Power Generation Subsea – Calculation of the Energy Conversion by Multiphase Twin-Screw Motors

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ABSTRACT

Today, surplus energy in the deposits of high pressure oil and gas reservoirs is controlled by chokes at the wellhead. In this way, the energy is dissipated by throttling the high-pressure stream. For years, twin-screw multiphase pumps (MPPs) are used to convey oil and gas by adding energy to the well-stream (1). Contrary, the mentioned technology may be applied to generate electrical power from the surplus energy in the well-stream by employing MPPs as a multiphase twin-screw motor (MTM). In order to predict the potential of this technology, the first and second law of thermodynamics are regarded and further theoretical calculations are carried out. The results of the mathematical simulation are compared to first experimental data.

NOMENCLATURE

General symbols

c	heat capacity, kJ/(kg K)
d	diameter, m
E	energy, J
F	force, N
g	gravitational acceleration, m/s²
h	specific enthalpie, kJ/kg
m	mass, kg
MPP	multiphase pump
MTM	multiphase twin-screw motor
M	torque, Nm
n	rotational speed, 1/min
P	power, W
p	pressure, bar
$\dot{Q}$	heat flow, W
s	specific entropy, kJ/(kg K)
T	temperature, K
t	time, s
u	specific internal energy, J/kg
$\dot{V}$	volume flow, m^3/h
v	specific volume, m^3/kg
w	velocity, m/s
z	geodetic altitude, m
α	gas volume fraction, -
β	average pitch angele, °
ω	angular speed, 1/s
η	efficiency, -

Subscripts

1	status 1
2	status 2
amb	ambient
ax	axial
dis	dissipated
g	gas
i	component i
in	inlet

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irr	irreversible
isen	isentropic
iso	isothermal
l	liquid
loss	loss
out	outlet
p	isobar
rev	reversible
s	shaft
sys	system
tan	tangential
theo	theoretical
vol	volumetric

1 INTRODUCTION - MOTIVATION

The production of oil and gas fields is divided in several stages characterised by permanently varying properties of the reservoir. Initially, the static pressure of the massif on the oil and gas solved in the porous rock is very high (> several 100 bar). The deposit pressure is significantly higher than the bottomhole pressure inside the wellbore. Therefore, in the primary recovery stage the multiphase mixture of liquid and gaseous hydrocarbons is displaced to the surface driven only by the natural pressure difference. Because of the high pressures and the corresponding energy level of the fluid the pipeline system has to be protected by a choke in order to reduce the pressure at the wellhead (2), see figure 1. Unfortunately, this is accompanied by dissipating the total pressure energy of the fluid.

The idea of this article is to make use of the dissipated energy. By assembling a multiphase twin-screw motor (MTM) at the wellhead, dissipation of energy in the choke may be avoided, see figure 2. A generator is powered by the MTM and, like this, the mechanical energy is converted in electrical energy (3).

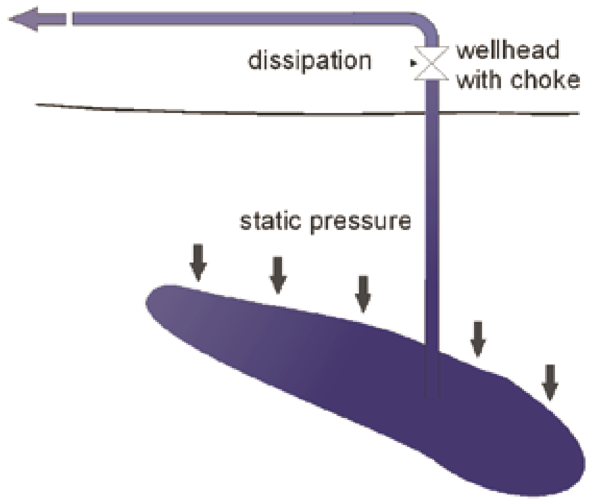

Figure 1: Dissipation of energy at the outlet (wellhead)

Subsequently, in further recovery stages the MTM operating mode may be changed and used as twin-screw multiphase pump (MPP) to convey crude oil and natural gas.

