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## THE DESIGN OF A SMALL OIL FLOODED ROTARY AIR COMPRESSOR

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### ABSTRACT

There have in the past been few successful small oil flooded rotary air compressors, due in the main to the cost of manufacture and the problem of internal condensation with intermittent operation. The paper describes a new approach to overcoming these problems, and reports a successful outcome.

### INTRODUCTION

Thirty four years ago the first oil flooded sliding vane rotary air compressor appeared on the British market. The new invention of oil flooding (otherwise known as oil injection or oil cooling) had such a dramatically beneficial effect on the performance and reliability of rotary compressors that in due course oil flooded vane and, more recently, screw compressors were to dominate much of the world market in the 7 - 10 bar (100 - 150 psig) output pressure range. The exceptions were at the top end of the output league, above 40 cmm (1500 cfm) approximately, where dynamic compressors had advantages and at the small end, typically below 10 l/s (20 cfm) where the reciprocating compressor was almost unchallenged.

It is interesting to observe that the earliest oil flooded compressors were not merely adaptations of the well-established rotary compressor with means for oil flooding and oil separation added, but entirely new designs bearing little or no resemblance to existing compressors. It is equally interesting to note that some of those early compressors were still in production 30 years later - and still considered in the market to be up-to-date designs from both technical and aesthetic viewpoints. Great credit must go to the designers of the day, but it is difficult to envisage history repeating itself in today's fast changing world.

Although oil flooded rotary vane compressors are manufactured in a wide range of sizes extending from 1.1 - 150kW (1.5 - 200hp) their strongest impact has been in the 4 - 75kW (5.5 - 100hp) area. There have been very few successful attempts at producing oil flooded rotary compressors of below 4kW (5.5hp). The author's company manufactured 1.5 and 3kW (2 and 4hp) compressors in the 1950's and 1960's and a 1.1/2,2kW (1.5/3hp) machine in the 1970's but although these were successful and sold in fairly large numbers, their penetration of the overall market was very small indeed. There is no doubt as to the main cause of this: however competitive the larger rotary machines, these smaller compressors were simply too expensive. Furthermore they were not particularly well suited to automatic stop/start operation, which is a popular requirement with small compressors, as light duty running could lead to problems with internal condensation.

#### PREVIOUS DESIGNS

The 1.5 and 3kW (2 and 4hp) designs already referred to were typical of 'first generation' oil flooded rotary air compressors, being known as the Hydrovane 9 and 18 PU respectively: Figure 1 shows the 9 PU 'air end'. Although excellent innovative thinking had created these compressors, there was not the same emphasis placed on design for economic manufacture that exists today. For example many of the main components were fairly heavy and complex, designed for sand rather than die casting. Access to some of the internal components for maintenance purposes was not easy and the compressor required careful attention by the user and regular servicing by the specialist. This somewhat demanding attention was the price that had to be paid for the benefits of low noise, no vibration and the ability to operate continuously without an air receiver.

The second attempt to produce a cost-effective oil flooded small vane compressor is shown in Figure 2. This 'second generation' machine, known as the Hydrovane 6/12 PU, was designed to reduce costs, increase efficiency and reliability, and be more accessible for maintenance. Designed as a 1.1kW unit when driven by a 4 pole motor (the 6 PU), the same compressor (with a larger cooler) absorbed 2.2kW when driven by a 2 pole motor (12 PU). However this dual-speed concept could only be applied to 50Hz electrical supplies as the 60Hz 2 pole speed of some 3500 rpm was too fast, giving rise to excessive noise and frictional losses. Therefore with a 60Hz supply, a 4 pole 1.5kW (2hp) version was the sole machine.

The 6/12 PU compressor represented a considerable step forward compared with the older 9/18 models. One major change was to mount the motor and compressor rotors on a common shaft, and therefore the motor and compressor end plates could become one component. This arrangement had the advantage of eliminating the

need for couplings and connecting flanges but in the end the benefits proved to be more than offset by the fact that the motor, being a non-standard item, became relatively expensive. Unlike the parallel of the hermetic refrigeration compressor where volume of manufacture justifies a special design, the numbers of the 6/12 were never large enough to capitalise on the concept and it eventually became a major disadvantage as motor manufacturers moved towards a greater degree of product standardisation.

