1976

Development of Large Refrigerant Compressors

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DEVELOPMENT OF LARGE REFRIGERANT COMPRESSORS

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The size range "large refrigerant compressors" is understood hereinafter to mean compressors with displacements of about 300 m^3/h to 7,000 m^3/h. Several different compressor types in this size range have been studied, but those with which the greatest success has been achieved are reciprocating compressors and screw compressors. The reciprocating compressors are constantly being developed, but the development is largely a matter of improvements to familiar and wellproven components. The technique for development of new valve systems, for instance, has become much faster and more reliable in consequence of the simulation methods nowadays available. A detailed account of the latest experience of this technique will quite certainly be presented during this conference.

The machine type on which by far the most development work has been carried out in recent times is the oil-injected screw compressor. Compressors of this type were introduced to the market as refrigerating compressors about 15 years ago and farily soon found a natural size range of about 750 m^3/h to 2,500 m^3/h, i.e. bigger than most common sizes of reciprocating compressors and smaller than most common centrifugal compressors. These displacements correspond to rotor diameters of 200 to 250 mm with direct drive from 2-pole AC motors.

The most important development phases of the last five years can be summarized as follows:

- asymmetrical rotor profile
- economizer system
- auxiliary systems such as oil separation, oil cooling and capacity control

ASYMMETRICAL ROTOR PROFILE

The asymmetrical profile (Fig. 1) is built up with a line-generated leading flank and a point-generated trailing flank. Line-generated means that the entire flank of the female rotor (point D to E) generates the leading flank of the male rotor (point I to J), while point-generated means that a point on one rotor (B on the female rotor, H on the male rotor) generates a flank on the other rotor. The major advantages of the new profile are that:

- an increased displacement is obtained from given dimensions.
- a shorter sealing line is obtained in the rotor mesh, i.e. the sealing area between high pressure side and low-pressure side will be smaller.
- there will be a smaller blow hole. The blow hole is a leakage area giving leakage from an interlobe space with higher pressure to the next interlobe space with lower pressure. This has little influence on volumtric efficiency, but affects the power consumption of the compressor.
- better meshing properties, i.e. the profile works even better as a gear than the earlier symmetrical profile. The reason for this is the line generated leading flank.

The profile thus gives both better volumetric efficiency and better total efficiency. It is characteristic that the profile gives substantial improvements at low tip speeds, i.e. permits the introduction, from the standpoint of efficiency, of smaller screw compressors than are common today.

ECONOMIZER SYSTEM

This system has been described earlier in some articles, as indicated in the list of references. In principle, the system is based on the liquid after the condensor in the plant being either throttled in two stages or subcooled in a heat exchanger (Fig. 2). The flash gas formed in course of throttling is subsequently introduced at an intermediate-pressure port in the compressor, positioned in the rotor housing where a suitable intermediate pressure prevails.

By this means, the thread volume is supercharged with the amount of flash gas formed.

By employing this system on a given compressor 20-40% higher capacity is gained, depending on operating conditions. A further benefit being an improvement of 5-30% of the specific capacity (refrigerating capacity/power consumption). This improvement of the specific capacity is gained mainly by the two-stage effect, which means that the increase in refrigerating capacity due to subcooling of the liquid increases more than the power consumption due to compression of the flash gas. The flash gas must of course be compressed from intermediate pressure to high pressure. Further gains are obtained because a large part of the compressor losses is fixed and related to a higher refrigerating capacity.

The system has been used on a large scale in recent years. The background is that it allows a very simple installation build-up in comparison with a two-stage system. The efficiency in normal freezing applications is roughly between that of a single-stage and a two-stage plant.

AUXILIARY EQUIPMENT

When the oil-injected screw compressors first were introduced considerable problems were encountered in some installations with the oil separation and the entire design of the plant in order to adapt this to an oil-injected compressor. Over the years, great investments have been made by different manufacturers of screw compressors in order to improve the efficiency of the oil separators. Today, a great deal more is known about oil separation, which is no longer a problem.

Oil separation systems can often be selected according to plant type, i.e. depending on the demand on the degree of oil content in the refrigerant. Even a standard oil separator nowadays gives such a low oil carry-over that, counted for instance in kg/h of circulated refrigerant, the oil consumption is as low as in a modern reciprocating compressor.

