Opportunities for Innovation with Screw Compressors

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Abstract

Screw compressors are at present a widely used means of compressing air, process gas and refrigerants. Computer simulation programs, verified by experimental data and improved by recent advances in mathematical analysis, form a powerful tool for process analysis within these machines and hence for a comprehensive check of innovative suggestions and design optimisation. One consequence of this is that screw rotors lobe profiles have been developed over the past few years which lead to enhanced machine performance. Although efficient operation of screw compressors is mainly dependent on the rotor profile and clearance distribution, other components, such as housing ports, bearings, seals and the lubrication system must be designed to take full advantage of their potential if maximum performance gains are to be achieved. A review of the most recent achievements in application of innovative details into the screw compressor practice is given at the end of this paper.

Introduction

Positive displacement compressors are today a mature product. They are subjected to the highly competitive market, especially in air compression and refrigeration. Orchestrated efforts of a large number of companies driven by market forces resulted in a compact and efficient compressor machine. Every detail counts today. Even virtually a negligible difference will give a small, but distinctive improvement, which may be individual advantage. Although evolutionary improvements may only be expected today to move the compressor performance forward, there is still place left for more or less breakthrough methods and procedures to result in a better product. New rotor generation, optimisation of the rotors and compressor design for a certain compressor duty and a specialized design are tips for innovation, improvement and development in area of positive displacement compressors. Although efficient operation of screw compressors is mainly dependent on the rotor profile and clearance distribution, other components, such as housing ports, bearings, seals and the lubrication system must be designed to take full advantage of their potential if maximum performance gains are to be achieved.

Mathematical modelling, experimental validation, design of critical components, complete machine design, product development, training in machine design, advanced computerized design tools, machine process modelling, 2-D and 3-D Computational Fluid Dynamics, modern experimental technique, computerized data acquisition, rotor and compressor optimisation, all are essential element stages needed for appropriate screw compressor development.

Ever increasing market demand for efficient screw compressors requires today that compressor designs are tailored upon their duty, capacity and manufacturing capability. A suitable procedure for optimisation of screw compressor shape, dimension and operating parameters, which results in the most appropriate design for a given compressor duty, is also presented. It is based on a rack generation algorithm for rotor profile generation combined with a numerical model of the compressor fluid flow and thermodynamic processes. Compressors thus designed achieve higher delivery rates and better efficiencies than those using traditional approaches. It is shown that the optimum rotor profile parameters, compressor speed, oil flow rate and temperature may significantly vary with different gases or vapours in various compressor modes.

Background

Screw compressors are rotary positive displacement machines of simple design with the moving parts comprising only two rotors rotating in four to six bearings. They capable of efficient operation at high speeds over a wide range of operating pressures and flow rates. They are therefore both reliable and compact and consequently they comprise a large portion of all positive displacement compressors now sold and of those currently in operation. Their remarkable success is mainly due to improvements in high accuracy profile milling and grinding which now make it possible to reduce linear tolerances to below 10 μm. This permits rotors to be manufactured with interlobe clearances of 30-50 μm at an economic cost. Internal leakages are thus far less than in earlier machines of this type and as a result, screw compressors are becoming more efficient than other types.

Screw machines are used today for different applications both as compressors and expanders. They operate on a variety of working fluids which may be gases, dry vapours or multi-phase mixtures with phase changes taking place within the machine. They may operate with oil flooding, with other fluids injected during the compression or expansion process, or without any form of internal lubrication. Their geometry may vary depending on the number of lobes in each rotor, the basic rotor profile and the relative proportions of each rotor lobe segment. It follows that there is no universal configuration which would be the best for all applications. Hence, detailed thermodynamic analysis of the compression process and evaluation of the influence of the various design parameters on performance is more important to obtain the best results from these machines than from other types which could be used for the same application. It follows that a set of well defined criteria governed by an optimisation procedure is a prerequisite for achieving the best design for each application. Such guidelines are also essential for the further improvement of existing screw machine designs and broadening their range of uses.

Typically, refrigeration and process gas compressors, which operate for long periods, must be designed to have a high efficiency. In the case of air compressors, especially for mobile applications, efficiency may be less important than size and cost.

For both dry and oil free air compression, screw compressors are almost exclusively used. This is also gradually becoming the case for process gas compression. In the field of refrigeration, the use of reciprocating and vane compressors is decreasing and a dramatic increase in favour of screw machines is expected in the forthcoming years.

Screw compressors in normal commercial usage today have main rotors whose outer diameters vary between 75 mm and 620 mm. These deliver between 0.6 m3/min and 600 m3/min of compressed gas. The normal pressure ratios attained in a single stage are 3.5:1 for dry compressors and up to 15:1 for oil flooded machines. Normal stage pressure differences are up to 15 bars, but maximum values sometimes exceed 40 bars. Typically, for oil flooded air compression applications, the volumetric efficiency of these machines now exceeds 90 % while specific power inputs, which are both size and performance dependent, have been reduced to values which were regarded as unattainable only a few years ago.

Screw Compressor Practice

Svenska Rotor Maskinen (SRM), a Swedish company, was the pioneer and they are still being leaders in the field of screw compressor design and development practice. Other companies, such as Compair in the U.K, Atlas-Copco in Belgium, Ingersoll-Rand and Denver Gardner in the USA and GHH in Germany are the main air compressor manufacturers. York, Trane and Carrier in the USA lead in refrigeration and air conditioning applications. Japanese screw compressor manufacturers, such as Hitachi, Mycom and Kobe-Steel are also well known. In recent years there have been a number of mergers and buy outs of smaller companies by these larger ones. New markets in China, India and other developing countries in the Middle and Far East have led to new screw compressor companies being founded there as well as factories being built there by the major manufacturers. Although they do not manufacture compressors, Holroyd, a British company is the world's largest screw rotor manufacturer. They are also world leaders in tool design and machine tool production for the manufacture of screw compressor rotors.

Despite the rapid growth in screw compressor usage, public knowledge of the scientific basis of their design is still limited. Several classical screw compressor textbooks were published in Russia in the early nineteen sixties. *Sakun 1960* gives a full analysis on how to generate circular, elliptic and cycloidal rotor profiles as well as a Russian asymmetric profile later named SKBK. The method of profile generation in this book was based on an envelope approach. *Amosov et al 1977* in his handook on screw compressors presents this theory and, in addition, makes a contribution to the field of rotor tool profile generation and gives a reproducible presentation of how to generate the SRM asymmetric profile almost at time when it was patented, as well as the classical Lysholm profile. There are two German textbooks. *Rinder, 1979* used a profile generation method based on gear theory to reconstruct the SRM asymmetric profile. *Konka, 1988* published some detailed engineering aspects of screw compressors. Only recently have any textbooks on screw compressors been published in English. *O'Neil, 1993* was on industrial compressors and *Arbon, 1994* was on rotary twin shaft compressors. Very recently, *Xing, 2000* publishes his comprehensive textbook on screw compressors in Chinese.

A few compressor manufacturers' handbooks on screw compressors and a number of brochures give useful information, but they are either classified or very limited accessibility. Some, like the SRM Data Book, although cited in screw compressor literature, are available only to SRM licensees.