Another feature which had a major impact on the design of the 6/12 was the oil cooler. At the time, the whole range of Hydrovane compressors up to 18.5kW (25hp) utilised circular oil coolers produced from sheet steel pressings, seam welded on the inner and outer perimeters to form toroidally-shaped cooler tubes which were then 'stacked' together to form a cooler. Production tooling existed for two diameters of cooler tube and it was therefore highly convenient to use the smaller of these, being a known route and a low cost item; moreover, suitable alternatives were simply not available at the time. However, the use of this cooler tube with its outside diameter of 350mm (14ins) led to a main casing of similar diameter, partly for ease of oil connections and partly for aesthetics. But this diameter was somewhat larger than it needed to be for its prime purposes of (1) containing the compressing elements (known as the rotor stator unit); (2) containing an adequate quantity of oil; (3) providing space for primary oil separation. The main disadvantage of the larger diameter arose from the higher wall loadings leading to a thicker and more expensive casting which also had to be relatively complex in internal shape to accommodate the separator and other components. Functionally the compressor was very similar to existing larger models.

The end result of the design was an excellent compressor which achieved its functional objectives admirably and which was a great improvement over its predecessor. It gained many friends, being smooth, quiet and very reliable and opened up new markets because of its ability to operate at full load for 24 hours per day. However increasing downward pressure on prices, particularly from imported die-cast reciprocating compressors, meant that sales of the 6/12 gradually became more polarised towards the very small sector of the market where quality and reliability were the main considerations.

#### THE NEW DESIGN (5 SERIES)

#### Marketing Specification

The decision to re-design the 6/12 came as a result of the fact that it had become possible to obtain a reciprocating compressor of equivalent output for less than half the price.

In order to halve the cost of the 6/12 it was obvious that a fundamentally new approach would be required as no amount of trimming and refinement of existing designs would be adequate.

But before design could be considered, the marketing specification had to be obtained to ensure that the various features (as well as price) were what the customer really wanted. The marketing specification abbreviated to the following:-

- \* Range of sizes: 1.1 & 2.2kW (1.5 & 3hp) -- 50Hz  
1.5kW (2hp) -- 60Hz
- \* Working pressure: 7 & 10 bar (100 & 150 psig)
- \* Output: 2.5 & 4.5 l/s at 7 bar -- 50Hz  
(5.2 & 9.5 cfm at 100 psig)  
3.3 l/s at 7 bar -- 60Hz  
(7 cfm at 100 psig)  
10% less at 10 bar (150 psig)
- \* Noise level: 60 dBA for 1.1 kW, up to 68 dBA for  
2.2kW measured at 1 metre
- \* Oil carryover: Less than 10 ppm by weight
- \* Method of drive: Direct
- \* Motor: Standard metric TEFC IP54, 1 & 3 phase
- \* Electrical supply: All international single and three  
phase light industrial and domestic  
voltages
- \* Packaging: Receiver mounted and stand mounted.  
Receiver 50 - 100 l (13 - 26 US gal)
- \* Control: Automatic stop/start on receiver  
mounted; continuous run with output  
modulation, push button start, on stand  
mounted
- \* Maintenance: Oil changes and filter cleaning only.  
Minimum long-term internal maintenance.
- \* Styling: Compatible with existing products
- \* Cost: Target figures given (approximately  
half that of the 6/12)

## Functional Analysis

An early observation was that all existing compressors manufactured by the company were functionally similar to each other, being different primarily on physical size and shape and the way in which the components were arranged. In order to stand a chance of making the cost breakthrough, the existence of every component and feature would have to be challenged from the functional point of view. The main areas examined and questions deriving were as follows:-

Rotor: Obviously essential but variation possible in number of vane slots and length/diameter ratio. Also not necessarily integral with shaft.

Stator: Equally essential but considerable simplification possible e.g. manufactured from tubing with either no ports or drilled ports as distinct from the conventional casting with cast ports. How essential is the secondary bore which is normally incorporated to provide a long narrow gap between stator and rotor as a seal between high and low pressure?

Vanes: Not much variation possible except in materials. One vane per slot desirable.

End Covers and Bearings: End covers are essential but not necessarily with conventional plain bearing bushes. Do other forms of support bearing offer advantages? Are support bearings necessary? Could a suitable bearing surface be machined in the end cover? End covers could possibly be combined with other components. Porting could be in the stator as opposed to end covers.

Oil Flooding: The concept of oil flooding is so essential that it had to be retained but in view of the fact that the normal quantity used for flooding is 10,000 times greater than that required for lubrication, might smaller quantities give adequate results and perhaps simplify or reduce the cost of oil separation?