A great deal of work has also been devoted to the development and simplification of oil cooling systems. Formerly, the oil was always cooled with water in an ordinary oil cooler, which is still done today. In order to avoid using water, refrigerant cooling systems have been developed and there are now a number of such systems available. One method is to use a heat exchanger and to lead the boiled-off refrigerant back to the condenser. Another system is to inject liquid refrigerant directly into the compression chamber at a pressure lower than the discharge pressure. This system is widely used, but should nevertheless be used with some caution in certain operating conditions. A more reliable system is to mix the warm oil and liquid refrigerant in a mixer, and inject the cooled oil and flash gas into the compression chamber. The discharge temperature is hereby kept so low that the oil in the sump can be used for lubrication of bearings, shaft seal, etc.

Electronic capacity regulating systems have been developed which are directly adapted for the functions required in the screw compressors in order to fully utilize the continuous and highly sophisticated capacity control capabilities of this compressor type. Also built into these systems are secondary functions such as minimum switch for unloaded starting, motor current limitation, time delay for oil pressure cut-out, etc., in order to simplify plant engineering.

LARGE SCREW COMPRESSORS

The development work carried out has made it possible from an economical standpoint to extend, both upwards and downwards the natural size range for screw compressors. Over the last 2-3 years, several screw compressor manufacturers have introduced larger screw compressors in the range of about 7,000 m^3/h displacement.

The design philosophy behind the development of these compressors has been to attain the greatest possible operational reliability and robustness. The cost aspect is not equally dominant for these sizes as it is in the case of smaller compressors. Figure 3 shows an example of such a compressor from one manufacturer. This compressor has a rotor diameter of approximately 330 mm and a rotor length of approximately 590 mm. The compressor gives a refrigerating capacity of 5300 kW at +25/-10°C with NH_3 as refrigerant.

SMALL SCREW COMPRESSORS

The range of the screw compressors has also been extended downwards and as a result the screw compressor is now definitely in the range covered by the reciprocating compressor.

As mentioned earlier, the asymmetrical profile has enabled a competitive efficiency to be attained even at relatively low tip speeds. On the other hand, radically new approaches have been necessary in the design development work in order to get an economically defendable product. Scaling down of the layout of the bigger compressor units does not give an adequate cost reduction, despite the fact that the large reciprocating compressors compared with are heavy-duty units intended for industrial operation.

Figure 4 shows an example of a screw compressor which was introduced to the market a good two years ago. As can be seen, it has completely new design concepts. The displacement is 600 m^3/h with 60 Hz drive. The compressor is built up with a vertical shaft and the rotor unit is centrally sited in the compressor housing. The latter serves also as an oil separator. The vertical construction means that space requirements are very small.

In the rotor unit itself, the simplifications in design are far-reaching. The rotors are carried in roller bearings, which has the advantage of enabling the oil pump to be eliminated. Further, the slide valve arrangement and the hydraulic system in the capacity control device have been drastically simplified, with few tolerance chains and parts with simple machining. The number of parts in this compressor is only two thirds of that in a conventional compressor. When the entire compressor unit is studied, i.e. including also the necessary auxiliary equipment like oil system and oil separation, it is of the utmost importance for the systems here too to be simplified. The cost of the auxiliary equipment is more than 50% of the cost of the entire unit.

Considerable efforts have been devoted to the development of effective and small oil separators. The example shown is based on the very latest experience, as evident from the dimensions of the oil separator. Oil cooling is achieved with either water or refrigerant. In the case of refrigerant-cooled oil, oil and liquid refrigerant are mixed in a small mixer outside the compressor and the cooled oil and the flash gas are then injected into the compressor at intermediate pressure. The amount of liquid refrigerant is regulated by an injection valve. The system is far simpler than a system with a water-cooler oil cooler. There will, however, be some increase in the power consumption of the compressor, as the flash gas formed has to be compressed from intermediate pressure to condensing pressure.

The example discussed here illustrates one possible way to simplify and integrate systems into small screw compressors to be able to compete with modern high-speed reciprocating compressors.

Several manufacturers are working on the development of even smaller screw compressors. Here, simplification must be brought still further, as the range concerned is the same as for the medium-sized, high-volume and inexpensive reciprocating compressors.

ZIMMERN-TYPE COMPRESSOR

A completely new compressor type - the ZIMMERN type - was introduced by two companies in Europe during 1975. This compressor type was already available as an air compressor, but has now been developed as a refrigerant compressor. The compressor is volume-controlled with oil injection and should therefore have largely the same properties as the established SRM-screw compressor.

As can be seen from Fig. 5, the compressor has a single 6-thread helically grooved rotor and two identical gearwheels engaging therein. The task of the gearwheels is to seal in the rotor meshes in order to obtain internal compression. The compressor works on the principle of built-in volume ratio and thus has no valves. The suction and discharge ports are located in the rotor housing on both the top and the bottom of the single rotor, meaning that both sides are utilized for compression and that the gas forces on the rotor are counterbalanced. The sizes so far presented in catalogue data have displacements in the range of 800 m^3/h to 2400 m^3h and thus agree with the common screw compressor range.