Literally thousands of patents have been awarded on screw compressors in the past thirty years and SRM alone has 750. They cover various aspects of screw compressor design but are mainly for rotor profiles. SRM patents *Nilson, 1952* for symmetric, *Shibbie, 1979*, for asymmetric and *Astberg, 1982* for the "D" and *Ohman, 1999* for the “G” profiles are classical examples of state of the art screw compressor profile generation. Other succesfull profiling patents are those of Atlas-Copco and Kaeser, *Bammert, 1979*, Compair, *Hough et al, 1984*, Gardner Denver, *Edstroem, 1974*, Hitachi, *Kasuya et al, 1983* and Ingersoll-Rand, *Bowman, 1983*. In recent years, several highly successful patents such as *Lee, 1988* and *Chia-Hsing C, 1995* have been granted to relatively small companies, Fu Sheng and Hanbel respectively. A fresh approach to profile generation based on the use of a rack for the primary curves was published in *Rinder, 1987* and *Stosic, 1996*.

Although all patented profiles were generated by well defined procedures, so little of the methods on which they were based was published that it was very difficult to reproduce them. Thus details of how to derive the circular symmetric profile, were not published until 26 years after it was patented. Similarly, a derivation of the SRM asymmetric profile based on classical gear criteria, was not published until 9 years after the patent appeared. It is even more interesting to note that one SRM licencee funded a university research programme to obtain an analytical derivation of the SRM "D" profile in 1995, after 13 years of its manufacturing.

Most of the well known characteristics of screw compressors, such as oil flooding, making the suction and discharge ports follow the rotor tip helices, axial force compensation, unloading, the slide valve and the economizer port, were also patented, mainly by SRM. However other companies were equally keen to file patents. It appears that, in the field of screw compressors, patent experts are as important as engineers.

There are surprisingly few journal publications on screw compressors in the technical literature. Almost exceptions are *Hanjalic and Stosic and Stosic et al, 1997*. In recent years, journal publication of screw compressor material has been encouraged by the International Institution of Refrigeration *Stosic et al 1992* and *Fujiwara and Osada, 1995*, the IMechE *Tang et al, 1994* and *Fleming et al 1999*. This led to more information on screw compressors in journals during recent years than in all previous years together.

There are three compressor conferences which deal partly or exclusively with screw compressors. These are: the Purdue University compressor engineering and refrigeration conferences in the U.S.A, the IMechE conferences on industrial compressors in England and the "VDI Schraubenkompressoren Tagung" in Germany. Despite the wealth of screw compressor papers which these contain, very few give useful information on rotor profiling procedures and compressor design.

Purdue papers are frequently cited, examples of such a variety of papers are: *Sing and Shwartz, 1990 , Stosic et al, 2000 and Kovacevic et al 2002*. The Dortmund proceedings contain some interesting screw compressor papers. *Rinder, 1984* presents a rack method of rotor profile generation, based on classical gearing procedure, which is fully reproducible. Unfortunately, although soundly conceived, profiles produced by this method were not commercially successfull because of their poor tightness. *Sauls, 1998* gives more details on profiling and manufacturing control. *Kauder and Harling, 1994* is a typical example of a successful university research applied to solve real engineering problems. Although relatively infrequent, the London compressor conferences included some interesting papers, such as *Venumadhav et al and McCreath et al, 2001*.

Reference textbooks on gears give a useful background to screw rotor profiling. However all of them are limited to the classical gear conjugate action condition without a direct connection to the screw compressor profiling. *Litvin, 1994* is an exception to this practice which contains gear theory directly applicable to screw compressor profiling.

Screw Compressor Developments

The efficient operation of screw compressors is mainly dependent on proper rotor design. An additional and important requirement for the successful design of all types of compressor is the ability to predict accurately the effects on performance of the change in any design parameter such as clearance, rotor profile shape, oil or fluid injection position and rate, rotor diameter and proportions and speed.

Now that tight clearances are achievable, internal compressor leakage rates become small. Hence, further improvements are only possible by the introduction of more refined design principles.

The main requirement is to improve the rotor profiles so that the internal flow area through the compressor is maximised while the leakage path is minimised and internal friction due to relative motion between the contacting rotor surfaces is made as small as possible.

Although rotor profiling procedures may appear to be fully defined, substantial improvements are still possible. The most promising approach for this seems to be through rack profile generation which gives stronger but lighter rotors with higher throughput and lower contact stress. The latter enables fluids with lower viscosity than oil to be used for lubrication.

Rotor housings with better shaped ports can be designed using multivariable optimization techniques. Flow losses may thereby be reduced permitting higher rotor speeds and hence more effective compressors.

Recent improvements in bearing design make process fluid lubrication possible in some applications. Also seals are more efficient today. All these developments can be utilised to produce more efficient, lighter and cheaper screw compressors.

Rotor Profiles

The normal procedure used to generate rotor profiles is to create primary profile curves on one rotor and, by use of an appropriate conjugate criterion, a corresponding secondary profile curve on the other. Any curve can be used as the primary one, but traditionally a circle is the most common. All circles with their centres on the pitch circle generate a similar circle on the other rotor. This is also true if the circle centres are at the rotor axes. Circles with centres not on the pitch circle and other curves, like ellipses, parabolae and hyperbolae require more elaborate curves to be generated, on the other rotor which are described as trochoids. Similarly, points located on one rotor will cut epi- or hypocycloids on the other rotor. For decades, the skill required to produce rotors was limited to the ability to choose a primary arc which will permit the derivation of an appropriate secondary profile.

The symmetric circular profile consist of circles only. Apart from the use of pitch circle centred circles, *Lysholm, 1967* introduced a set of cycloids on the high pressure side to form the first asymmetric screw rotor profile. The SRM, asymmetric profile employs an eccentric circle on the low pressure side of the gate rotor. Later the SKBK profile used the same curve for the main rotor. In both cases the curves evolved analytically from them were epi- or hypocycloids. The SRM "D" profile consists exclusively of circles, almost all of them eccentrically positioned on the main or gate rotor. All patents which followed specify primary curves on one rotor and secondary, generated curves on the other rotor. All are probably derived from classical gear contact theory or a similar alternative criterion. More recently, circles have gradually been replaced by other curves such as ellipsae in the Fu Sheng profiles parabolae in the Compair and Hitachi profiles, and hyperbolae in the 'hyper' profile. Replacing the circle and parabola by a hyperbola in the latest profile, seems to give the best ratio of rotor displacement to sealing line length.

Another method of rotor profile generation is to consider imaginary, or 'nonphysical' rotors. Since all gearing equations are independent of the coordinate system in which they are expressed, it is possible to define primary arcs as given curves using a coordinate system which is independent of both rotors. By this means, in many cases the defining equations may be simplified. Also, the use of one coordinate system to define all the curves, simplifies the design process. Typically, the template is specified in a rotor independent coordinate system. This is valid for a rotor of infinite radius, which is a rack. From this, a secondary arc on some of the rotors is obtained by a procedure, which is called 'rack generation'. The first patent on rack generation was patented earlier, but lacks practicality. Only, *Rinder, 1983* and more recently *Stosic, 1996* give a good basis for the rack generation of a rotor profile.

For a screw compressor to be efficient, the rotor profile must form a large flow cross section area, a short sealing line and a small blow-hole area. The larger the cross section area, the higher the flow rate for the same rotor sizes and speeds. Shorter sealing lines and a smaller blow-hole reduce leakages. Higher flow and smaller leakage rates both increase the compressor volumetric efficiency, which is the rate of flow delivered as a fraction of the sum of the flow plus leakages. This in turn increases the adiabatic efficiency because less power is wasted in the compression of gas which is recirculated internally.

