Screw expanders for small scale cogeneration

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Zusammenfassung

Verdrängermaschinen der Schrauben- und Kolbenbauart sind mit Turbinen kleiner Wellenleistung vergleichbar. Außer für Heizkraftanlagen kleiner Leistung (< 1 MW) werden Kleinturbinen auch industriell genutzt für den Antrieb von verschiedenen Maschinen. Optimale Drehzahl — Wellenleistungsverhältnisse für Schrauben- und Turbinenexpansionsmaschinen sind mit Drehzahl — Wellenleistungs-Kennfeldern für schnellaufende Hochfrequenzgeneratoren verglichen worden, um die Möglichkeit für einen direkten Generatorantrieb zu überprüfen. Allgemeine analytische Modelle sind für Expansionsmaschinen der Verdrängerbauart und insbesondere für Schrauben- und Kolbenmaschinen abgeleitet worden. Verschiedene thermodynamische Prozesse wurden diskutiert, aber dieser Beitrag beruht hauptsächlich auf dem Rankine-Prozess mit Wasserdampf als Arbeitsmedium. Ein Computer-Simulationsprogramm für Schraubenexpansionsmaschinen mit Naßdampf wurde entwickelt. Der Wirkungsgrad (Kraftstoff — Wellenleistung) einer Schraubenmaschine erreicht gerade 15 % - 20 %, was sich zurückführen läßt auf die begrenzte Druckfähigkeit und innere Volumenverhältnisse dieser Maschinen. Dieser Wirkungsgrad ist wesentlich besser als der für gewöhnliche kleine Industrieturbinen, die mit Drehzahlen unterhalb des Drehzahl-optimums betrieben werden. Expansionsmaschinen der Kolbenbauart können einen Wirkungsgrad von 25 % - 30 % bei Teillast erreichen. Dies läßt sich zurückführen auf die größere Druckfähigkeit, größere innere Expansion und kleinere Beeinflußung durch mechanische Verluste. Allerdings gibt es noch viele Konstruktionsschwierigkeiten, die die Lager und Dichtungen sowie die Werkstoffe und die Tribologie betreffen.

Abstract

Positive displacement expanders of screw and piston types have been evaluated and compared performance-wise to small turbines, for the application of small scale (< 1 MW) cogeneration. Optimum shaft speed — power relations for screw and turbo expanders have been compared to the shaft speed — power characteristics for high frequency generators, considering the possibility for direct generator drive. Generalized analytical models for displacement expanders of arbitrary type and for screw and piston expanders in particular have been derived. Different thermal cycles are discussed, but this evaluation has been

concentrated to the steam Rankine cycle. A computer program for simulation of wet steam expansion in twin-screw non-injected expanders, has also been developed. With a screw expander, the over-all steam cycle efficiency (fuel to mechanical power) will be fairly moderate, 15 % - 20 %, due to the limited pressure capability and the limited internal volume ratio of the screw machine. This is however much better than for conventional small industrial turbines, operating at speeds much lower than optimum. Piston expanders have a potential for higher efficiency, 25 % - 30 % at part-load, due to higher pressure capability, a higher degree of internal expansion and less influence of mechanical losses. A number of design problems concerning bearings, seals, materials and tribology remains to be solved, however.

Symbols/Symbole

A	$[m^2]$	area	Fläche
D	$[m]$	rotor diameter	Rotordurchmesser
e	$[J/kg]$	specific internal energy	spezifische innere Energie
h	$[J/kg]$	specific enthalpy	spezifische Enthalpie
m	$[kg]$	fluid mass	Masse
$\dot{m}$	$[kg/s]$	mass flow	Massenstrom
n	$[s^{-1}]$	shaft speed	Drehzahl
P	$[W]$	shaft power	Wellenleistung
p	$[Pa]$	pressure	Druck
$\dot{Q}$	$[W]$	heat flow	Wärmestrom
R	$[J/(kg\ K)]$	gas constant	Gaskonstante
s	$[m]$	blade length	Schaufellänge
T	$[K]$	thermodynamic tempe-rature	thermodynamische Tempe-ratur
u	$[m/s]$	peripheral speed (tip speed)	Umfangsgeschwindigkeit
V	$[m^3]$	displacement (volume/revolution)	Verdrängervolumen (Volumen pro Umdrehung)
V/D^3	$[-]$	specific displacement	spezifisches Verdränger-volumen
$\dot{V}$	$[m^3/s]$	volume flow	Volumenstrom
$\dot{W}$	$[W]$	mechanical power	Leistung
w	$[J/kg]$	specific work	spezifische Arbeit

