The influence of rotor designs suitable for industrial production on the performance of screw compressors

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Abstract

The very limited market presence of screw compressors in mass-produced vehicles can be put down to the complicated geometry and tight tolerances of screw rotors, which rule out economic production in large numbers by modern manufacturing methods. However, a range of innovative fabrication procedures for the manufacture of compressor components, in particular of the rotors, suggests that economic production cannot be far away. The solution is likely to be by means of deformation production procedures, along with suitably modified profile and rotor geometries.

The screw compressor we have developed, which can be considered as a well-designed machine for mass production using deformation techniques, but with untypical geometrical parameters, will be compared with a screw compressor which has thermodynamically favourable operating performance. The following evaluation of the screw compressor is based on the volumetric and power efficiency within the framework of a steady state and quasi-steady state analysis of the machine characteristics. The development and construction of a screw compressor using techniques suitable for deformation production techniques results in a machine with comparatively good thermodynamic performance, in spite of the untypical machine geometry parameters.

1. Introduction

The increasing importance of mass production, current problems such as the manufacturing costs of screw compressors, and pressure from the automobile industry for a promising downsizing concept for mechanical screw compressors all combine to provide motivation for the realization of innovative concepts for the production of screw rotors with suitably modified profile and rotor geometry.

The casing is already produced by means of deformation methods. The most significant other costs are then the synchronizing gears and the rotors themselves. In the case of screw compressor production, deformation methods are only employed for untwisted or lightly-twisted profiles. An extension of these techniques to meet compressor-specific requirements would be genuinely innovative.

Pure metal-working rotor manufacture involving milling and grinding procedures results in comparatively high costs because it is so complex and time-consuming. Production by means of castings, which then have to be finished, provides an acceptable solution in manufacturing terms, but the costs are still high in terms of efficient production.

Nowadays deformation manufacturing techniques are used to produce items with dimensions which are close to their final form, e.g. casing parts for screw compressors [1], [2], and extrusion methods, which are not very time-consuming, provide a viable alternative for the production of screw rotors.

2. Geometrical analysis of the screw compressor

Within the framework of a research project new variations of deformation manufacturing methods for screw rotors have been examined and analysed. For example, the following procedures for the production of screw rotors were employed: Conventional Twisting (CT), Twisted Profile Extrusion (TPE), and Helical Profile Extrusion (HPE), [3], [4], [5]. The relevant research was carried out at the Institute of Forming Technology and Lightweight at the Technical University in Dortmund.

The specifications for geometrical machine parameters such as rotor length, length/diameter ratio and wrap angle requirements were adapted for deformation manufacturing methods, using Conventional Twisting (CT) as an example, and subjected to close examination. A large wrap angle (180 or 200 degrees) is generally regarded as favourable from a thermodynamic point of view. However, this characteristic can only be achieved by means of greater accuracy in the production process.

2.1 Rotor wrap angle

Changing the wrap angle affects all the gaps in the machine, even if all other geometrical dimensions remain constant except scoop volume. In order to evaluate the effects of changing the wrap angle, the relative area ratios of the gap nearest to the polytropic working chamber were examined, i.e. on the high-pressure side of the machine, as these have the major influence on the energy conversion efficiency of the machine. **Fig. 1** shows the relative area ratios of the machine gaps as the male rotor wrap angle is changed by constant scoop volume. The reference point is a wrap angle of 200 degrees. Starting from this point, reducing the wrap angle results in an increase in the gap areas a) to c). This behaviour is caused by the increasing rotor gradient, which raises the angle of the profile flanks of male and female rotors to one another. As reducing the wrap angles has no influence on the size of the machine or on gap area d), only the wrap angles have an effect on gap changes. The degree of gradient change still depends on the gap in question.