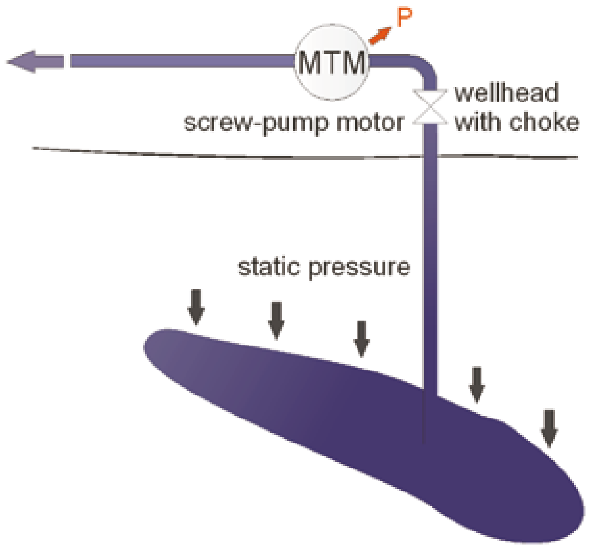

Figure 2: Application of a multiphase twin-screw motor (MTM)

2 REGARD ON THERMODYNAMIC BASICS

In the following, the choke at the wellhead is regarded as a "throttle", pictured in figure 3 (4). The throttle dissipates the energy of the flow and thus, the energy cannot be converted into technical work or power (5).

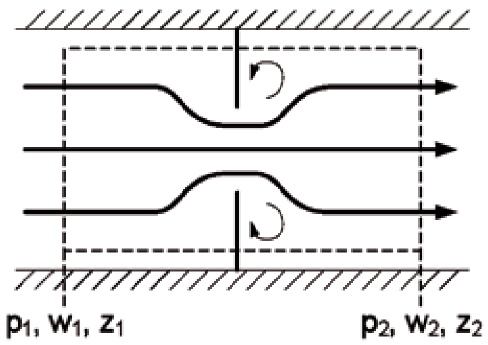

Figure 3: Sketch of the flow through an adiabatic throttle

The first law of thermodynamics for flow processes

$$\frac{dE_{sys}}{dt} = \dot{Q} + P + \sum_i \dot{m}_i \left(h_i + \frac{w_i^2}{2} + g \cdot z_i \right) \qquad (1)$$

is applied to the flow through the adiabatic throttle ($\dot{Q}$=0), where no work is provided (P=0). The system is considered to be at constant geodetic altitude ($z_1=z_2$) and the flow velocities are equal before and after passing the throttle ($w_1=w_2$). This results in

$$\frac{dE_{sys}}{dt} = 0 = \sum_i (\dot{m}h)_i \, . \qquad (2)$$

According to eq. 2, the isenthalpic change of state of the stationary flow ($\dot{m}_{in}=\dot{m}_{out}$) through the throttle can be stated as

$$h_1 = h_2 \, . \qquad (3)$$

In figure 4, the specific enthalpy h is shown as a function of the specific entropy s. Changes of state reducing the pressure from p_1 to the lower pressure p_2 are drafted, assuming the fluid as an ideal gas. According to eq. 3, the isenthalpic change of state appears as a horizontal line (1-2). The technical work

$$w_{t,rev} = h_1 - h_{2'} \qquad (4)$$

achievable by the change of state through an adiabatic, reversible turbine, can be metered directly on the ordinate. No entropy is produced (isentropic) and the maximum of the specific technical work is reached.

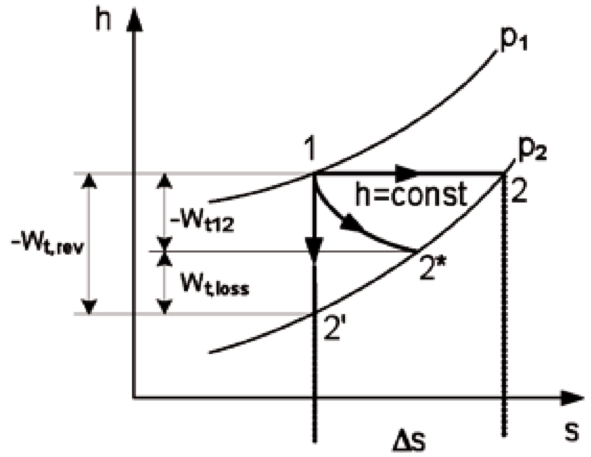

Figure 4: Isentropic (1-2'), polytropic (1-2*) and isenthalpic (1-2) changes of state of an ideal gas in a h,s-diagram

Referring any change of state to the reversible case results in the isentropic efficiency η_{is}

$$\eta_{isen} = \frac{w_{t,12^*}}{w_{t,rev}} = \frac{h_1 - h_{2^*}}{h_1 - h_{2'}} \, . \qquad (5)$$

The isentropic efficiency varies between η_{is}=1 for the isentropic expansion (1-2') and approximately zero approaching to the isenthalpic change of state.