This compressor has not yet had a chance to prove itself in the field, but one could expect that it will be more talked about in the future.

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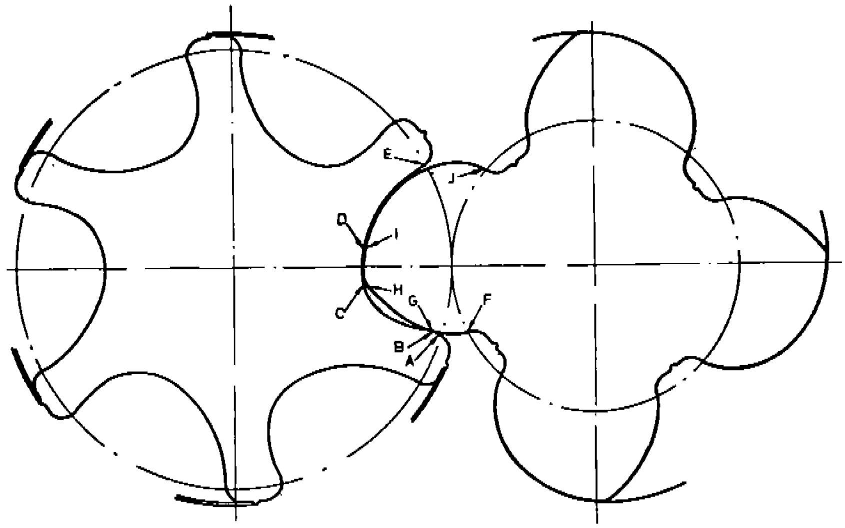

