion of energy are replaced by a new single element, the cellular rotor in which a direct pressure interchange takes place and no external mechanical work is needed for compression or given out during the expansion process.

The cellular rotor consists of a radially bladed wheel which

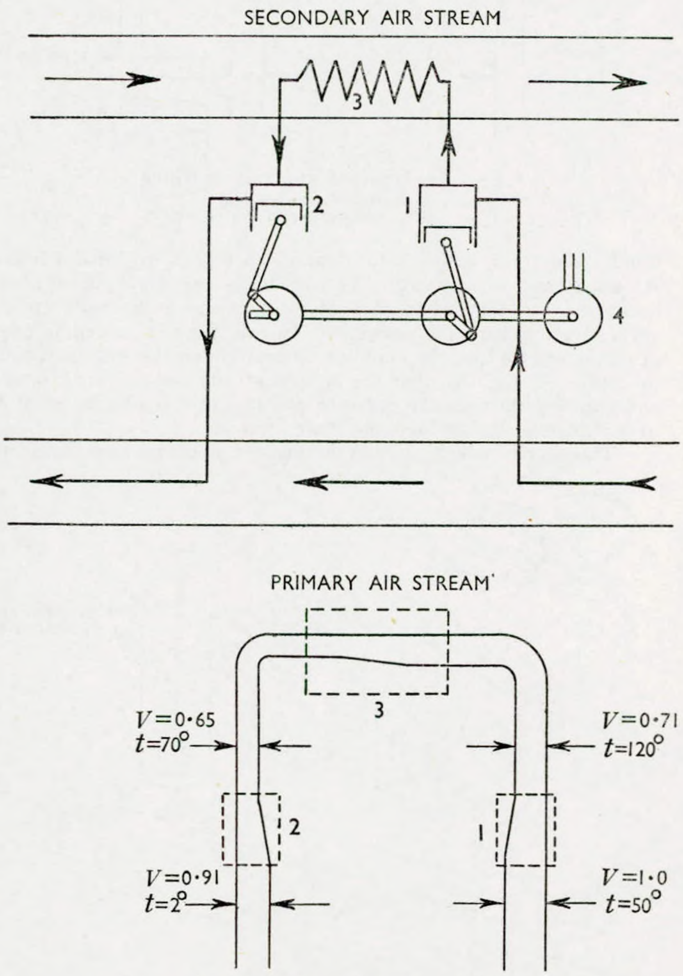


FIG. 7.—*The cold air machine*

1—Compressor. 2—Expansion machine. 3—Heat exchanger. 4—Motor.

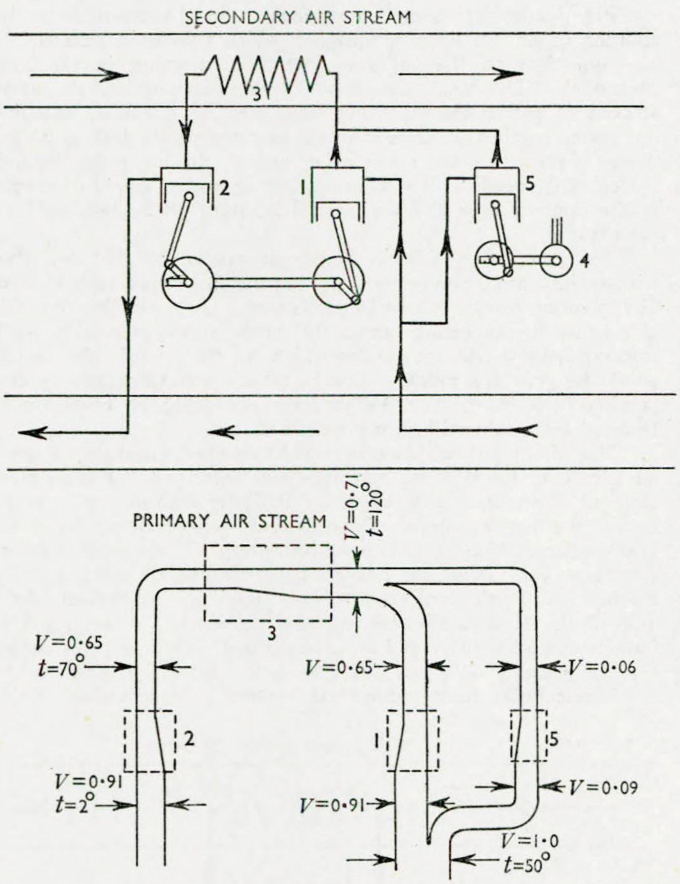


FIG. 8.—*Modified cold air machine*
5—Auxiliary compressor.
Other symbols as in Fig. 7.

divides the rotor casing into a number of cells of equal volume. At either end of the casing two openings are arranged opposite each other so that when the rotor is revolving the cells appear successively before the openings. In this position a cell is open at either end and can be supplied with air from the source of heat by means of a fan so that the air already in that cell is displaced and expelled through the opposite opening, thus a continuous flow of air through the rotor takes place.

Those cells which are not before the openings are sealed off

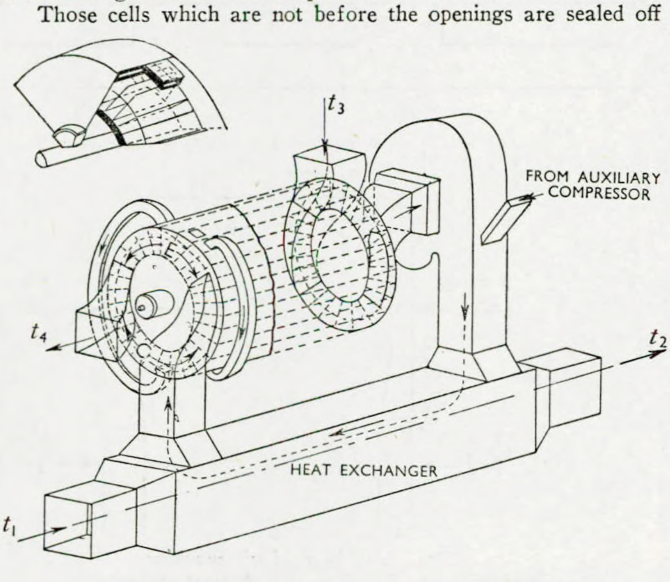


FIG. 9.—*Lèbre's air cycle heat pump*

and the air in them is compressed during the revolving of the rotor. After completion of the compression section the cell in question appears before the other pair of openings where the compressed and hot air is evacuated by another fan which delivers it to the heat exchanger to be cooled and, still under pressure returns to the same cell at the opposite end of the rotor. This return air, therefore, replaces the warm air which previously occupied the cell. The revolving rotor seals off this cell and the air therein expands. After passing the expansion section the cell filled with cold expanded air reappears before the first mentioned openings when the cool air is expelled to atmosphere by the fresh air from the source of heat. In revolving the rotor starts the cycle just described anew.