The optimum choice between blow hole and flow areas depends on the compressor duty since, for low pressure differences, the leakage rate will be relatively small and hence the gains achieved by a large cross section area may outweigh the losses associated with a larger blow-hole. Similar considerations determine the best choice for the number of lobes since fewer lobes imply greater flow area but increased pressure difference between them.

As precise manufacture permits rotor clearances to be reduced, despite oil flooding, the likeliehood of direct rotor contact is increased. Hard rotor contact leads to deformation of the gate rotor, increased contact forces and ultimately rotor seizure. Hence the profile should be designed so that the risk of seizure is minimised.

The search for new profiles has been both stimulated and facilitated by recent advances in mathematical modelling and computer simulation. These analytical methods may be combined to form a powerful tool for process analysis and optimisation and thereby eliminate the earlier approach of intuitive changes, verified by tedious trial and error testing. As a result, this approach to the optimum design of screw rotor lobe profiles has substantially evolved over the past few years and is likely to lead to further improvements in machine performance in the near future. However, the geometry and processes involved are so complex that numerous approximations are required for successful modelling. Consequently, the computer models and numerical codes reported in the open literature often differ in their approach and in the mathematical level at which various phenomena are modelled. A lack of comparative experimental verification still hinders a comprehensive validation of the various modelling concepts. In spite of this, computer modelling and optimization are steadily gaining in credibility and are increasingly employed for design improvement.

Fig. 1 shows a comparison of several pairs of screw compressor rotors. Each pair is labeled by a number in brackets which denotes its source.

Fig. 1 Review of common screw compressor rotor profiles

The first group gives rotors with 4 lobes on the main and 6 lobes on the gate rotor. This rotor configuration is the most universally acceptable for almost every application. The review starts with the symmetric rotor profile and continues with the SRM asymmetric profile, which historically was the most successful. They are presented on the top. The next one is the SRM "D" profile. The ‘Cyclon’ profile in 4/5 configuration shows how attempts were made to produce more displacement from the same compressor size.

The largest group of rotors presented has a 5/6 configuration. This is becoming the most popular rotor combination because it permits a good compromise between large displacement and large discharge ports with favourable load characteristics in small rotor sizes. These rotors are equally successful for air compression and refrigeration and air-conditioning applications. The group starts with the Lysholm asymmetric profile, followed by the SKBK profile, Fu Sheng, and "Hyper" profiles. All of the profiles presented up to now are 'rotor-generated' profiles and the main difference between them is in the leading lobe which is mainly an eccentric circle followed by a line, ellipse and hyperbola respectively. The hyperbola appeared to be the best possible geometrical solution for that purpose.

Finally, rack generated, Rinder's and 'N' rotors are presented. The selection and distribution of primary curves on the rack which was used to create these rotors, gives a larger cross section area with stronger gate rotor lobes than any other known screw compressor rotor pair. The last of these shows the best delivery for its size. it also has two additional features. Firstly, the rotors retain a seal over the entire contact length while maintaining a small blow-hole. Secondly, two contact bands in the proximity of the pitch circles are straight lines on the rack which form involutes on the rotors. Hence the relative motion between the rotors is close to pure rolling which is the best possible. The last rotors are elaborated in more detail in *Stosic and Hanjalic 1997*, where a sequence of generating rack curves is given.

Compressor Design

Screw compressor design has gradually evolved through the screw compressor history and the trend is to get as small as possible machine to meet the required performance. This means that rotor tip speeds are as high as possible within the limits imposed by efficiency requirements. Regular practice is to use rolling element bearings wherever possible because these permit smaller clearances than journal bearings. Similarly, the ports are made as large as possible to minimize suction and discharge gas speeds. These features lead to great similarity in all designs for any specified application. Although advanced rotor profiles are a necessary condition for a screw compressor to be efficient, all other components must be designed to minimise losses if maximum improvements are to be achieved. Thus the rotor to housing clearances must be properly selected, especially at the high pressure end. This in turn requires either expensive bearings with small clearances or cheaper bearings with their clearances reduced to an acceptable value by preloading.

A screw compressor, especially of the oil flooded type, operates with high pressure differences. These create large axial and radial bearing forces. Rolling element bearings are normally chosen for small and medium screw compressors and these must be carefully selected to obtain a satisfactory design since, inter alia, the distance between the rotor centre lines is in part determined by them. It must be added that recent advances in the development of advanced low friction rolling element bearings greatly contributes to this choice, as presented by *Meyers, 1997*. Usually two bearings are fitted at the high pressure end of the rotor shafts in order to absorb the radial and axial loads separately.

The contact force between the rotors is determined by the torque transferred between them and is very significant when they make direct contact. It is relatively small when the compressor drive is through the main rotor. However, when the drive is through the gate rotor, the contact force is substantially larger and, as far as possible, this arrangement should be avoided.

The oil, which is injected into the compressor for flooding is also used for bearing lubrication but, to minimise friction losses, the bearing feed and return system is separate. The position in the compressor chamber where the oil is injected is set at the point where thermodynamic calculations show the gas and oil inlet temperatures to coincide. It is defined on the rotor helix with the injection hole located so that the oil enters tangentially in line with the gate rotor tip in order to recover as much as possible of the oil kinetic energy.

To minimise flow losses in the suction and discharge ports, the following features must be included. The suction port should be positioned in the housing to let the gas enter with the fewest possible bends and the gas approach velocity kept low by making the flow area as large as possible. The discharge port size is first determined by estimating the built-in-volume ratio required for optimum thermodynamic performance. It is then increased in order to reduce the exit gas velocity and hence obtain the minimum combination of internal and discharge flow losses.

The casing should be carefully dimensioned to minimize its weight, containing reinforcing bars across the suction port to improve its rigidity at higher pressures.

Recent Examples of the Innovative Screw Compressor Design

Only publicly available literature resources are used in this part of the paper to review the recent screw compressor designs. Therefore, due to corporate limitations, not all the recent innovative designs are presented here.

Rotor Retrofit for Efficient Screw Compressors

Stosic et al, 2000 pulished a paper which described a retrofit of 'N' rotors into an existing family of oil-flooded compressors. Since the market for oil-flooded screw air compressors is highly competitive, new designs are continually being introduced which are more efficient and cost effective than their predecessors. However, because of the high cost of development of new machines, manufacturers seek to maintain their existing designs for as long as possible. Closer study of many of the older designs has shown that in the majority of cases, all that is required to bring them up to date is to change the rotor profile to one of more up to date type.

Fig. 2 Rotor retrofit for efficient oil-flooded screw compressors, Stos*ic, 2000*

That paper describes the design of a family of highly efficient screw compressor rotors based on rack generated profiles which were used to replace standard asymmetric 4/6 rotors in sizes between 102 and 204 mm and thereby maximise the compressor efficiencies. The old and new rotors are compared in Fig. 2. The new rotors were developed with the aid of a software package which takes into account almost every aspect of thermodynamic and geometric modelling. All the definitive dimensions of the original rotors, such as centre distance, outer and root diameters, rotor length and average clearance were retained. However the favourable features of the new 'N' rotors permit their clearances to be reduced, if required, with no increase in manufacturing cost. All the other compressor components remained unchanged and no changes of any kind were required in the housing and bearing design.

Experimental tests showed that in all cases the introduction of the new rotors increased the compressor displacement by more then 6.5 % and raised the adiabatic efficiency by not less than 2.5 %. This led to an increased power input requirement of 2.6 %. However, the existing drive motors were sufficient to meet this additional need. The high efficiencies obtained on test confirm their validity. It should be noted though that attention to the improvement of every detail of design such as the ports, the oil injection system and the bearings would all additionally contribute to the potential for improvement offered by the novel high throughput involute rotors.

