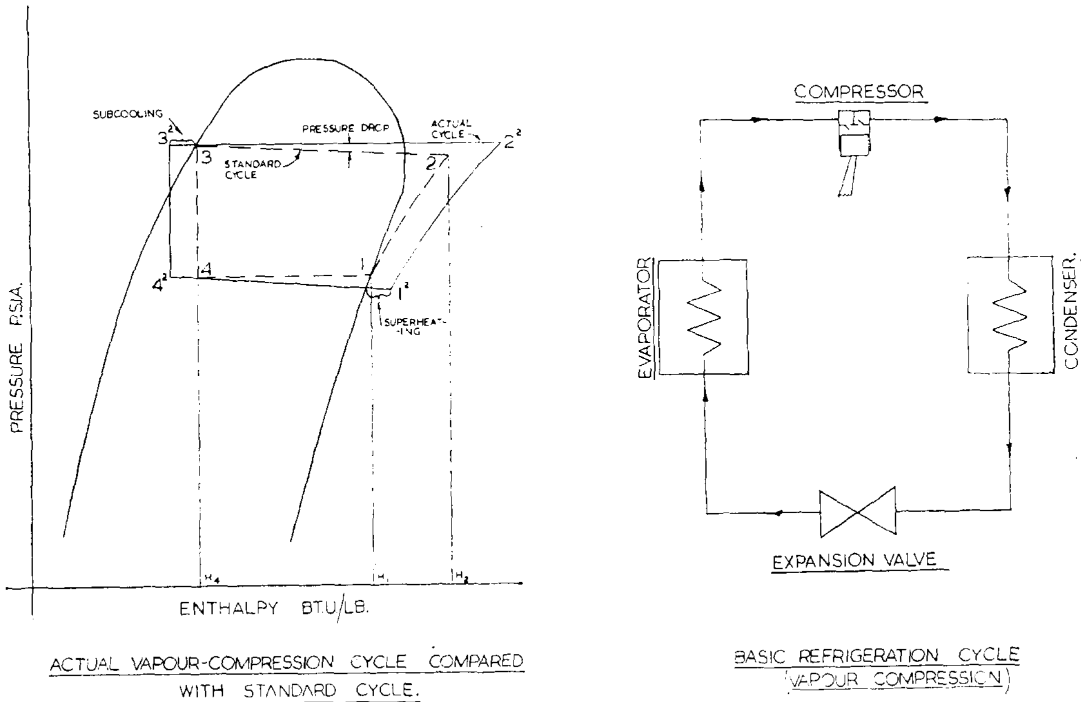

FIG. 1—THE VAPOUR COMPRESSION CYCLE

PART II

AIR CONDITIONING MACHINERY

BY

D. G. GREAVES, C.ENG., M.I.MECH.E., R.N.E.S.

With the implementation of the 'repair by replacement' concept and the need to maintain an intact gas circuit between major overhauls to avoid contamination problems, all new air-conditioning plants are being designed in package form. It then becomes possible to remove the complete refrigeration plant intact to a suitable overhaul facility at regular intervals when the specified life has been achieved. With the rapid growth of air-conditioning requirements and the need to restrict the number of equipments fitted in any one vessel, these sets are now becoming extremely large and the physical problem of removing such a set from the ship requires careful planning in the early design stage of the ship.

Although a variety of designs of compressor and other circuit components are available, all current naval plants are of the same basic vapour compression cycle.

Vapour Compression Cycle

The cycle is illustrated in FIG. 1. The essential processes in this cycle are:

1—2 Compression
2—3 Rejection of heat in condenser
3—4 Expansion
4—1 Addition of heat in evaporator.

The coefficient of performance of the cycle, which is a measure of its efficiency, is defined as the 'useful refrigeration' divided by the 'net work'. It is calculated from the expression:

$$\frac{H_1 - H_4}{H_2 - H_1}$$

Maximum efficiency is achieved with a low condensing temperature and a high evaporating temperature but in practice the former is limited by the specified sea-water temperature and the latter by the chilled-water temperature selected.

Some modification of the cycle is necessary to ensure that the gas returning to the compressor is dry and slightly superheated and that the liquid entering the expansion valve is completely gas free and slightly sub-cooled. Pressure drops in the refrigerant pipes must also be allowed for. The resultant actual vapour compression cycle is indicated by 1^2, 2^2, 3^2, 4^2.

Typical Vapour Compression System

The gas flow diagram of a typical vapour compression system is shown in FIG. 2. The principal components are:

Compressor, the function of which is to raise the pressure and temperature of the refrigerant gas to a point when it can be condensed by sea water at the highest temperature likely to be encountered—usually specified as 95 degrees F.

Oil Separator, to remove any lubricating oil carried over with the gas discharge, returning the oil to the compressor.

Condenser, which provides the heat transfer surface to pass the heat in the hot gas, plus the latent heat, through to the sea water.

Liquid Line Drier, charged with silica gel or activated alumina, to remove any moisture present in the system.

Heat Exchanger, to ensure that the refrigerant passing back to the compressor is completely gasified and slightly superheated.

Thermal Expansion Valve, which meters the correct flow to the evaporator and reduces the pressure of the liquid refrigerant to enable it to flash off at the conditions existent in the evaporator.

Evaporator or *Chiller*, which provides the heat transfer surface to pass the heat from the chilled water back into the refrigerant.

Control

In earlier plants no capacity control devices were fitted. As the thermal expansion valves closed under reduced load conditions, the suction pressure of the compressor fell until the machine stopped by operation of the low-pressure cut-out.