The dissipation is described by the second law of thermodynamics

$$\frac{dS_{sys}}{dt} = \sum_i \frac{\dot{Q}_i}{T_i} + \sum_i \dot{m}_i \cdot s_i + \dot{S}_{irr} \qquad (6)$$

and the fundamental equation of Gibbs

$$Tds = du + pdv \, . \qquad (7)$$

As seen before, for an adiabatic and stationary throttling process all technical work

$$w_{dis} = Tds_{irr} = -vdp > 0 \qquad (8)$$

is dissipated. The amount of the dissipated energy is visualized quantitatively in figure 5, where the temperature T is plotted vs. the specific entropy s.

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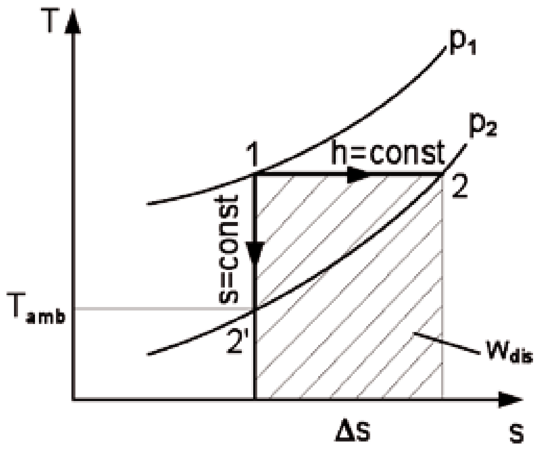

Figure 5: Isentropic (1-2') and isenthalpic (1-2) changes of state of an ideal gas in a T,s-diagram

Due to the caloric equation of state for the ideal gas

$$\Delta h = c_p \cdot \Delta T \qquad (9)$$

the isothermal change of state coincides with the isenthalpic line. According to eq. 8 the area below the isothermal change of state from high to low pressure corresponds to the specific energy wasted during throttling.

Exploiting the dissipated energy in primary oilfield recovery by employing MTMs would contribute significantly to save natural resources by "generating" energy.

3 SIMUS

The numerical program SiMuS (simulation of multiphase screw pumps) is developed in order to predict the delivering behaviour of multiphase pumps (5). Posterior, SiMuS is extended for the special conditions of MTMs.

The design of the intermeshing twin-screws working within the MTM is presented in figure 6. Together with the enclosing casing the screws form a c-shaped chamber (6). Due to the rotation of the screws the chamber moves from the inlet along the screw axis until the fluid is released through the outlet (7).

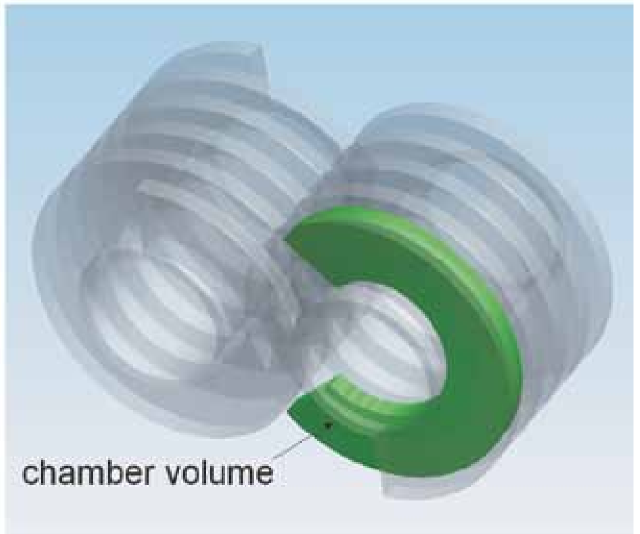

Figure 6: C-shaped conveying chamber in a twin-screw pump or motor

Gaps are located between the screws themselves and between the screws and the casing to provide contactless rotation of the screws. Three types of gaps appearing are shown in figure 7. The circumferential gap is located between the tip circle of a screw and the inner wall of the casing. The radial gap is placed between the tip circle of one screw and the root circle of the opposed screw. The flank gap is a lenticular gap, which is placed between the flanks of adjacent screws (8).