β	[rad]	male rotor angle of rotation	Rotationswinkel der Hauptläufer
Δ	[-]	difference	Differenz
ε	[-]	internal volume ratio	inneres Volumenverhältnis
η	[–]	efficiency	Wirkungsgrad
κ	[–]	isentropic exponent	Isentropenexponent
Π	[–]	pressure ratio	Druckverhältnis
ρ	[kg/m³]	local gas density	Dichte
σ	[–]	specific speed	spezifiche Drehzahl
ϕ	[--]	flow coefficient	Durchflußzahl
ψ	[–]	pressure coefficient	Druckzahl
ω	[rad/s]	angular velocity	Winkelgeschwindigkeit

Subscripts/Indizes

boiler	boiler	Dampfkessel
cond	condenser	Kondensator
e	control volume exit	Auslaß, Arbeitsraum
i	control volume inlet	Eintritt, Arbeitsraum
in	expander inlet	Eintritt, Expansionsmashine
m	mechanical	mechanisch
out	expander outlet	Auslaß, Expansionsmashine
pump	feed pump	Speisepumpe
R	Rankine	Rankine
s	isentropic	isentrop

1 Small scale cogeneration, state-of-the-art

There is no well-defined upper limit for small-scale cogeneration. In this paper, the approximate power range $50\ kW_e$ — $1\ MW_e$ has been considered. Small back pressure steam turbines have been used since very long for local generation of electricity and for mechanical drive of pumps, compressors, fans and other equipment, in process industries utilizing steam for process heating. An increased use of small scale cogeneration could now be foreseen, e.g. for local district heating and buildings using bio-fuel. Other applications for small expanders are bottoming cycles for utilization of exhaust heat from gas turbines or diesel engines, solar power and geothermal plants, as well as pressure reduction units in gas pipelines. Turbo expanders, screw expanders and to some extent also diesel or otto engine "total energy packages" compete within many of these applications. Piston type expanders for high-pressure steam have an interesting potential, but a number of qualified design problems remains to be solved.

The main reason for discussing alternatives to small turbo expanders, is the very poor efficiency for present types of industrial turbines, operating at shaft speeds significantly lower than optimum. The screw expander has a more favourable and rugged mechanical design, as well as a lower optimum speed. It is tolerant to wet fluids. With the ongoing introduction of high-speed electrical drives and generators based on power electronics, the screw machine has a very interesting potential for simplified direct drives. The screw machine power — speed relation is well adapted to that of the high-speed electrical drives. Being a clearance-sealed machine, the screw expander is however not feasible for applications with large speed variations. At low speed, the screw machine efficiency will be poor due to the increased influence of internal leakage. Internal combustion engines in stationary base-load applications have the disadvantage of frequent need for maintenance.

2 Comparison of displacement and turbo expanders

All expansion processes for power generation could in practice be considered adiabatic, i.e. the heat transfer will be negligable. The specific work will then be equal to the enthalpy drop. The enthalpy drop is defined by the application, i.e. admission conditions and condenser pressure. For volumetric machines, the specific work and enthalpy drop will also be influenced by the degree of internal expansion (section 4.1 in this paper). For a rough comparison of turbo and displacement expanders, the following equations could be used, independent of expander type. Thermodynamic losses related to throttling, leakage and heat tranfer, as well as mechanical losses, have been neglected.
Specific work, per unit mass of working fluid:

$$w = \Delta h \tag{1}$$

Shaft power output

$$P = \dot{m} \cdot \Delta h \tag{2}$$

Expander dimensions will usually be constrained by the exit conditions (high specific volume of working fluid). The massflow should then be expressed in terms of volume flow at expander exit condition. Due to the possibility for incomplete internal expansion in displacement machines, the exit condition is not nescessarily the same as the down-stream (condenser) condition.

$$\dot{m} = \dot{V}_{out} \cdot \rho_{out} \tag{3}$$

Rotor speed — shaft speed relation:

$$u = \pi \cdot D \cdot n \tag{4}$$

2.1 Displacement expanders

The exit volume flow is proportional to the expander displacement and the shaft speed:

$$\dot{V}_{out} = V \cdot n \tag{5}$$

A class of geometrically similar displacement machines could be characterized by their spcific displacement V/D^3 /7/. For displacement machines, the enthalpy drop will be virtually independent of the tip speed. The tip speed will have an optimum value for each machine geometry and application. The tip speed is hence a characteristic parameter, /7/. The optimal tip speed for screw expanders is also different for oil-injected and dry (non-injected) screw machines. In the following, the analysis will be restricted to dry expanders. The following explicit expression for shaft speed vs´ shaft power will be obtained from the previous equations:

$$n = \sqrt{\frac{1}{\pi^3} \cdot \left(\frac{V}{D^3}\right) \cdot \frac{u^3 \cdot \rho_{out} \cdot \Delta h}{P}} \propto \sqrt{\left(\frac{V}{D^3}\right) \cdot \frac{u^3 \cdot \rho_{out} \cdot \Delta h}{P}} \tag{6}$$

2.2 Turbo expanders

For turbo expanders, the well established non-dimensional performance parameters /2/, /11/ for characterization of a geometrically similar class of turbo machines could be used.