To avoid this continual starting and stopping on low load with consequential damage to the starter contacts and disturbance to the main supply system, the hot gas by-pass system was introduced. This system, which is sensed off the chilled-water outlet temperature, meters hot gas from the compressor discharge into the evaporator thus maintaining sufficient flow back to the compressor to avoid the low-pressure cut-out operating. The TSA10 valve, which controls the movement of the MSA valve, commences to operate when the chilled-water

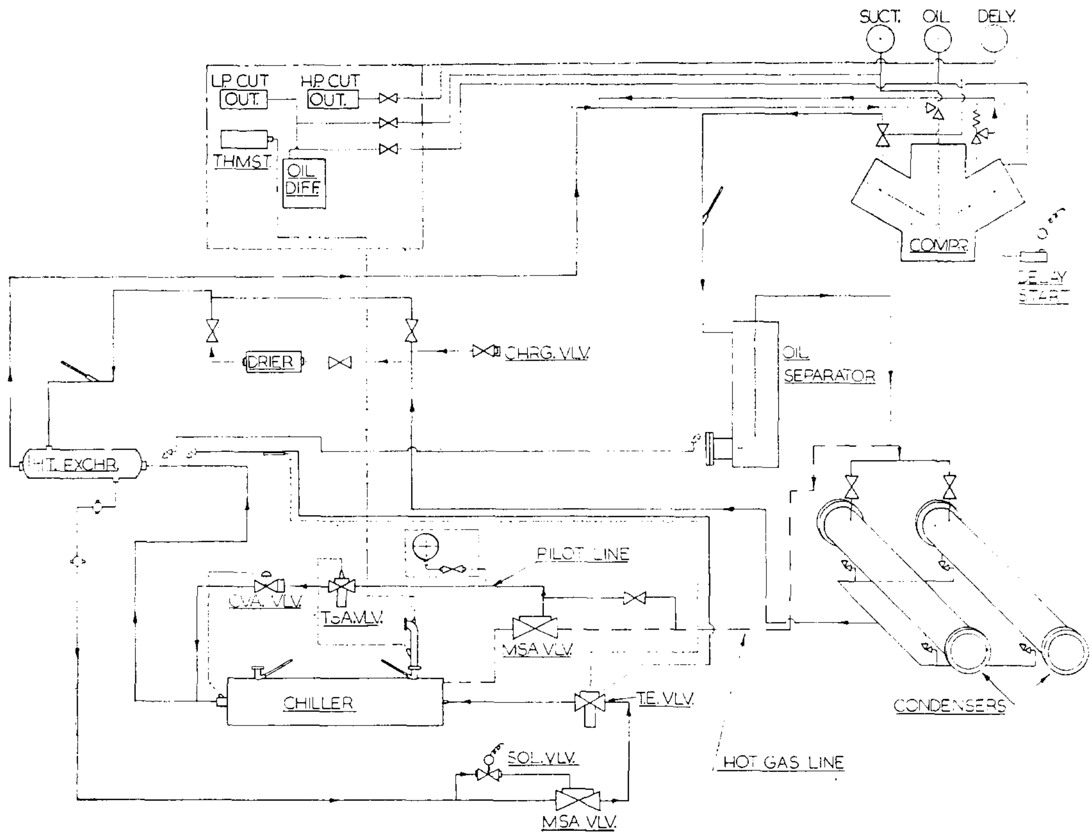

FIG. 2—REFRIGERANT FLOW DIAGRAM—880 000 BTU/HR AIR-CONDITIONING PLANT

outlet temperature falls from 44 to 43 degrees F, and is closed at 40 degrees F when the MSA valve is wide open. The CVA10 valve is independent of the TSA10 and operates the MSA valve only on the occasion of start up when low suction pressures occur in the system which could cause the LP trip to operate. The CVA10 valve will start to admit hot gas at 29 p.s.i.g., full by-pass being achieved at 27 p.s.i.g. The LP trip will operate at 26 p.s.i.g.

In the case of the screw compressor, described later, the control system is simplified by the internal design of the unit and no hot gas by-pass system is considered necessary.

Control and Protection Panel

Previously, protection and indication have been provided by separate standard pressure and temperature gauges and thermal and pressure cut-outs. On all new plants control, indication, warning and shut-down will be centralized into a single Teddington Mk IV panel mounted on the plant (FIG. 3). The panel is built up of any required number of standard 6 in. cube solid state modules (or multiples thereof), which plug into sockets permanently mounted on the enclosing case. Temperature sensing is achieved by the use of platinum resistance thermometers, and pressure sensing by diaphragm-operated transducers. Multipoint selector switches are utilized where possible to reduce the number of temperature/pressure indicators required. The panel mounted on a SYMES 300kW packaged air conditioning plant can be seen in FIG. 9. Repair by replacement of individual modules was a design feature of this panel.

Selection of Refrigerant

The ideal refrigerant should be non-toxic, non-flammable and non-explosive, have physical and thermal characteristics which will result in the maximum

