Screw-Type Supercharger without Timing Gear – Simulation and Verification

Schraubenlader ohne Synchronisationsgetriebe – Simulation und Verifikation

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Abstract

Engine downsizing as a new combustion engine concept is leading to increased interest in mechanical superchargers. The screw-type supercharger is a tried and tested solution, but today only used as an add-on tuning component or in the high-price segment of the automobile industry.

This paper presents some new ideas in the design of screw-type superchargers. The aims are better thermodynamic performance and overall efficiency associated with less parts and simpler assembly which will reduce the complexity and costs of such a supercharger. Characteristic diagrams of the new supercharger without timing gear are presented and discussed.

The second main topic is the verification of the simulation tool *KaSim*. Comparisons between simulation and measurement are analysed. The influence of friction on seals and bearings and the possibility of reducing this friction is discussed. To take friction into account in the simulation a friction model is implemented. Furthermore the intake flow into the supercharger affects the performance and is analysed.

Zusammenfassung

Das Downsizing von Verbrennungsmotoren hat in letzter Zeit wieder ein verstärktes Interesse an mechanischen Verdrängerladern ausgelöst. Der Schraubenlader ist gerade in diesem Anwendungsbereich eine praktikable und erprobte Lösung, derzeit allerdings nur als Nachrüstsatz in Tuning-Bereich oder im Premiumsegment (z.B. AMG/Daimler Chrysler) der Automobilindustrie.

Dieser Beitrag stellt einige neue Ideen in der Konstruktion von Schraubenladern vor. Zielsetzungen sind neben der Verbesserung der thermodynamischen Energiewandlung und der Erhöhung des Gesamtwirkungsgrades vor allem weniger Bauteile und eine einfachere Montage. Dies ermöglicht die Reduzierung der Komplexität und der Kosten eines Schraubenla-

ders. Der Schraubenlader ohne Synchronisationsgetriebe ist ein Schritt in diese Richtung. Kennfelder dieses neuen Laders werden diskutiert.

Des weiteren ist die Verifikation des Simulationswerkzeuges *KaSim* Gegenstand dieses Beitrages. Simulierte und gemessene Kennfelder werden verglichen und analysiert. Der Einfluss der Reibung an Dichtungen und in Lagern und die Möglichkeit diese Reibung zu reduzieren werden diskutiert. Ein Model zur Berechnung der Lagerreibung wird vorgestellt. Weiterhin wird der Einfluss der Einlassströmung in den Schraubenlader auf sein Betriebsverhalten näher betrachtet.

1. Introduction

The use of rotary displacement machines in automotive applications has been experiencing a renaissance. At the International Motor Show 2005 in Frankfurt Volkswagen AG presented a new motor design, the TSI motor [1]. The main technical innovation is the use of a twin supercharger concept with a roots blower and a turbocharger [2][3]. This design combines good performance with economical fuel consumption due to the application of downsizing, twin supercharging and direct injection into a small size engine.

The screw-type supercharger is an alternative to the roots blower or to the combination of roots supercharger and turbocharger. Due to the internal compression of this machine type it is especially suited for downsizing concepts with high boost pressure.

So far the screw-type supercharger is not widely accepted. Possible reasons are on the one hand the previous requirements of lower boost pressure for supercharged engines. On the other hand the complex geometry, the high degree of accuracy required during production and high noise levels cause problems in the acceptance of screw-type superchargers.

The screw-type supercharger without timing gear, developed at the FG Fluidenergiemaschinen, is perhaps one step towards making this machine type more attractive, especially for downsizing concepts. The compact design allows simple and fast assembly without any adjustment. Also higher efficiency can be assumed, due to the direct contact between the coated rotors with lower gap areas.

The steady state and the transient operational behaviour of this new machine is calculated with the simulation tool *KaSim*. The program has been enhanced to encompass bearing forces and drive torque, and to analyse effects of geometric and material parameters on the transient performance [4][5].

The next development steps are the verification of the simulation tool (steady state and transient), characteristic diagrams of the new machine, transient behaviour and investigations into the wear characteristics of the coated rotors.

2. Screw-type supercharger and combustion engine

To simulate the interaction of the supercharger and the combustion engine a new model has been created in *KaSim*. The schematic function is shown in Fig. 1. The concept is comparable to the Volkswagen TSI engine, but the roots and the turbo superchargers are replaced by a screw-type supercharger. Air is compressed by the supercharger and flows through the intake valves into the engine after intercooling. At partial load part of the compressed air flows through the bypass valve back to the suction side of the supercharger. The bypass is connected behind the intercooler to avoid high temperatures on the suction side as a result of air flow from the discharge side. At partial load when no boost pressure is required, the supercharger can be switched off by an electro-magnetic clutch.