The cells which are in the compression section are interconnected with the corresponding cells in the expansion section by fixed ducts arranged in the casing walls. This allows a flow of air to take place in each duct from a cell in the expansion section to one at slightly lower pressure in the compression section. Thus the air coming from the heat exchanger, in expanding, is used to compress the air in the respective cell which is intended to give up its heat in the heat exchanger. The continuous energy interchange through the interconnecting ducts neither liberates mechanical energy, nor is such required from an external source. The necessary power for the operation of the rotor is only needed

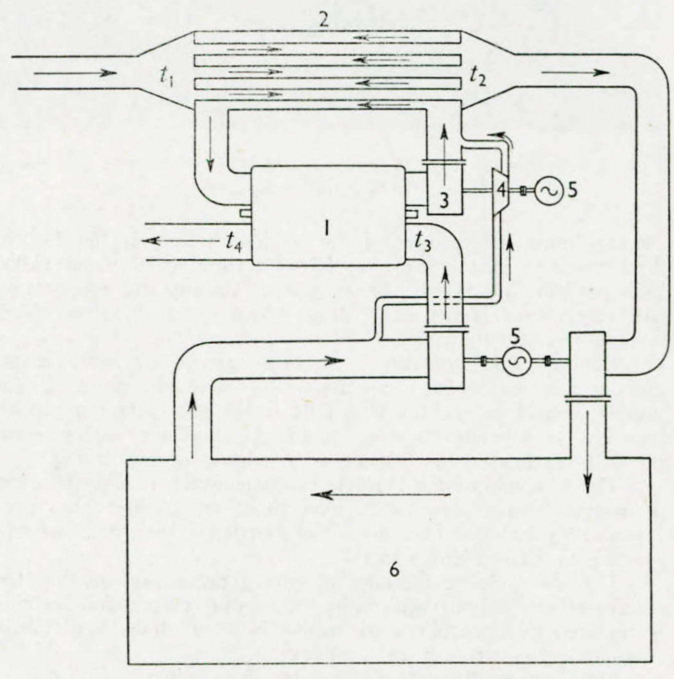


FIG. 10.—*General arrangement of an air cycle heat pump*
1—Cellular rotor.
2—Heat exchanger.
3—Fans.
4—Auxiliary compressor.
5—Motors.
6—Conditioned space.

to overcome the friction in the bearings. The auxiliary compressor is the chief consumer of power. Fig. 10 shows schematically the general arrangement of an air cycle heat pump, and the construction of the cellular rotor.

The majority of heat pumps so far installed are driven by electric motors and a very few of the larger ones by steam turbines. As already mentioned at the beginning, internal combustion engines could usefully be employed for this purpose. Apart from supplying power they also yield considerable quantities of waste heat. From 35 to 40 per cent of the calorific value of the fuel consumed by Diesel engines can be recovered in heat exchangers and used to boost the output from the heat pumps. Fig. 11 shows such an arrangement proposed for the production of hot water for one particular process in a paper mill.

The source of heat is cooling water at a temperature of 70 deg. F. from the condenser (1) of the steam turbine of the

Table I

Type of plant		Operating hours per annum	Actual c.o.p.	Conversion factor c.o.p. × 3412, B.Th.U. per kW.	Power consumption per 10^6 B.Th.U. delivered, kW.	Specific installation costs per B.Th.U., d.	Installation costs per 10^6 B.Th.U. delivered, s. d.	Capital charges per 10^6 B.Th.U. delivered: Interest, per cent	Interest, s. d.	Depreciation, per cent	Depreciation, s. d.	Operating costs per 10^6 B.Th.U. delivered: Maintenance, per cent	Maintenance, s. d.	Power rate 0·4d. per kW., s. d.	Power rate 0·6d. per kW., s. d.	Total costs per 10^6 B.Th.U. delivered: Power rate 0·4d. per kW., s. d.	Power rate 0·6d. per kW., s. d.
1		2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
(1) *Air Cycle*																	
Heating, air conditioning	1	1,500	2·5	8,500	118	1·1	61 0	4	2 5	10	6 1	1½	0 11	3 11¼	5 10¾	13 4½	15 3¾
Drying ...	2	3,600	2·5	8,500	118	1·1	25 6	4	1 0	10	2 6½	1½	0 4½	3 11¼	5 10¾	7 10¼	9 9¾
Drying ...	3	8,000	3·0	10,200	98	1·1	11 6	4	0 5½	10	1 2	1½	0 2	3 3¼	4 10¾	5 0¾	6 8¼
(2) *Auxiliary Vapour Compressor*																	
Space heating, normal temp.	4	1,500	2·5	8,500	118	1·1	61 0	4	2 5	10	6 1	2	0 11	3 11¼	5 10¾	13 4¼	15 3¾
Space heating, low temp. ...	5	1,500	4·5	15,350	65	1·1	61 0	4	2 5	10	6 1	2	0 11	2 2	3 3	11 7	12 8
Hot water F. process work	6	3,000	6	20,450	49	0·7	19 6	4	0 9½	0	1 11½	2	0 3½	1 7½	2 5½	4 8	5 6
Hot water F. process work	7	6,000	7·5	25,550	39	0·7	9 8	4	0 4¾	10	0 11¾	2	0 1¾	1 3½	1 11½	2 9¾	3 5¾
Hot water F. process work	8	8,000	9	30,700	32·5	0·7	7 4	4	0 3½	10	0 8¾	2	0 1¼	1 1	1 7½	2 2½	2 9
(3) *Thermo - compression*																	
Evaporation, etc.	9	3,000	12	40,900	24·5	0·35	9 8	4	0 4¾	10	0 11¾	1½	0 1¾	0 9¾	1 2½	2 4	2 8¾
Evaporation, etc.	10	6,000	15	51,200	19·5	0·35	4 10	4	0 2½	10	0 6	1½	0 1	0 7¾	0 11¾	1 5¼	1 9¼
Evaporation, etc.	11	8,000	18	61,400	16·3	0·35	3 8	4	0 1¾	10	0 4½	1½	0 0¾	0 6½	0 9¾	1 1½	1 4¾
(4) *Coal Fired*																	
Space heating	12	1,500	—	7,200*	—	0·29†	21 0	4	0 10	10	2 1	3	0 7½	3 8¾‡		7 8¼	

*Calorific value of fuel 12,000 B.Th.U. per lb., boiler efficiency 60 per cent. †Capital cost of boiler plant £1,200. ‡Cost of fuel 60s. per ton.

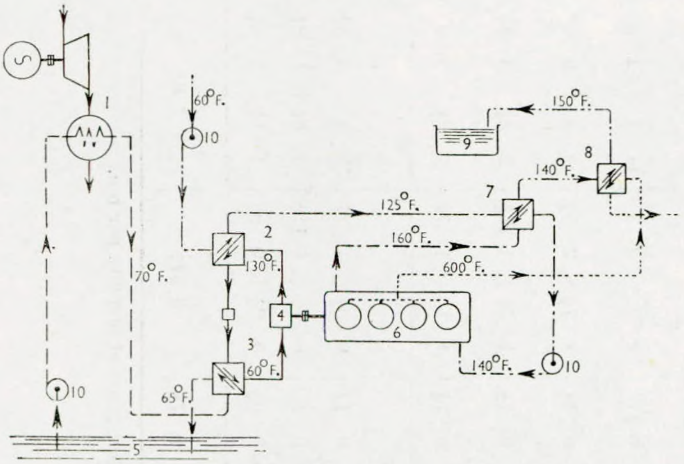


FIG. 11.—*Proposed arrangements for hot water production for one paper mill process*

1—Steam turbine condenser.
2—Condenser of heat pump.
3—Evaporator of heat pump.
4—Compressor.
5—Source of heat (canal).
6—Diesel engine.
7—Heat exchanger for engine cooling water.
8—Heat exchanger for engine exhaust.
9—Hot water storage tank.
10—Circulating pumps.
—— Refrigerant.
—·· Fresh water.
– – – Source of heat.
----- Engine exhaust.
—· Engine cooling water.

works power station, which, instead of passing directly into the canal adjoining the works, is first led through the evaporator (3) of the heat pump (4) where it has its temperature lowered by about 5 deg. F. The fresh process water entering the condenser of the heat pump (2) at 60 deg. F. is heated to 125 deg. F. In the first boosting stage, consisting of a heat-exchanger (7) the temperature of the water is raised to 140 deg. F. by the engine cooling water, which in this case would be circulated in a closed circuit, i.e. the same cooling water is used over and over again with only a small amount of make up due to leakages and evaporation. In the second boosting stage, also a heat exchanger (8) of a different type, the water is heated by the engine exhaust gases to its final temperature of 150 deg. F. If the heat pump had to cope with the whole temperature lift of 150-60=90 deg. F. the theoretical coefficient of performance would have been 6·5. By utilizing part of the heat contained in the cooling water and exhaust gases the temperature lift of only 125-60=65 deg. F. to be supplied by the heat pump will raise the coefficient of performnce from 6·5 to 8·4. Allowing for all the losses the actual coefficient of performance would be about 5·5.