Figure 1 Asymmetrical Rotor Profile

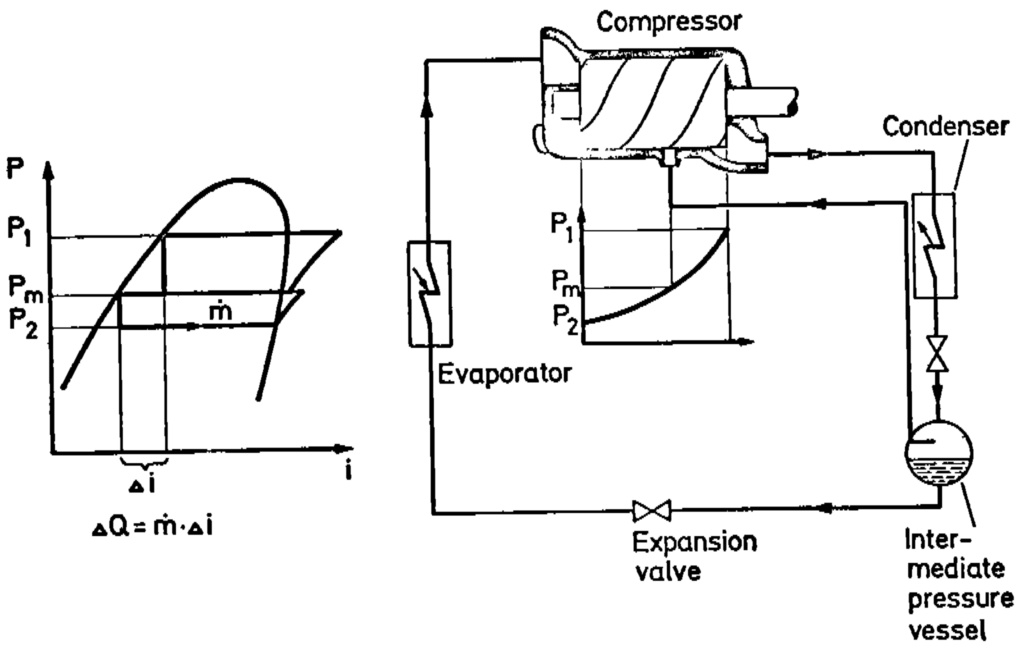

Figure 2 Economizer Arrangement With
Two-Stage Throttling

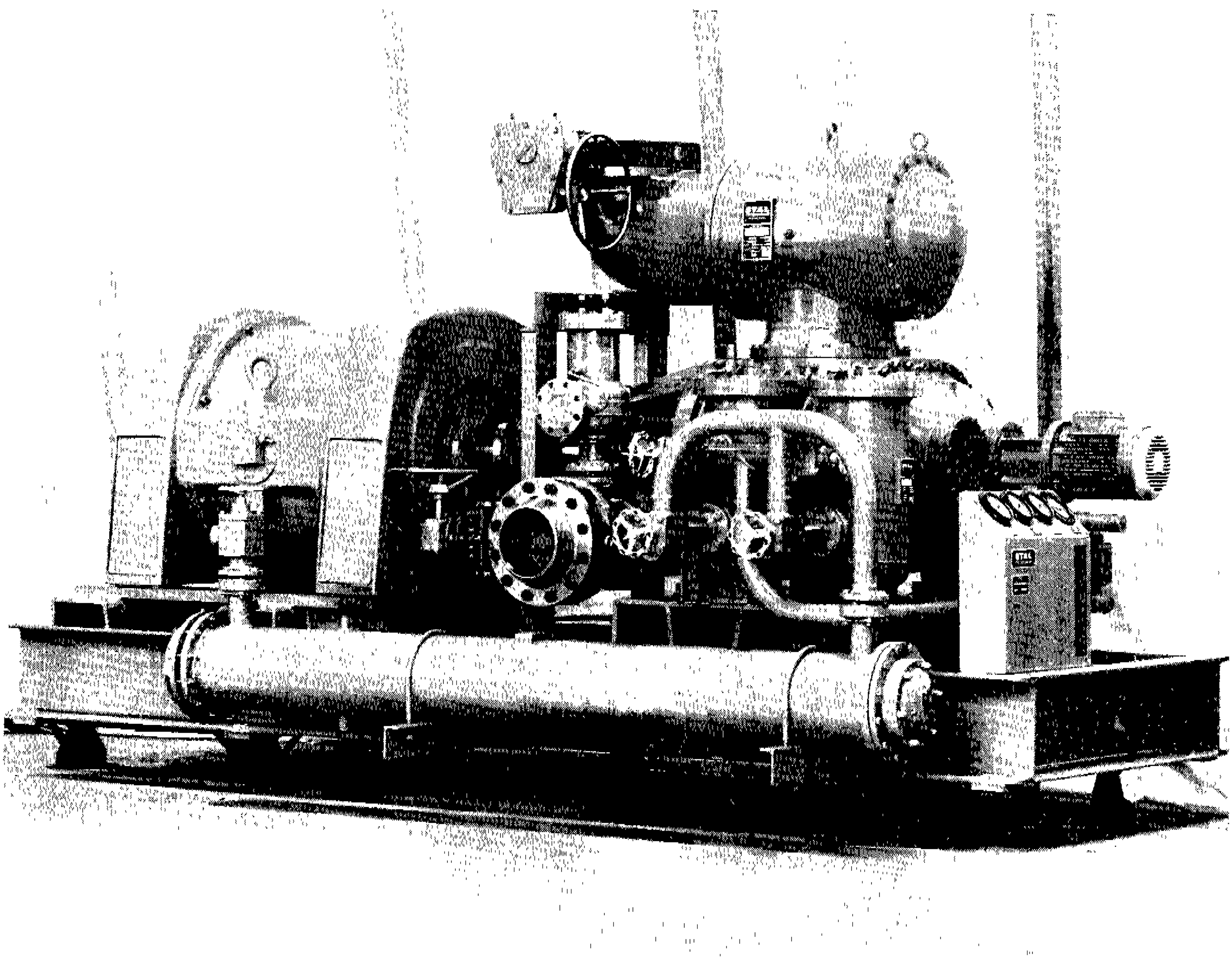

Figure 3 STAL Screw Compressor Type S93 Having 6900 m^3/h Displacement