Screw Compressor for Delivery of Dry Air

McCreath et al, 2001 published a paper which describes two high efficiency oil-free screw compressors designed for dry air delivery. Their design is based on the rack generated 3/5 rotor profiles, presented in Fig. 3.

Fig. 3 Rotor profile for delivery of dry air, *McCreath, 2001*

The optimum rotor size and speed, together with the shape and position of the suction and discharge ports, were determined by mathematical modelling, taking full account of the limitations imposed by bearing and seal selection required to maximise endurance and reliability. Together, the two machines cover the discharge range of 350-1000 m^3/h. Prototype tests showed that both the volumetric and adiabatic efficiencies of these machines were higher than the published values of any equivalent compressors currently manufactured. This confirmed the advantages of both the rotor profile and the design procedure.

Fig. 4 Compressor for delivery of dry air, McCreath, 2001

A family of two compressors has been developed for dry air delivery by Drum-International to cover air delivery in the range of 300-1000 m^3/min. One of the compressors is presented in Figs 4 and 5. Extensive testing has shown that their performance is superior to that of all known compressors of similar application and similar size.

Fig. 5 Layout of the compressor for delivery of dry air, *McCreath, 2001*

Design of a Family of Oil-Flooded Compressors

The design of a family of efficient oil-flooded twin screw air compressors is presented by *Venumadhav et al, 2001*. It was carried out using a software package, which included almost every aspect of the rotor profiling and compressor thermodynamic and geometric modelling with the capacity to transmit calculated output directly into a CAD drawing system. The package was used to determine the optimum rotor profile, size and speed, the volume ratio and the shape and position of the suction and discharge and oil ports. Rack generated rotors of the 4/5 configuration were applied to 5 screw compressors of 73, 102, 159, 225 and 284 mm rotor diameter which covered deliveries between 0.6 to 60 m^3/min. They are presented in Fig. 6. Other modern concepts, such as late closing of the suction port and early exposure of the discharge port were included in the design, together with improved bearing and seal specifications, to maximise the compressor efficiency, durability and reliability.

Fig. 6 Screw rotors for the oil-flooded compressor family, *Venumadhav, 2001*

The compressor family is being gradually introduced by manufacturing prototypes, pre-production compressors and finally, production units. The compressor prototypes tests showed that at delivery pressures between 5 and 13 bar gauge the volumetric and adiabatic efficiencies of the prototypes were very high when compared to published data on the best compressors currently manufactured. The predicted advantages of both the new rotor profile and the design procedure were thereby confirmed.

Fig. 7 The smallest compressor of the compressor family, *Venumadhav, 2001*

Morvover, it should be noted that, apart from the improvements made due to the advanced rotor profile and computer optimisation, attention was paid to every detail of the designs such as the ports, the oil injection system and the bearings in order to derive the full benefit possible from this approach. The smallest compressor of the family, 74 mm outer diameter is presented in Fig. 7.

The high efficiencies obtained on test, as well as the reduced size and weight of these machines, compared to the models which they have been designed to replace, all confirm the validity of this approach.

Expressor

Brasz et al, 2001 published an introductory report on a novel means of replacing the throttling process in vapour compression systems. Power is recovered from the two-phase expansion process and used directly to recompress a portion of the vapour formed during the expansion. Both the expansion and recompression processes are carried out in a twin screw machine with only one pair of rotors. The principle of the simultaneous expansion and compression on the same pair of screw rotors is presented in Fig. 8. One of features which the rotors must satisfy for that purpose is their sealing on both sides, which is not a necessary requirement for the compressor rotors. Such the rotors are presented in Fig. 9.

Fig. 8 The expressor principle, Brasz, *2000*

These rotate, without the need for timing gear, in a process-lubricated, totally sealed unit which the authors have called an "expressor". First test results indicate that the overall expansion-compression efficiency of the expressor is of the order of 55%. This corresponds roughly to 70% expansion efficiency and 80% compression efficiency. The simplicity of the expressor design, together with its promising performance give clear indications that it should be a highly cost effective component in large commercial chiller systems.

Fig. 9 The expressor two-side sealed rotors, Brasz, *2000*

The single-pair screw rotor expressor is derived from a continuing investigation into the potential of twin screw machines to expand two-phase fluids efficiently which has been conducted over a period of eighteen years. The resulting machine, comprising a high efficiency sealed unit with only one pair of rotors, process lubricated and without the need for any seals or timing gear is presented in Fig. 10. It is potentially a highly cost effective means of power recovery from the throttling process in vapour compression cycles. Further studies are needed to determine the best built in volume ratios for the expansion and compression processes and to improve liquid-vapour separation during low pressure discharge.

Fig. 10 The expressor, Brasz, ***2000***

Balanced Rotors

Stosic et al, 2002 responded to recent interest in natural refrigerants, which has created a new impetus for studies of CO_2 as a working fluid in vapour compression systems for refrigeration and air conditioning. Two major drawbacks to its use are the very high pressure differences required across the compressor and the large efficiency losses associated with the throttling process. To overcome the throttle losses, a number of proposals have been made for various types of positive displacement machine, mainly of the vane type, which combine compression with some recovery of work from the expansion process. However, how well they operate with high pressure differences across the vanes has not been confirmed.

For many years, the authors have been investigating the use of twin screw machines to fulfil both the expansion and compression processes when using more conventional halocarbon refrigerants. These have many potential advantages over other types of positive displacement machine. Unfortunately, when applied to CO_2 the huge bearing forces associated with the pressure distribution within them have hitherto made them appear to be unsuitable.

In that paper, it is shown how the rotor forces created by the compression and expansion processes can be partially balanced in order to eliminate the axial forces and reduce the radial bearing forces. The disadvantages of twin screw compressors for such high pressure applications are thereby reduced. The balanced rotor concept, similar to the principle presented in Figs 8 and 9 is also applicable to vapour compression systems using more conventional refrigerants and even for high pressure gas compression.

The pressure range over which a single stage screw compressor may be readily made to operate is limited to 65 bar and in special cases up to 85 bar. At high pressures, axial rotor loads have been reasonably successfully reduced by the inclusion of balancing pistons or their equivalent. However, high radial loads have up to now still been unavoidable. These must be reduced for use in high pressure CO_2 applications to avoid excessive rotor deflection and premature bearing wear. The combined compressor expander, as described in this paper fully balances the axial loads and reduces the radial bearing loads. Design problems associated with high bearing loads in screw compressors for CO_2 systems are thereby reduced. The concept is presented in Fig. 11.

Fig. 11 Compressor – Expander with balanced rotors, Stosic, *2002*

Apart from reducing pressure loads, the simultaneous compression and expansion processes reduce throttling losses in CO_2 vapour compression system plants significantly.

Although, the balanced rotor configuration described in this paper was conceived specifically for high pressure CO_2 refrigeration systems, it may also be used with more conventional refrigerants, or indeed, wherever there is a need for combined expansion and compression processes. For a small penalty in overall efficiency, it may also be used in non refrigeration applications to permit high pressure compression in a single stage system with reduced rotor loads, even where there is no system requirement to recover power from the expansion process.