Pressure coefficient:

$$\psi = \frac{\Delta h}{u^2/2} \tag{7}$$

Flow coefficient:

$$\phi = \frac{\dot{V}}{u \cdot A} \tag{8}$$

The cross sectional reference area could be expressed in terms of outer diameter and blade length:

$$A = \pi \cdot D^2 \cdot \frac{s}{D} \cdot (1 - \frac{s}{D}) \tag{9}$$

Dimensionless specific speed:

$$\sigma = 2^{1/4} \cdot \pi^{1/2} \cdot \frac{n \cdot \dot{V}^{1/2}}{\Delta h^{3/4}} = \frac{\phi^{1/2}}{\psi^{3/4}} \tag{10}$$

For turbo machines, the tip speed is strongly dependent on the enthalpy drop. The tip speed should then be expressed in terms of pressure coefficient and enthalpy drop, according to eq. (7). Utilizing theese equations, the following explicit expression for shaft speed vs´ shaft power for turbo expanders will be obtained:

$$n = \frac{2^{3/4}}{\pi^{1/2}} \cdot \sigma \cdot \sqrt{\frac{s}{D} \cdot (1 - \frac{s}{D}) \cdot \frac{\rho_{out} \cdot \Delta h^{5/2}}{P}} \propto \sigma \cdot \sqrt{\frac{\rho_{out} \cdot \Delta h^{5/2}}{P}} \tag{11}$$

2.3 Shaft speed vs´ power for screw and turbo expanders

For comparison of speed — power characteristics, the Curtis impulse type turbine has been taken as a representative example. Expansion of superheated steam of 20 bar, 480 Celcius, to 1 bar condenser pressure has been assumed. For the screw, an optimum tip speed of 120 m/s, an internal volume ratio of 5.3 and a specific displacement of 0.5 has been assumed. For the 2-row Curtis turbine, a specific speed of $\sigma \approx 9$ is representative. The left diagram in fig. 1 illustrates the speed — power relations for this screw expander and turbine. Despite anticipating data for a specific cycle and for specific machine types, the conclusion concerning speed — power relations should be fairly general. For other types of displacement expanders the shaft speed will be lower than for the screw.

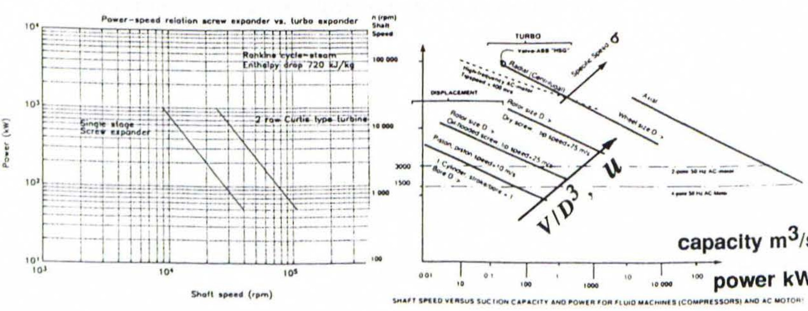


Fig 1 Comparison of power-shaft speed relations for displacement and turbo expanders

Bild 1 Vergleich der Kennfelder von Drehzahl-Leistung für Expansionsmaschinen von Verdränger- bzw. Turbobauart

2.4 High-speed generators

In the low power range, turbo as well as screw expanders should be operated at a shaft speed considerably higher than for normal AC-generators of 4- or 2-pole type (1500, 3000 rpm) at 50 Hz mains frequency. With the development of power electronics and static frequency inverters, high-speed generators have been introduced during the recent years /4/, /5/. These AC generators operate at higher frequency and shaft speed. A solid state inverter will be used to modify the generator output frequency to the 50 Hz (or 60 Hz) mains frequency.

With direct drive of a high-speed generator, the generator gets smaller and the reduction gear can be omitted. A representative shaft speed — power relation for high-speed generators (and motors) of different design /4/ is shown in fig. 2.