The use of sea water as an inexhaustible source of heat for a heat pump on passenger and cargo vessels is obvious. This is a potential field of heat pump applications particularly where the main propulsion machinery is either a Diesel engine or a gas turbine where steam raising plant is not available. This would mean that the customary steam heating would be superseded by a hot water or direct electric heating installation. In spite of the many advantages of the latter system, such as ease of control and operation, cleanliness, etc., it has, however, one serious draw-back, namely that of risk of fire in the event of failure of a control thermostat. It is, therefore, quite possible that in future hot water heating may become the preferred method of heating spaces other than those which are dealt with by air conditioning plants. Most of the large ships have refrigerating equipment for their air conditioning plants and its conversion into heat pumps could be affected with a minimum of additional capital expenditure. All that is required is that the heat exchangers of the air conditioning plant be constructed and arranged in a manner that they can alternatively be used as condensers for heating purposes and as evaporators when cooling of the air is required. The combination of heat pumps-cum-refrigerator and air conditioning plant would ensure a fuller use of the existing refrigerating machines and thus result in increased overall efficiency and lowered operating costs. The same remarks apply to refrigerated cargo ships where cargo spaces have to be heated and cooled to maintain predetermined temperatures for the conservation of the cargo and to prevent the formation of condensation.

Economic Considerations

The economic justification of the heat pump depends on the following factors:—

(*a*) High cost of fuel.
(*b*) Low cost of power.
(*c*) High annual operating factor.
(*d*) Low cost of heat source.

It must be said at once that in Great Britain (*b*) is impossible of attainment with (*a*) since power is almost exclusively generated in thermal stations, and increased costs of fuel immediately reflect themselves in corresponding higher power rates. (*c*) is only possibly in certain industrial plants and in some measure in combined heating and air conditioning installations with refrigerating equipment. Item (*d*) is possible when the plant is situated near a river or a lake. Where these are at some distance from the heat pump onerous pumping charges can be avoided by putting the evaporator of the heat pump directly into the source of heat. The relatively high cost of town water rule it out as a heat source.

Table I shows the relative costs per million B.Th.U. transmitted to the heat consuming system for various types of heat pump installations. The table also shows the adverse influence which a small number of operating hours per annum has on the operating costs. For a space heating installation with low temperature panels with a power rate of 0·6d. per unit and operating during 1,500 hours the table (line 5) shows these costs to be just over 12s. For a coal-fired plant of similar capacity in place of the heat pump the relative costs would be only about 8s. with fuel reckoned at 60s. per ton. High annual operating hours together with the higher possible coefficient of performance i.e. small temperature lifts, are a necessity for the heat pump to compete successfully with modern fuel fired plants.

Conclusions

The design of heat pump installations requires careful consideration, particularly from the economic angle, and unless thorough investigations on the various aspects of the problem are made, disappointments will be certain and will bring the heat pumps into discredit amongst its users. It should be borne in mind that heat pump problems are a matter of co-operation between heating engineer and the refrigerating specialist. The former knows the requirements a plant has to fulfil, but it will be for the refrigerating engineer to select and provide the most suitable equipment. The slow progress which the heat pump is making in this country (there is as yet only one commercial installation for space heating in this country, namely that at Norwich) is not entirely due to lack of enterprise, but the realization by the interested parties of the economical limitations of the heat pump compared with modern fuel fired installations.

Discussion

Professor S. J. Davies (Visitor) who opened the discussion, said that the paper was timely, and he would like to congratulate the author. As he had, in collaboration with Mr. F. G. Watts, recently presented a paper on this subject to the British Association* he would confine himself to certain small points in the present paper.

He protested against the use of the term "coefficient of performance" for a heat pump. That term was used for a refrigerator, and to use it for both the heat pump and the refrigerator could only lead to confusion. Dr. Faber† used the term "advantage". The French used the term "coefficient of amplification". He was not converted to either of those expressions, and preferred "performance energy ratio". One measured the performance, using the ratio of the energy delivered as heat to the energy in mechanical work in the compressor. The heat pump was, from the practical point of view, in its early stages in this country, and it was desirable to adopt straight away a term which explained itself.

This country had a ready source of low temperature heat in the sea, a fact that was of importance in connexion with the use of heat pumps on shipboard. The author showed the small difference of temperature between his working substance and his heat-conveying media of 10 deg. F. and the fact should be emphasized that to get good performance from a heat pump this difference must be kept small; the overall temperature range was otherwise extended, and the performance energy ratio reduced in consequence.

The figures for the overall efficiency in the second column on the second page of the paper were of particular importance, because it was not easy to get actual figures. As the author pointed out, the performance energy ratio involved three efficiencies: the efficiency of the cycle, the efficiency of the compressor, and the efficiency of the motor or driving unit. The total of all those was involved in the term "overall efficiency".

The author said that about 90 per cent of all the heat pumps installed in the last twenty-five years belonged to the group of thermal compressors, the special form of heat pump used for evaporation processes. That was true if the author meant the actual amounts of heat delivered, but not if the number of installations was taken.

He did not agree with the author's sub-division of the subject into thermal compressors, complex industrial installations, and space-heating. The author surprisingly dismissed what he called the classical heat pump which worked on a refrigerator cycle but on a higher temperature range, although in fact the largest number of heat pumps worked on that cycle.

The author also suggested the combination of a boiler with a heat pump. This was clearly a sound combination in certain types of installation, but not in the way to which the author referred. The boiler was certainly available to bridge any gaps when the electric power was cut off, but that was not the primary need for the boiler, since it was equally possible to bridge those gaps by suitable thermal storage, which was in fact done in certain smaller installations.

He asked the author to give figures for the measured performance energy ratio of the Lèbre machine, a most interesting invention.

The author mentioned that Diesel drive had advantages, and gave an interesting example. Table 2 was taken from his own recent paper and referred to a building used as a swimming bath for twenty-two weeks in summer and as a dance and assembly hall for thirty weeks in winter. It was a new building, with modern coke-fired boilers, and the actual consumption was as in line (*a*). The boiler consumption was $1{\cdot}25 \times 10^6$ B.Th.U. per hour. If the boilers were in good hands, an efficiency of 50 per cent might be assumed. Line (*b*) was for heat pumps driven by electric motors, with an assumed performance energy ratio in winter of 3·5, when the heat pumps were heating the building, and of 6 in summer, when they were heating the swimming baths. Those were conservative values.

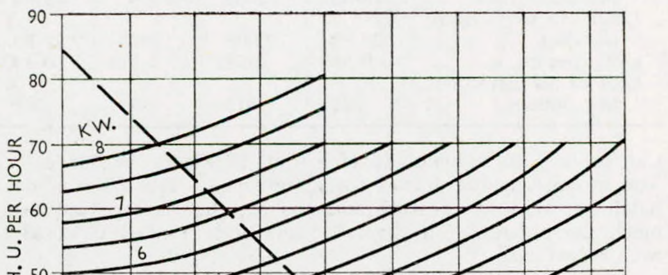
To supply the same amount of heat, electric motors of 141 h.p. would be necessary as in line (*b*). With the heat pumps, driven by oil engines, and thus independent of the central station, and using the waste heat from the engines, 99 b.h.p. would be necessary as in line (*c*). Under (*a*), the consumption of coke per annum was 90 tons. With (*b*) the consumption at the central station would be 39 tons of coal per annum. With oil engines the consumption would be 17 tons of fuel per annum. The annual saving for fuel, therefore, provided an ample margin for any increased capital costs involved in fitting the heat pumps.