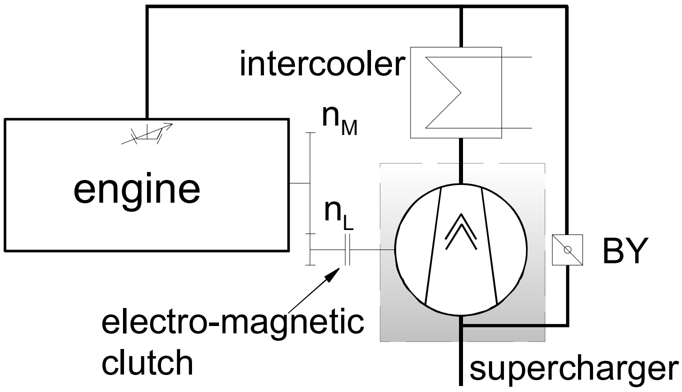

Fig. 1 Schematic function of interworking between supercharger and combustion engine (BY = external bypass)

An existing screw-type supercharger model [4] is enhanced to include an intercooler, an external bypass and the engine (without combustion, only as an ideal reciprocating machine).

To realise a downsizing concept with high boost pressure even at low engine speed a small sized engine model with displaced volume of 1100 cm^2 is connected to the supercharger model. A simulated steady state interaction between supercharger and engine with varying external bypass areas is shown in Figure 2.

The thick black line displays the full throttle boost pressure curve of a small downsized engine. To realise a downsizing concept high boost pressure at low engine speed is necessary. Therefore the design point of the interaction is located at 2000 RPM engine speed. Because the displacement of the supercharger is fixed, only the transmission ratio between supercharger and engine can be varied. With a ratio i = 6.0 this leads to high maximum shaft speeds of the supercharger.

One advantage of the new concept without timing gear are the smaller clearance areas of the profile meshing gap. This is taken into account in transmission ratio above specified. With “standard” synchronised rotors transmission ratio and supercharger shaft speed would be even 10% higher to get the same boost pressure at the design point. This also means approx. 10% more required power at this operation point.

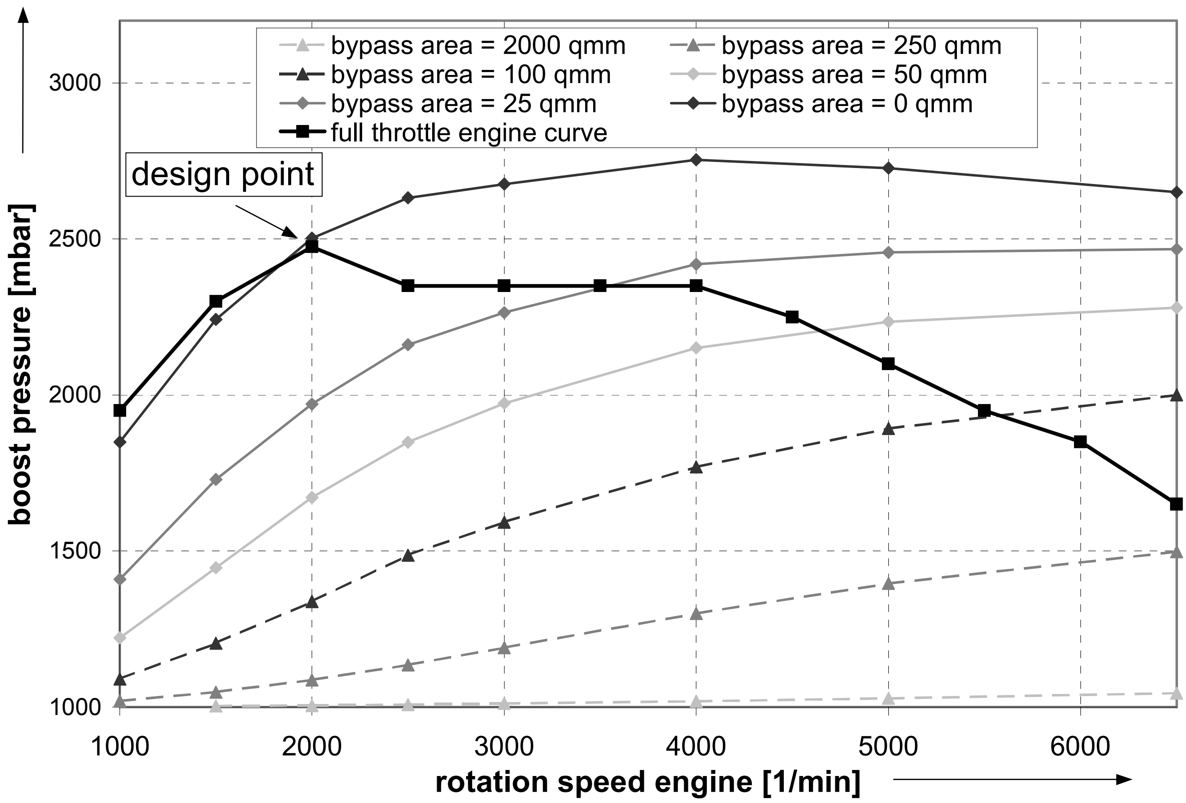

Fig. 2 Characteristic diagram of interaction between screw-type supercharger and engine with varying external bypass area, engine displacement 1100 cm^2, transmission ratio supercharger/engine i = 6.0