Conclusions

With the new millenium, the screw compressor is becoming a mature product. As a result of frantic efforts by a number of companies, driven by market forces, it has become a compact, efficient and reliable machine. Every detail counts today. Even small advances in any feature will give distinctive improvements which may be used to gain commercial advantage. Hence, despite its now established role in industry, efforts continue to make advances in every aspect of its design, manufacture and mode of operation.

Although improvements so gained are most likely to be evolutionary, there is still scope for revolutionary methods or procedures to achieve a better product. The most promising development appears to be rack profile generation to produce stronger but lighter rotors with higher displacement and lower contact stress.

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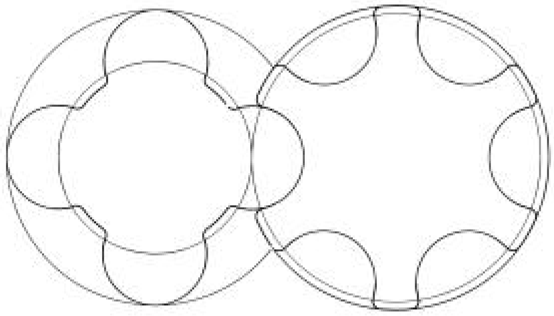

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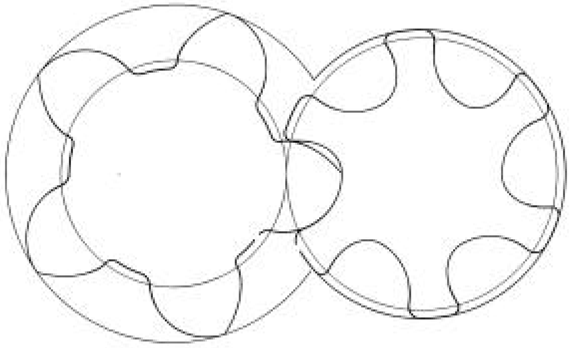
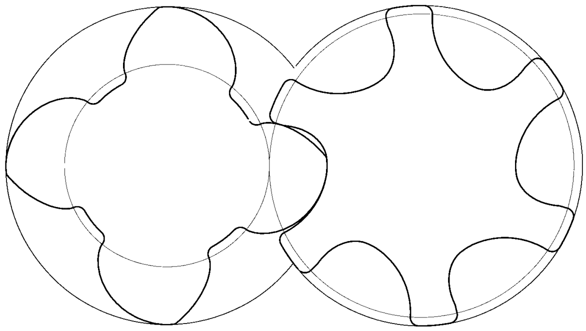
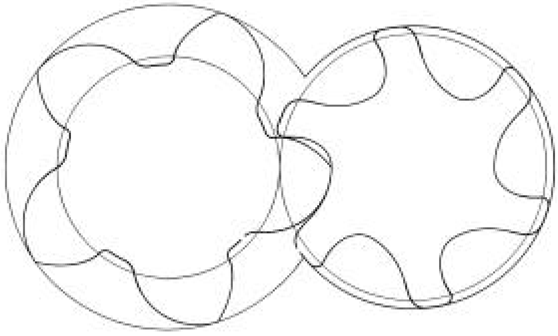