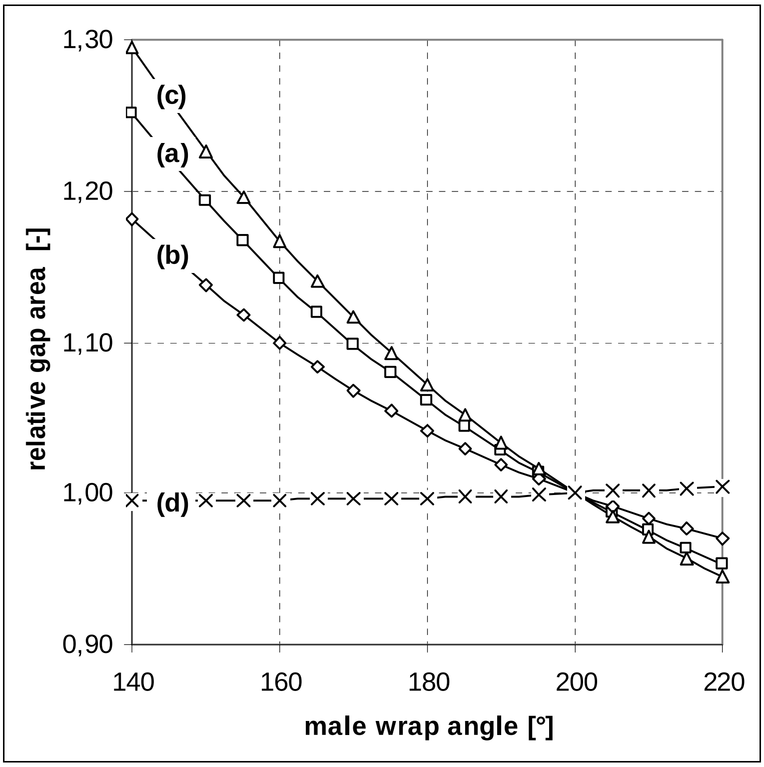

Fig. 1: Relative gap area as a function of the male rotor wrap angle.
Reference values: tooth count pairing 3/5 (male rotor/female rotor), 200° (MR), L/D=1.4, constant scoop volume, profile meshing gap (a), housing gap (MR) (b), housing gap (FR) (c), front gap (HP) (d)

Raising the male rotor wrap angle above the reference point brings about a further reduction in the gap area, while the gradient becomes flatter. As the maximum chamber volume is successively reduced as the wrap angle increases, scaling of the rotor diameter is necessary if a comparative framework (constant scoop volume) is to be maintained. This leads to an increase in the gap areas, with the axial extension of the gap predominating, and this in its turn results in a smaller change in gradient. The increase in rotor diameter can be seen in the progression of the front gap d), as this has a small, but rising relative gap area.

2.2 Rotor length-diameter ratio

A variation in the length-diameter ratio results, if constant scoop volume is maintained, in changed gap geometry within the machine. In **Fig. 2** the relative gap areas are represented as a function of the L/D ratio by constant scoop volume. Starting from an L/D ratio of 1.4, a reduction in the L/D ratio leads to an increase in rotor diameter with a concurrent reduction in rotor length. A rising L/D ratio, on the other hand, reduces the rotor diameter and leads to an increase in rotor length.