Oil Circuit: In addition to supplying the injectors, oil has to be distributed to the bearings and rotor end faces for sealing and lubrication purposes. Normally all oil passes through the cooler before being distributed (except during the warming up period when it is bypassed). Functionally only the injected oil needs to be cooled and therefore the other oil circuits could be fed directly from the sump as long as the bearings are designed for uncooled oil.

Oil Chamber/Main Casing: Functionally it serves to provide a sump for the oil and space for primary oil separation. Conventionally it also houses the rotor stator unit and as such may play a functional part in noise attenuation. Under conventional arrangements it is pressurised to system pressure but whereas this is necessary for oil separation, is it necessary for its function as an oil sump?

Oil Separator: In view of the quantity of oil that is discharged with the compressed air from the stator port, some means of separation is essential. Two stages, one mechanical and one through a coalescing filter are required: mechanical alone is not adequate and a coalescing filter would be grossly overloaded with oil without primary separation. However smaller separator systems may be possible.

Rotary Shaft Seal: A rotary shaft seal is essential but whether or not this is subjected to pressure in the running or stopped condition depends on design. Unpressurised operation is simple to achieve by venting the inside face of the seal to the inlet port, but most recent practice has subjected the seal to pressure when stopped, made possible by either multiple sealing lips or single lip PTFE seals. Whichever system is used, reliability is essential.

Oil Filter: The function of the oil filter is twofold: to prevent debris causing mechanical damage, and to prevent premature blockage of the oil coalescing separator. The progressive oxidation of mineral lubricating oil does not create deposits which can usefully be removed by filtration. The only ways in which debris can be present are via the air intake filter, oil filler plug, or inadequate cleaning during manufacture. For many years strainers, as opposed to filters, had been used successfully and therefore there might be a case for no oil filters or strainers - given adequate quality control and suitable positioning of pick up points. Also coalescing separators could be protected by integral pre-filters.

Cooler: Unlike a reciprocating compressor, the heat generated within an oil flooded compressor cannot be dissipated by forced convection from the casing. The reasons for this are partly the lower internal temperatures in the flooded compressor, and partly the fact that much of the heat of compression is transferred to the oil; thus an oil cooler is required and forced ambient air is the preferred cooling medium (as opposed to water) by universal market demand. However, current practice on other current Hydroyane models dictated a maximum sump oil temperature of 100° C. (212° F.) in an ambient temperature of 40° C. (104° F.); if the compressor could be allowed to run hotter then a smaller cooler may be adequate.

Controls: All Hydrovane compressors, past and present, have had modulating control (i.e. variable output) which has enabled them to run continuously irrespective of air demand. The particular system used has consisted of an intake throttling valve and a servo valve to control its position. A vacuum limiting valve has then usually been necessary to prevent vane instability when running with the intake closed. This system of control causes the off load power to be reduced to about 70% of full load power, and this figure can be reduced further by decompressing the main casing. However in the case of small compressors, any off load power savings are minimal. Also, market predictions showed that the main demand for small compressors was for receiver mounted automatic stop/start units where there need not be an off load mode. There was therefore a good case for eliminating the somewhat complex and expensive reduced-power unloading system although the specification called for modulating control for stand mounted versions.

Packaging: Receiver mounted and stand mounted versions of the compressor were specified. There was little previous experience of receivers mainly because the modulating control rendered them unnecessary for most applications. Apart from size, the main issue with receivers was whether or not they could be designed to incorporate or avoid the various international regulations on construction which can have the effect of severely limiting possibilities for export. Otherwise the receiver, as the stand, merely provides a mounting for the compressor.

#### Preliminary Experimental Work

Before the design could be started it was necessary to carry out some experimental work to validate the concepts suggested as a result of the functional analysis. The main areas of interest were means of generating the minimum necessary oil flow and its subsequent separation; and whether the rotor could be mounted on the motor shaft thus avoiding the need for bearings. In addition, experimental data was required on vane and end cover materials and the effectiveness of the traditional secondary sealing bore.

The results proved invaluable and showed that whereas conventional approaches to the rotor, bearings, vane materials and secondary bore were preferable, aluminium was quite suitable for end covers and that reduced oil flows could be used without impairing efficiency.