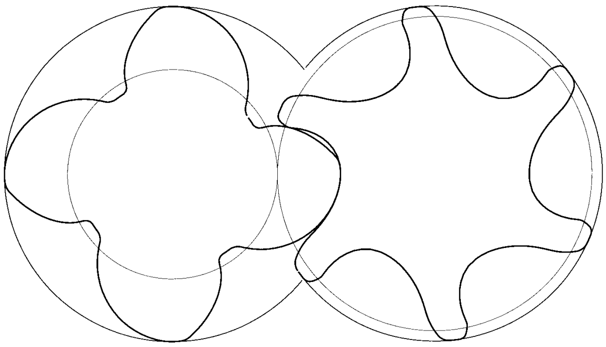
SRM 'A' profile, *Shibbie 1979*



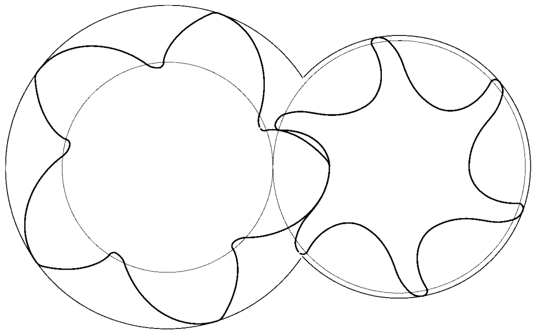
SKBK profile, *Amosov, 1977*



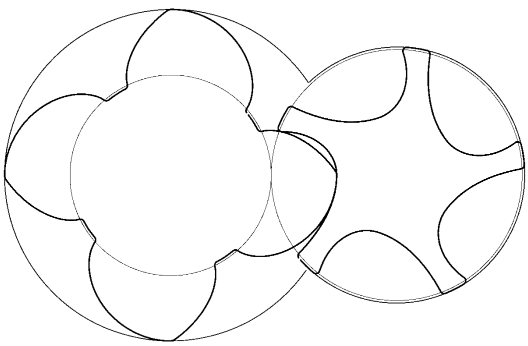
SRM "D" profile, *Astberg, 1982*



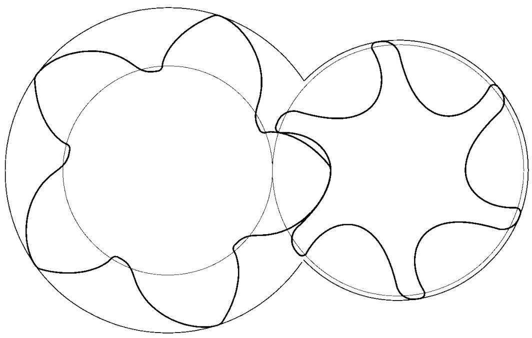
FuSheng profile, *Lee 1988*



Compair profile, *Hough et al, 1984*



Hyper profile, *Chia Hsing, 1995*



Rinder's profile, *Rinder, 1987* **'N' profile,** *Stosic, 1996*

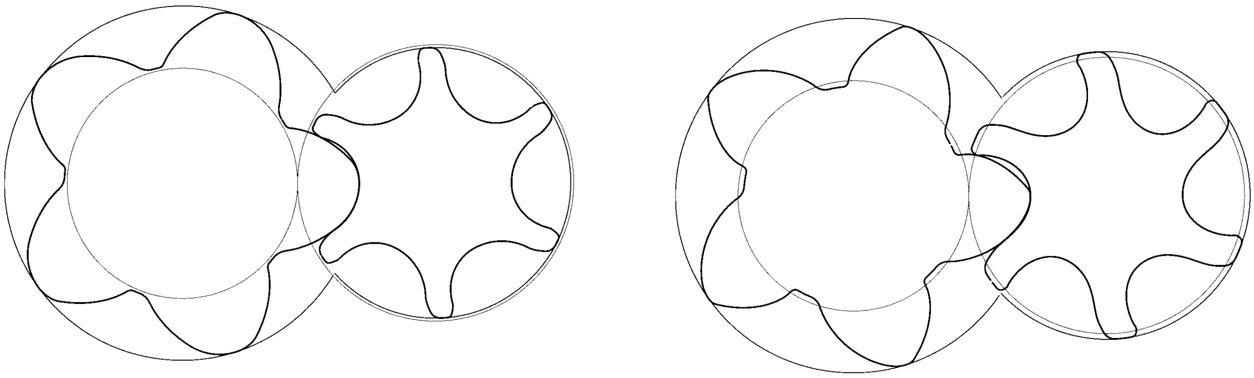


Fig. 1 Review of common screw compressor rotor profiles

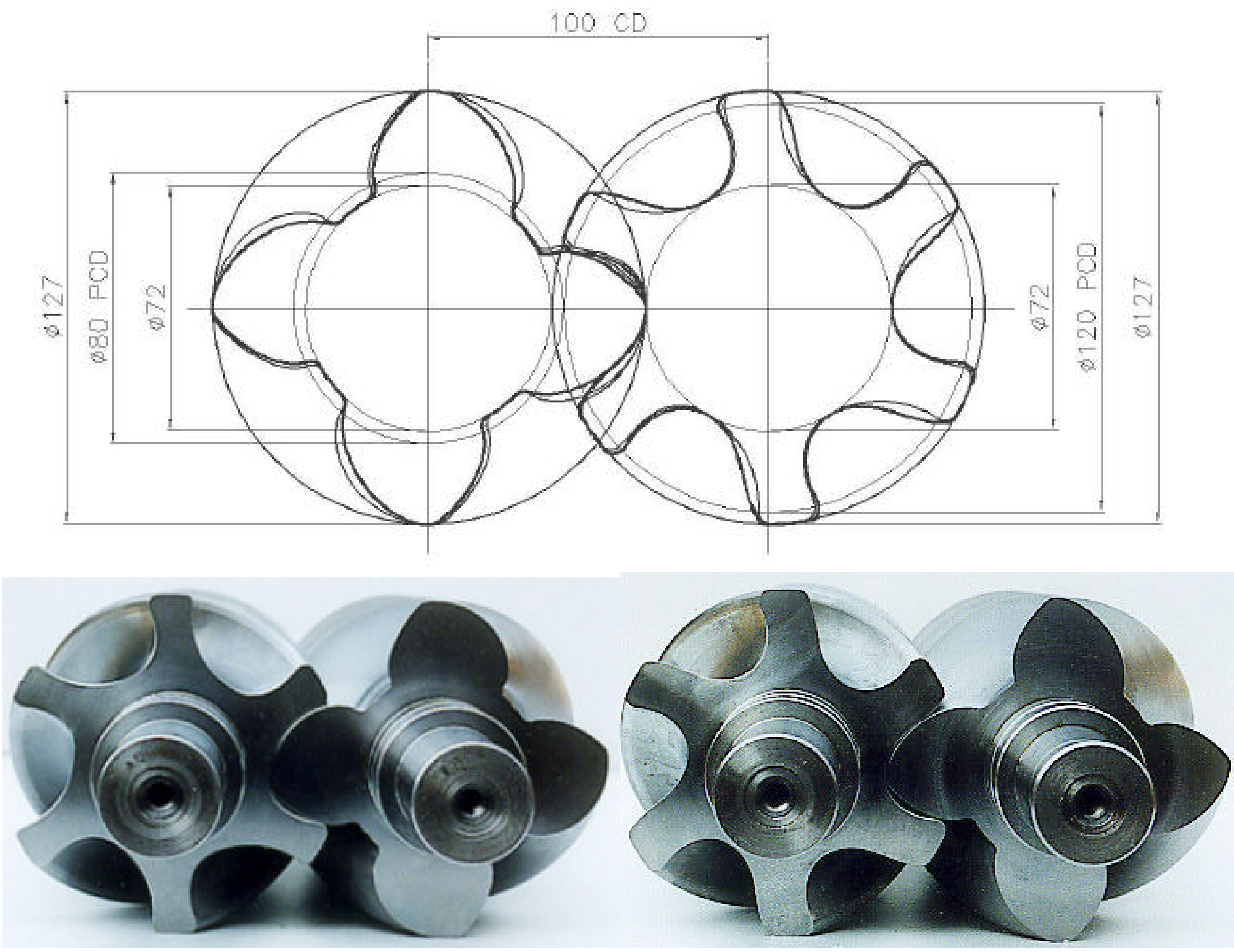


Fig. 2 Rotor retrofit for efficient oil-flooded screw compressors, Stos*ic, 2000*