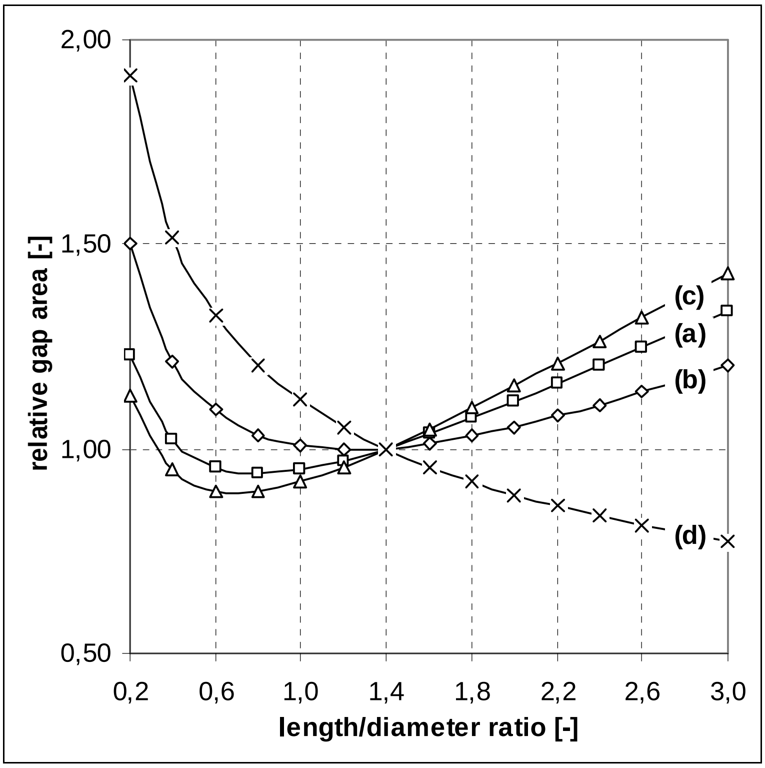

Fig. 2: Relative gap area as a function of the length-diameter ratio.
Reference value: tooth count pairing 3/5(MR/FR), 200 (MR), L/D=1.4, constant scoop volume, profile meshing gap (a), housing gap (MR) (b), housing gap (FR) (c), front gap (HP) (d)

This behaviour can also be observed in the relative area development of the front gap. The changes in the gap areas a) to c) each have a minimum area, which varies according to the type of gap. As the wrap angle remains constant during L/D variation, only changes in the rotor diameter and length affect the relative gap proportion. The configuration of the axial and radial gap constituents results in a gap minimum in the area of L/D ratios of 0.8 to 1.0, depending on the type of gap, with the exception of the front gap. In the case of very small L/D ratios, a major rise in rotor diameter causes a significant rise in the radial gap component, as is made clear by the relative gap area c). This reduces down to L/D ratios of 1.2-1.6, at which point, with the rotor length continuing to increase, the axial gap component becomes the main factor in the total gap area.

The untypical geometrical machine parameters which appear to be suitable for rotor production by deformation methods are the smallest possible wrap angle and a high length/diameter ratio. Low rotor gap angles, e.g. 160° compared with the more typical 200°, result in a relative increase in the meshing and casing gaps of about 10-16%, see Fig. 1. Raising the L/D ratio to e.g. L/D= 2.2, compared with 1.4, results, at a constant wrap angle, in a further relative increase in the main gap areas of 8-20%, see Fig 2. Geometrical changes of this kind influence the energy conversion of the machine to varying degrees, allowing comparative geometrical performance values to be defined, thus permitting different machine variations to be compared.

Employing machine parameters of this kind leads, with constant scoop volumes, to an increased influence of the gap under examination on the operating behaviour. The extent of this kind of geometrical configuration on operating behaviour will be examined below.

3. Geometrical configuration of the screw compressor

In the present case, the geometrical configuration of the screw compressor is not based on the research results of Weckes [6], who has analysed the influence of geometrical components such as tooth count, wrap angle, L/D ratio and moment of inertia on volumetric and power efficiency.

Instead, the design of the compressor is based on the following requirements:

- small wrap angle (suitable for deformation techniques), which means that the gap factor of the machine increases (see Fig. 1),
- high L/D ratio (formation of a closed working chamber due to small wrap angle), also resulting in an increase in gap area (see Fig. 2),
- Reduction in weight/moment of inertia of the rotors (material choice for the rotors: Aluminium),
- synchronised screw compressor (synchronising gears),
- Coating of the rotor pairs via the Eloxal process (electrolytic oxidation of aluminium, which provides protection from contact during operaion),
- high rotor revolutions (reduction of gap influences on the volumetric efficiency, compensation for production tolerances,
- modular construction (breaking down of the machine components in the interests of deformation techniques and mass production),

- inexpensive components (use of standard components such as bearings and seals of types already available),
- rapid assembly (serial production aspect).

The modular construction of the screw compressor AL45 which has been developed (see **Fig. 3**) has an axle spacing of 45mm. The rotor pair (Fig. 3 LHS) was produced by deformation methods, using extrusion followed by conventional twisting. After the twisted blank has been shrunk onto the compressor shaft, it is ground down to its final profile. Coating the rotors via the Eloxal process produces a protective layer for the aluminium rotors in case there is accidental contact between them.