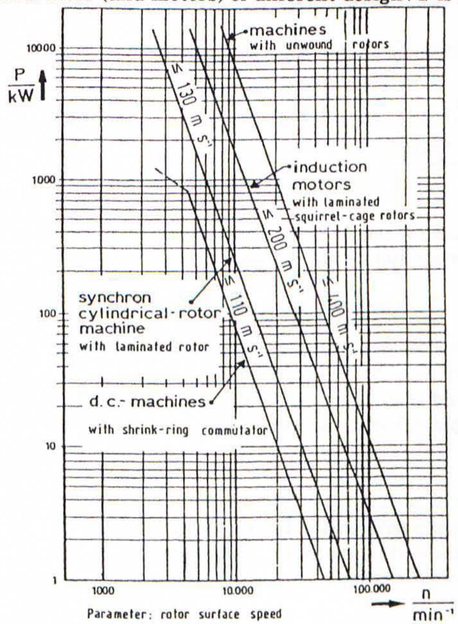


Fig. 2 Power vs´ shaft speed for different types of high-speed generators /4/
Bild 2 Kennfelder Drehzahl-Leistung für schnellaufende Generatoren verschiedener Bauart /4/

These generator characteristics should be compared with the corresponding speed-power relations for screw and turbo expanders. Generators of induction (asynchronous) type with unwound rotors are still at the speed limit for direct drive with turbines. With screw expanders, generator performance would be feasible for direct drive. However, the Swedish gas turbine / PM-generator (with a high strength NdFeB rotor) for an automotive hybrid system should be mentioned /5/. Nominal data for this unit are 40 kW at 100 000 rpm. Gas turbine and generator have been integrated into a common high-speed unit.

3 Thermal cycles

The screw expander could be utilized in various thermal cycles /1/, /3/. ORC-cycles for low temperature heat recovery and for solar plants have been studied theoretically as well as experimentally since the energy-crisis during the seventies. Expansion of high-temperature gas in a Brayton cycle would be very interesting, but high-temperature materials, i.e. ceramics must then be used. Problems to be solved for the high-temperature screw expander application are manufacturing of ceramic screw rotors, resistance to brittle fracture, thermal distorsion and heat losses. The following of this paper has been concentrated to expansion of steam in a non-injected screw expander.

3.1 Rankine cycle (water vapour)

The total thermodynamic efficiency for the Rankine cycle is defined as follows:

$$\eta_{tot} = \frac{\eta_s \cdot \eta_m \cdot \Delta h_s - \left(\frac{\Delta p}{\rho}\right)_{pump}}{\Delta h_{boiler}} \qquad (12)$$

For steam, when the feed pump accounts for a minor share of the total mechanical power, the following approximation is justified:

$$\eta_R = \frac{\Delta h_s}{\Delta h_{boiler}} \qquad (13)$$

The total cycle efficiency could then be approximated by the product of Rankine efficiency, expander isentropic efficiency and expander mechanical efficiency:

$$\eta_{tot} \approx \eta_R \cdot \eta_s \cdot \eta_m \qquad (14)$$

The Rankine efficiency for steam is illustrated in fig. 3, vs´ admission temperature and admission pressure. It is clearly seen that high pressure will be required to obtain an acceptable thermal efficiency.

4 Screw expander performance

4.1 Analytical model for rough performance evaluation

For adiabatic rotary displacement expanders of arbitrary type, with a varying degree of internal expansion, the following expression for specific work vs´ external pressure ratio and built-in volume ratio can be derived /10/. A perfect gas has been assumed.

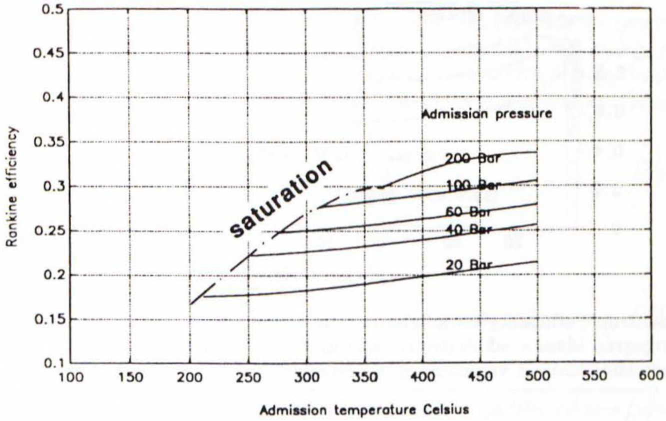


Fig. 3 Rankine efficiency. Steam, condenser temperature 120^0 Celcius
Bild. 3 Wirkungsgrad für den Rankine-Process, mit Wasserdampf und einer Kondensationstemperatur von 120^0 Celcius

$$w = |\Delta h| = R \cdot T_{in} \cdot \left[1 + \frac{1}{1-\kappa} \cdot (\varepsilon^{1-\kappa} - 1) - \frac{\varepsilon}{\Pi} \right] \tag{15}$$