Furthermore the diagram reveals a fundamental problem of the interaction in a downsizing concept. Due to the characteristic torque curve of the engine and the high boost pressure at low speeds the air flow at high engine speeds is higher than needed due to the constant transmission ratio. The boost pressure and the air flow must be adjusted by the bypass valve. This is a simple and approved, but inefficient way of adapting the air mass flow.

The most efficient adjustment concept would be a variable speed drive for the supercharger, by means of a variable transmission or an electric motor. Unfortunately both possible solutions are not yet state of the art. A first step in this direction could be a two-stage gear or belt drive between engine and supercharger with a high transmission ratio for the high pressure sector at low speeds and a smaller ratio at higher engine speeds.

3. Friction on seals and bearings

The high supercharger shaft speed leads to an increasing interest in mechanical efficiency. Especially in the partial load energy consumption is increased significantly by friction in seals and bearings. State of the art are radial seals between working chamber and bearing case.

At high shaft speeds the seals cause a high frictional loss. Takei and Takabe measured friction of up to 2 kW on a screw-type supercharger [6].
In the following a model for calculating the friction loss in bearings and a sealing concept with labyrinth seals is presented and discussed.

3.1. Friction model bearings

In the literature and catalogues of bearing manufacturers a calculation model of friction in bearings is commonly used [7]. In this model the total friction moment M_R is classified in a rotation speed depending part M_0 and load depending part M_1:

$$M_R = M_0 + M_1 = f_0 \cdot 160 \cdot d_M^3 \cdot 10^{-7} + f_1 \cdot P_1 \cdot d_M \,.$$

Both parts are strongly influenced by bearing type, oil temperature and oil viscosity. Using a detailed analysis of the design and the ambient conditions, the friction in the bearings of the supercharger can be calculated.

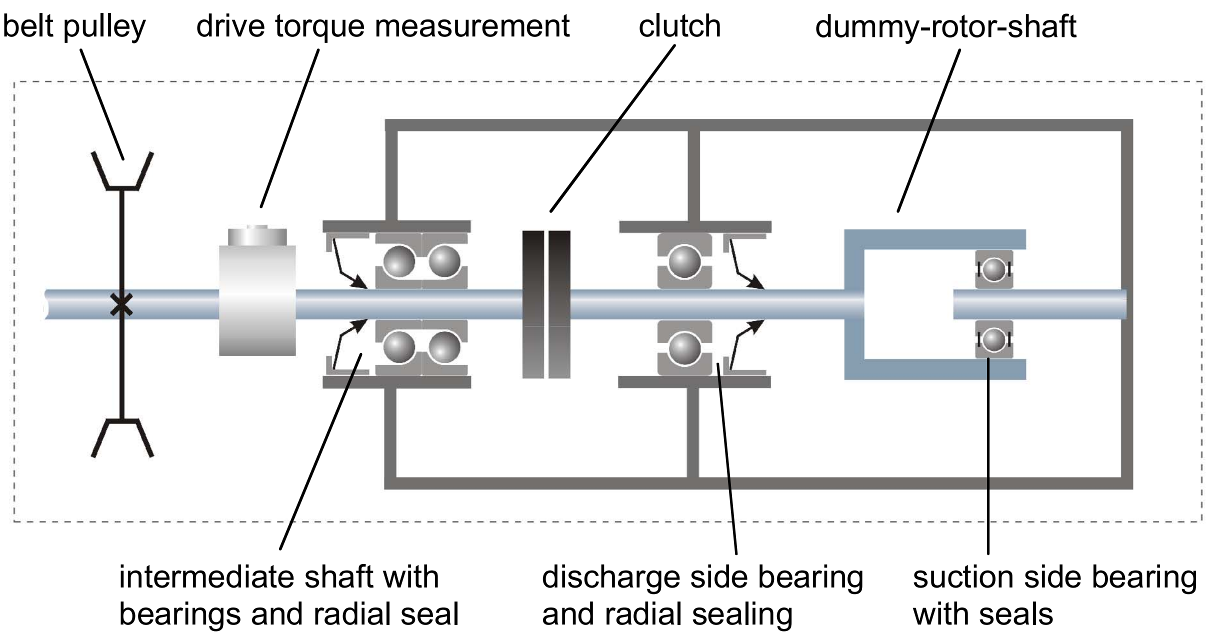

Fig. 3 Schematic setup of drive torque measurement with a dummy-rotor-shaft on the male rotor side (female rotor not driven)

3.2. Friction on radial seals

An abstract model to describe friction in radial seals does not exist in the literature. The influence of the different parameters (type of sealing, contact area, oil lubrication, etc.) on friction seems to be quite strong.
Measurements with a dummy rotor instead of the male rotor in the supercharger (cp. Fig. 3) and radial PTFE seals show a friction loss of the same magnitude as measured by Takei and Takabe [6].