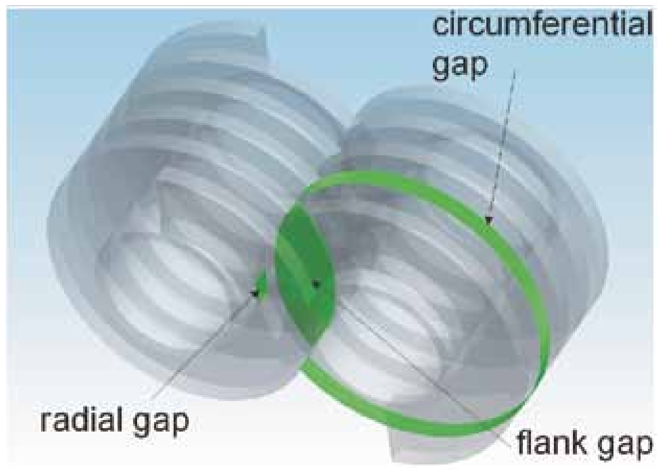

Figure 7: Different gap types in a twin-screw pump or motor

During the movement of a chamber along the axis of the screws the pressure decreases. The emerging pressure differences between the

Power Generation Subsea – Calculation of the Energy Conversion by Multiphase Twin-Screw Motors

chambers lead to internal loss flows of the conveyed fluid through the gaps. The conveying characteristic is strongly affected by the loss flows.

All mentioned gaps are considered within the enhanced SiMuS program developed by Scharf et al (9). The movement of the chambers along the rotating screws is described by time intervals. For every time interval mass and energy balances are established for each closed conveying chamber, which is considered as open system like displayed in figure 8. Within the model, gap flows act as connection between the chambers by allowing the transport of mass and energy between the chambers (10).

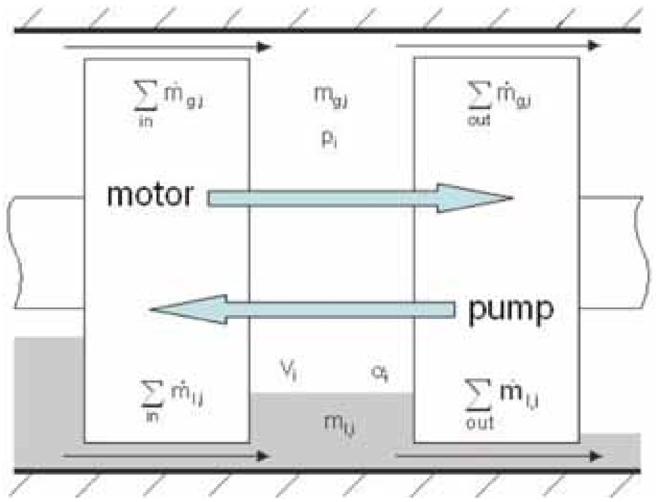

Figure 8: Visualization of the energy and mass balance inside the conveying chambers considering the gap flows for a multiphase pump and motor

SiMuS still contains a number of assumptions. Principally, there have to be mentioned:

- constant pressure within each chamber,
- equalisation of the temperature between the liquid and gas phase in each chamber due to good mixing,
- the gas is assumed as ideal gas,
- liquids are considered as incompressible,
- only single phase gap flows.

The simplified procedure of the simulation is presented in the overview in figure 9.