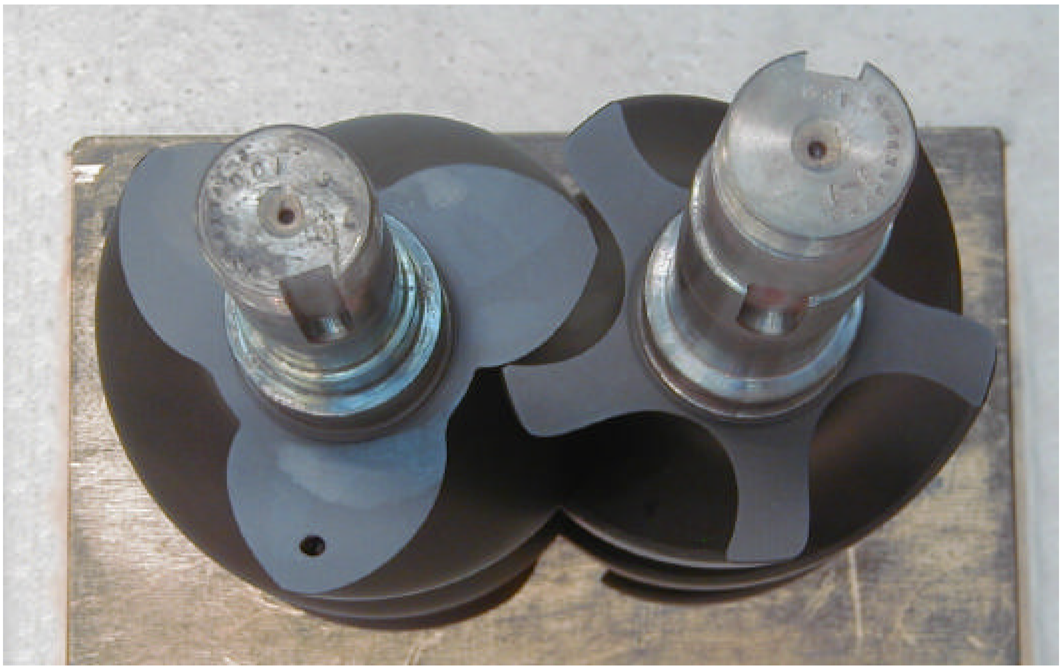


Fig. 3 Rotor profile for delivery of dry air, *McCreath, 2001*

Fig. 4 Compressor for delivery of dry air, *McCreath, 2001*

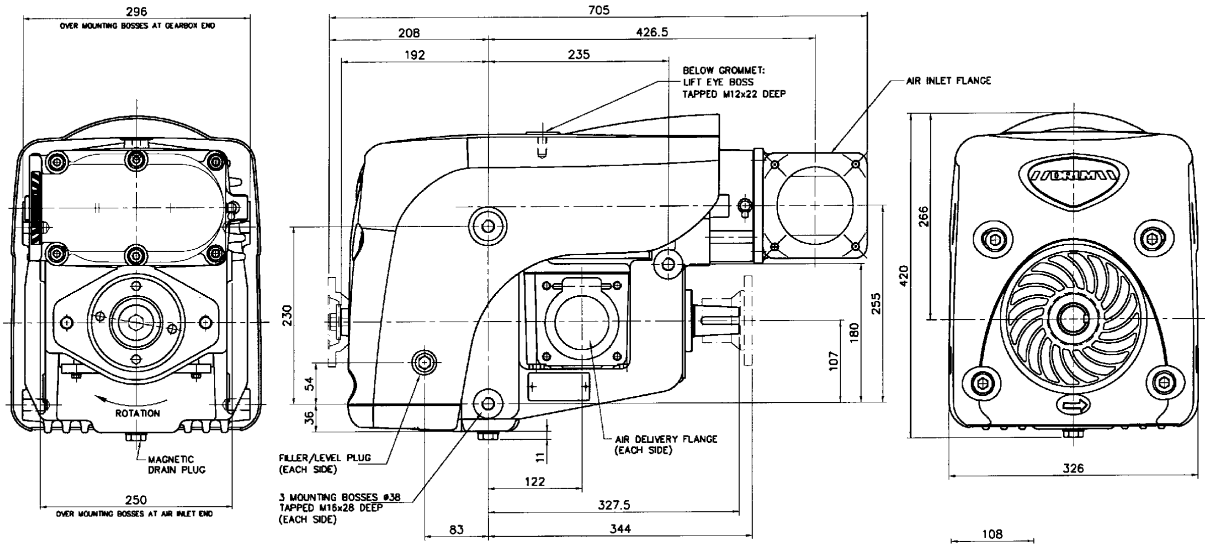


Fig. 5 Layout of the compressor for delivery of dry air, *McCreath, 2001*

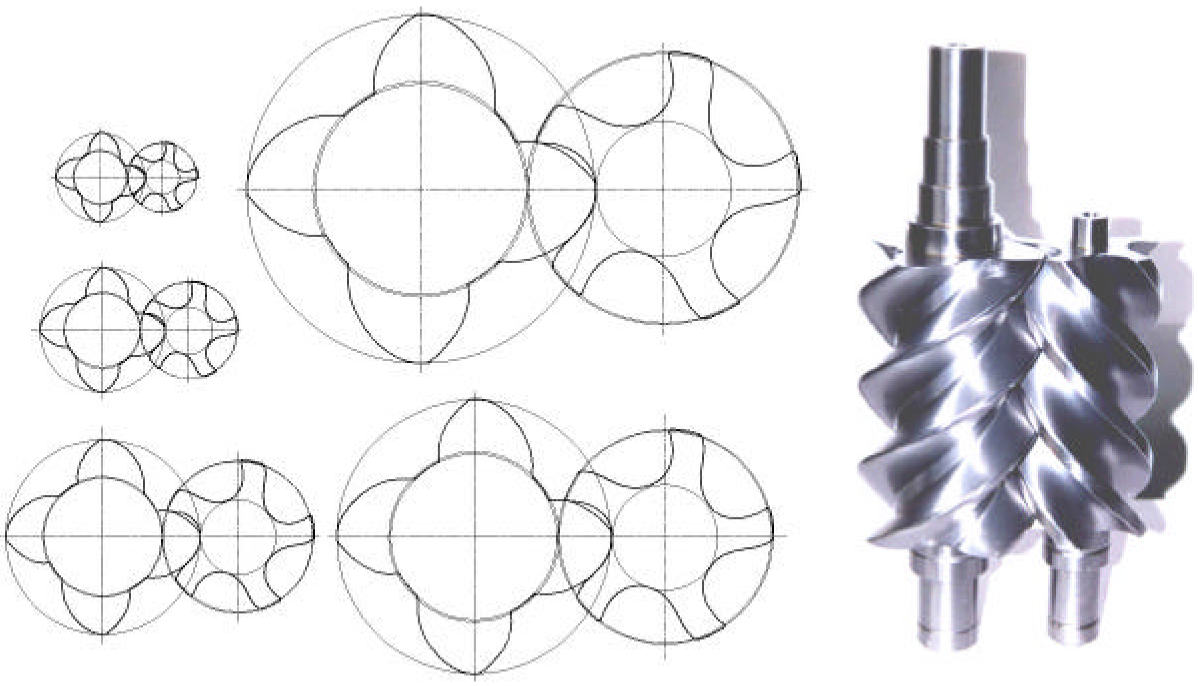


Fig. 6 Screw rotors for the oil-flooded compressor family*, Venumadhav, 2001*

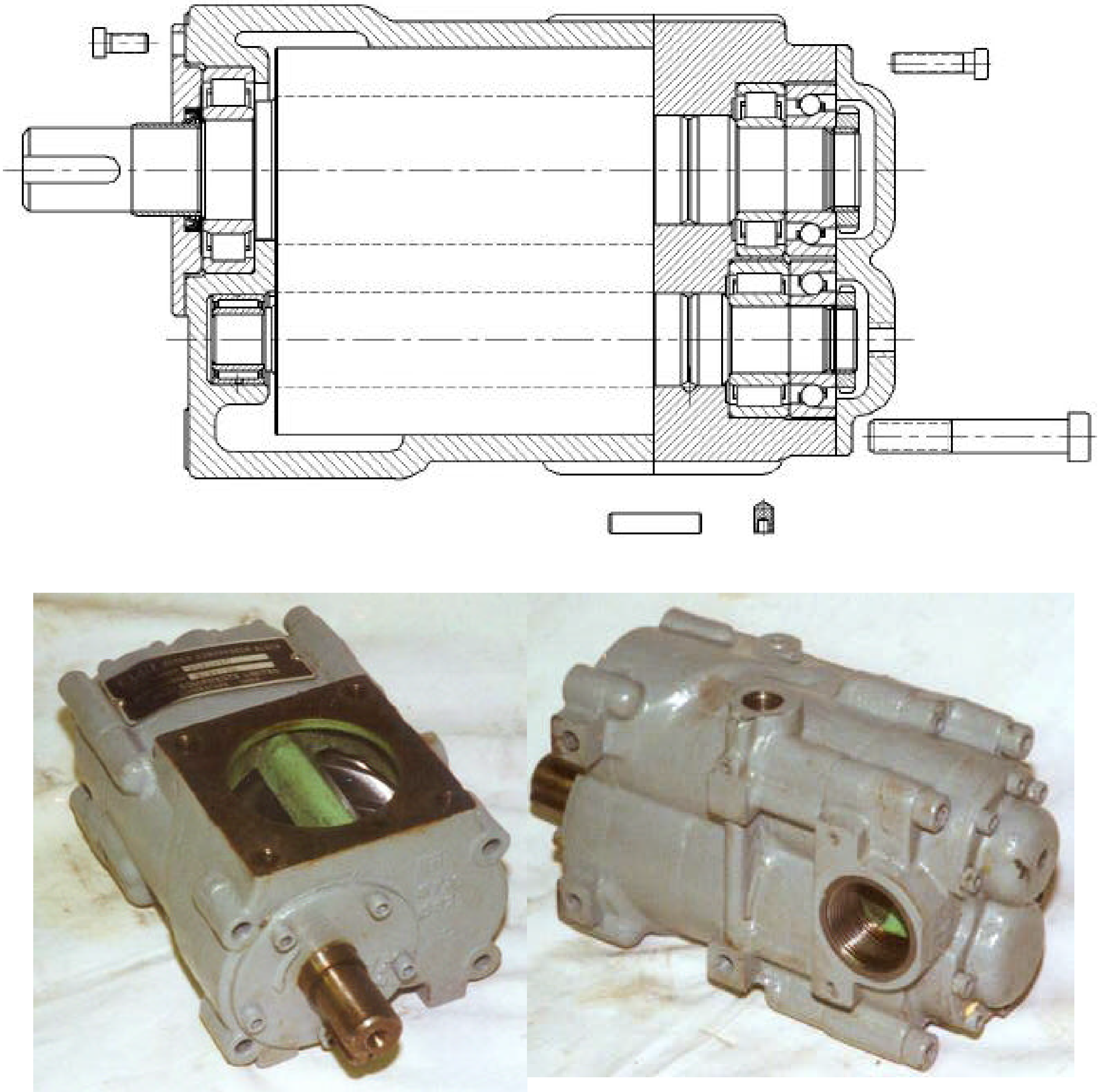