Together the literature and the measurements indicate that the friction in the radial seals dominates the friction in the bearings. This leads to the idea of abandoning radial seals and designing a new sealing concept with labyrinth seals.

3.3 Model of leakage through labyrinth seals

Labyrinth seals offer a number of advantages compared to radial seals [8][9]:

- suitability for high rotor shaft speeds,
- no friction loss, no wear,
- no heat input into the shaft,
- no lubrication required,
- simpler design of the connection facilities.

The disadvantage of labyrinth seals is the gap flow. Because there is no contact between the rotating and stationary part of the seal some leakage is unavoidable. In the case of the screw-type supercharger this leakage is a mass flow from the discharge side into the bearing case. It reduces the air mass flow and volumetric efficiency of the supercharger. It is now necessary to find out if the benefit of reduced friction is higher than the loss in volumetric efficiency.

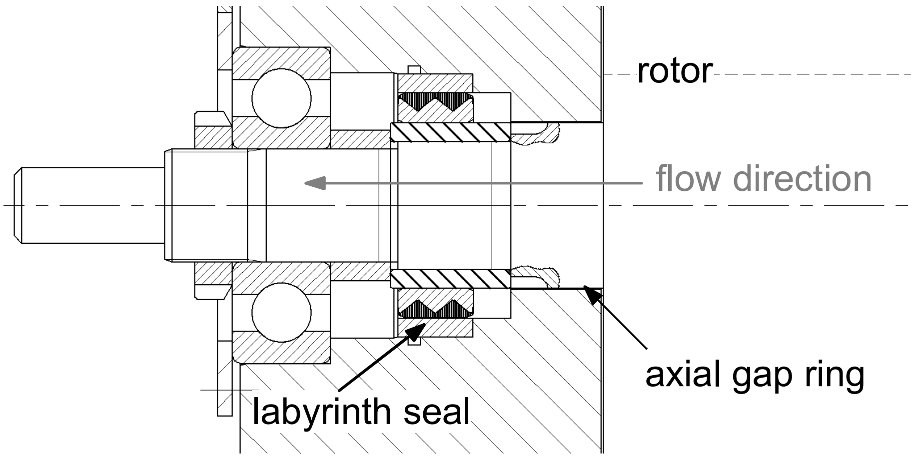

Fig. 4 Geometrical position of labyrinth seal and gap ring in the supercharger

To build up a model of the gap flow, first the leakage path has to be analysed. The radial seals are replaced by a labyrinth seal, so the supercharger now has a labyrinth with a gap ring between rotor shaft and housing, Fig. 4. The leakage at every labyrinth or gap ring is given by:

$$\dot{m} = A_s \cdot \mu \cdot \psi \cdot \sqrt{\rho_0 \cdot p_0} = A_s \cdot \varepsilon \cdot \sqrt{\rho_0 \cdot p_0} \; .$$

To calculate the leakage it is necessary to find either the nondimensional flow rate ε or the flow coefficient μ of each seal.

A model to calculate the combined flow rate of the two seals cannot be found in literature, but the gap ring and the labyrinth can be calculated in isolation.

Egli [10] and Shapiro [11] have presented a calculation method of axial gap rings. This method gives the nondimensional flow rate of axial gap rings depending on geometrical dimensions.

The flow coefficient of the labyrinth can be approximated by a squared labyrinth seal [12]. Besides the geometrical dimensions the number of teeth of the labyrinth influences the flow coefficient.

To calculate the leakage of each seal the pressure between the seals is necessary, but unknown. Certainly the leakage of the combined seals is lower than that of a single seal. So a first estimate of the nondimensional flow rate of the combination can be specified with:

$$\varepsilon_{ges} = 1{,}5 \cdot \varepsilon_{gap} \cdot \varepsilon_{lab} \,.$$

The calculated flow rate is between 2.0 and 5.0 kg/h (depending on the discharge pressure). Over a wide range of the characteristic diagram of the supercharger this means a leakage of less than 3% of the total compressed air mass flow, so the loss in volumetric efficiency seems to be acceptable compared to the higher mechanical efficiency due to reduced friction on labyrinth seals.

3.4. Measurement and model verification

To verify the model for bearing friction a special test application was built, Fig. 3. To measure the friction without any influence from actual compression, a rotor-dummy-shaft was designed. Also labyrinth seals were used so that only the influence of bearings and the performance of these seals could be evaluated. Because there is no timing gear between the rotors only the friction on the male rotor is taken into account.