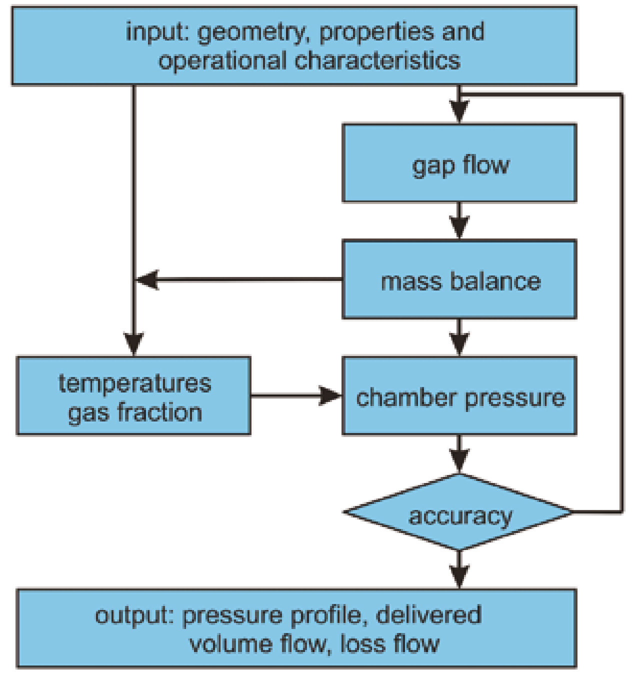

Figure 9: Procedure of the simulation program SiMuS

The geometry of the twin-screws, the thermophysical properties of the involved fluids and the operating parameters have to be provided as input data. Gap flows are calculated and mass balances of the chambers containing gas and liquid are established. Along with the computed temperature and the void fraction, the pressure change due to the expansion inside of the chambers is determined. When accuracy is adequate, the iterative process is finished. The output data includes the pressure profile, the temperature distribution, the conveying volume and the loss flow.

4 MODELLING OF MULTIPHASE MOTOR

The decrease of the pressure as function of the axial coordinate of the screw is demonstrated qualitatively from the inlet to the outlet of the MTM in figure 10. The pressure drop results in gas expansion and thus increasing gas volume. Liquid is displaced by the gas phase. In case of only gas remaining within the chamber the single-phase gap flow consists of gas only. There is no gap sealing by the liquid phase for high gas fractions and the pressure profile in figure 10 falls abruptly to the discharge pressure P_{out}.

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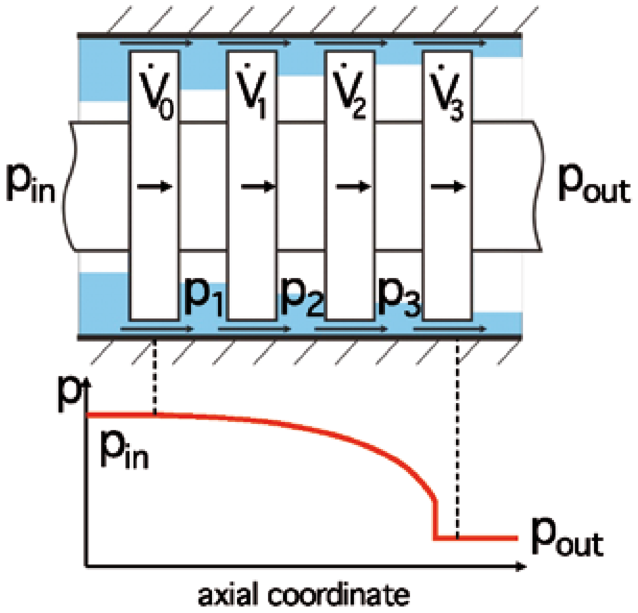

Figure 10: Qualitative pressure profile along the axial coordinate of the screw

The gap flows influence strongly the volume flows within the MTM, as the total volume flow of the fluid at the outlet

$$\dot V = \dot V_{theo} + \dot V_{loss} \qquad (10)$$

is determined by adding the theoretical volume flow $\dot V_{theo}$ and the loss flow over the gaps $\dot V_{loss}$. The theoretical volume flow is a function of the geometry of the chambers and the rotational frequency of the MTM. In order to estimate the performance of the motor the volumetric efficiency

$$\eta_{vol} = \frac{\dot V_{theo}}{\dot V} \qquad (11)$$

is defined as the ratio of the theoretical volume flow and the actual volume flow of the MTM according to eq. 10.

The conversion of power from the pressurised fluid to the motor shaft is modelled in figure 11 by a force balance on the screw thread resulting from the pressure difference between two adjacent chambers.

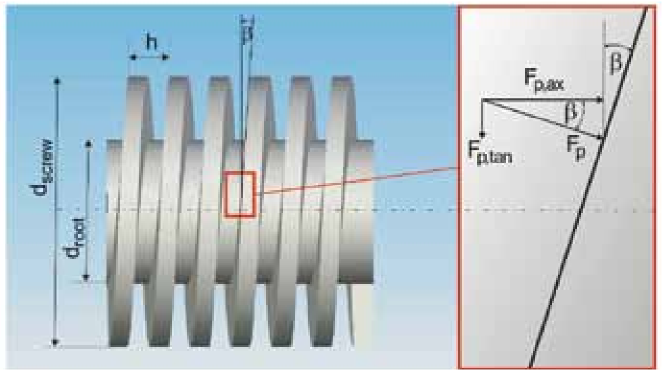

Figure 11: Demonstration of the force balance due to the pressure inside of the chamber

The tangential force on the screw due to pressure is

$$F_{p,tan} = F_p \sin\beta \,. \qquad (12)$$

The average pitch angle β

$$\beta = \mathrm{atan}\left(\frac{2h}{\pi(d_{screw} + d_{root})}\right) \qquad (13)$$

results from the pitch h, the root diameter d_{root} and the diameter of the top circle of the screw d_{screw}, see figure 11.