For the case of complete internal expansion, the specific work will be equal to the isentropic enthalpy drop defined by pressure ratio and inlet temperature:

$$w_s = \Delta h_s = R \cdot T_{in} \cdot \frac{\kappa}{\kappa - 1} \cdot (1 - \Pi^{\frac{1-\kappa}{\kappa}}) \tag{16}$$

The expander isentropic efficiency considering degree of internal expansion, but neglecting throttling, leakage and heat losses, can then be calculated as (fig. 4):

$$\eta_s = \frac{\Delta h}{\Delta h_s} \tag{17}$$

For an expander, the specific power related to machine dimensions, tip speed and inlet pressure, will be strongly dependent on the built-in internal volume ratio as well as on the external pressure ratio:

$$P / \left[p_{in} \cdot \frac{u}{\pi} \cdot (\frac{V}{D^3}) \cdot D^2 \right] = \frac{1}{\varepsilon} \cdot \left[1 + \frac{1}{1-\kappa} \cdot (\varepsilon^{1-\kappa} - 1) - \frac{\varepsilon}{\Pi} \right] \tag{18}$$

With increasing internal expansion, the inlet port area and hence the massflow and output power will be reduced. This is illustrated in fig. 5:

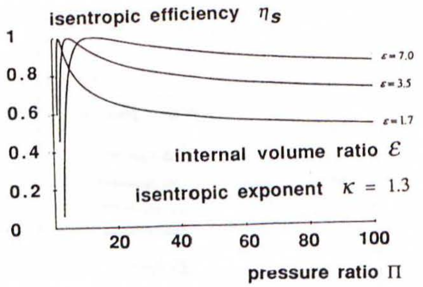


Fig. 4 Ideal isentropic efficiency for adiabatic rotary expanders
Bild 4 Wirkungsgrad idealer adiabater schadraumfreie Drehkolben-Expansionsmaschinen verschiedener Bauarten

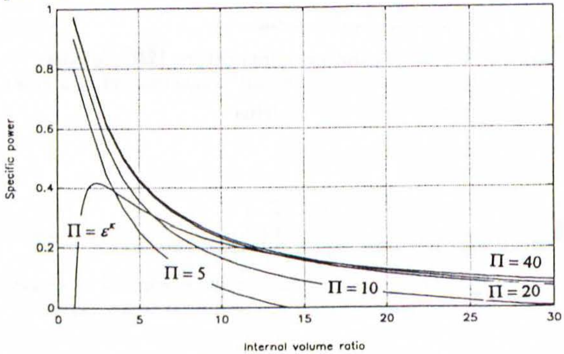


Fig. 5 Specific power according to eq. 18, for ideal adiabatic rotary expanders
Bild 5 Spezifische Leistung idealer adiabater Drehkolben-Expansionsmaschinen verschiedener Bauarten

4.2 Computer simulation of steam screw expanders

In order to account for speed dependent losses and real steam properties, a computer simulation program for non-injected screw expanders operating in steam cycles was developed /10/. The conventional quasi-sationary approach for simulation of volumetric fluid machines /6/, /1/, /3/, utilizing equations for stationary one-dimensional isentropic flow in leakage clearances and ports for each time step, was applied. Geometry data (volume curve, port areas and leakage path legths vs´ male rotor angle) were generated from a separate mathematical model of the screw machine geometry. In fig. 6, the control volume (one working chamber) and related mass and energy flows are indicated. By conservation of mass and by application of the first law of thermodynamics for energy conservation, the ordinary differential equations for integration of control volume mass (density) and internal energy can be derived. All masses entering or leaving the control volume (the working chamber studied) will be represented by their enthalpy h (including work for change of volume) in the energy equation, while working fluid within the control volume is represented by the internal energy e. The first law for energy conservation, in non-stationary formulation:

$$\dot{Q} = \sum \dot{m}_e \cdot h_e - \sum \dot{m}_i \cdot h_i + \frac{d}{dt}(m \cdot e) + \dot{W} \tag{19}$$

Conservation of mass:

$$\frac{dm}{dt} = \sum (\dot{m}_i - \dot{m}_e) \tag{20}$$

Combining these equations and utilizing geometry data (the volume curve) yields:

$$\frac{de}{dt} = \frac{1}{m} \cdot [\dot{Q} + \sum (\dot{m}_i \cdot h_i - \dot{m}_e \cdot h_e) - e \cdot \sum (\dot{m}_i - \dot{m}_e) - \omega \cdot \rho \cdot \frac{dV}{d\beta}] \tag{21}$$