There was a further point to be considered. Not only were 90 tons of coke consumed at present, but it brought with it dirt and ash which had to be disposed of.

Table 2. Comparison between coke-fired boilers and heat pumps, electrically and oil-engine driven.

Installation	Capacity	Mean fuel consumption, tons per week		Thermal efficiency × heat delivered
		Solid fuel	Oil	Heat in fuel
(*a*) Boilers, coke-fired ...	12·5 therms per hour	(Coke)		
Winter		1·67	—	50 per cent
Summer		1·82	—	50 per cent
(*b*) Heat pumps driven by electric motors	Power of motors, 141 h.p.	(Coal)	(Boiler oil)	
P.E.R., 3·5 winter		0·88	0·64	87 per cent
P.E.R., 6 summer		0·56	0·41	150 per cent
(*c*) Heat pumps driven by oil engines	Power of engines, 99 h.p.		(Diesel oil)	
P.E.R., 3·5 winter		—	0·36	155 per cent
P.E.R., 6 summer		—	0·27	228 per cent

* *Engineering*, 17th and 24th September 1948.
† Faber, O. 1946 Proc.I.Mech.E., vol. 154, p. 144, "Value of Heat with Special Reference to the Heat Pump".



——Heat pump output, B.Th.U. per hour.
— —Building heat losses, B.Th.U. per hour.
– – –P.E.R. earth heat pump.
–·–·P.E.R. air heat pump.
Room temperature 70 deg. F.

FIG. 12.—*Electric power required to heat a well-insulated six-room house*

Professor Penrod had reported on the American position. Industrial applications were not mentioned, and the installations here were divided between office buildings, in the range from 40,000 to 350,000 cu. ft., and houses, from 20,000 cu. ft. down to quite small houses. Two manufacturing companies were producing package unit installations, similar to household refrigerators, on a scale approaching quantity production, one company having installed 150 sets in 1946, with motors ranging from 3 to 10 h.p. Fig. 12 showed the electric power required to heat a well-insulated six-roomed house, heating the whole house to 70 deg. F., with an air heat pump, i.e. one which took its low-temperature heat from the air. It would be seen that with an outdoor temperature of 20 deg. F. the amount of heat to be supplied was 45,000 B. Th.U. per hour., and 4·5 kW., with a performance energy ratio of just under 3, was sufficient. In America experiments had been made in taking the heat from the ground, and in that case the performance energy ratio at that outside temperature was nearly 4, which reduced the necessary power consumption to 3·6 kW., which gave very economical heating.

Lastly Table 3 gave data for four office buildings in Southern California with volumes ranging fom 50,000 to 160,000 cu. ft. The installation and annual costs are given. The loads in these buildings were for heating and cooling, the cooling load being much the greater. The overall load factor ranged from 23 to over 30 per cent.

Table 3.—Heat air-conditioning plants in District Offices of Southern California, Edison Co. Ltd. (Penrod).

District Office and year of installing	Whittier, 1937	San Bernardino, 1937	Montebello, 1938	Santa Ana, 1940
Volume air-conditioned, cu. ft.	75,300	73,800	50,100	116,700
Floors	1	1 plus Mezzanine	1	2
Approximate installation cost, dollars ...	5,500	5,400	6,500	8,900
Average annual values—				
Heating, kWh. ...	7,996	11,440	8,556	21,800
Cooling, kWh. ...	16,100	27,460	17,111	30,526
Total air-conditioning, kWh.... ...	24,096	38,900	25,667	52,326
Load factor, heating, per cent	7·94	9·42	6·75	11·35
Load factor, cooling, per cent	15·88	22·63	13·5	15·95
Load factor, total, per cent	23·82	32·05	20·25	27·30
kWh. per cu. ft. ...	0·318	0·527	0·513	0·499
Cost of air-conditioning, dollars ...	531	672	530	901

Mr. J. K. W. MacVicar (Associate) said that while the author had indicated fuel economy by utilizing the heat pump principle for air conditioning installations on board ship, and while the principles themselves were undoubtedly well known to air conditioning engineers, he thought it must be admitted that there were certain practical difficulties which existed and which tended to prevent this application of the heat pump, owing to the complications which would be introduced. He thought that it should perhaps be explained that, while there were many air conditioning plants on board ship at present, and while there would be more very shortly, in almost every case the cooling medium was delivered to the air conditioning plant from the standby refrigerating machine on board ship.

According to Lloyd's Register regulations, the owners must carry a standby refrigerating plant, and in many cases, from the point of view of economy in equipment, this standby plant was used to deliver brine to the air conditioning units. Immediately a breakdown occurred in one of the machines in operation, obviously air conditioning was of secondary importance, and the standby machine must at once go on to its main job. He felt that the complications which would exist in piping were such that the average marine engineer would hesitate to introduce them from the point of view of simplicity of running. In fact, the tendency nowadays was to avoid all complications and to reduce change-overs, which might lead to trouble and injury to valuable cargo.

Apart from ships, which had not reached the stage of utilizing all that the author had to offer, there was a very wide field of application for the heat pump principle, and he himself was concerned with a particular application at the moment which, unfortunately, he was not allowed to describe. In the early days of the war, his firm had been faced with a quite unusual air conditioning job. In one very important strategic position the German threat caused the headquarter staff to go underground. They burrowed into the rock and found very large caverns there. The temperature of the rock was more or less constant, and rather cooler than was desirable for human comfort, and so the problem was primarily to heat the place; but coupled with that there was the problem that in a large number of the rooms there was a great deal of heat-generating electrical equipment and there was quite a definite variation in load. There were large rooms for high-ranking officers and smaller rooms for lower rating, together with these heat-producing rooms.

His firm were faced with the problem of supplying cold conditioned air to certain rooms and warm conditioned air to other rooms, or a mixture of both, depending on the requirements of the occasion. They had already installed refrigerating plant to deal with the cooling down load, and they first of all adopted the methods of delivering to each room cold conditioned air and warm conditioned air, with a mixing device. To obtain the heat, they did not want to run a very heavy electrical load with electric heaters, which apart from anything else would be cumbersome to control, and so they inserted in the warm conditioned air duct an auxiliary air condenser in series with the water cooled condenser. If no one wanted warm air, the water cooled condenser did its full job, but when people called for warm air it was automatically provided. It was not highly efficient, and in installing it they had no thought of the performance energy ratio, but it was very effective from the point of view of comfort and it worked very well.

There was another aspect of the matter which he would like to mention, though it was really only a side-line to the paper. As a ventilating engineer on board ship, he frequently met the condition of refrigerating plant put in for cold cupboards, the tendency being to go in for air-cooled condensers. The refrigerating engineers knew their own problems, but very often the shipbuilders did not know them, and that little refrigerating plant was put in a tiny cupboard which might be more or less hermetically sealed, and all the heat of the condenser was given to the place, so that the plant did not work and the conditions were extreme. The author had added greatly to the knowledge available by pointing out the temendous heat loads which had to be encountered.

Professor Davies had referred to the standby boiler operating with the heat pump. The author suggested that the standby boiler could readily be used when the electric power failed and the heat pump shut down, but personally he found that in his own works when the electric current failed, so did the oil-burning equipment.