In view of the conclusion that conventional reduced-power modulating control was unnecessary, it was decided that the variable output facility could best be met by a bypass system recirculating air from high to low pressure. However there was a need to ensure that such a system operated smoothly and progressively and with a maximum pressure differential of 0.5 bar

(7 psi) between open and closed positions. Experimental work on various designs of valve was undertaken and confirmation obtained that these characteristics were achievable and also that overheating would not occur in the bypassed condition.

### Condensation

An oil flooded rotary compressor (vane or screw) will be subjected to internal condensation until the air paths within the compressor have reached a temperature above dew-point. As a rule the condensate will collect in the oil sump. Operation for a period above dew-point is then necessary to evaporate any condensed water. The fact that most oil flooded compressors take at least 10 minutes to reach a temperature high enough to drive off any condensate, despite bypassing of the oil cooler, originally tended to steer their application towards continuous or semi-continuous duties. Today automatic stop/start control is commonplace but generally requires minimum-run timers or similar devices which ensure that a high enough temperature is reached before the compressor stops. Simple stop/start control with no such override has in the past been unsuitable for light duty applications but for the new design it was seen as essential.

The problem of condensation was approached in two ways: first to let it happen and then take steps to detect and drain off the water; second to try to prevent it. The former would rely on using an oil, such as a turbine oil, which would not emulsify with water and allowing the level of water to reach a point where it could be detected, either visually or automatically. The thinking behind this approach was that it was impossible to eliminate condensation under all conditions and therefore means had to be found to live with it. However the solutions propounded were neither elegant, reliable nor inexpensive and therefore attention was directed towards minimising condensation. Various ways of achieving this were considered: reducing the thermal mass by minimising the amount of metal and oil in the air end; isolating the cooler whilst the unit was warming up; retaining the heat in the discharge airstream prior to leaving the compressor. Other ideas were to use a fairly large air receiver and a wide stop/start pressure differential in order to maximise the run times. And finally the use of conventional compressor oils which emulsify in the presence of water was seen to be beneficial because more water could be condensed without causing problems, and in passing emulsified oil through the rotor stator unit, there would be a greater chance of evaporating the water.

### Cooling

A major problem in the design of small compressors is the cooler. As previously mentioned, there was a wish to avoid excessively large cooler diameters. As the basic motor diameter for the new model was about 200mm (8 ins), it was felt that the

cooler should be of a similar diameter. The existing toroidal cooler sections were, at 350mm (14 ins) too large and it was felt that quality and maintenance problems caused by sealing of adjacent sections merited an integral cooler for 'third generation' compressors. Although the experimental units had used coiled coil coolers located in the air stream being induced into the motor, the cooling was inadequate for a production design and therefore an alternative system was required. A range of highly efficient rectangular panel coolers were available, designed for automotive applications, and constructed as integral units from salt-bath brazed aluminium. Experiments were conducted using small motor-fan units separate from the air end and whereas adequate cooling could be obtained the solution was neither inexpensive nor elegant. After a number of alternative schemes had been considered it became clear that a cooler would have to be found which was toroidal in shape so that it could be located in the conventional Hydrovane position, utilising a centrifugal fan located between air end and motor. Coiled coil coolers were tried but the relatively low air velocity from small diameter fans caused the downstream side of the cooler tube to be 'blanked off' with resulting low efficiency. The only alternative was to consider a toroidal version of the brazed aluminium cooler and, working with a cooler manufacturer, a suitable one was developed. It was considered highly desirable to draw rather than push air through the cooler in order to achieve more uniform air distribution, and ensure that contaminating dust would collect on the outside of the cooler from where it could be more easily removed.

#### Final Design (see Figures 3 and 4)

General Configuration: The range of sizes, voltages and frequencies spanned by the new compressor could be covered by a single frame of motor, the standard metric D90, with a 200mm (8 ins) flange diameter. From the air compression and oil separation aspects this dimension was considered to be highly suitable as a main casing outside diameter. The significance of this decision is shown in Figure 5. A motor compressor unit shaped essentially as a horizontal cylinder of this diameter was considered to be aesthetically attractive and in keeping with the general appearance of the existing larger Hydrovane compressors.

The arrangement of main components, as shown in Figure 3, consists of a main casing, cast closed at the drive end, and closed at the opposite end by the separator casing. The main casing closed end serves as a stator end cover and houses one bearing. The stator and inlet end cover are clamped between the main and separator casings. As well as closing the main casing, the separator casing provides a chamber for the oil separator element; this chamber is closed by an end cover which contains a discharge minimum pressure valve. The air end is flange

connected to the motor and the drive to the rotor is via a flexible coupling. The motor half coupling also comprises a centrifugal fan which draws air through the adjacent cooler.