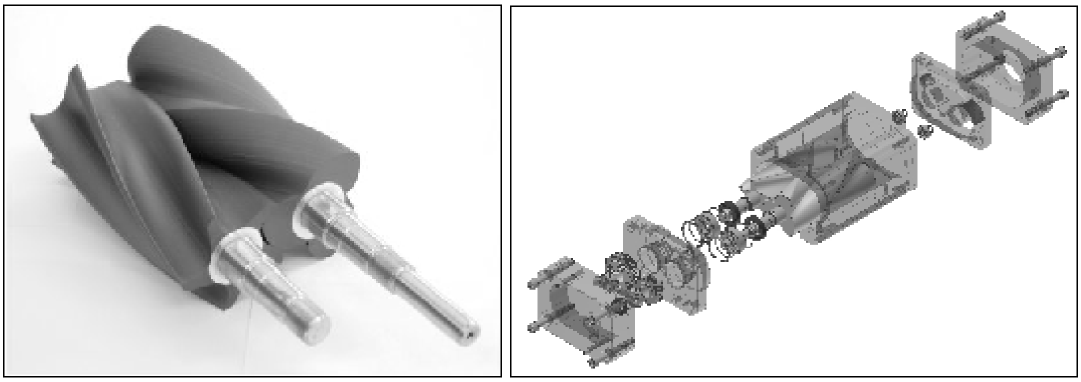

Fig. 3: Construction of the screw compressor AL45.
(Aluminium compressor, axle spacing: 45mm), LHS: rotor pair of the AL45 (from the left, female rotor, male rotor), RHS: exploded drawing of the compressor.

The machine data resulting from the profile requirements can be seen in **Table 1.** The analyses of compressor AL45 are compared with a control machine (VSL) which has the same internal volume ratio of 1.25. The L/D ratio of compressor AL45 has the value 2.20, which corresponds with a rotor of small diameter but extended length, cf. relative gap changes, see Fig. 1 and Fig. 2.
According to Weckes [6], ideal L/D ratios are in the area of 1.2 to 1.6. On the basis of the maximum pressure ration of $\Pi = 2.4$ for combustion motor supercharging, and the resulting compressive forces inside the machine, rotor contact as a result of deflection under load can be ruled out.
Additional data for the VSL compressor are available. Its parameters were selected primarily on the basis of good thermodynamic process management. The geometrical parameters of the VSL compressor are given in standardised form. The AL45, which possesses the same

internal volume ratio, serves as a control machine. The VSL has an L/D ratio of 1.20 and the same tooth count, which results in a 25% increase in the diameter of the male rotor, and a reduction in the rotor length by a factor of 0.69. The wrap angle is increased by 23% compared with the AL45.

Table 1: Machine date of the AL45 (reference machine) and the control machine (VSL)

	AL45	**VSL**
Tooth pairing (MR/FR)	3/5	3/5
Internal volume ratio	1,25	1,25
Standard scoop volume	1,00	1,15
L/D ratio [-]	**2,20**	**1,20**
Standard male rotor diameter	1,00	1,25
Standard rotor length	**1,00**	**0,69**
Standard wrap angle	**1,00**	**1,23**
Standard axial inlet area	1,00	1,75
Standard axial outlet area	1,00	2,05
Standard radial outlet area	1,00	1,43

The effects of the basic geometrical parameters on the inlet and outlet areas of the two compressors are also shown in Table 1. Compared with the control compressor VSL, the AL45, as its specification includes a lower L/D ratio and a considerably higher wrap angle, has a 75% greater inlet area, a 105% larger axial outlet area and a 43% greater radial outlet area.

The almost identical gap height configuration of the two machines ensures that the two sets of engine characteristics can legitimately be compared, **Table 2**.

Table 2: Gap heights of the AL45 (reference machine) and the control machine (VSL)

	AL45	**VSL**
Profile meshing gap	0,16 mm	0,17 mm
Casing gap, male rotor side	0,14 mm	0,14 mm
Casing gap, female rotor side	0,14 mm	0,14 mm
Front gap, high pressure side	0,20 mm	0,20 mm
Front gap, low pressure side	0,30 mm	0,30 mm

A comparative assessment of the machine is carried out by means of the thermodynamic performance values, i.e. volumetric and power efficiency, at a maximum pressure of $\Pi = 2.2$, on the basis of the ambient pressure on the suction side. Volumetric efficiency is essentially a measure of the general efficiency of a machine, and it is expressed by the relationship between the mass actually transported and the theoretical mass flow. An assessment of the total energy conversion of a screw machine is carried out with the help of the isentropic efficiency factor, which is defined as the ratio of isentropic to actual performance, taking into account the mass flows transported by the machine. The screw compressors referred to here operate in the range from 9,000-33,000 min^{-1}.

