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# Speeding Up Diaphragm Compressors

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## Introduction

One of the specific features of diaphragm compressors is that static seals are employed which ensures virtually perfect leakproofness. There are no sliding parts requiring lubrication inside the gas space of a diaphragm compressor. On account of this particular design concept,

- diaphragm compressors are hermetic tight with respect to the ambient. The entire gas space of the compressor is enclosed by metallic, static seals. Leakage rates of 10 $^{4}$mbar l/s can be achieved easily; if a specific modification is applied, even leakage rates of $10^{-8}$ mbar l/s into the ambient are feasible. Because of these low leakage rates, applications in the "hot" areas of nuclear facilities or for the compression of highly toxic gases are possible.

- the working space of diaphragm compressors remains absolutely free of lubricants, i.e. the gas to be compressed does not come in contact with any lubricant, and there is no need to dispose of exhausted and contaminated lube oil or grease. Critical gases like oxygen or chloric gas can be compressed without problem even at high pressures.

- there are no particles released due to abrasion on piston rings or gland packings. There are no flushing and seal gas systems required. When the compressed gas is discharged from the diaphragm compressor, its percentage purity is still the same as when it was taken in. Even if the gas has a defined quality at the compressor intake, it can be supplied without aftertreatment after compression, e.g. as respiration air, in the semiconductor industry as dopant, or as hydrogen in food industry applications.

- the gas is only in contact with metallic materials. Depending on the requirements and the type of gas, different material grades can be selected. The right choice of material allows to achieve an excellent corrosion resistance. The available materials range from standard constructional steel to stainless special steels and high alloy materials e.g. Hastelloy. [1]

These considerable advantages with respect to other compressor conceptions are contrasted by relatively low delivery capacities. The delivery capacity, also designated as mass flow or volumetric flow, is determined by the diaphragm head's geometrical volume displacement per stroke and the compressor speed. Presently, the compressor speed is limited to approx. 400-500min$^{-1}$ for big belt-driven compressors and approx. 750min$^{-1}$ for small direct coupled units (up to about 40kW driving power). The volumes per stroke range from 4.5 to 8000cm$^{3}$.

# 1 Design and working principle of diaphragm compressors

## 1.1 Function

The gas compression is effected by a sandwich diaphragm that oscillates in a space delimited by two concave surfaces. The diaphragm acts together with a static metal o-ring as a seal and separates the gas space hermetically from the drive section. While it is clamped between the diaphragm head cover and the flange together with the perforated plate along its circumference, an oil-hydraulic drive forces it to perform oscillatory flexions (figure 1).