The motor shaft power P is calculated by

$$P = \omega \cdot M = 2\pi \cdot n \frac{d_{screw} + d_{root}}{4} \Delta p \, A \sin(\beta) \qquad (14)$$

from the angular speed ω and the torque M, which depends on the geometry of the screw.

Numeric calculations of the named values are carried out for two different motor geometries and discussed in the following.

5 RESULTS OF CALCULATIONS AND EXPERIMENT

All experiments in the MTM mode are realised on a SLH-80/34 with 1.85 chambers. Involved working fluids for the multiphase flow are water as liquid and air as gaseous element. The discharge side pressure p_{out} is set constant to 5.3 bar and the pressure difference Δp, the gas volume fraction at the inlet α_{in} and the rotational speed n are

Power Generation Subsea – Calculation of the Energy Conversion by Multiphase Twin-Screw Motors

varied. Experimental results are compared to the calculations at equivalent conditions.

In figure 12 the motor shaft power P is plotted vs. the pressure difference between the inlet and the outlet of the MTM for only liquid flows and n=500…1500/min. As expected, for all rotational speeds the shaft power increases linearly with the pressure difference (12). According to eq. 14, higher rotational speed leads to growing shaft power. The calculated values coincide satisfactorily with the experimental data.

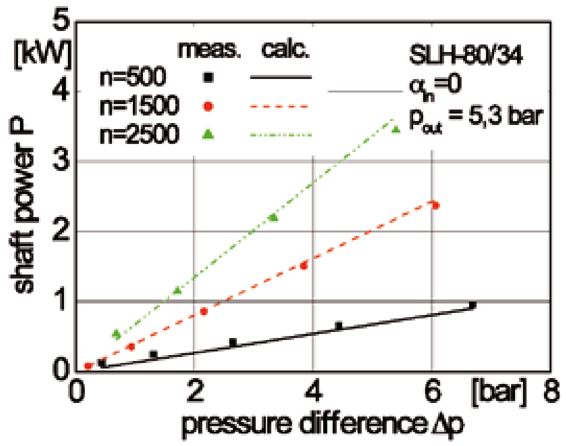

Figure 12: MTM shaft power vs. pressure difference for different rotational speeds at only liquid flow

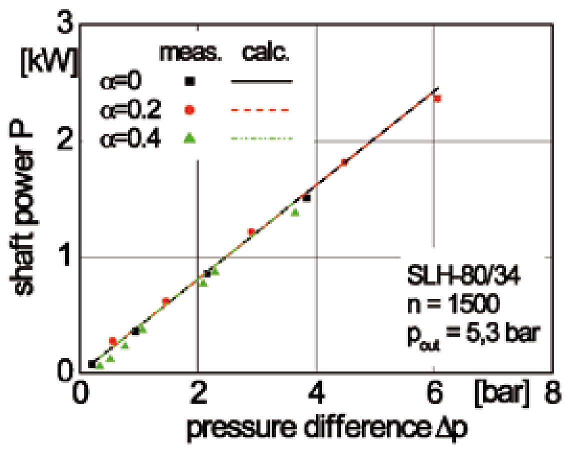

Figure 13: MTM shaft power vs. pressure difference for different gas volume fractions at n = 1500 /min

Figure 13 shows the released shaft power in dependence of the gas volume fraction for low α up to 0.4. For both, experimental and calculated results, there is no dependence of the shaft power on the air fraction. Again, no difference is detected between computed and measured values.

Next, the efficiencies of the multiphase twin-screw motor are regarded. The isentropic efficiency for the MTM

$$\eta_{isen} = P_s \left(\dot V_l \cdot \Delta p_{MTM} + \dot m_g c_p T_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right] \right)^{-1} \qquad (15)$$

is given in figure 14 for only liquid flow and n=1500/min. For rising MTM pressure differences the measured and calculated values of the isentropic efficiency fall degressively. Calculated values tend to be to high, but approach the measurements for growing pressure differences.