Fluid density will be determined by mass and chamber volume:

$$\rho = m / V(\beta) \tag{22}$$

From numerical integration of equations (19), (20), and from equation (22), specific internal energy e and fluid density ρ of the working fluid in the control volume will be obtained for each time step. These two state variables will then be used for evaluation of all other thermodynamic state variables. Within or near the wet region, the perfect gas approximation is not justified for steam. Polynomial fits for steam data are available only with pressure and temperatutre as arguments. In simulation of volumetric machines however, the reverse computation of pressure, temperature and steam quality from density and internal energy will be required. Interpolation in a computerized steam table with internal energy and density as arguments, from /9/, was then considered to be the most efficient solution, to minimize computation effort.

A predictor-corrector method of integration was used. For programs where a large number of production runs is expected, it can be useful to adapt the integration time step according to the typical time scales for different phases of the working cycle, in order to minimize execution time, which was not implemented in this case. For screw machines, with a number of phase-shifted working chambers interacting by leakage, as indicated in fig. 7, the simulation must be iterated, however. For the first cycle, initial state variable distributions vs´ rotor angle (or time), approximated by data for an ideal irreversible process, must be calculated. Algebraic equations for isentropic change of state,

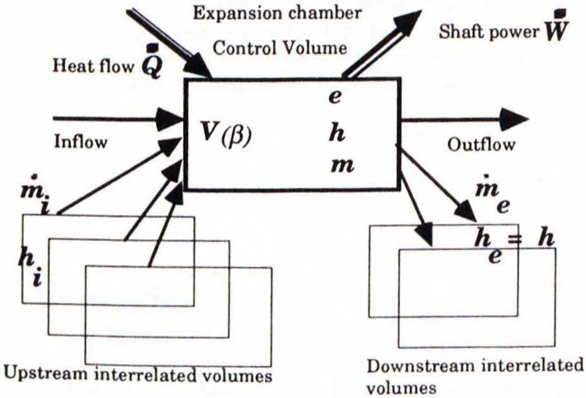


Fig. 6 Lumped parameter model of expansion volume
Bild 6 Berechnungsmodelle des Arbeitsraums mit zusammenwirkender Kammer verschiedener Drücke

$pV^{\kappa} = const.$, can then be used for initialization of state vectors. After simulation of the the first cycle, these distributions are updated by the simulation results and the process will normally converge after a few iterations. An acceptable convergence was obtained after 4 - 8 cycles.

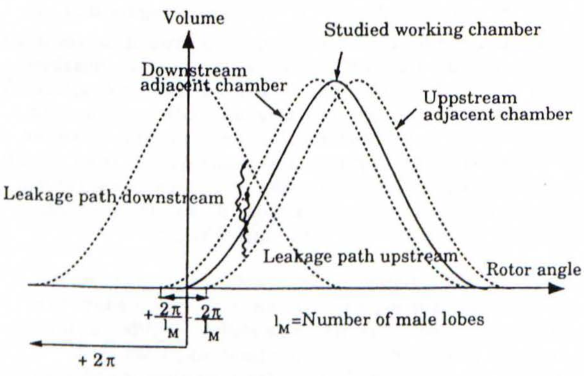


Fig. 7 Interaction between the studied working chamber and its adjacent working chambers
Bild 7 Beziehung zwischen den untersuchten und den angrenzenden Arbeitsräumen

One common type of numerical problem is the initial filling of a chamber volume starting from zero volume. During the first time steps, the additional mass due to inflow will be of the same order of magnitude as the accumulated mass in the chamber, and stability problems will usually occur. One solution is to calculate initial filling from chamber volume increase and upstream fluid conditions, instead of using the differential equations for pressure driven flow.

For this steam expander simulation program, heat transfer to rotors and casing has been neglected. This is an acceptable approximation due to the high speed and short cycle time in a high-speed rotary machine /8/. In the case of low-speed operation however, heat tranfer must be considered especially for steam, as there is a risk for loss of pressure due to condensation at the casing and rotor surfaces. Some problems are related to the handling of heat transfer:
One problem is how to estimate steady-state rotor and casing temperature distributions. The time constant for heating of solid machine components is normally several orders of magnitude larger than the cycle time. The author has previously used a separate simplified model for stationary heat tranfer from hot gas sections to machine parts and from machine parts to cold gas sections, including heat conduction between hot and cold ends of the machine parts. The stationary heat balance gives an equation system for the unknown temperatures. Another approach is to extrapolate machine part temperatures calculated during simulation of a number of cycles.
Determination of accurate correlations for the heat transfer coefficient (or Nusselt number, Nu(Re, Pr)) is another problem due to the complex geometry and local flow pattern.