Fig. 7 The smallest compressor of the compressor family*, Venumadhav, 2001*

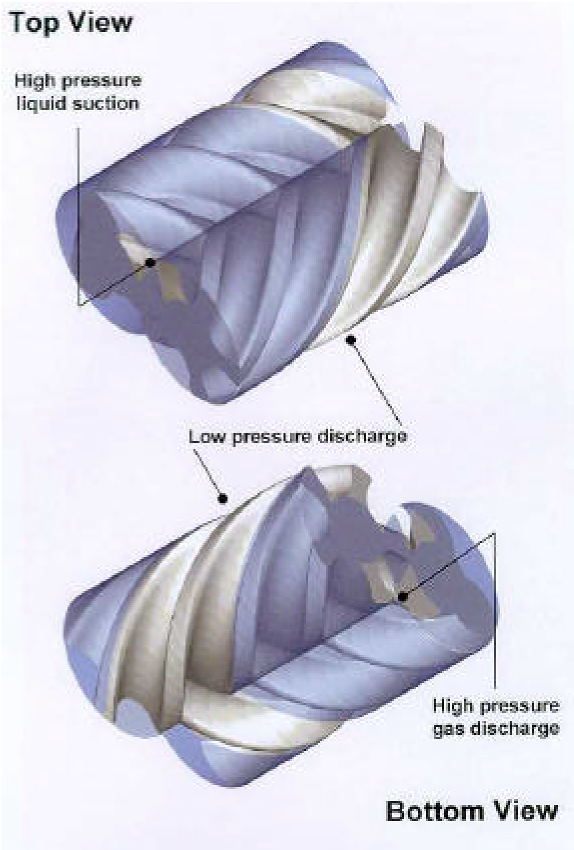


Fig. 8 The expressor principle, Brasz*, 2000*

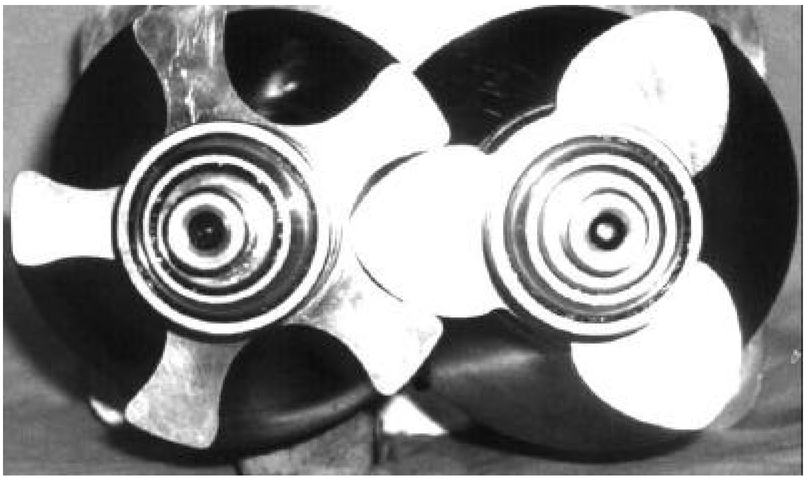


Fig. 9 The expressor two-side sealed rotors, Brasz*, 2000*

Fig. 10 The expressor, Brasz*, 2000*

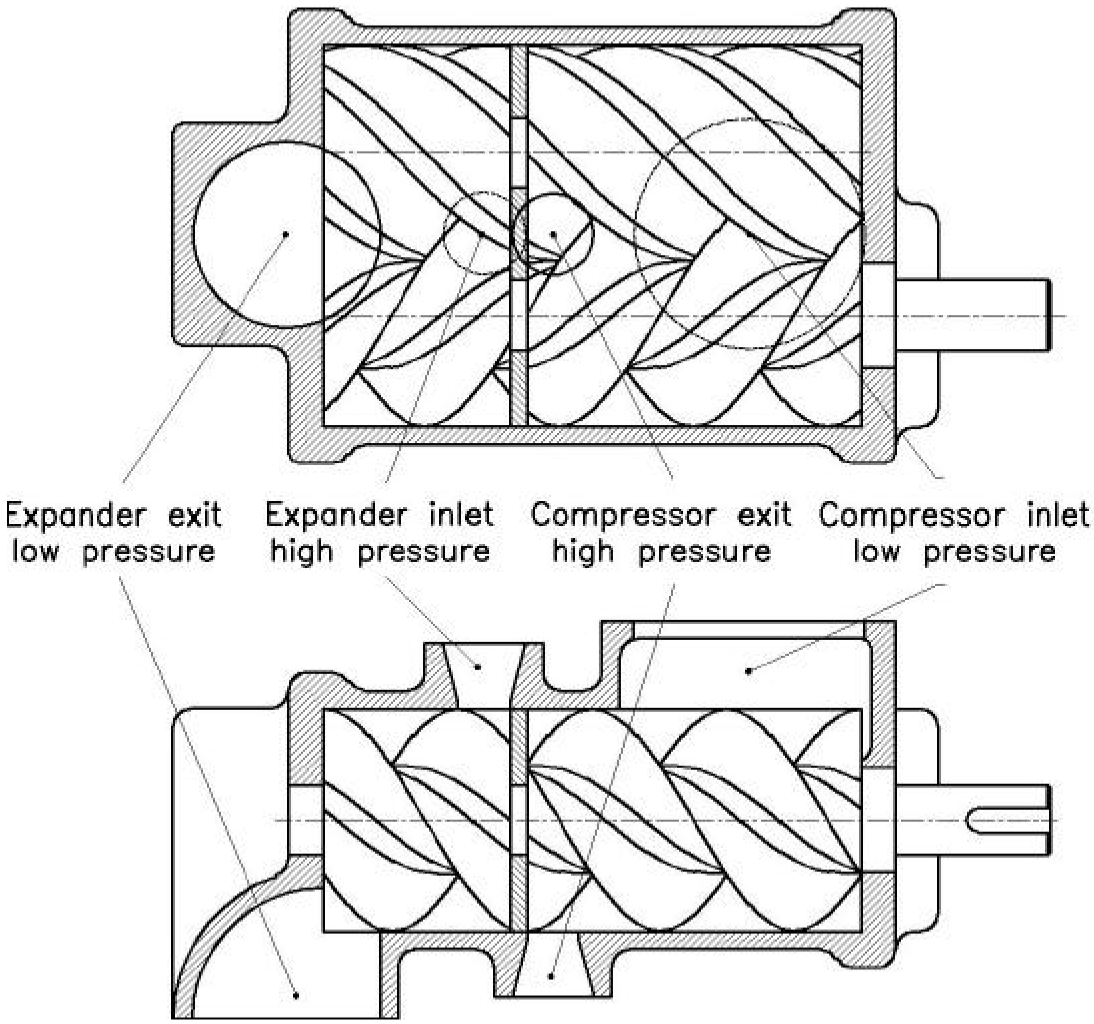


Fig. 11 Compressor – Expander with balanced rotors, Stosic*, 2002*