Mr. E. G. Russell Roberts (Member) said that because of the steady increase in recent years in the size of refrigerating plants fitted on board ship, and particularly because of the use of refrigerating plants for air conditioning becoming increasingly common in passenger vessels, the possibility of using such equipment as a heat pump was a matter which clearly sooner or later should be investigated. He decided, therefore, to give the matter further thought the first time that he came to deal with an air conditioning plant of a size and type which might make such an application a possible proposition. What appeared to be such a case was when he was called upon to consider the refrigerating machinery for air conditioning in a very large passenger vessel building in this country and now approaching completion. The plant as now being installed was capable of extracting 3,500,000 B.Th.U. per hour when cooling fresh water from 52 to 46 deg. F., and for this purpose required two single-stage Freon 12 compressors with motors of 160 b.h.p.

From the point of view of capital outlay the plant appeared ideal, as it could very easily be designed, with almost no additional cost, to reverse the cycle when used as a heat pump, and the only substantial difference would be the need for rather larger compressor motors, owing to the increased power necessary with the higher condensing temperature required to heat the circulating water to a sufficiently high temperature to give air at about 100 deg F. from air heaters.

The air coolers fitted by the ventilating contractors could equally well be used for heating by the circulation of a warm medium through them instead of a cold, and little or no additional cost would be involved in this part of the equipment. The slight additional power required for the plant would not affect the large generators for which provision had already been made, and the only point which caused him some qualms at that stage was what would happen when the ship most required heating, i.e. in New York harbour in the winter, where the water around the vessel would be at its freezing point or too close to it to be possible to cool it down any further without freezing in the evaporator side, which in cooling conditions was the condenser. He did not worry about that point unduly, as it was reasonable to suppose that sufficient low-grade heat could be obtained from the condenser circulating water of the turbo-generators, one at least of which would always have to be running.

He calculated that the ventilating contractors would want an air temperature of 100-105 deg. F. leaving their heaters, and since these, being designed for air conditioning, had ample surface and would therefore work with a small temperature difference, it was thought likely that it would be possible to get the condensation temperature of the plant down to about 130 deg. F. As the amount of waste heat to provide a medium for the plant to cool might be limited, he worked on the assumption that it would not be possible to rely on the plant working at an evaporation temperature above 35 deg. F. Each of the machines fitted would be capable under these conditions of extracting 1,250,000 B.Th.U. per hour.

An equivalent amount of heat would be rejected to the condenser, together with the heat equivalent of the i.h.p. of the compressor motor, which in turn was about 90 per cent of the shaft horse-power. Under these conditions, this was 195 h.p. To that figure, therefore, must be added $195 \times 0{\cdot}9 \times 2{,}545$, equalling about 450,000 B.Th.U. per hour, giving, with the 1,250,000 B.Th.U. per hour referred to in the previous paragraph, a total heating effect per machine of 1,700,000 B.Th.U. per hour. Two machines would therefore give 3,400,000 B.Th.U. per hour. To this must be added the heat rejected to the circulating water by the circulating water pumps, which were estimated to have a total shaft horse-power of 16, equivalent to a total of, say, 40,000 B.Th.U. per hour, giving a total potential heating duty of 3,440,000 B.Th.U. per hour.

The total power consumption of the plant would be:

	b.h.p.
Two compressors at 195 b.h.p.	390
Two fresh water circulating pumps at 8 b.h.p. ...	16
Two sea water circulating pumps at 8 b.h.p. ...	16
Giving a total of	422

The equivalent of this at 2,545 B.Th.U. per b.h.p. was approximately 1,070,000 B.Th.U. per hour, so that this gave a performance factor of $\frac{3{,}440{,}000}{1{,}070{,}000}$, or about 3·2. This, incidentally, compared very closely with the author's calculations. In Fig. 3 the author gave a theoretical coefficient of performance under these conditions of about 6·2, and an overall efficiency for a plant of the size in question of between 45 and 55 per cent. The performance factor of 3·2 was about 52 per cent of the author's theoretical factor of 6·2.

So far, the proposition looked promising. The next question to be considered was that of running costs, i.e. the relative fuel consumptions of a heat pump of the kind in question and of direct heating. It was not necessary to go right back to the cost of fuel since, the vessel being a steamer, it was possible to get a ready comparison on the assumption of steam as the basis for both; in the case of the heat pump providing electrical energy via a turbo-generator, and in the case of direct heating by steam being used direct in a calorifier to heat the water to be circulated to the existing air conditioning units. Assuming an overall motor efficiency for the compressors and pumps of 85 per cent, the total consumption of the plant of 422 s.h.p. would amount to 371 kW. Assuming the turbo-generators to require 12·5 lb. of steam per kW., which was about as low a consumption as could be expected, the heat pump would need $371 \times 12{\cdot}5 = 4{,}650$ lb. of steam per hour.

Ignoring sensible heat, and assuming that one would take advantage only of the latent heat of steam, it should be possible to obtain rather over 1,000 B.Th.U. per lb. of steam when used in the calorifier, in which case the same quantity of steam used for direct heating would give $4{,}650 \times 1{,}000 = 4{,}650{,}000$ B.Th.U. per hr. In other words, the use of steam direct from the boiler for heating the water would give 35 per cent more heat than the heat pump cycle considered, having a performance factor of 3·2. To show sufficient advantages to compensate for the necessity of running additional machinery, a heat pump performance factor of somewhere about 5 would have to be achieved.

There was no opportunity in the case of that particular vessel to pursue the matter further, principally because the refrigerating plant was installed on the tank top, and it was necessary to decide all the details with the greatest of urgency in order that the overall space required, seatings, etc., could be settled, so as not to delay the construction of the ship. It would appear, however, as if a sufficiently high factor of performance to make a heat pump a practicable proposition in a vessel of the description in question would certainly require to be arranged for two or more stage compression. That entailed additional complication, undesirable in itself and possibly prejudicial to the equipment when working as a straightforward air conditioning plant on a low ratio of compression.

The author mentioned that he considered the heat pump more likely to be a proposition in the case of motor ships. Personally, while he had not had the occasion to study any specific example in this category, he would have thought that potentially there was less likelihood of such a plant showing up to advantage in this class of vessel. In the first place, very efficient wasteheat boilers were obtainable in which steam was produced at no additional fuel cost while the main engines were running. If the exhaust gases were to be used merely for heating water to a temperature of the order of 150 deg. F., and not at steam-raising temperature, even more heat could be extracted from the exhaust gases, as they could be cooled to a lower temperature in the heat exchanger. He had only a few figures to go on about waste heat from Diesel engines, but the author would be fully *au fait* with this question and could correct his figures if they were wrong.

He believed, however, that usable heat from the exhaust gases, when passed through a waste heat boiler, amounted to something of the order of 1,700 to 2,000 B.Th.U. per kW. Furthermore, a great deal of heat at a temperature of about 150 to 160 deg. F. could be obtained from the cylinder jacket and manifold cooling systems which could presumably be used for direct circulation to the air heaters particularly as most marine engines worked with a closed fresh-water cooling system. He had particulars of a 40 kW. Diesel generating set where 280,000 B.Th.U. per hour were rejected to the cooling water, giving about 7,000 B.Th.U. per kW. It would appear, therefore, as if there was potentially somewhere about 9,000 B.Th.U. per hour of usable waste heat per kW. generated, at a temperature which could be used for direct heating, available from Diesel engines. This would give a performance factor of about 2·5.

Another point which must be borne in mind when comparing a heat pump with, say, direct steam heat was the question of capital cost incurred in the heating units themselves. In a cabin heating system, for instance, for a room temperature of about 70 deg. F. there was a mean temperature difference between the room temperature and that of saturated steam condensing at 212 deg. F. of some 140 deg. F. If, however, the water were to be circulated at 150 deg. F., this temperature difference would be reduced to about 80 deg. F., entailing an increase in heating surface in the radiators of about 80 per cent. Even so, to produce water at this temperature would almost certainly require a multi-stage plant, which would mean that any normal refrigerating plant fitted in the vessel for other purposes, such as air conditioning, would probably require extensive modifications to make it suitable for such a purpose. It might well entail sufficient additional capital cost to preclude such modification being an economic proposition.