4. Operating data analysis of the AL45 screw compressor

The experimental examination of the AL45 with reference to its production by means of deformation techniques and its thermodynamic properties, is carried out on the basis of measurements of the operating values during steady state and quasi-steady state operation of the machine, which is run briefly and as far as possible without heat expansion becoming a factor, i.e. in a cold state.

4.1 Steady state and quasi-steady state operating measurements

In the case of a downsizing concept, the behaviour of a compressor in both continuous operation (steady state) and in short bursts (quasi-steady state), plays a significant role. A compressor operating at steady state behaves, as a result of heat expansion of the rotors and housing, differently from a machine operating in short bursts. The comparative measurement data for the AL45 compressor are shown as a function of the volume flow superimposed on the pressure ratio, **Fig. 4**.

For both modes of operation, steady state and quasi-steady state, an increase in motor speed leads to improved volumetric efficiency, as one would expect Fig. 4 (top). A rise in pressure ratio leads to reduced volumetric efficiency because of the higher back pressure, which is particularly marked at low r.p.m. However, this can be compensated for by an increase in motor speed.

At very high motor speeds, improvements in volumetric efficiency fall away sharply, which can be put down to less effective chamber filling during the suction phase. Comparing the volumetric efficiency of the two operating modes shows that quasi-steady state operation of the compressor is more efficient under identical volume flow and pressure ratio conditions.