Fig. 5 illustrates the simulated and measured friction torque and friction of intermediate and rotor-dummy-shaft. The model of the bearing friction matches up with the measurements, in particular above 18000 rpm. At lower speeds a small gap between simulation and measurement can be detected. A possible reason can be found in the temperature dependant viscosity of the grease in the suction side bearing. The temperature is unknown and estimated as a constant parameter. But the viscosity varies extremely with temperature and influences the friction torque.

With the new model verified, a calculation of the overall friction of the supercharger is possible. At 24000 RPM male rotor rotation speed a friction of approx. 500 W is calculated. The measurements and simulation results also demonstrate the advantage of labyrinth seals. The simulation without any friction in sealing components corresponds to the measurement. And compared to the friction measurements of Takei and Takabe the friction could be reduced by more than 50%.

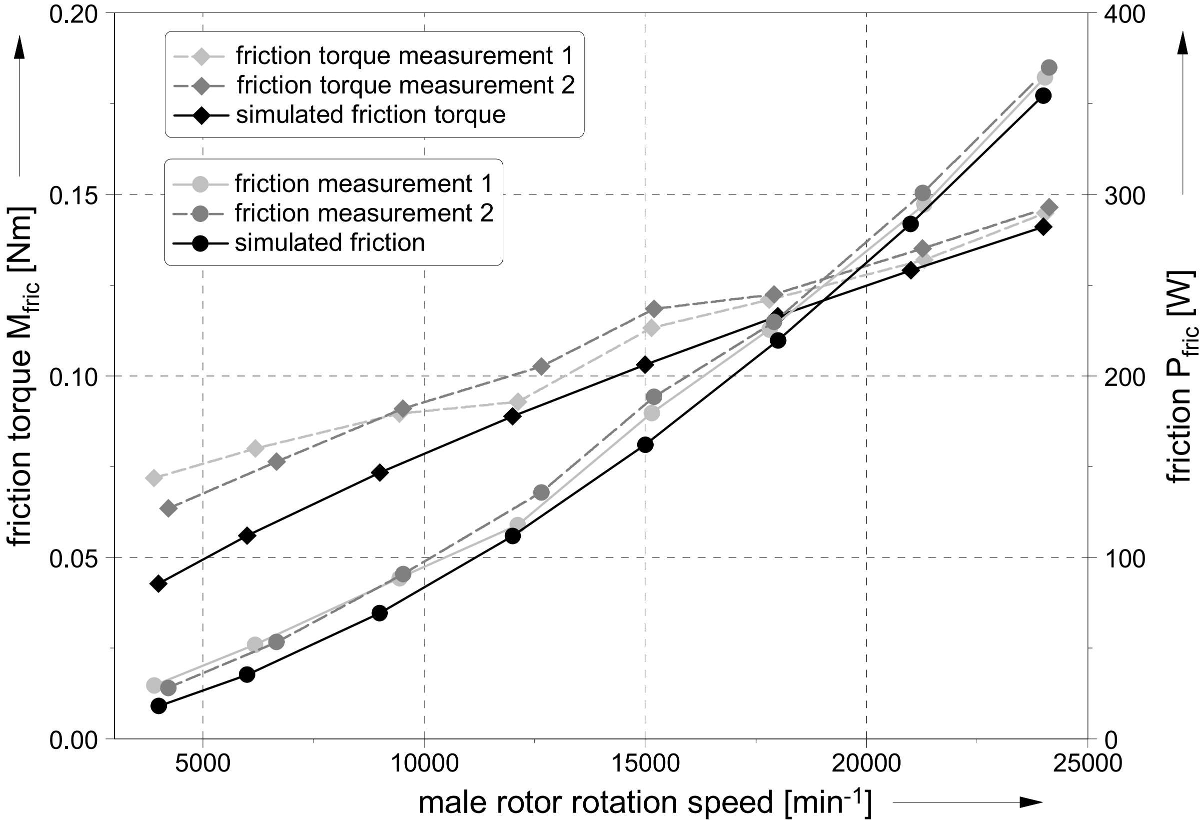

Fig. 5 Measured and calculated friction torque and friction with male rotor-dummy-shaft

But to achieve a higher overall efficiency the low calculated leakage through labyrinth seals also has to be verified. The leakage reduces the volumetric efficiency and influences the delivered air mass flow. With the small simulated leakage a higher overall efficiency can be estimated over a wide range of the characteristic diagram. Unfortunately at low rotation speeds the loss in volumetric efficiency dominates the higher mechanical efficiency. Here a detailed analysis of the overall efficiency in the interaction of screw-type supercharger and engine is necessary. First measurements validate the results of the model, but a detailed measurement and analysis of the leakage is still in progress.