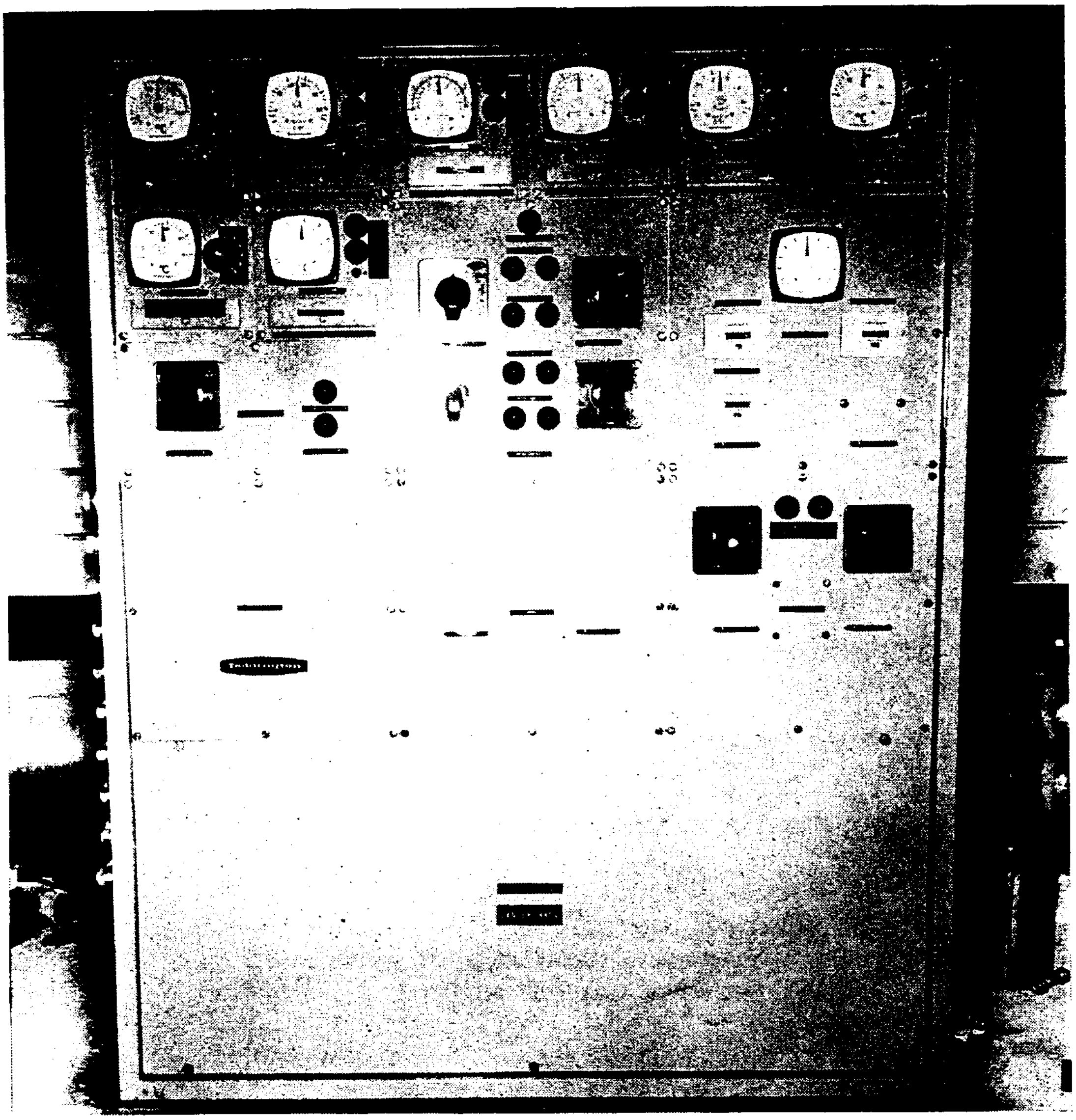

FIG. 3—TEDDINGTON MK.4 CONTROL PANEL

economy of plant weight and space, will not attack plant construction materials either alone or in the presence of water, and is not miscible with oil.

In the early days of refrigeration, application was limited and refrigerants were restricted to ammonia and carbon dioxide. Later, methyl chloride was developed particularly for use in centrifugal compressors. The search for a completely safe refrigerant with good thermal properties led to the development of the fluorinated hydrocarbons in the late 1920's and this range is now practically the only world-wide source of refrigerant. The field of choice is therefore limited by practical considerations of replenishment.

FIG. 4 illustrates some of the pertinent properties of the more commonly used refrigerants arranged in descending order of the evaporator pressure required for similar duties, high operating pressures being considered disadvantageous to mechanical design. Further considerations are given below.

Carbon Dioxide

Although desirable from its chemical properties and small compressor capacity, carbon dioxide has a small coefficient of performance (COP) and therefore requires twice the power of other modern refrigerants. It also requires a robust design of equipment due to the high circuit pressures.

COMPARATIVE PERFORMANCE PER TON REFRIGERATION									
R. REFRIGERANT	EVAP. PRESSURE PSIG.	CONDSG. PRESSURE PSIG.	REFGT. CIRCU'L'D. LBS/MIN.	COMP. DISPMT. FT³/MIN.	CO EF. OF PERFMC.	HP/TON	TOXICITY.	COMP. DISCHARGE TEMPS. °F	FLAM-MABLE LIMITS.
744. CARBON DIOXIDE	317·5	1031	3·62	0·96	2·56	1·84	5	151	NON FLAM.
22. MONO-CHLORO-DIFLUORO-METHANE.	28·3	159·8	2·89	3·60	4·66	1·011	5A	131	NON FLAM.
717. AMMONIA.	19·6	154·5	0·422	3·44	4·76	·989	2	210	16-25%
12. DICHLORO-DIFLUORO-METHANE.	11·8	93·2	3·92	5·81	4·70	1·002	6	172	NON FLAM.
11. TRICHLORO-MONO-FLUORO-METHANE.	24" HG.	3·6	2·96	36·32	5·09	·927	5	113	NON FLAM.
718. WATER (40°F & 86°F)	29·7 HG.	28·6 HG.	0·195	476·7	4·10	1·125	—	282	NON FLAM.

NOTE. ABOVE CONDITIONS ARE FOR 5°F EVAPORATION AND 86°F CONDENSATION

FIG. 4—COMPARISON OF REFRIGERANTS

Ammonia

Ammonia is considered undesirable for shipboard use in enclosed compartments being toxic and somewhat flammable. It does hower have the highest refrigerating effect per pound of any refrigerant and requires only a relatively small compressor.

Water

Water is acceptable from the chemical view point but the necessary compressor capacity is extremely high and the COP is lower than the halogen compounds. Its use is therefore limited to applications where large quantities of low quality steam are readily available and where leakage of other refrigerants is unacceptable.

Halogens

The remaining refrigerants all belong to the halogen group. The choice of which halogen refrigerant to be used will depend upon the application. R12 would be the preferred gas as it is most commonly available, is completely safe, is highly stable and relatively inexpensive. The operating pressures are above atmospheric for air-conditioning applications so that no possibility exists of air or moisture leaking into the plant.

Where space restrictions are of prime importance R22 will be selected as indicated by the smaller compressor displacement required. It is however more expensive than R12, is slightly more chemically active and has a lower miscibility with oil. R22 is highly suitable for hermetic gas systems (where the refrigerant passes through the motor) having a high volume resistivity—100 times that of R12—and high dielectric strength.