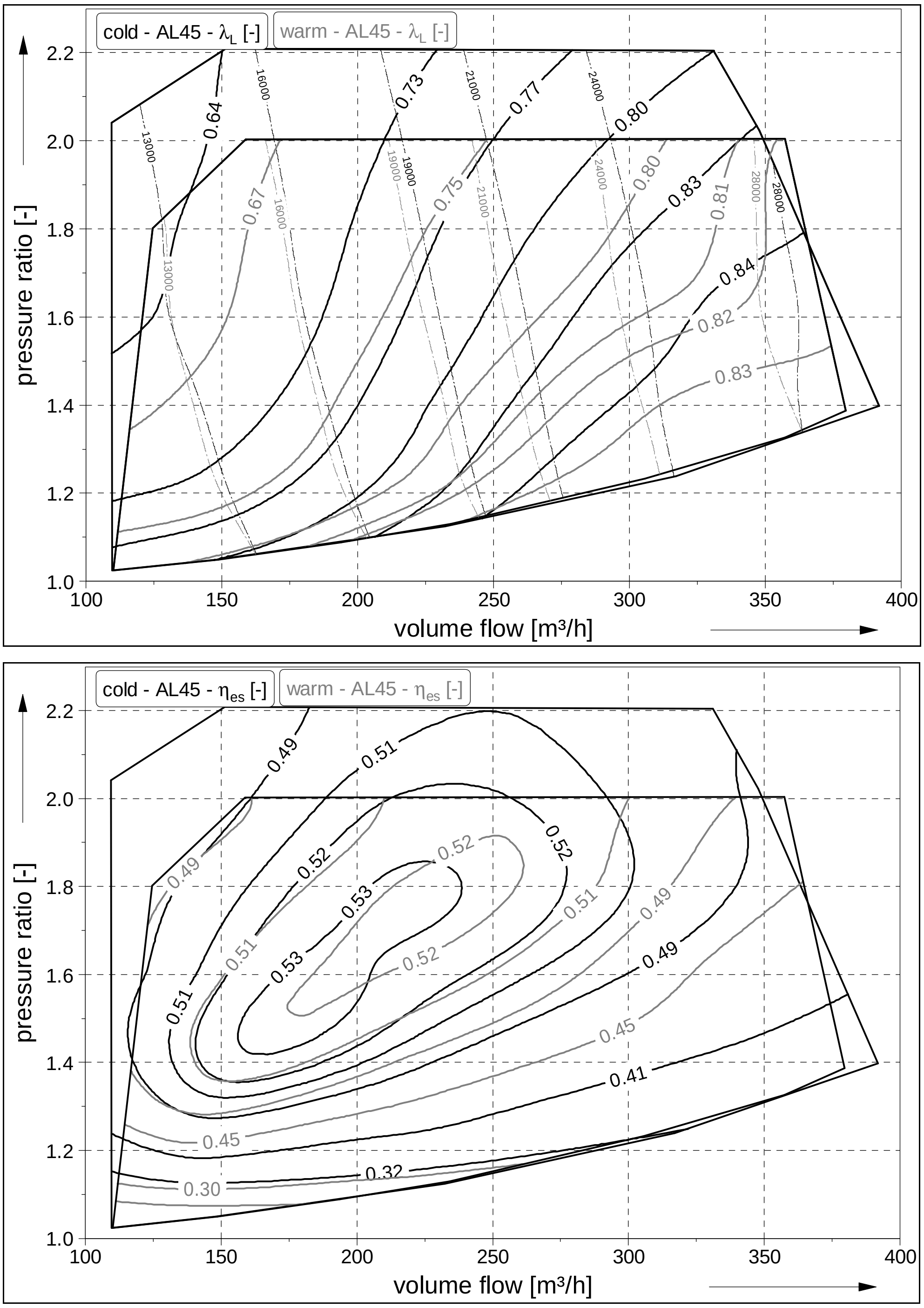

Fig. 4: Steady state and quasi-steady state volumetric efficiency (above) and effective, isentropic efficiency (below), plus male rotor speed

Initial conditions: $940 < p_{in}$ [mbar] < 1020, $19 < \vartheta_{in}$ [°C] < 22,

warm (steady state measurements), cold (quasi-steady state measurements),

dotted lines (male rotor speed, in min^{-1})

At low speeds, the cold machine achieves approximately 3% better volumetric efficiency than the warm machine. This difference reduces to about 1% as revolutions and pressure ratios rise.

Both operating modes achieve a virtually identical degree of isentropic efficiency, with the optimal values of 0.53 (quasi-steady state) and 0.52 (steady state) being achieved at a volume flow of 200-225 m^3/h and a pressure ratio of Π = 1.5-1.9. As the degree of efficiency corresponds to the relationship between isentropic and indicated working area ratios, efficiency is reduced as revolutions increase, because of increased choking effects during the outlet phase.

4.2 Comparing operating date of the two screw compressors AL45 and VSL

The volumetric efficiency of the control machine VSL is about 5% higher than that of the AL45 at low revolutions and rising pressure ration, **Fig. 5**. As revolutions and pressure ratios rise further, the difference between the two machines only reduces slightly, so that a minimum efficiency deficit of 3-4% remains. The optimum efficiency range for both machines is more or less the same. This is reached at a volume flow of 200-250 m^3/h and a pressure ratio of Π = 1.5-1.9. So the optimal efficiency factors of the VSL and the AL45 work out at 0.61 and 0.52 respectively.

These differences in volumetric efficiency are the result of differing machine parameters, i.e. the wrap angle and the L/D ratio. In the case of the AL45 this results in an unfavourable gap situation compared with the VSL. This can be attributed to back flow at the gaps. Over and above this, at identical volume ratios, the VSL has considerably larger inlet and outlet areas so that, on the one hand, any tendency towards choking at higher revolutions is reduced, and on the other hand, the expulsion process is also more effective for the same reason. This factor also brings about a rise in working area ratios and general efficiency, Fig. 5

5. Summary

Among the many influences on the operating behaviour of a screw machine, adding the aspect of deformation production techniques leads to geometrical gap changes inside the machine. Reducing the wrap angle and increasing the length/diameter ratio of the rotors has a negative effect on the geometric gap situation of the machine, which inevitably results in reduced energy conversion data for the machine. In addition to the gap situation, the inlet and outlet areas, which have a direct influence on thermodynamic processes, are considerably reduced.