4. Simulation and measurement

4.1. Steady state characteristic diagrams

Fig. 6 describes an adiabatic, steady-state characteristic diagram of the screw-type supercharger without timing gear simulated with the tool *KaSim*. The boundary conditions at intake (p_{in}, T_{in}) depend on experimental setup and measurements. Gap areas in the simulation are measured after assembly with components at ambient temperature. Under operating conditions the gap areas will change due to heating and thermal expansion. Also the friction in bearings is taken into account according to the validated model described above.

The influence of non synchronised rotors due to a reduced profile meshing gap leads to an acceptable volumetric efficiency even at low rotation speeds. At 6000 RPM a pressure ratio of 2.0 can be realised without any thermal overload of the machine parts. In a downsizing engine concept this performance is necessary to get the desired drivability. From medium (12000 to 15000 RPM) to maximum rotation speeds the volumetric efficiency varies only with a small gradient. The time of the operation cycle is shortened, so leakage through the gaps has a decreasing influence.

The positive performance of volumetric efficiency also can be found in the effective isentropic efficiency (= overall efficiency). Over a wide range of the characteristic performance diagram efficiency is higher than 60%. The maximum efficiency is simulated at 21000 RPM and a pressure ratio of 2.2. The theoretical optimal pressure ratio depending on the internal volume ratio of this supercharger is 1.6. But increasing rotation speed leads to throttling of the discharge process and therefore increasing efficient optimal pressure ratios.

This throttling also reduces the efficiency at low pressure ratios and high rotation speeds. Due to an over-compressing process during the working cycle the drive torque and drive power rise. Hence the effective isentropic efficiency decreases with high gradient to lower pressure ratios.

Compared to the simulation results, Fig. 7 shows a measured, steady-state characteristic diagram of the new supercharger. The positive influence of the new rotor profile can also be detected in the measurement. Even at very low rotor speeds pressure ratios from 1.6 up to 2.0 are possible due to the volumetric efficiency. The discharge temperature reaches a stable value after a short time (approx. 10 to 15 minutes).

Analogous to the simulation, maximum efficiency can be found at a higher pressure ratio than expected from the internal volume ratio of the supercharger. In combination with an engine a wide range of the interaction area is located at lower pressure ratios. To reduce power consumption in this part load area and to move the high efficiency area to lower pressure ratios, it will make sense to reduce the internal volume ratio of the supercharger.