He had mentioned earlier the problem which would arise in certain circumstances in finding a source of heat for the cooling side of the heat pump. Most vessels at some time or other found themselves in ports where the water temperature was approaching the freezing point, and in such an event a source of low grade heat must be found from within the vessel itself to enable a heat pump to function. He would suggest that that point had been overlooked by the author when he suggested that sea water provided (to use the words in the paper) "an inexhaustible source of heat for a heat pump", and any heat pump propositions would have to be very closely considered in relation to the water temperatures likely to be encountered by the vessel in the course of its existence.

His own limited examination of the field certainly bore out the author's conclusion that a very great deal of thought would have to be given to any proposal for fitting a heat pump on board ship, and he would suggest that very close liaison would be necessary between not only the ventilating engineer and the specialist in refrigeration but also the ship and engine builders. It would, in his view, be essential to establish close co-operation with the engine builder.

It was highly desirable on all counts that the plant should be as simple as possible and should be equally suitable for use either as a heat pump or for normal refrigeration purposes such as air conditioning. This strongly suggested a single-stage installation, which in turn would mean achieving a high evaporation temperature in order to obtain an economic coefficient of performance of the order of 5, as suggested earlier. For instance, in the case which he investigated a coefficient of performance of approximately this figure could be achieved by the plant fitted in the vessel if a minimum evaporation temperature of 60 deg. F. could be assured under all conditions.

It seemed to him that the engine builder was the one person who would know whether low grade waste heat in sufficient quantity could be found from sources already existing on board, so as to make the achievement of such an evaporation temperature possible.

He would very much like to know whether the author had as yet worked out any specific proposals for a heat pump to be fitted on board ship, and whether the results of his investigations had yielded results comparable with his own small excursion into this field.

Mr. E. S. Green (Visitor) said it might be of interest to the meeting to know that there were some heat pumps being installed and in course of construction on shore in this country. Most engineers had heard and read of the Norwich heat pump, which was a pioneer of its type and was said to have worked very effectively.

A small plant was installed during the war for air-conditioning a storage room for museum pieces in a quarry at Westwood. It was very like the one which had been mentioned earlier. After installation it was modified so that the refrigerating plant provided not only the cooling and dehumidifying but also the reheating necessary to give the required temperature and humidity within the storage space.

This small installation led one of the privately owned electricity undertakings to order three different types of plant for experimental purposes, and these were being started up at the present time. They were installed at one of the large generating stations, where they would be used to heat an office block and workshop. These plants had a similarity to those which might be installed in a steam-driven ship, as the low-grade heat was taken from water after it had passed through the steam condensers.

It was expected that this low-grade heat would be at a much higher temperature than at sea, with a minimum figure of 66 deg. F. and a normal of around 80 deg. F. Under those conditions, a coefficient of performance of between 5 and 7 was anticipated. One of the plants heated water for a panel-heating system; another was used as a direct heating unit for air circulation, and the third would be used in winter as a direct air heating plant and in summer as a cooling unit for air conditioning.

Another and larger plant on order and in the early stages of design also had some connexion with marine engineering, as it would be installed in an ice factory supplying ice to a fishing fleet. In the process of ice manufacture, quite a considerable amount of heat was used to thaw and free the ice blocks from the cans, and, while some of this could be obtained from the heat contained in hot ammonia gas as it was discharged from the compressors, this was insufficient when the ice frozen over a full 24-hour run by the plant had to be thawed during the 6 hours or so in which the ice was harvested.

To provide this heat for thawing was therefore the function of the heat pump, and, rather than take its low-grade heat from the brine in the ice tanks at 15 deg. F., the plant would be used to cool the water used for ice-making on its way to fill those ice cans which had just been harvested, cooling this water from 52 to 38 deg. F. and heating the thawing water from 64 to 85 deg. F. This arrangement had the somewhat unusual feature of performing its function under conditions which would give a coefficient of performance of over $5\frac{3}{4}$ to 1, while relieving the main ice-making plant of cooling the ice water under much less favourable operating conditions. To cool the ice water with a normal ice plant would take around 2·1 kW. per ton of ice, while with the heat pump it took around 1·8 kW. per ton. With the heat pump one also got thawing heat at the rate of 19,000 B.Th.U. per kW., which was about 115 per cent of the calorific value of the coal used to produce this at the power station, a double gain which, if it were not for the capital cost, would be more than something for nothing and an ideal heat pump.

Mr. W. F. Jacobs (Member) wrote that with regard to Fig. 11 was there not an error in stating that the source of heat was the canal? As far as he could see, the canal was the source of the circulating water for the turbine condenser and the probable initial temperature of the canal would be about 60 deg. F. or a little less.

In fact the text bore this out, and he assumed that somehow, the statement on the figure had appeared from those cases where heat has been "pumped up" from such a source.

At the same time it showed one method of recovering some of the energy lost overboard in the circulating water, which had always troubled him.

The heat pump would no doubt come into use for space heating ashore, but with regard to use at sea it was a question of would the saving in fuel pay for increased capital cost, increased space taken up, maintenance charges and similar items.

The Author's Reply to the Discussion

Mr. F. Joder wrote in reply to Professor Davies that he had used and preferred the term "coefficient of performance" because, in spite of its inaccuracy, it was in common use in this country and in America for both the heat pump and the refrigerator. Similarly, the French used the term "coefficient d'amplification" while the Swiss and Germans used "Leistungsziffer" for both types of plants.

Professor Davies was correct in assuming that 90 per cent referred to the total output of $2{,}000 \times 10^6$ B.T.H. per hour and not to the number of plants installed.

Professor Davies mentioned the thermal storage principle as a means for covering peak loads. He himself did not know of any actual heat pump installations which made use of this method, and to the best of his own knowledge peak loads were, at least on the Continent, taken care of by standby boiler plant.

Mr. Russell Roberts appeared to be under the impression that water near freezing point would be useless as a source of heat. He would like to point out that most heat-pump installations for space heating working with river water, particularly those in Zurich, were at times operating under conditions when the temperature of the water was only 3 deg. above freezing point, yet no icing up troubles in the evaporators had been experienced. In the case of a large swimming bath the evaporator was placed directly in the river, yet the plant worked satisfactorily with water near freezing point. It should be borne in mind that the temperature of the water of rivers, lakes or the sea at a depth of a few feet was always slightly higher than at the surface.

With regard to Mr. Jacobs' question, he was inclined to regard the canal as the source of heat and consider the turbine condenser merely as a temperature boosting stage. Taking Mr. Jacobs' argument to its logical conclusion the omission of the canal would automatically remove the source of heat.

Minutes of Proceedings of the Ordinary Meeting Held at the Institute on 12th October 1948

An Ordinary Meeting was held at the Institute on Tuesday, 12th October 1948 at 5.30 p.m. A. F. C. Timpson, M.B.E. (Past Chairman of Council and Convener of the Papers and Transactions Committee) was in the Chair. A paper entitled, "The Heat Pump with Some Notes on its Application to Air-conditioning in Land and Marine Service" (published in this issue of the TRANSACTIONS) by F. Joder was read and discussed. Fifty-five members and visitors were present and four speakers took part in the discussion.

On the motion of W. J. Ferguson, M.Eng. (Member) seconded by H. T. Meadows, D.S.C. (Member) a vote of thanks was accorded to the author for this paper. The meeting terminated at 7.15 p.m.