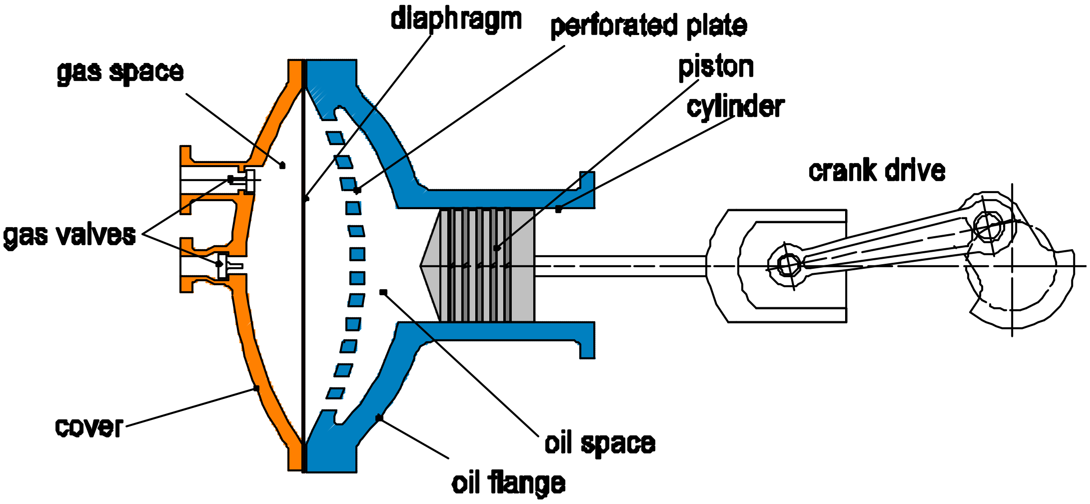


**Figure 1:** Diaphragm head and crank drive [1]

This spatial flexion enlarges and reduces the gas space between diaphragm and diaphragm head cover cyclically in size. As the gas space starts to enlarge, gas is taken in from the suction line through the inlet valve installed in the diaphragm head cover; when the gas space shrinks, it is delivered through the discharge valve - which is as well mounted in the diaphragm head cover - into the pressure line.

The oil pressure required to bend the diaphragm is generated by a crank drive with a reciprocating piston. The volume swept by this piston corresponds roughly to the diaphragm head's volume displacement per stroke.

During the compression stroke, the piston pushes oil from the cylinder into the diaphragm head where it flows through the perforated plate to the back side of the diaphragm. The diaphragm is thus forced to bend into the concave diaphragm head cover surface. As the piston moves back, it pulls the diaphragm against the surface of the perforated plate which is also concave. So the oscillation frequency of the diaphragm corresponds to the compressor speed. [1]

## 1.2 Design of the diaphragm head

The diaphragm head consists essentially of diaphragm head cover, triple sandwich diaphragm, perforated plate and flange. The compressor valves are arranged one beside another in the diaphragm head cover. They are sealed by metallic seal rings and kept in place by thrust pieces.

The sandwich diaphragm consists of 3 individual non profiled metal sheets. Along its circumference, it is gastight clamped between diaphragm head cover and perforated plate. A metallic o-ring seal keeps the system gastight.

Since only...

- the diaphragm head cover,
- the gas-side metal sheet of the sandwich diaphragm,
- the compressor valves with thrust pieces and seal rings, and
- the metallic o-ring for sealing the diaphragm head

... come into contact with the gas, leakproofness can be ensured at low cost. The static seals, especially their geometry and installation place, are field-proven and have turned out reliable in practice over decades [2]. Only the diaphragms which are subject to alternating load cycles need to be monitored. Such a monitoring allows to detect the failure of an individual diaphragm and to switch off the compressor immediately.

## 1.3 Design of the hydraulic drive

The diaphragm head is screwed to the flange and thus connected to the cylinder. During the compression stroke, the reciprocating piston pushes the hydraulic oil from the cylinder into the flange. In there, the oil flows through the perforated plate to the diaphragm backside.

Since a little oil quantity leaks through the piston seals back into the crank drive with every piston stroke, this oil loss needs to be compensated continuously.

This is done by means of the so-called compensating pump (figure 2). This oil pump is driven directly by the crankshaft and each time the main piston performs the intake stroke, it injects a small oil quantity into the space behind the diaphragm. Since the oil loss rate is not precisely known, the oil injection rate must in any case exceed the leak rate. Therefore, the oil space behind the diaphragm is slightly overfilled prior to the compression stroke.

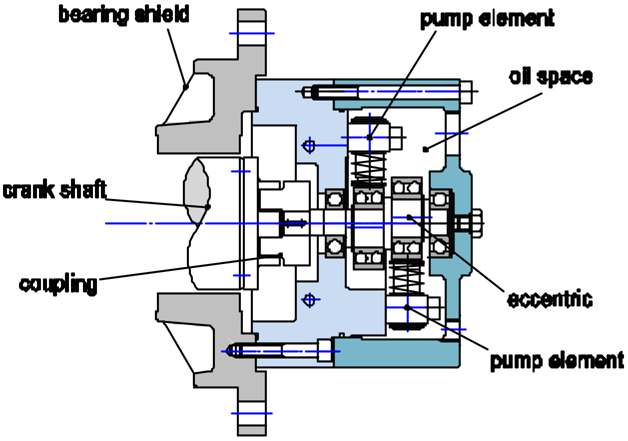


**Figure 2:** Compensating pump [1]

The oil injected in excess needs to be removed from the system. To this end, an overflow valve is installed at the highest point of the oil space. This valve discharges the excessive oil injected by the compensating pump.