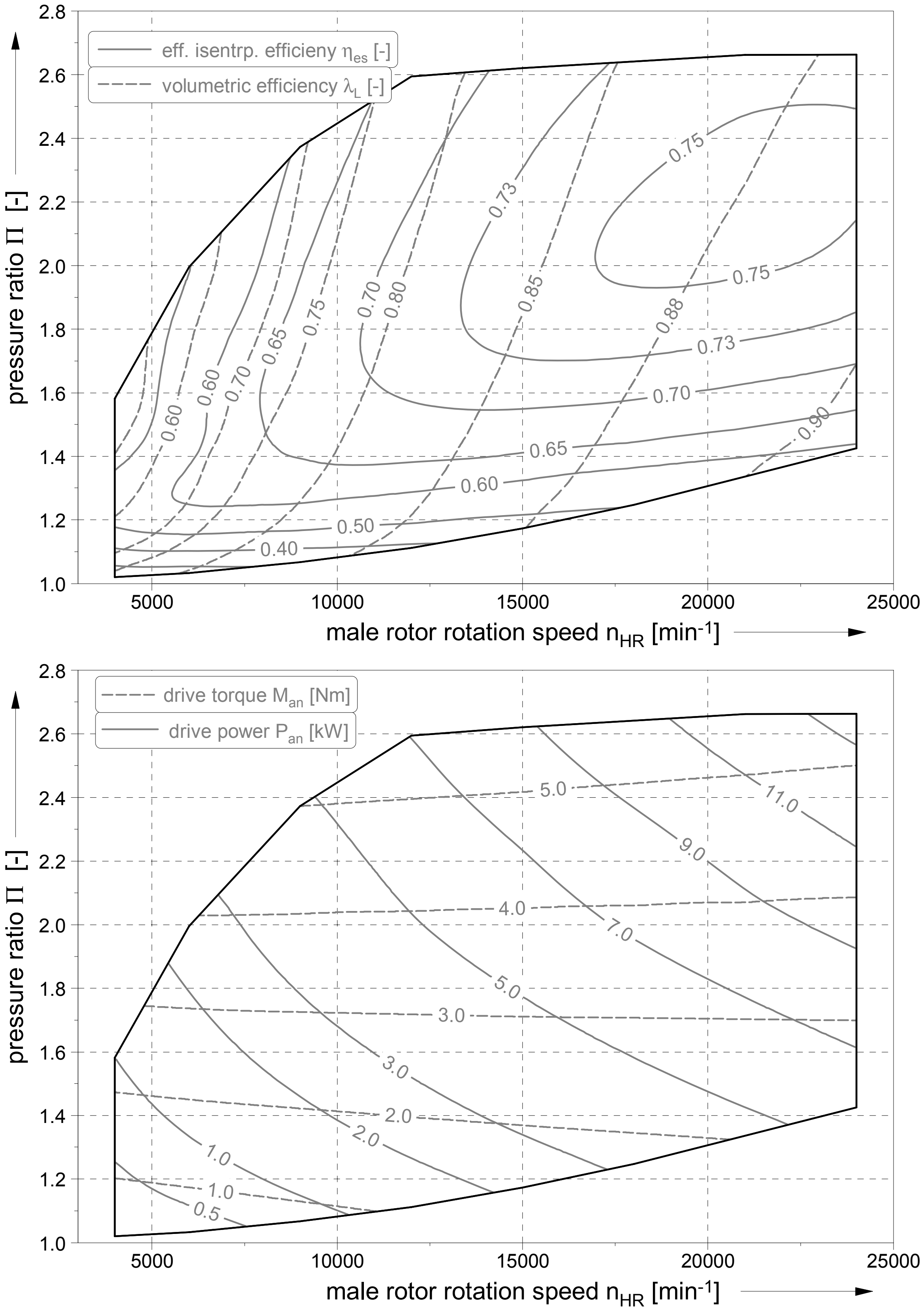

Fig. 6 Simulated adiabatic characteristic diagrams of the screw-type supercharger without timing gear
(p_{in} = 910 – 1010 mbar, T_{in} = 20 – 23°C, housing with internal volume ratio of 1,4)