CARDIFF LOCAL SECTION

Inaugural Meeting

The first meeting of the Cardiff Local Section was held on Monday, 22nd October at 7.15 p.m. when Mr. Lewis M. Scaife (Member) of the London Salvage Association read a spirited paper entitled "The Ship Repair Works Manager". The meeting was held at the South Wales Institute of Engineers, Park Place. The paper covered a variety of materiel, pyschological and practical questions in connexion with the repair of ships which had seldom been dealt with in a single paper before.

Group Captain C. B. Bailey, O.B.E., D.F.C., Managing Director of Messrs. Bailey's Dry Docks, Ltd., was the guest Chairman and the Council was represented by Mr. A. C. Hardy, B.Sc. (Associate Member of Council).

Invitations to the meeting had been extended to members of the local engineering societies and the audience was as large as it was enthusiastic—numbering about 150. Of these there were forty-nine corporate members of the Institute residing in the Cardiff area.

After the reading of the paper the author answered numerous questions and it was unfortunate that the time available for discussion was not longer. A vote of thanks to the author was proposed by Mr. J. Blackmore (Member) and seconded by Mr. C. Moffatt (Chairman of the Section). Mr. Hardy proposed on behalf of the Council and members present a vote of thanks to the Chairman which was seconded by Mr. Ivor J. Thomas (Local Vice-President).

At the termination of the meeting at 9.15 p.m. an invitation was extended by the Chairman to all present to partake of light refreshments in the Park Hotel, a kindness and welcome surprise which was greatly appreciated and enjoyed by all.

JUNIOR SECTION

Lecture at Dartford Technical Institute

Mr. J. K. W. MacVicar gave a lecture at Dartford Technical Institute on 26th October entitled, "Air Conditioning". This was a very interesting talk about the use of air conditioning on board ship with particular reference to its use in large passenger liners. The difficulties due to the various conditions required by individual passengers and also the different climatic conditions encountered by the ship were stressed.

This lecture was very well attended, there being about sixty present and Mr. J. D. Farmer (Member of Council) represented the Council.

A very instructive evening closed with a vote of thanks to the author for his lecture.

Lecture at Paddington Technical Institute

A lecture was given by Mr. C. P. Harrison (Member) at Paddington Technical Institute on 7th December 1948 and an audience of about eighty much enjoyed this talk on the variety of uses of electricity at sea. The lecturer explained at the beginning of his talk that he proposed to exclude consideration of propulsion. With the aid of a number of good slides Mr. Harrison gave an interesting general outline of the use made of electricity on board a ship taking as his example a passenger liner. In passing he mentioned some of the difficulties peculiar to marine use in relation to pitching, rolling and exposure.

Lecture at West Hartlepool Technical College

A lecture on gas turbines was delivered at the West Hartlepool Technical College on Friday 3rd December 1948 at 7.30 p.m. by Dr. A. T. Bowden (Member), Chief of Research at Messrs. C. A. Parsons and Co., Ltd.

Mr. R. Kagan, M.Eng., Principal of the College, who occupied the chair, introduced the lecturer to his audience of one hundred and fifty students and engineers.

The lecture, which was given in an interesting manner, started from earliest ideas and led up to present designs; it was also well illustrated by lantern slides.

The lecture described the open and closed cycle types mentioning their advantages and disadvantages; the metallurgical and thermodynamic considerations affecting the design of the gas turbine were also reviewed. The different fuels that could be used were described, and slides shown of the effect on turbine blades of various fuels which had been tried. An experimental turbine which Messrs. Parsons had been testing for three years was also described, various details being given, and the difficulties they had had to overcome were pointed out. Photographs of this machine, both complete and in section were shown.

After the talk, the Principal invited questions from the audience, and Dr. Pugh, Head of the Engineering Section of the College opened the discussion; subsequent questions showed a full appreciation of the clear way in which the subject had been dealt with by the lecturer. At the end of the proceedings, which lasted nearly two hours, the Principal proposed a vote of thanks to the lecturer, which was accorded with enthusiasm. The Council was represented at the meeting by Mr. W. A. Alton.

Visit to De Havilland Aircraft Company

On Tuesday, 12th November a visit arranged by the Junior Section to the works of the De Havilland Co., Ltd. took place. This visit unfortunately lacked the expected support of graduates and students.

The party was met by Mr. Gearing who was an excellent guide. Firstly, films depicting the aircraft manufactured by the company were shown. The programme arranged featured the De Havilland association with the Naval Air Arm. The party was then taken on a tour of the apprenticeship training section. One was impressed by the initial tuition that the apprentices underwent in using the file and chisel. Some of the exhibits of work using only these tools were outstanding examples of skill and craftsmanship, especially in these times when such manipulation by hand appears only to belong to the past age. Other aspects of the training were shown before passing to the general machine shop. Here there were on view "mock-ups" of the engine design which are made and passed to the air-frame section before finalizing the design of the engine. Various components for the gas turbine were seen in the many stages of machining although only prototypes are manufactured in this factory.

Gas turbine and piston type engines were then seen in various stages of being dismantled after having been run for a specified period for testing purposes. Detection of flaws by X-rays and magnetic apparatus are used on the various components of the engines after being run and this type of test is also carried out on the various materials before being used for the production of engines.

The party then toured the tool room where the latest addition was a jig borer of Swiss manufacture. From here the party visited the test house where different types of reciprocating engines were seen on test and also inspected under running conditions.

The party then proceeded to the production works where the gas turbine engine is produced in quantity. Here the various members of the party showed great interest in the balancing of the rotary components of the gas turbine by use of electronics, and the tests the fuel injectors were subjected to also proved enlightening.

Thanks were expressed to the management through the guide, Mr. Gearing, on behalf of the party by Mr. F. Everard (Member).

Visit to The National Physical Laboratory

A visit took place to this laboratory on 27th November when sections of the Engineering Division were visited where creep tests at elevated temperatures and fatigue tests were being performed. Some of the research and routine testing was described which the establishment carries out on the strength of manufactured parts and on new materials. A 50-ton dead-weight primary standard testing machine was seen which is used for calibration of proving rings and testing machines throughout the Commonwealth.

The party then inspected two scale models of parts of the rivers Forth and Wyre. These enable silting problems to be investigated by setting up artificial "tides" and observing the results in a much shorter time than would be possible on site.

One of the wind tunnels in this Division occupies an entire building and is used for testing structural models under wind loading. A supersonic tunnel was seen which was employed during the war period for projectile research.

The party also saw the massive compressed air tunnel in the Aerodynamics Division where air flow is studied under pressures up to 25 atmospheres.

A brief visit was paid to a section of the Metrology Division where standards are maintained of the units of length, mass and time. The visitors were shown apparatus capable of checking time measurements with an accuracy of 0·00001 second.

The ampere balance was inspected in a section of the Electricity Division where the units of current and secondary electrical standards are maintained on behalf of the Government.

This division contains the impressive high voltage laboratory where insulation breakdown tests are performed. It houses equipment for this work capable of developing two million volts.

Of particular interest was the Ship Division. Here were seen the two main tanks for the testing of ship models. The moulding and shaping equipment was seen which produces the wax models from the shipbuilders' drawings. There was opportunity to inspect the towing carriage and instruments used to record resistance data during runs along the tanks, the longer of which measures 678 × 20 feet.

Finally, the Lithgow water tunnel was shown to the party. The underwater behaviour of propellers can be observed in this section which has done valuable work in the investigation of cavitation and associated problems.

Thanks are due to the Laboratory Administration for the facilities afforded and to the guide who piloted the party over the premises under trying weather conditions.