The overflow valve is adjusted to an invariable opening pressure by means of a pre-loaded spring. The oil pressure is approximately 10% higher than the maximum gas pressure generated in the compressor section at the end of the compression stroke.

Shortly before the piston reaches its upper dead centre, it has already pushed the diaphragm completely against the diaphragm head cover. The oil which is displaced over the remaining few millimetres of piston stroke will now be returned through the overflow valve into the oil tank. At the same time, any air that may have accumulated at the highest point due to hydraulic oil degassing and which also impairs the compressor efficiency is removed as well.

In the oil tank, the oil can degas. Furthermore, the compensating pump takes the oil in from this vessel. This arrangement ensures that the hydraulic system is always optimally filled and the oil space is always perfectly vented.

Above a certain diaphragm head size, the oil inside the diaphragm head is cooled. Cooling water that flows through a cooling coil dissipates a part of the compression and frictional heat already at this point.

The employed hydraulic fluids are special mineral oils that simultaneously act as lubricant for the crank drive. These oils must meet the following requirements:

- good lubricating properties,
- low compressibility,
- good antifoam characteristics, and
- viscosity index that is as high as possible

Special applications or operating conditions may require the use of other hydraulic fluids such as special synthetic oils or water with antirust agent for high-pressure oxygen compressors (which requires however a separate lubricating system for the crank drive). The use of such special hydraulic fluids is a mere precaution. It prevents destruction by internal fire even in the extremely unlikely case that the entire set of diaphragms fails concurrently and the hydraulic fluid then comes into contact with the gas. [1]

## 2 Possibilities to increase the delivery capacity of diaphragm compressors

At first sight, it may seem simple to increase the delivery capacity of a diaphragm compressor: after all, it just needs more volume per stroke or more strokes per unit of time. However, physics places restrictions on such increase.

### 2.1 Increasing the displacement per stroke

An increase in the displacement per stroke can be achieved by increasing the diaphragm deflection or increasing the diaphragm diameter. The diaphragm deflection is ruled by specific curve geometries as these determine the life span of the diaphragms as well as the maximum deflection. Since these are decisive factors for the reliability of the compressor no one will change a tried and tested geometry without getting a great improvement.

In contrast to this, increasing the diaphragm diameter is the easier method that allows to standardize the product range and to offer different delivery capacity levels. Furthermore, the volume per stroke is proportionate to the cube of the diameter ($D^3$) and has therefore the biggest effect. Changes to the diaphragm head geometry and to the piston dimensions in the hydraulic drive section can be obtained at relatively low engineering cost. The maximum diameter is determined by the sheet metal sizes available on the market (presently 1400mm).

Above a certain dimension there are obvious disadvantages. The sandwich diaphragm will gradually become "softer" as the diameter increases. This effect can be counteacted by increasing the number of individual diaphragms that work together in a "sandwich" set. Of course the manufacturing costs will increase considerably as the material to be cut away multiplies when producing a big diaphragm head. And also exchanging the diaphragm increase the cost, not least because the significantly increased number of bolts will lengthen the time needed for the assembly of the diaphragm head.

Another, but physical, factor that should not be underestimated is the quantity of oil that needs to be moved in a diaphragm head above certain dimensions. It exceeds by far the displacement per stroke (factor 3 and more).

The efficiency of a diaphragm head depends crucially on the flow dynamics, i.e. on the oil flow and how the pressure builds up in the hydraulic section of a diaphragm head. Only a flow pattern that makes the pressure build up from the circumference towards the centre will ensure that the sandwich diaphragm rolls suitably off the head cover surface, also starting out at the circumference and moving towards the centre, so that the gas is completely expelled from the diaphragm head. If not so, gas bubbles will occur and reduce both efficiency and delivery capacity and that will furthermore cause local strain peaks which shorten the life span of the material considerably.