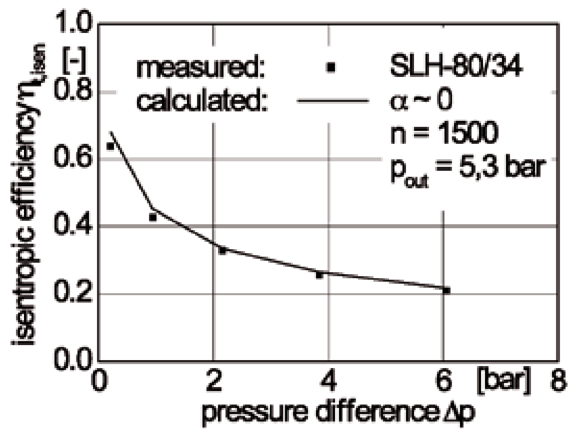

Figure 14: Isentropic efficiency vs. pressure difference for only liquid flow and n = 1500 /min

According to eq. 11, the volumetric efficiency in figure 15 shows also degressively falling curves. Calculated values are overvalued due to loss flows predicted much to high.

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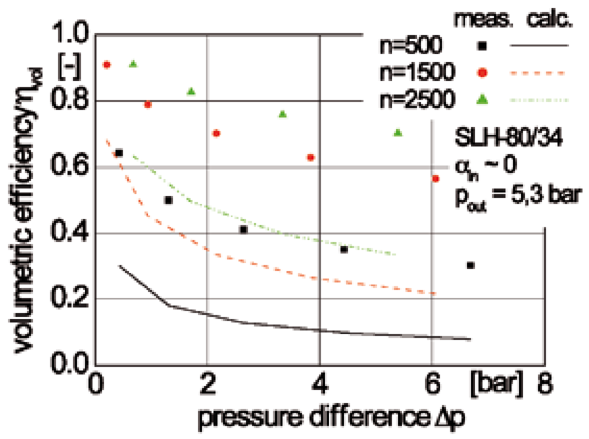

Figure 15: Volumetric efficiency vs. pressure difference for only liquid flow and n = 500…2500/min

Besides, further calculations are realised for a MPC 133-20 with 5.1 chambers for varying operational parameters in order to compare the thermodynamic efficiencies at different inlet pressures. In figure 16 the isentropic efficiency according to eq. 15 is compared to the isothermal efficiency

$$\eta_{iso} = P_s\left(\dot{V}_{liqu}\cdot\Delta p_{MTM} + \dot{V}_{gas}\cdot p_{in}\cdot\ln\frac{p_{out}}{p_{in}}\right)^{-1} \quad (16)$$

for two inlet pressures at α=0.5.

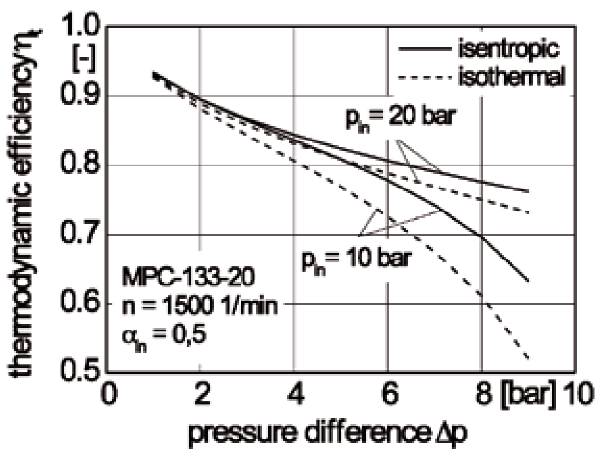

Figure 16: Isentropic efficiency vs. pressure difference for only liquid flow and n = 1500 /min

The isentropic efficiency is always the upper limit, the isothermal one is the lower limit. The efficiencies increase with the inlet pressure in both cases and the distance between the isothermal and isentropic efficiency increases with the pressure difference and decreases with the inlet pressure. The diverging trends for the two different inlet pressures result from the pressure ratio included in eq. 15 and eq. 16.

5 CONCLUSIONS

The calculation model SiMuS allows obtaining the delivery flow of a multiphase twin-screw motor in dependence on the screw design and the operating conditions. By means of experiments, the theoretically calculated data is validated. The comparison is done considering all kinds of gaps for calculations. The calculated values match the experimental data satisfactorily in most cases. Only the loss flow is calculated to high and leads to diverging volumetric efficiencies.

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