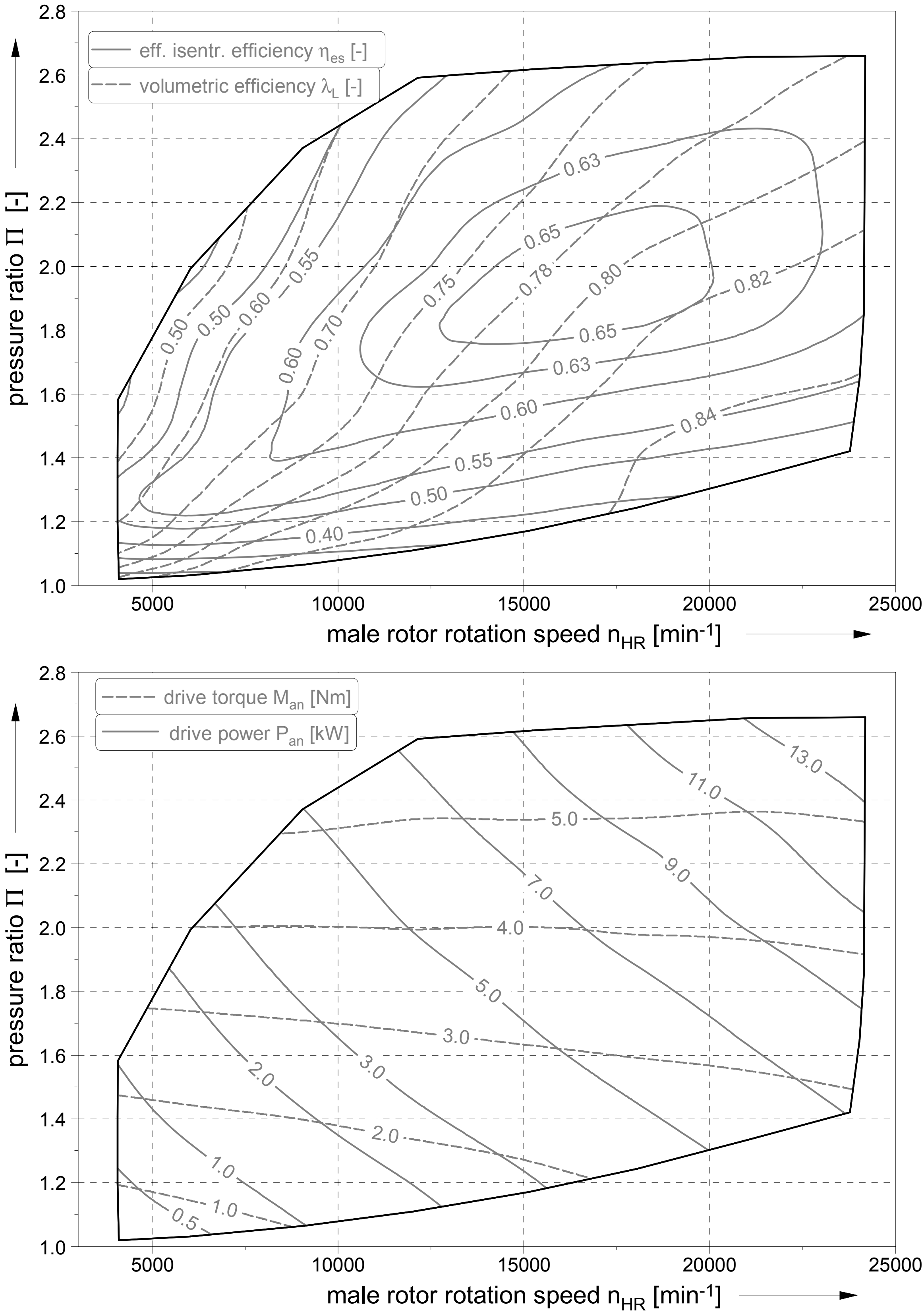

Fig. 7 Measured characteristic diagrams of the screw-type supercharger without timing gear, labyrinth seals mounted at discharge side
(p_{in} = 910 – 1010 mbar, T_{in} = 20 – 23°C, housing with internal volume ratio of 1,4)

At low rotor speeds simulation and measurement also correspond with only small deviation. With increasing rotation speeds simulation and measurement do not correspond more and more. The gradient of volumetric efficiency curves and also the absolute values are higher in the simulation, so the delivered air mass flow of the supercharger also rises.

At the same time the measured drive torque and drive power increase with higher rotation speeds compared to the simulation. In combination, both air mass flow and drive power affect the effective isentropic efficiency. Due to the divergence of these values the simulated efficiency varies from the measured one at high rotor speeds.

A reason for the increasing divergence in air mass flow and drive torque can be found in the intake air flow process of the supercharger. The supercharger possesses only axial intake areas. At high rotor speeds the chamber volume must be filled in a short time; so the flow velocity through the intake area and into the chamber increases. During the filling process this leads to a lower pressure in the chamber compared to the pressure in the suction port. This pressure drop also could be measured with a transient pressure transducer. This effect increases the drive torque resp. the indicated work required for the filling process and reduces the air mass flow.

In the simulation this effect was also detected, but at a lower magnitude. Maybe the conception of the tool *KaSim* reaches its limits at high rotor speeds. The tool *KaSim* is based on a chamber model with homogeneous states of pressure and temperature in the chambers; dynamic effects and varying pressure (e.g. at the inner and crown diameter of the rotor) are not taken into account.

On the other hand the measurement gives only one pressure value at one point in the chamber. To get some idea of dynamic effects in the chamber, measurements with more pressure transducers at different points are necessary.

To recapitulate, the screw-type supercharger without timing gear is a possible solution for a downsized engine concept. The supercharger offers high volumetric and overall efficiency at low speeds. In simulation the efficiency would rise with increasing rotor speeds up to 75%, but this could not be validated by measurements. A possible reason for this behaviour may be the suction process or other dynamic effects, which increase at higher rotation speeds. To analyse these effects more detailed measurements with different housings are planned.

5. Summary and outlook

The results of the steady state simulations and measurements of the new supercharger can be summarised as follows:

- interaction of a screw-type supercharger and a small combustion engine to realise a downsizing concept is possible. To achieve a high part load efficiency new adjustment concepts (electric drive, variable transmission) should be developed;
- reduction of the profile meshing gap due to the non-synchronised rotors has a positive influence on the volumetric and overall efficiency, specially at low rotor speeds. This is an advantage for using the supercharger in a downsizing concept;
- less complexity and number of parts of the supercharger, simpler and faster assembly (no adjustment of timing gear and rotors to set up the profile meshing gap);
- integration of labyrinth seals reduces the friction. This has a positive effect on the overall efficiency at higher rotor speeds. The loss in volumetric efficiency due to leakage through the seal seems to be acceptable, but needs further investigations;
- to get a higher part load efficiency at lower pressure ratios the internal volume ratio of the supercharger should be reduced;
- the intake process seems to have a significant influence on volumetric and overall efficiency at high shaft speeds. It also influences the accuracy of the simulation tool *KaSim* compared to measurements.

Further investigations are planned to achieve a higher accuracy between simulation and measurement. The detailed analysis of the intake process will be another step on the way. To find out more about the influence of the intake process on the thermodynamic behaviour of the supercharger, different housing geometries are being tested. With an additional radial intake area the flow velocity is reduced. This should minimise the pressure drop, if it depends on dynamic effects. Additionally the verification of the transient simulations of *KaSim* will be carried out in the near future.

On the experimental side new bearings will be tested to reach male rotor rotation speeds of 30000 and more. Furthermore, wear on the coated rotors will be tested under long duration conditions to analyse the wear characteristics and verify a lifetime of 5000 h.

Symbols

A	area [m^2]	ε	flow rate [-]
m	mass flow [kg/s]	η	efficiency [-]
M	drive torque [Nm]	λ	volumetric efficiency [-]
n	rotation speed [RPM]	μ	flow coefficient [-]
p	pressure [Pa]	Π	pressure ratio [-]
P	power [W]	ψ	flow rate function [-]
T	temperature [°K]		

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