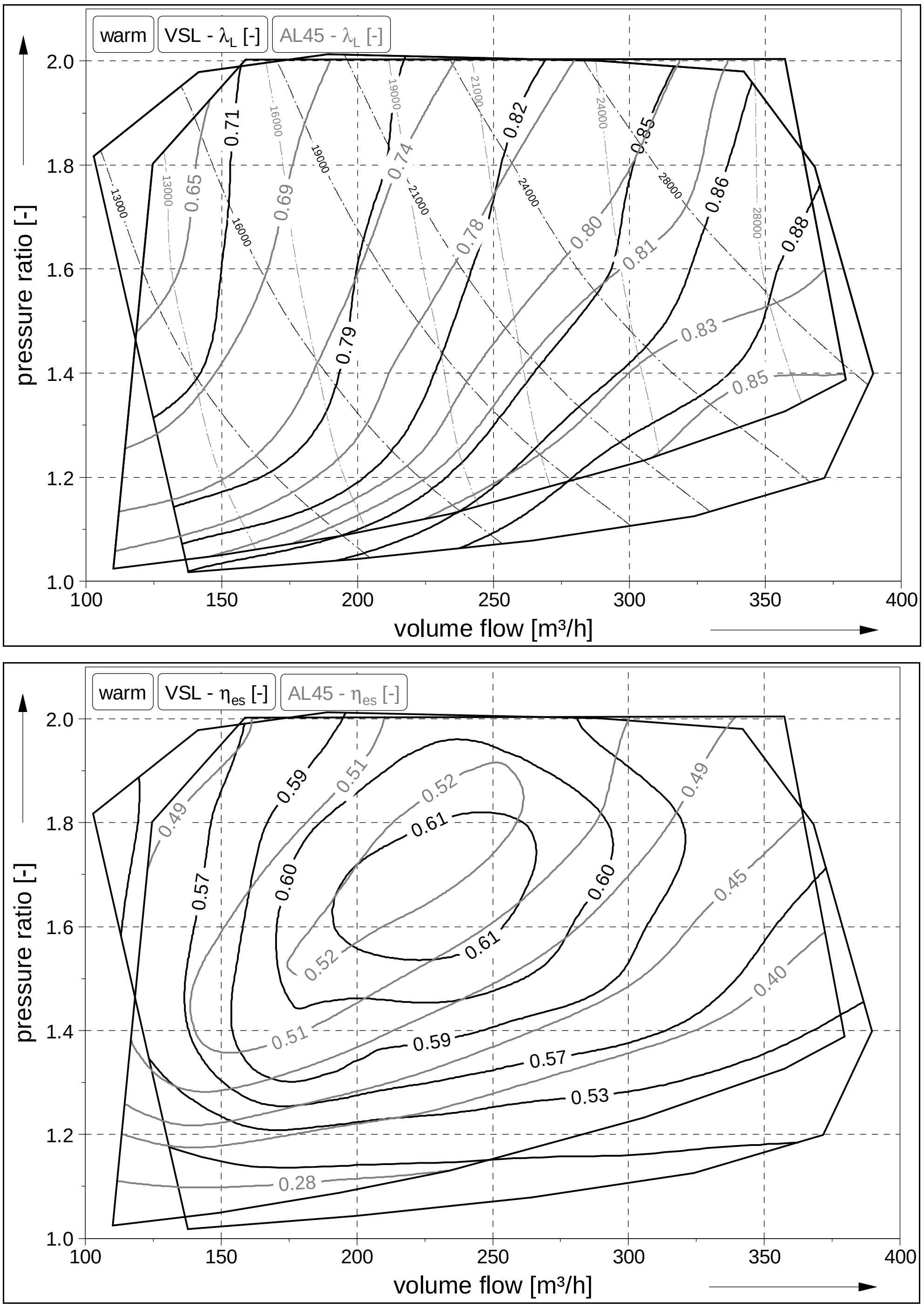

Fig. 5: Efficiency factor at steady state (above) and effective isentropic efficiency factor (below) for the screw compressor AL45 and the control screw compressor VSL.
<u>Initial conditions:</u> $940 < p_{in}$ [mbar] < 1020, $19 < \vartheta_{in}$ [°C] < 22,
Dotted lines (male rotor speed, in min^{-1}),

The development of the screw compressor AL45 provides information, within the framework of steady state and quasi-steady state operating data, on the influences of modified wrap angles and L/D ratios on the thermodynamic performance values, i.e. volumetric and overall efficiency. The volumetric efficiency of the screw compressor AL45 is slightly lower than that of the control machine, amounting to 9% at the optimal operating point.
From the point of view of producing screw rotors suitable for manufacture by means of deformation techniques, which would open the door to mass production, the screw compressor concept presented here combines characteristics suitable for mass production with comparatively good performance.

Symbols

FR	female rotor	MR	male rotor
HP	high pressure	p	pressure [mbar]
in	inlet	ϑ	temperature [°C]
λ_L	volumetric efficiency	η_{es}	effective, isentropic efficiency

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