Castings: In order to minimise cost and thermal mass, all castings were designed for manufacture from pressure die aluminium of minimum wall thickness using simple straight pull dies. The choice of LM24 (Al-Si8Cu3Fe) as the casting alloy gave the required mechanical properties although chromate finishing treatment was considered necessary to prevent corrosion.

Rotor: This was initially chosen with the conventional 8-slots and half-hole vane venting. The possible reduction in number of slots, and venting by vane grooving, were deferred for future action, being considered of minor importance. Material S.G. (nodular) iron bar, fully machined.

Bearings: Conventional white metal lined shell bearings were chosen as being low cost and highly effective from both bearing performance and pressure sealing aspects. Elimination of separate bearings by machining a bearing surface in the end covers was regarded as a future development.

Stator: Designed for simple machining from either thick-walled cast grey iron tubing, bar or castings, the stator has a single drilled discharge port. The intake port was located wholly in the end cover. The secondary bore was retained because it provided certain locational advantages.

Machining and Assembly: The design was such that all critical inter-dependant surfaces could be machined at one component setting. Most of the machining is turning, boring and drilling, with minimum milling. Contrary to previous practice, and because of minimum accumulated tolerances, the use of pre-finished bearing bushes was possible. Simple vertical assembly was provided by being able to stack components in their correct sequence, starting with the motor connection flange and finishing with the separator end cover.

Air Path: Air enters the compressor through the intake filter to the intake cavity between the separator and intake end cover. Entering the stator through the inlet non-return valve, the air is compressed and discharged from the port, and then directed towards the separator by a short and discrete path which provides just enough time and space for primary oil separation. In this way minimum heat is lost prior to separation. Downstream of the separator element, the air leaves the compressor via the minimum pressure valve.

Oil Path: Oil is circulated by pressure differential, being fed to the bearings and end faces direct from the sump. In this way the oil is at maximum pressure where its sealing function is particularly important. Oil is directed to the single injector from the sump via the cooler. This path is controlled by a thermal valve which opens when the oil temperature reaches a pre-determined value. In this way the only oil introduced to the compression cells during the warm-up period is that which has passed the end faces and this ensures a rapid temperature rise of discharge air whilst still providing adequate lubrication and sealing. Primary separation takes place in the vicinity of the separator casing, thus ensuring maximum transfer of heat to the casing, another factor in minimising condensation. Final removal from the airstream of any remaining oil takes place by coalescence in the separator element (which also embodies a dirt-holding pre-filter). The coalesced oil drains to the bottom of the separator chamber from where it is returned to the intake cavity via a filtered orifice; recirculation through the intake valve then follows.

Valves: The only valves are the intake non-return, discharge minimum pressure, and pressure control valves. The first two employ lightweight stainless steel valve plates sealing against brass seats and the pressure control valve utilises a spring-loaded stainless steel ball on a brass seat. These materials were chosen to prevent corrosion and eliminate maintenance.

Sealing: Sealing was an important criterion. Apart from the rotary shaft seal, which is a single-lip PTFE type capable of withstanding full pressure when the compressor stops (non-pressurised when running), there are only two 'O' rings which seal the castings externally and 'O' ring-sealed connectors join the cooler to the main casing. Where appropriate, Viton elastomer is used.

Safety: Very great care was taken to cater for all possible modes of failure. Although a pressure relief valve is incorporated within the compressor to guard against the most unlikely event of excess internal pressure, the design of bolted connection of the main casings is such that on increasing pressure the bolts would stretch, the sealing faces would part and total internal relief from high pressure to atmosphere would occur. This ultimate relief pressure is still only 40% of the burst pressure of the main casing components. High temperature shut-off devices are fitted where appropriate.

Air Receiver: Various schemes of receiver were considered, aimed at overcoming the problems caused by international regulations. However the final outcome was a simple standard receiver which could be sourced locally without difficulty.