The low operating pressures and the very high compressor displacement required by R11 necessitates the use of a centrifugal compressor. R11 is therefore suitable only for plants of 1 million BTU capacity and above, and leakage

OUTPUT B.T.U/HR	MOTOR H.P.	COMP. R.P.M.	REFGT.	OVERALL DIMENSIONS
700,000	75	575	R. 12	82"; 96"; 111"
880,000	100	750	R. 12	82"; 91"; 114"
$1 \cdot 1 \times 10^6$	135	638	R. 22	80"; 92"; 116"
$1 \cdot 2 \times 10^6$	150	710	R. 22	67"; 104"; 121"

FIG. 5—GROWTH OF LEANDER AIR-CONDITIONING PLANT

of air, oil and moisture into the refrigerant would be hazards of using this refrigerant.

Selection of Compressor

The only circuit component which is broadly open to selection in a vapour compression system is the compressor. Chillers, condensers and expansion valves are to a basic design for the air-conditioning application and few alternatives are available for a given duty. The three main types of compressor available are:

(*a*) Reciprocating
(*b*) Rotary
(*c*) Centrifugal.

Reciprocating Compressors

The reciprocating compressor is by far the most commonly used machine and is found in naval service in plants ranging from small self-contained air-conditioning units of 7000 BTU/hr up to the Ikara 'conversion' plant of $1 \cdot 1 \times 10^6$ BTU/hr. It is a simple piece of machinery and easily maintained, although with many moving parts the maintenance cost is high. A single compressor can be easily adapted to provide a wide range of duties by varying the speed of drive and, within limits, changing the refrigerant. In general the compressor duty will vary directly as the speed and replacing the R12 refrigerant by the R22 will increase the available duty by a factor of 1·6.