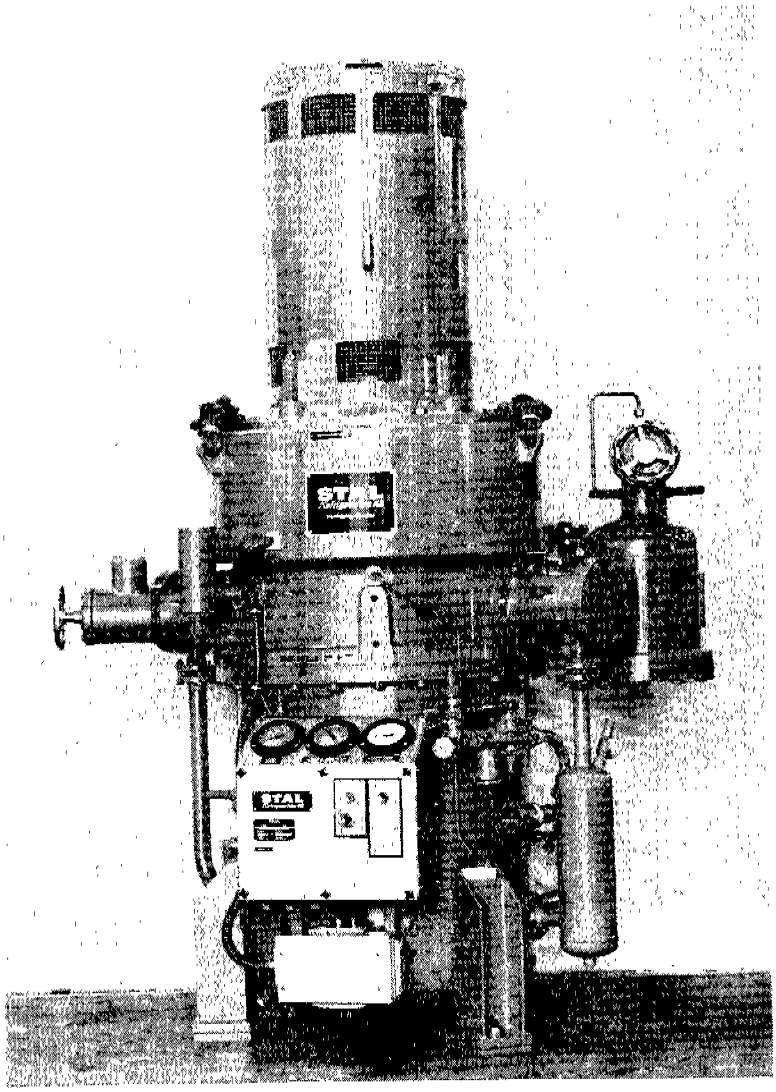

Figure 4 STAL Screw Compressor Type S33 Having 600 m^3/h Displacement

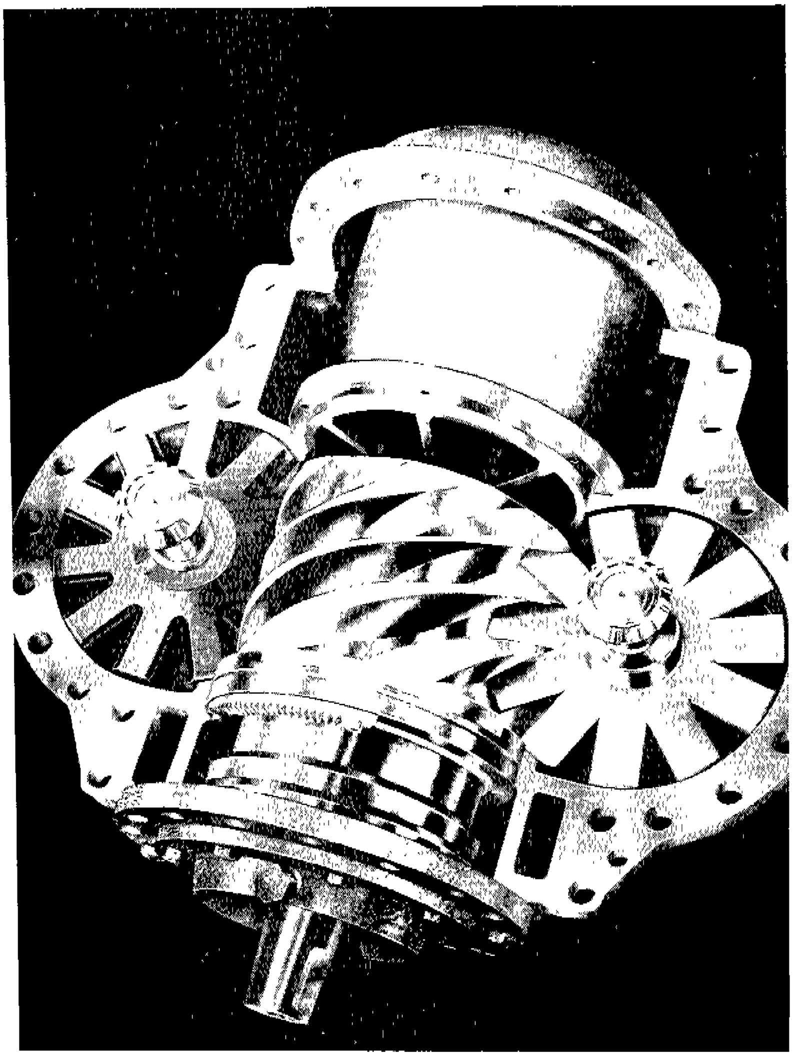

Figure 5 GRASSO "Monoscrew" Compressor