### Comparison with the 6/12

The final design showed the following comparison with the earlier 6/12 models:-

	<u>6/12</u>	<u>New Design</u>	<u>% Reduction</u>
Number of components:	209	145	30
Number of different components:	125	107	15
Number of specially manufactured, i.e. non-proprietary components:	93	43	56
Weight of oil chamber equivalents:	12.7 kg	3.8 kg	70
Weight of air end:	31.8 kg	11.2 kg	65
Overall cost:			50

### Performance

The performance of the new 5 Series compressor was assessed in two main areas: the flow, power consumption and noise characteristics, and the resistance to condensation. The target figures for the former were reached and in certain cases exceeded. Resistance to condensation proved to be excellent and can be explained by the test results shown in Figures 6 and 7.

Figure 6 shows the warm-up characteristics of the 5 PU at various locations within the compressor, and the effect of the thermal valve; also a comparison with the previous model 6. Despite the improved rate of warm-up compared with the 6, the oil separator region stays noticeably cooler than the other parts of the compressor, due no doubt to the cooling effect of the incoming air. Therefore condensation is more likely to take place in the separator, the condensate being returned to the intake of the compressor via the oil return orifice and thus either re-cycled through the separator or deposited in the sump. Although the separator condensation could be reduced by fitting a thermally insulating internal sleeve, tests have shown this to be unnecessary.

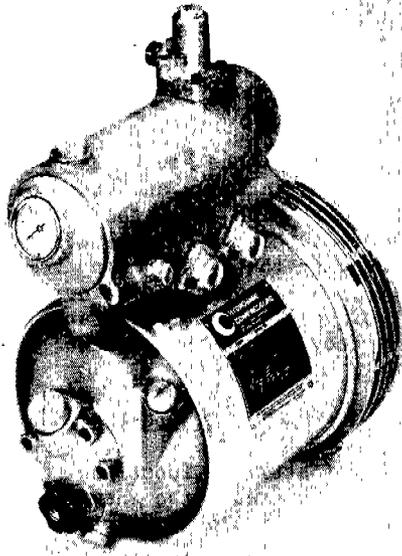
Figure 6 also shows that with the thermal valve closed a significant temperature difference exists between the top surface and lower regions of the oil sump. This occurs as a result of the relatively small quantity of oil which then circulates, being only that which lubricates and seals the bearings and end faces.

This flow is insufficient to cause significant turbulence in the sump and layering takes place as the hot oil drips down on to the sump surface. With the thermal valve open the increased turbulence leads to a more uniform temperature throughout the sump. The peaks on graphs 2 and 3 show the opening of the thermal valve.

Figure 7 demonstrates the significance of different light duty cycles on sump temperatures and the volume of condensed water remaining in the sump, with and without the thermal valve. This valve is seen to be highly effective in combatting condensation but the graphs suggest that the main reason for this is the indirect effect of the valve in allowing the formation of a hot surface layer of oil which evaporates the descending water droplets before they can reach the lower and cooler region of the sump.

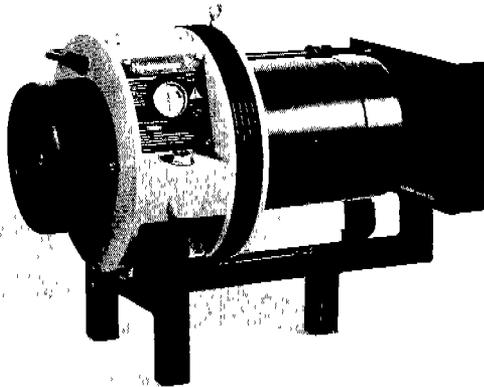
#### CONCLUSION

As a result of strictly functional design and departure from many established practices, the new 5 Series compressor was created. Being designed for production from pressure die castings with minimal and simple machining and assembly, the performance and cost objectives were met, the latter representing a 58% reduction over the previous compressor of equivalent output. The problem of condensation inherent in stop/start oil flooded compressors was successfully overcome.