FIG. 5 illustrates the development of the *Leander* Class air-conditioning plant

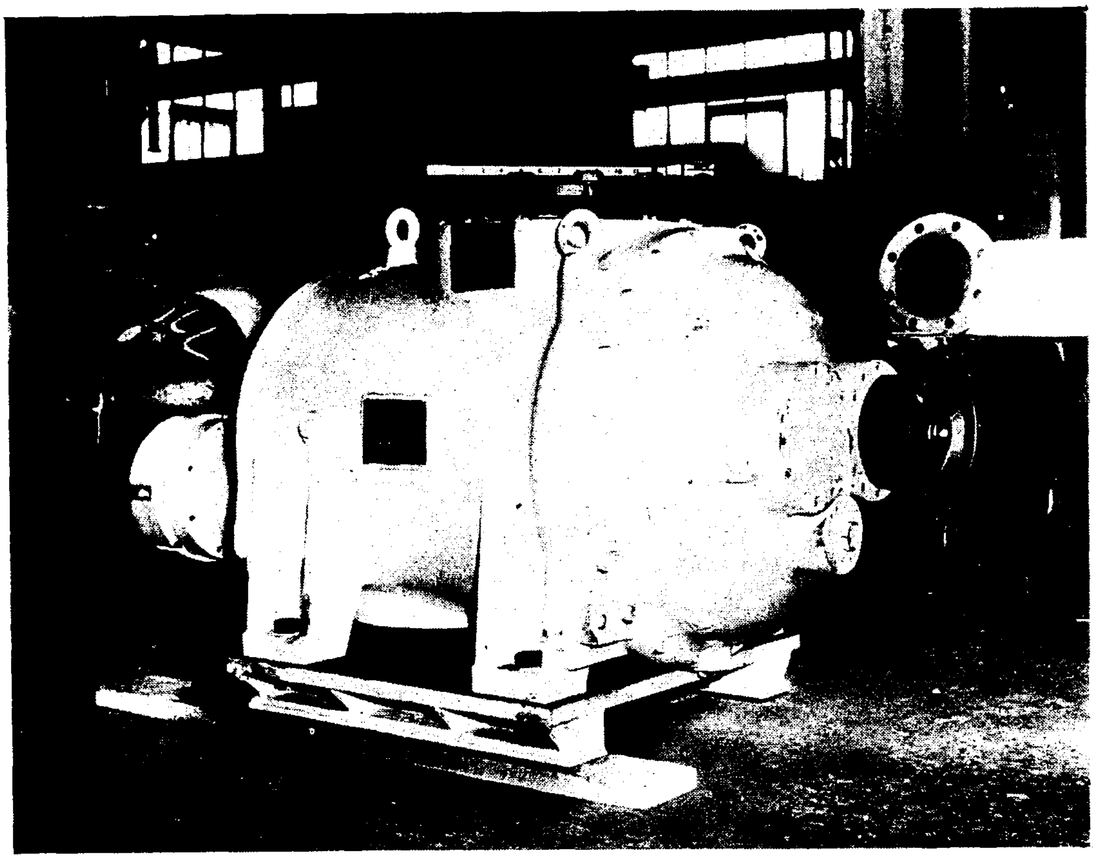

Fig. 6—Howden 204 screw compressor

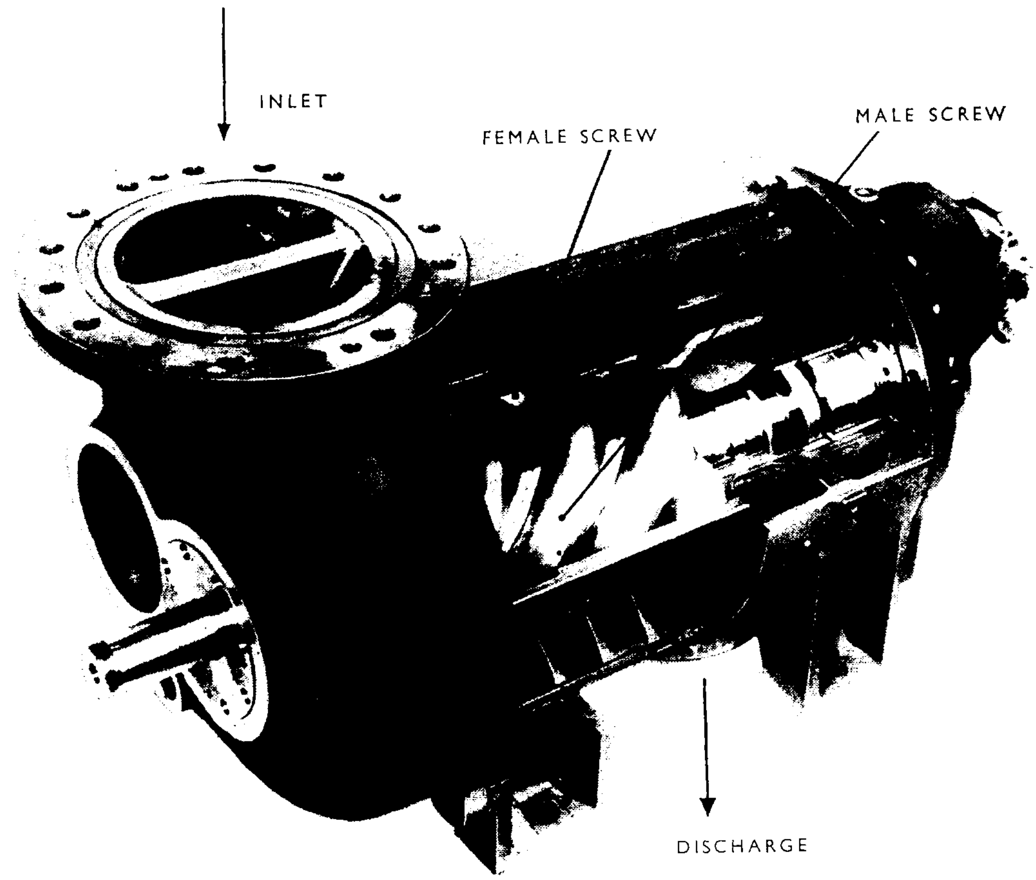

Fig. 7—Section through screw compressor

based on the Lightfoot F56 compressor. The plant was originally designed by Lightfoot Refrigeration for 700 000 BTU/hr at a compressor speed of 575 rpm using R12 refrigerant. An increase in speed to 750 rpm, still using R12, increases the guaranteed output proportionately to 880 000. A further improvement in output to $1 \cdot 1 \times 10^6$ BTU/hr was achieved by replacing R12 with R22 refrigerant and permitted a reduction in compressor speed to 638 rpm.

The possibility of converting this compressor to direct drive is now under active consideration with a target output of $1 \cdot 2 \times 10^6$ BTU/hr. However, the maximum running current limitations imposed by the ship's system are now likely to limit this final stage of development.

Rotary Compressors

Rotary compressors of the vane or eccentric-rotor type have not been developed for air-conditioning plants generally due to the difficulties of obtaining an internal seal at the relatively low pressures required by modern refrigerants.

The Rotary Screw Compressor, manufactured by James Howden and Company has however shown considerable promise since the successful application of oil injection into the meshing rotors to provide a more effective seal, reduction of the temperature of compression by absorbing some of the heat into the oil, and the elimination of rotor timing gears. The advantages claimed for this type of compressor are:

(*a*) Extreme reliability
(*b*) Low maintenance costs—mean time between failures (MTBF) of 4000 hours and a sub-assembly life of 25 000 to 40 000 hours is expected
(*c*) Absence of wearing parts
(*d*) No inlet or outlet valves
(*e*) Internal capacity control down to 10 per cent
(*f*) Liquid refrigerant can pass through the compressor without causing damage
(*g*) Stable surge-free running under all operating conditions
(*h*) Lack of vibration
(*j*) Can operate on a variety of refrigerants
(*k*) Discharge temperature never exceeds 100 degrees C.

The only known disadvantages are a relatively high noise emission and the need for a large oil separation vessel.

A commercial version of this type of compressor operating on R12 refrigerant and running at 1800 rpm (shown in FIG. 6) has been selected to power the air-conditioning plants now being built for the Type 42 with an output of 300kW (1 million BTU/hr). This plant has been designed to meet the latest interchangeability requirements. Control of this type of compressor is achieved by adjusting the inlet slide valve, thus altering the point of gas entry into the mating male and female scrolls. FIG. 7 shows the internal arrangement of the compressor. The slide valve is shown in the fully loaded position in FIG. 8(*a*) and in the unloaded position in FIG. 8(*b*).

A navalised version of modified design incorporating shock-proof materials, running at 3600 rpm and incorporating an integral lubricating oil pump, has been under development for the past three years and is now available to plant designers. The output of this compressor is expected to be $1 \cdot 15 \times 10^6$ BTU/hr on R12 refrigerant and $1 \cdot 85 \times 10^6$ on R22. In future, the screw compressor will be recommended for all plants of 1 million BTU/hr and above and only these plants will be included in the SYMES range.