## 2.2 Increasing the speed

As compressors are commonly driven via a V-belt transmission, an increase in the compressor speed is a much more simple and cost-efficient option. This can be achieved using a larger motor pulley of suitable dimensions or, in case of variable speed operation, employing a frequency converter. The achievable increase in delivery capacity is however limited (wear, stress by inertia loads).

A new motor pulley and V-belts are cheap and can normally also easily be upgraded after the machine has been already installed. Attention must be paid to ensure sufficient excursion of the sliding motor rest (belt tension!) and sufficient dimensions of the belt protective case. In any case, the dimensioning of the compressor valves and pipes must be checked if the compressor speed is increased for upgrading purposes.

Frequency converter operation is much more expensive. It requires a control system and, in hazardous areas, motors with explosion-proof enclosure. In case of big delivery capacities, all these components together will normally cost less than a bigger diaphragm head. Furthermore, the machine can be kept compact which additionally saves manufacturing costs.

The most serious disadvantage is an increased alternating bending stress to the diaphragms. Furthermore, any malfunction in the oil hydraulic or gas bubbles inside the diaphragm head will now reduce the life span of these components more seriously. This means the oil flow dynamic in the hydraulic section remains of crucial importance when speeding up a compressor. Recent research [2] shows that in contrast to assumptions made in the past, this does not only apply to large diaphragm diameters ($D_N$ > 350 mm) but also to smaller ones.

## 3 Prerequisites for an increase in delivery

The prerequisites for increasing the delivery capacity beyond the present limits have basically already been presented in the two preceding chapters. Here, these prerequisites are explained once more in detail and a summary is presented.

It is important that the diaphragms start to make contact with the head cover surface from the outer circumference to the centre. During this flexion movement, the diaphragm must contact the cover surface in all points; it may not warp. Otherwise, gas residues are enclosed and will either remain in the diaphragm head or be "squeezed" out of the diaphragm head at much higher oil pressure than normal. This reduces both efficiency and delivery capacity; the power consumption increases. Furthermore, high local bending stress is created around these gas inclusions which reduces the life span of this component considerably.

To ensure an optimised diaphragm movement under all conditions, the oil hydraulics that actuates the diaphragm needs to be optimised in such a way that the pressure will build up from the outer circumference towards the centre in controlled manner. The pressure drop during the return stroke should accordingly move from the centre to the circumference.

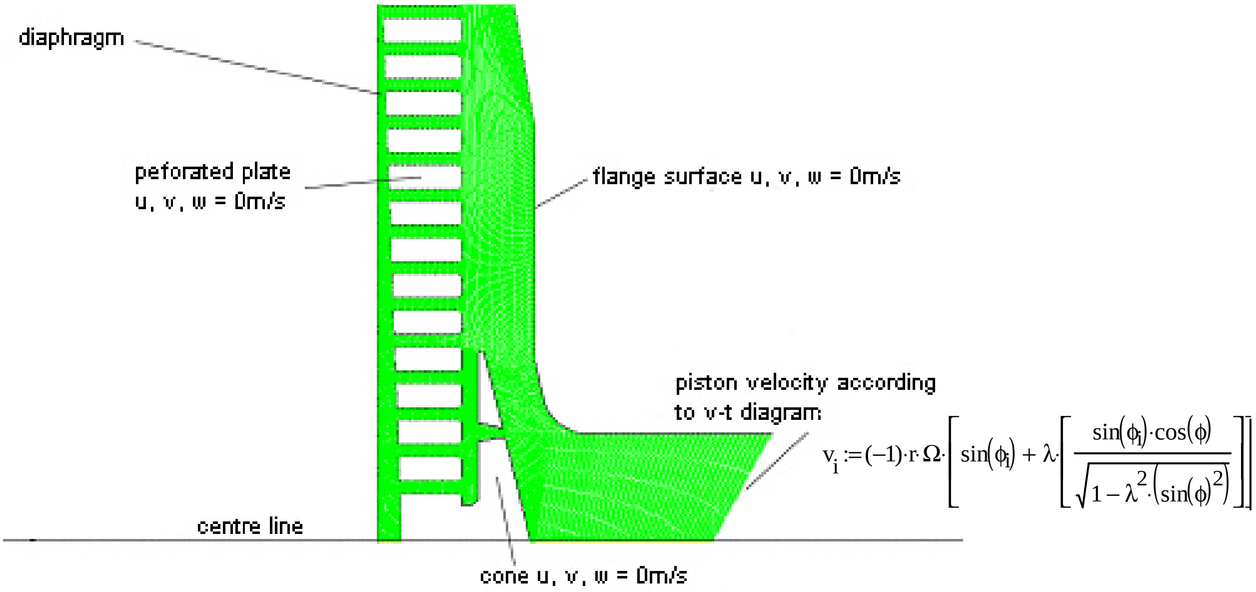
## 4 Basic research into the oil-hydraulics