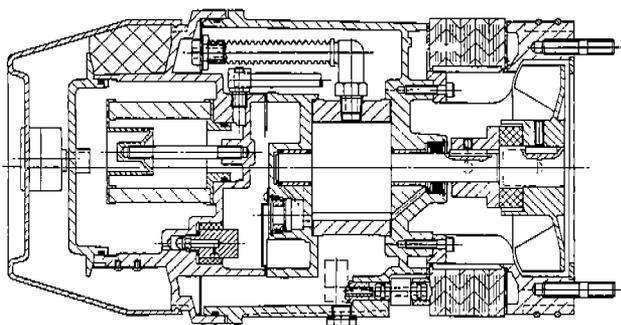
9 PU AIR END

FIGURE 1



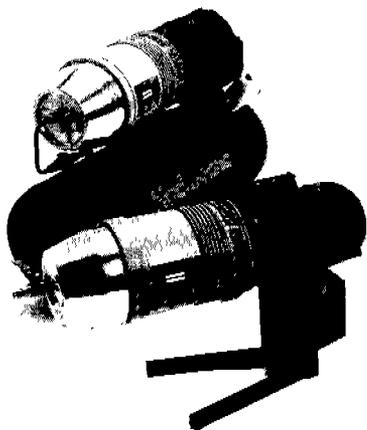
6 PU COMPRESSOR

FIGURE 2



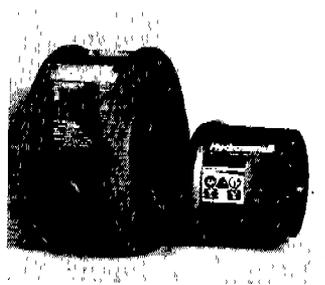
SECTION THROUGH 5 SERIES AIR END

FIGURE 3



5 SERIES COMPRESSORS

FIGURE 4



MAIN CASINGS OF  
6 (LEFT) AND 5 SERIES

FIGURE 5

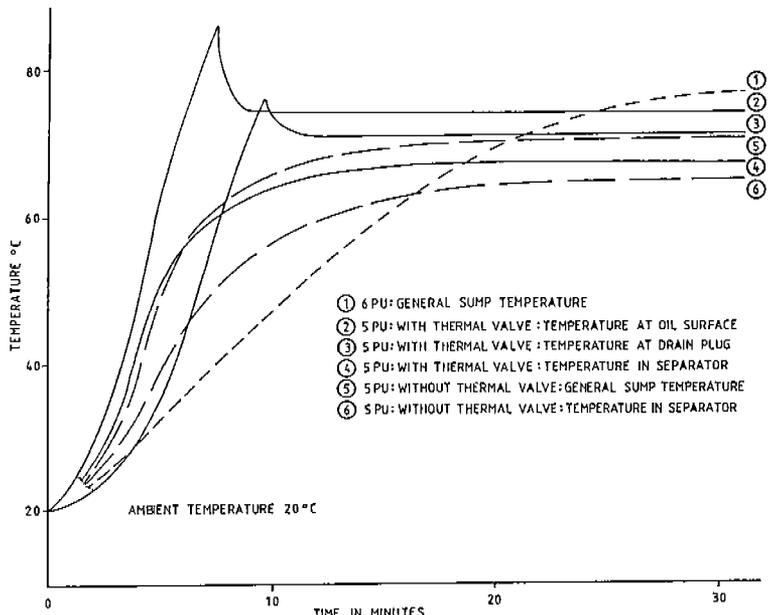


FIGURE 6.

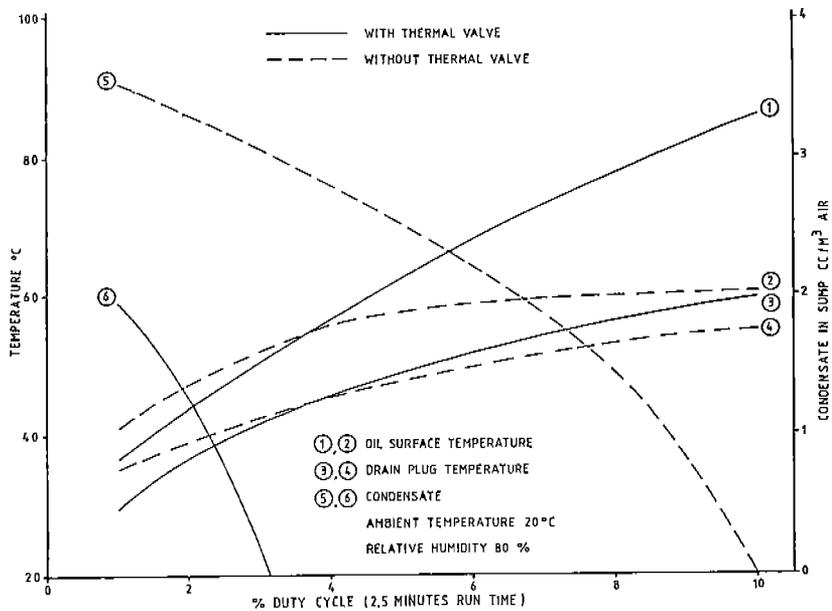


FIGURE 7