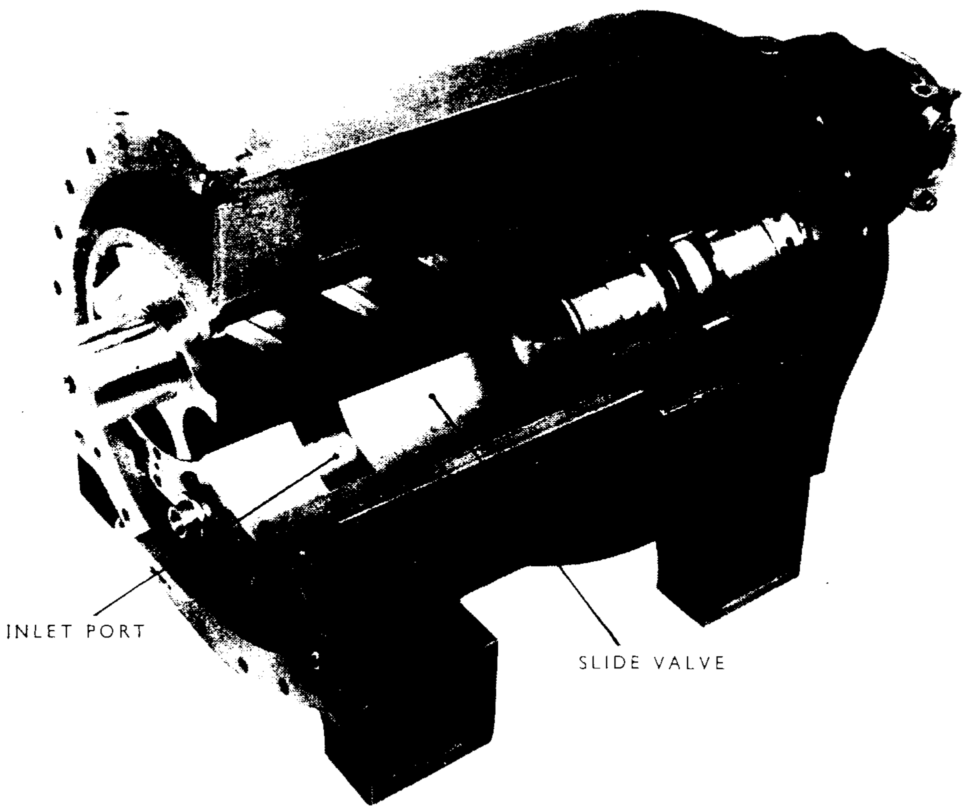

FIG. 8(*a*)—SCREW COMPRESSOR—LOADED

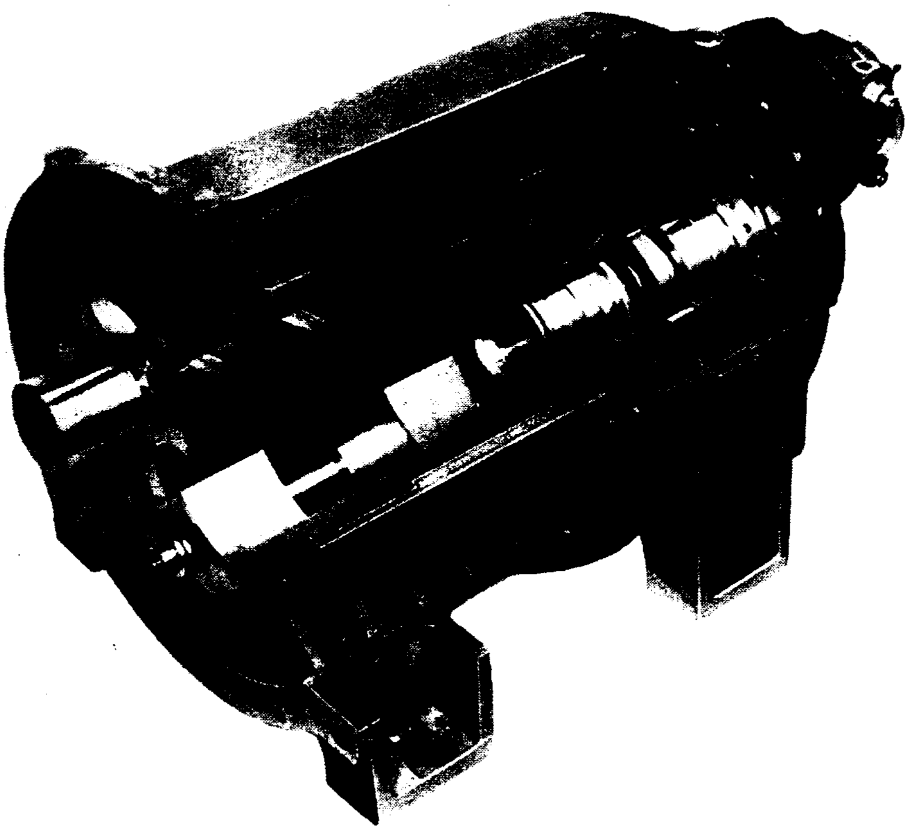

FIG. 8(*b*)—SCREW COMPRESSOR—UNLOADED

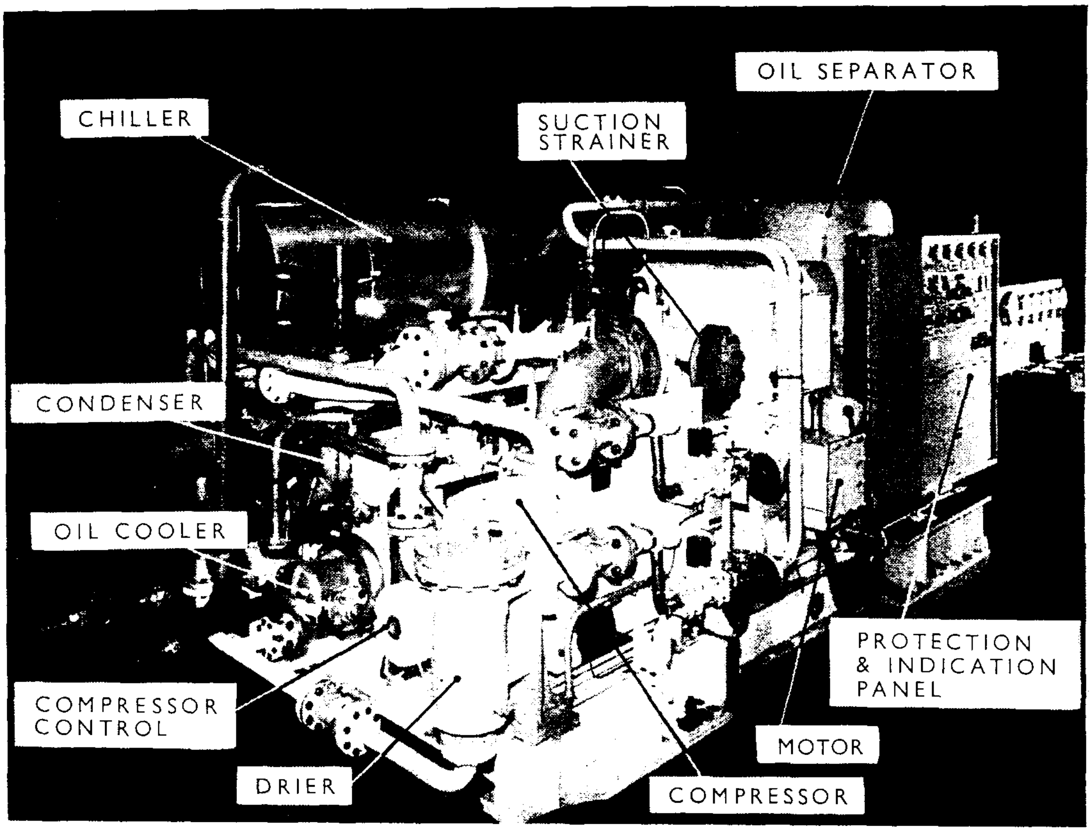

FIG. 9—300 kW PACKAGED PLANT

The size of the packaged 1 million plant, shown in FIG. 9, and designed by Hall Thermotank International is:

164 in. long x 74 in. wide x 79 in. high

compared with the 1·1 million *Leander* Ikara plant dimensions of:

116 in. long x 92 in. wide x 80 in. high

i.e., 4 ft longer, although 18 in. narrower, which illustrates the penalty of oil injection necessitating oil separation; the additional length is required to house the substantial oil separator. The oil separation problem can be simplified by fitting a hermetic motor to drive the compressor and using the motor as a first-stage separator. With such a design however, an electrical failure of the motor would distribute debris throughout the gas system and necessitate breaking the gas system to fit a replace unit. The design of a simple compact separator is currently under active consideration.

Centrifugal Compressors

As mentioned previously, these machines are suitable only for large capacity plants of more than 1 million BTUs output using R11 refrigerant and have been used in the 1, 2 and 3-million sizes in aircraft carriers and cruisers.

Necessarily operating under sub-atmospheric conditions, contamination of the refrigerant system with moisture and lubricating oil is a very real problem. Purge units are provided and a system cleaner has recently been added to improve reliability. Additionally, considerable quantities of refrigerant can be absorbed by the lubricating oil and may result in compressor journal and thrust bearing failure, particularly on starting up. If the life and reliability of the oil-injected screw compressor is proved in Fleet service, it will be adopted as the preferred compressor for larger plants, eventually replacing the centrifugal compressor.

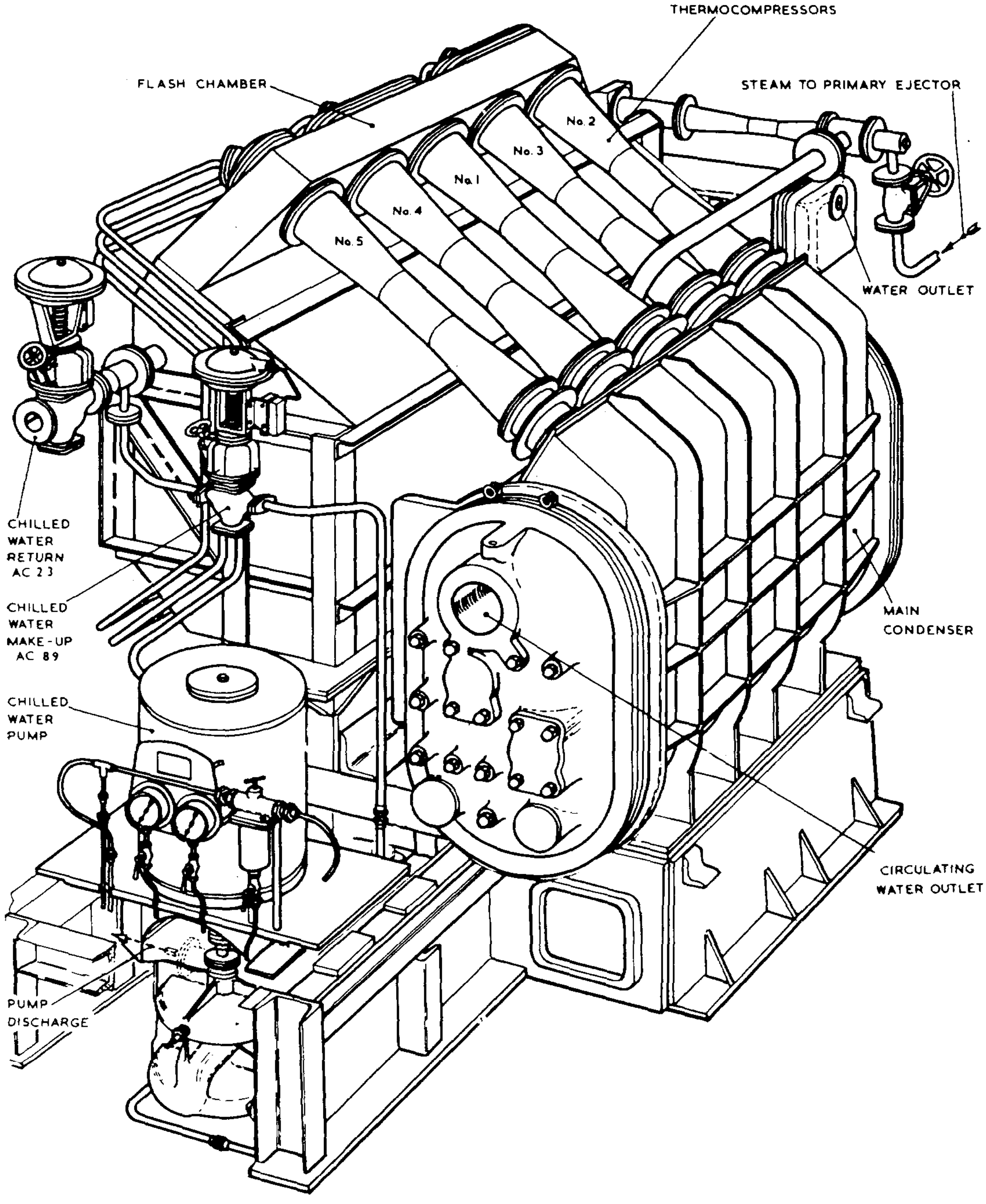

FIG. 10—720 000 BTU/HR STEAM VACUUM AIR-CONDITIONING PLANT

Other Refrigeration Systems

Jet Refrigeration Plant

The only other type of air-conditioning plant currently in use in the Fleet is of the steam vacuum vapour compression type shown in FIG. 10. These plants were selected for installation in a number of nuclear submarines on the basis of the availability of large quantities of low quality steam and the elimination of the hazard to the crew should a leak occur. These plants have been extremely successful in service apart from the difficulties of controlling the evaporator water level and hence the output when operating in parallel, and the release of considerable quantities of wild heat.