In the context of a joint research project with the technical academy "Fachhochschule Niederrhein" (Krefeld, Germany), basic research into the fluid dynamics and pressure rise inside a diaphragm head has been carried out. The related study has the title "**Entwicklung eines Membrankompressors mit neuem Hydraulikkammersystem" [3]** (Development of a diaphragm head with new hydraulic chamber system).

In this study, a Finite Volume Model has been used to model the geometry of a diaphragm head and to simulate the flow dynamics created by the piston movement. Similarity studies involving dimensionless characteristic values have been carried out with the aim to transfer the results to other operating parameters and other diaphragm diameters.

In a first phase, the research was focused on the operating points Upper and Lower Dead Centre; the entire intake and compression cycle was then computed using the best geometry approach.

The following figure 3) shows the model after the modifications and the related constraints.



$$v_i := (-1)\cdot r\cdot \Omega\cdot\left[\sin(\phi_i) + \lambda\cdot\left[\frac{\sin(\phi_i)\cdot\cos(\phi)}{\sqrt{1-\lambda^2\cdot\left(\sin(\phi)^2\right)}}\right]\right]$$

**Figure 3:** Simulation model [3]

## 5 Results

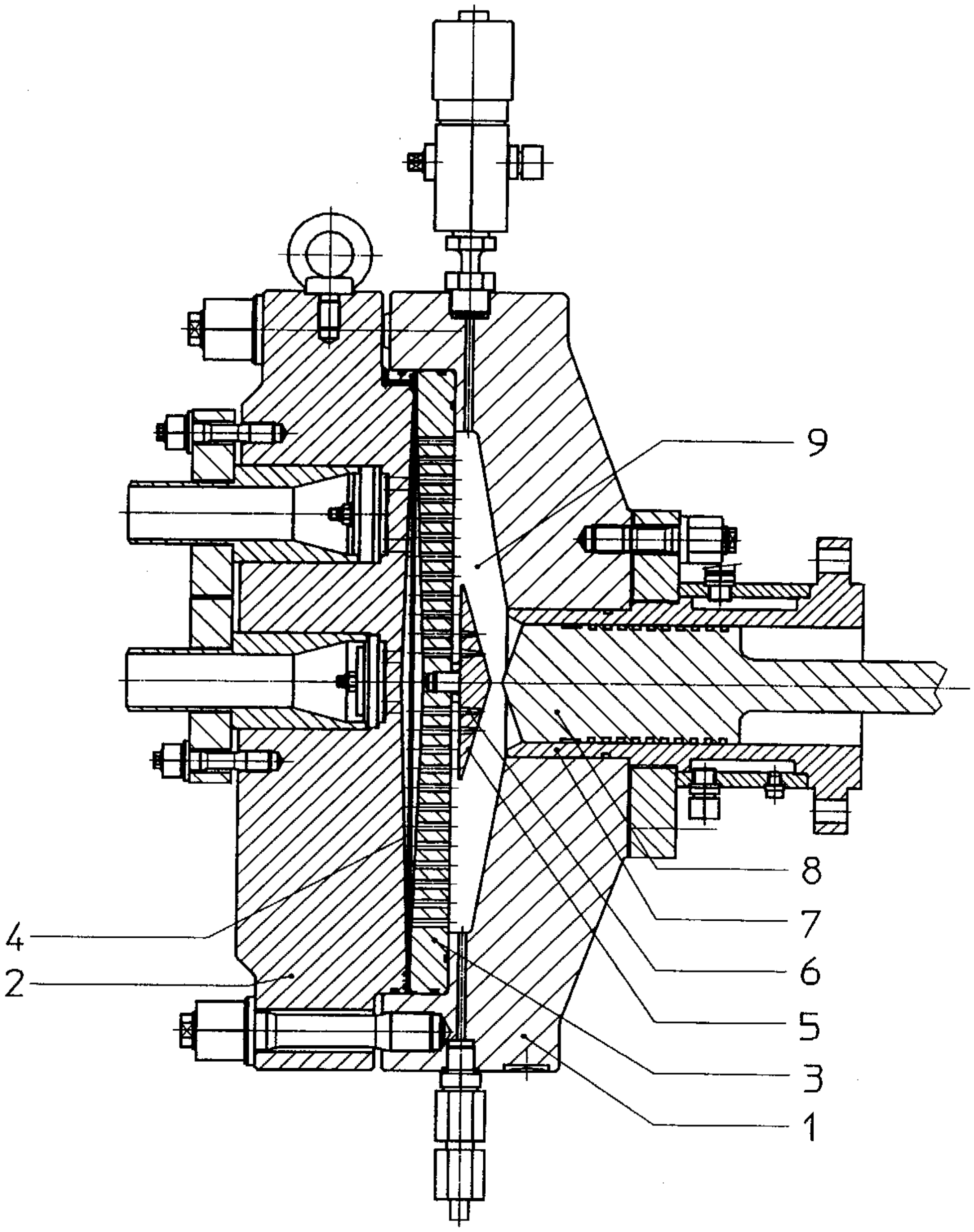
The main component of the new diaphragm compressor is unquestionably the distribution cone with bores according to figure 4 (patent pending).

**Figure 4:** New and conventional design conception of the compressor distribution cone [3]

This new cone has the function of a flow limiter (similar to a non-return valve), however without the need of mechanically moving components as in prior designs. This offers the following advantages with respect to conventional diaphragm head constructions:

- increased efficiency due to reduction / prevention of dissipative boundary layer separation and the related flow turbulence (eddying);
- increased efficiency due to almost 100% utilization of the geometrical working space (volume swept by the diaphragm);
- no local increased bending stress in the diaphragms due to the prevention of gas accumulations, thus longer life span of the diaphragms and consequently, increased availability of the machines.

To give a brief survey, figure 5 shows the new diaphragm compressor prototype in a sectional drawing.



**Compressor components:**

(1) Flange, (2) Head cover, (3) Perforated plate, (4) Diaphragms

(5) New distribution cone with (6) bores

(7) Cylinder, (8) Piston, (9) New hydraulic space

**Figure 5:** New prototype of the diaphragm compressor [3]

## Future prospects

First experiments showed that the model calculations coincide with practice. In particular, much lesser intermittent malfunctions were observed on the delivery valves of the compressor. It could also be observed that the positive effects result both on small and large-diameter diaphragm heads. Statistic confirmation will require more measurements on different diaphragm heads.

## Literature

[1] Dehnen, Dipl.-Ing Manfred; Hofer Membranverdichter

[2] Werksinterna der Fa. Andreas Hofer Hochdrucktechnik GmbH

[3] Nassenstein, Prof. Dr.-Ing. Christoph; Habel, Dipl.-Ing. Heiko; "Entwicklung eines Membrankompressors mit neuem Hydraulikkammersystem"