4.3 Results, accuracy, need for experimental validation

Some simulation results have been presented in fig. 8. The "indicated efficiency" (excluding mechanical losses) was found to be fairly low, do to the limited pressure capability of screw machines. However, the efficiency of roughly 20 % is still much higher than for conventional small industrial steam turbines, operating at a shaft speed much less than optimum. For small commercial industrial steam turbines, the isentropic efficiency will usually be as low as 35 % - 40 %, giving an indicated efficiency of roughly 7 % -10 % only.

Results for various design parameter values and operational data for screw expanders were compared. Simulation results can usually be used directly for sensitivity analysis with respect to design parameters and for comparison of alternative solutions. If reliable predictions of absoute values e.g. for efficiencies are required however, the simulation model must be adjusted according to test results. Some parameters with a high level of uncertainty are throttling factors or contraction coefficients in the equations for one-dimensional nozzle flow /1/. Other uncertainties are actual rotor clearances at operating temperature considering thermal distorsions, and coefficients of heat transfer.

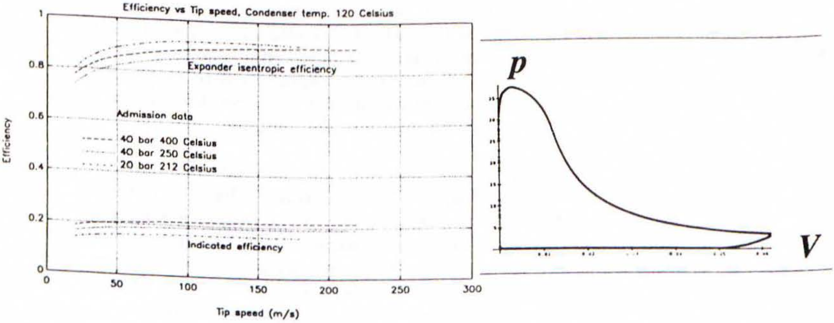


Fig. 8 Simulation results for a steam screw expander
Bild 8 Simulationsresultate für Wassswerdampf-Schrauben-Expansionsmaschinen

4.4 Future possibilities for integration of simulations to CAD

Presently there is a strong tendency in engineering design towards a higher degree of integration of various calculation programs and simulations to CAD. Separate geometry definition for various calculation purpuses can then be avoided, the user-interface will be improved and the simulation work could be done more interactively, with geometry definition and modification at the CAD screen. For the screw machine concept, an approach has been taken to generate the geometrical data required for simulation interactively by use of solid CAD-models (helical extrusions) from the rotor cross-section profiles. However, considering errors due to the spline representation of surfaces and intersections, truncation errors etc, the accuracy in rotor intersection curves (i.e. rotor mesh sealing lines) obtained is not yet (with present solid modellers) sufficient as compared to data generated directly from a mathematical geometry model with analytical expressions. A printout from the CAD-screen of rotor solids and rotor mesh contact line projections is presented in fig. 9.

5 Mechanical design and constraints for steam expanders

5.1 Screw expanders

Based on a study of bearing loads /10/, the maximum admission pressure was set to 40 bar. To increase the thermodynamic efficiency, an increased admission pressure would be required according to fig. 3. However, due to the geometry restrictions for the internal expansion ratio of a screw machine, an increased admission pressure would anyhow not give any significant improvement of the thermodynamic efficiency. For the same reason, a lower condenser temperature could not be utilized to improve the efficiency.

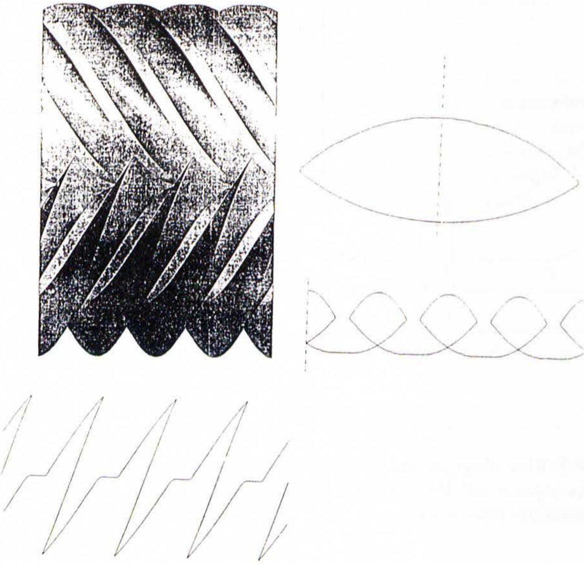
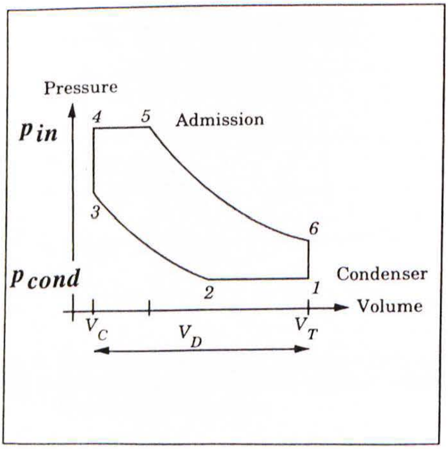