With the introduction of gas turbines for main propulsion, these sets are unlikely to be fitted in future surface ships due to the lack of suitable steam supplies.

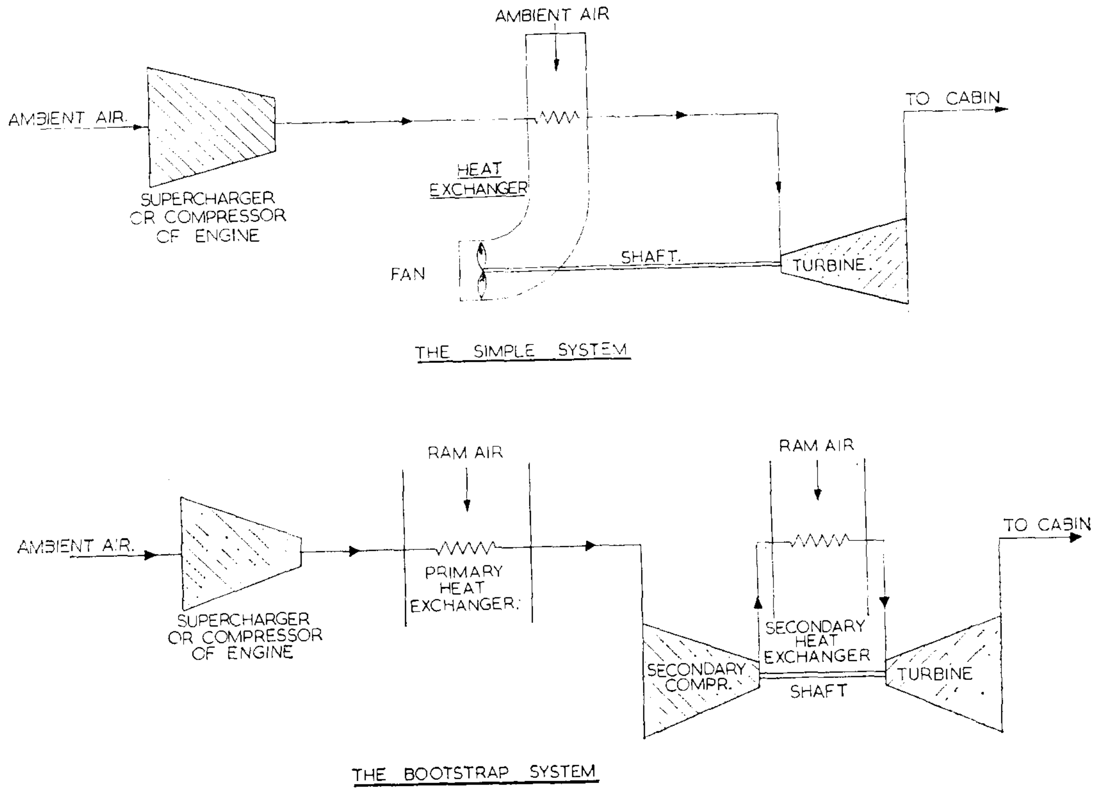

FIG. 11—AIR CYCLE REFRIGERATION

Absorption Plants

These plants operate on a similar principle to the Electrolux domestic refrigerator except that the ammonia solution is replaced by lithium bromide. Such a plant has been used in the United States Navy, but due to the requirement of large quantities of low quality steam or hot water, and excessive height and weight, this plant has not found favour in the Royal Navy. A 1·5 million BTU plant weighs 18½ tons as compared with 9 tons for a similar screw-compressor plant.

Thermo-Electric Plant

The Thermo-electric system is worthy of mention by reason of its many potential advantages: it has no moving parts, the motor compressor, condenser, throttling valve, evaporator and vapour refrigerant being replaced by thermo-electric couples and an electrical power supply; control is very simple; it is achieved by regulating the power supply to the thermo couples, and reversal of the power supply converts the unit into a heater.

Early development of such a system showed promise but was discontinued because of the heavy costs involved.

Air Cycle (FIG. 11)

Air cycle cooling is achieved by expanding pressurized air through a turbine thus converting heat energy into work. This cycle is commonly used to cool the cabins of commercial aircraft in which case the cycle is frequently modified by the addition of a compressor to reduce the load on the aircraft turbine. The advantages of such a system are that air is used as the working medium, and compact high-speed compressors and expanders may be used with considerable saving in weight and space. The life of these equipments is, however, relatively short compared with that required for marine equipments.

The COP of a straight air cycle is about 0·9 to 1·0, that is, the power requirements are at least four times that of a conventional vapour compression plant. In the aircraft system, the cooling plant power requirements are reduced by bleeding the compressed air off the main engine compressors and the cooling of the heat exchangers is achieved by the ram effect of the aircraft motion. In a warship the full power price must be paid.

The small high-speed turbines required to expand the compressed air operate at up to 100 000 rpm and noise levels of 130 dBs are to be expected. A feasibility study has been placed with Yarrow-Admiralty Research Department to compare such a system with a standard vapour compression system fitted in a nuclear submarine.

Air Conditioning Units

In conclusion, mention must be made of the small self-contained air-conditioning units ranging from 7000 to 24 000 BTU/hr capacity which are available 'off the shelf' and are a god-send to the designer who has made a mistake with his sums and underestimated the heating load. These units are fitted in most ships of the Fleet in varying numbers and are maintained on a repair-by-replacement basis.