Ministry of Transport Notice to Marine Engine Builders, Shipowners and Shipbuilders

Explosions in Crankcases of Compression Ignition Engines

(1) The high standards of design and construction of compression ignition engines in the United Kingdom have resulted in a correspondingly high standard of safety, and explosions in crankcases have been relatively rare. Those recorded in the Ministry are few in number, though when they do occur they may be attended by more or less serious risk of injury; in one case recently there were a number of fatalities.

(2) The Ministry have carefully examined the known facts of the explosions in the crankcases of compression ignition engines that have been recorded and have reached the conclusion that they have almost invariably been due to the overheating of an internal part and the consequent ignition of a particular mixture of lubricating oil mist and the atmospheric air present in every crankcase. The Ministry have come to the further conclusion that the risk of an explosion would be reduced if the following instructions were issued for the guidance of engineers in charge of this type of engine :—

(a) Early detection of overheating and the prompt slowing down or stopping of the engine as circumstances permit will prevent the occurrence of conditions favourable to fire or explosion.

(b) Crankcase doors or inspection doors should not be opened and the engine should not be restarted until the engine has cooled down.

(c) Oil should not be sprayed on any surface the temperature of which is above blue heat (about 550 deg. F.), as risk of fire is caused thereby.

(3) Owners and masters will be in the best position to decide how these instructions should be brought to the notice of their engineers; one method already adopted is to arrange for them to be printed on cards kept permanently and prominently displayed in the engine-room.

(4) The Ministry have also examined the extent to which the risk of explosion in crankcases of compression ignition engines might be reduced by attention to the design and equipment of the engines, and the following recommendations are made for the benefit of those responsible for the design and construction of compression ignition engines of more than about 500 b.h.p.

(a) The attachment of the crankcase doors to the entablature should be substantial.

(b) There should be fitted in or adjacent to each crankcase door an explosion valve or disk of sufficient area to relieve any abnormal pressure within the crankcase and designed to reseat itself to prevent admission of further air.

(c) In multiple engine installations measures should be taken which, in the event of a crankcase explosion, will be adequate to prevent the passage of flame by way of oil drain or vapour pipes from one engine to another.

(d) Means for the detection of overheating should be considered: it will be borne in mind that smoke from the oil in contact with the heated part is often the first indication of the existence of a hot spot.

(5) Designers and builders of compression ignition engines should also consider whether, in order to obtain complete immunity from possible fire or explosion in the event of the development of a hot spot, it would be practicable to arrange for the installation of equipment enabling the injection of CO_2 gas to all parts of the crankcase.

(6) The recommendations in paragraphs (4) and (5) of this notice need not be applied to existing ships except where there is reason to believe that a definite risk exists.

MEMBERSHIP ELECTIONS

Elected 13th December 1948

Members

Charles Weston Frederick Bass, D.S.C., Lt.(E), R.N.
John Henry Baston
William Bradley
William Brown Brown
George Frederick Butcher
Roberts William Butcher
Edward Callaghan
Michael Swainston Chambers, Lt.(E), D.S.C., R.N.
Evan Robert Cameron, B.Sc.
Oswald Arthur Colbeck
Frank Thornley Davies
Leslie Charles Edwards
Ardeshir Khodadad Enayati
Himangshu Kumar Gupta
George Hair
David Mitchell Harrison
Herbert Fred Hesketh
Claude Hitchins, Lt.(E), R.N.
Reginald Kagan
Annadurai Krishnan
Sydney Lewis Leech, Lt.(E), R.N.
Charles Littlewood, Capt.(E), O.B.E., R.N.
William George Mackie
Louis Mellard, Lt.(E), D.S.C., R.N.
James Moffat Morton
William Muckle, M.Sc.
Jehangir Meherwanji Munshi
Leslie Benjamin Perry
Phillip Paul Christo Pittas
Cyril Alfred George Ralfs
Frank Ernest Arthur Sarfas
Robert Steele
Hector Roy Comrie Stewart
Harold Brown Turner
Harold Purves Walter, Lt.(E), R.N.
Thomas Henry Killingsworth White

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Frederick Jack Bishop, Lt.(E), R.N
John Copeland
John Cludery Hardy Roberts

Associates

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Albert Buckler, Sub. Lt.(E), R.N.
Eric Bradshaw, Sub. Lt.(E), R.N.
George Wilkinson Bulman
Douglas Hamilton Cameron, A/Sub. Lt.(E), R.N.
Arthur Edward Coombes, A/Sub. Lt.(E), R.N.
Thomas Jefferson Davis
Capt. Henry Harold Fantham
Gordon Knewstubb
Ronald Thomas Lovell, Sub. Lt.(E), R.N.
Peter Lovering
John McCaig
Claude Robinson Maddick
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James Stephen Rowley
Kenneth Ernest Allen Shattock, Sub. Lt.(E), R.N.
Mahendra Paul Singh Sodhi
John Spence
Robert Joseph Stone, Sub. Lt.(E), R.N.
Leonard Sweeney
William Brock Thompson
George Newbegin Wallace
Stuart Benjamin Whitty, War. Eng'r, R.N.

Graduate

James Cadzow Smith

Students

Richard Bedforth Clayton
Reginald Charles Crouch
Norbert Dienes

Transfer from Associate Member to Member

Thomas Percy Gibbeson

Transfer from Associate to Member

John Sinclair Allan
John Andrew Beckwith
George Blakely
John McIntosh Breen
Alexander Coutts Brotherston
George Curphey
Mohamed Mostafa Aly El-deeb
Harry Gerrard
Sydney Walter Hammond
James Edward Francis Harris
Norman Joseph Harvey, M.B.E.
Ronald George Iliffe
Edgar Israel Kerridge
John Mackie
Leonard John McQuade
William Miller
Frederick Minnikin
Stephen Alexander Morrison
Jack Sherwood Nell
Arthur Edwin Plowes
William Edward Postlethwaite
Lessels Roy
Cyril Andrew Sinclair
James Snadden
John Michael Somerford
Sidney Slater Whitehead
Robert James Wilson
Arthur Woodbridge
Hugh Wynn-Davies
John Muir Young

Transfer from Associate to Associate Member

Thomas Henry Kennedy

Transfer from Student to Associate

Konstantin Herlofson, Lt.(E), R.Nor.N.
Frederic George Righton, Lt.(E), R.N.

OBITUARY

MR. GEORGE SCOTT (Member 7904) was born in North Shields in 1883. Educated at Kettlewell School and Manson Academy he served his apprenticeship with Messrs. Swan, Hunter and Wigham Richardson, Ltd. After beginning his seagoing career with the Prince Line he joined the British Tanker Co., Ltd. in 1919 and became superintendent engineer. Apart from service at Southampton, Glasgow and the Tyne, he was also associated with engine erection and installation of the first Swedish motor Diesel engine. He was also connected with a similar service in West Africa. He died on the 2nd November 1948 after thirty years' sea service and was buried at Preston Cemetery at North Shields.

LT. COM'R.(E) ERIC FRANCIS TRUSCOTT, R.N.V.R (Member 9344) was born in 1894. Educated at various private schools he served his apprenticeship with Coalbrookdale Iron Works and at the outbreak of the 1914-18 war was Sub. Lt.(E) R.N.R. serving with the auxiliary cruiser *Orbita*. From 1919 to 1923 he was assistant works manager and later works manager of Darling and Lloyd of Oswestry and in 1923 he joined the staff of the Anglo Iranian Oil Co., Ltd., and became Chief Inspecting Engineer, which post he held for twelve years broken only by service in the 1939-45 war in which he held the rank of Lt. Com'r(E) R.N.V.R. as assistant to the Admiralty engineer overseer, North-West Division and at the Admiralty, Bath. He turned to advantage his hobbies of model construction and inventing and patented several of his ideas. He was a Member of the Institution of Engineering Inspection and was elected a Member of this Institute in 1942. He leaves a widow and a son. He died on the 24th October 1948.