Fig. 9 CAD-geometry: solid rotors and rotor mesh contact line for a symmetrical point-generated profile without blow-hole
Bild 9 CAD-darstellung: Rotoren und Eingriffslinie für ein symmetrisches punkterzeugtes Schraubenprofil ohne Blasloch

With a high pressure difference over a screw machine stage, oil-lubricated hydrodynamic journal bearings will be required for the radial loads. The screw unit should be designed for maximum possible balancing of the axial load component. The mechanical loss for radial journal bearings was estimated to roughly 4 % for a 200 mm rotor and to less than 1 % for an 850 mm rotor, with an obvious scale influence.

5.2 Piston expanders, thermodynamics and mechanical design

In /10/, a theoretical study of piston expanders was also included. Piston expanders could take higher bearing loads and should then be capable of much higher admission pressure. A compact piston machine of axial piston type has been considered. With a small clearance volume and variable filling, the internal expansion ratio at part-load could be much higher than for a screw expander. With variable inlet and outlet valves and a cylinder clearance volume according to the definitions in fig. 10, the idealized adiabatic enthalpy drop for a piston expander has been derived, eq. 23. Dynamic losses have been neglected and a perfect gas has been assumed.



Filling factor $\Phi = \frac{V_5}{V_4}$

Recompression ratio $\chi = \frac{V_2}{V_3}$

Compression ratio $\varepsilon = \frac{V_6}{V_4}$

Cut-off ratio: $\varphi = \frac{\Phi - 1}{\varepsilon - 1}$

Pressure ratio $\Pi = p_{in}/p_{cond}$

Clearance volume $V_C = V_4$

Fig. 10 Idealized indicator diagram and definitions for a piston expander
Bild 10 Indikatordiagramm mit Definitionen für eine ideale adiabate Kolbenexpansionsmaschine

$$\Delta h = R \cdot T_{in} \cdot \left[\frac{\Pi \cdot \left[\frac{\Phi}{\kappa - 1} \cdot \left(1 - \left(\frac{\Phi}{\varepsilon}\right)^{\kappa - 1} + (\Phi - 1) \right) \right] - \frac{\chi}{\kappa - 1} \cdot (\chi^{\kappa - 1} - 1) - (\varepsilon - \chi)}{\frac{\Pi - \chi^{\kappa}}{\kappa} + \Pi \cdot (\Phi - 1)} \right] \tag{23}$$

An analytical expression for the increased end temperature after filling of the clearance volume from the admission line has also been derived. Inflow of the additional mass entering the clearance volume at top dead centre when opening the inlet valve and mixing with the recompressed gas in the clearance volume, has been assumed to take place instantaneously, at constant volume. A perfect gas has been assumed.

$$T_4 = T_{in} \cdot \frac{\kappa}{1 + \left(\kappa \cdot \chi \cdot \frac{T_{in}}{T_{cond}} - \chi^{\kappa} \right) \cdot \frac{1}{\Pi}} \tag{24}$$

Note, that for an infinite pressure ratio, i.e. initially vacuum in the clearance volume, the mixing temperature T_4 will be equal to κT_{in} , that is considerably higher than the admission temperature T_{in}. The explanation will be the work of volume change performed on the gas in the clearance volume, by the admission pressure.

Due to a high cylinder mean pressure, the influence of mechanical losses should also be reduced, as compared to conventional piston engines. The piston expander then has a potential for high efficiency, particularly at part-load with a high degree of internal expansion. For expansion of steam, with 200 bar, 500^0 Celcius admission, a total efficiency of roughly 25 % - 30 % might be possible at part-load. However, several very qualified design problems remain to be solved, e.g. concerning tribology and lubrication, valve design, materials selection etc. For the near future, screw expanders then represent the best alternative for small-scale expanders, as compared to both turbines and piston expanders.

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