Efficiency Potential of Dry Case Operation for Bent-Axis Motors

Dr.-Ing. Robert Rahmfeld, M.Eng. Sünje Marsch, Dipl.-Ing. Wilhelm Göllner

Sauer-Danfoss, Propel Division, Krokamp 35, 24539 Neumünster, E-Mail: rrahmfeld@sauer-danfoss.com

Prof. Dr.-Ing. Thorsten Lang, Dipl.-Ing. Thomas Dopichay, Dipl.-Ing. Johannes Untch

Institut für Landmaschinen und Fluidtechnik (ILF), TU Braunschweig, Langer Kamp 19a, 38106 Braunschweig, E-Mail: t.lang@tu-bs.de

Abstract

Providing high efficiency hydrostatic units is a key demand for pump/motor manufacturers today, and clearly the need for reducing losses will even grow in the future. Next to providing high efficient rotating groups, a further option to improve hydrostatic unit efficiency is the elimination of churning losses, preferably in hydrostatic bent axis motors due to the high speeds. This technology is known from wide angle (45°) bent axis units in powersplit tractor applications, which achieve the highest series efficiency today. To analyze the reduction of energy consumption by avoiding churning losses for standard hydrostatic drivelines, a bent axis motor has been tested and compared in full and dry case operation according to efficiency and losses. The generated measurement data was then used as base for different churning loss models. In a next step, the selected model was implemented in wheel loader and crawler drivetrain simulation models. Dynamic simulations predicted a noticeable fuel saving potential depending on the operating conditions when dry case instead of full case operation was used, when the motor is running at higher speeds. The efficiency component measurements and generated models were successfully verified with additional tests on a complete driveline test stand. These additional tests included investigations on tank designs and analyzed the basic abilities to avoid foaming.

KEYWORDS: efficiency, power losses, dry case, bent axis motors, oil foaming

1 Introduction

In mobile applications, the maximal power of the Diesel engine is often only needed to achieve maximal traveling velocities. Therefore, the motors in hydrostatic drivelines run here at low displacement and high shaft speeds. This leads to high churning losses, by the rotating group churning in the oil filled housing, combined with an increase in case temperature which need to be cooled down by additional loop flushing. By draining the case of the unit – dry case in contrast to full case – the churning losses are avoided and with that the overall efficiency increases, respectively. This supports downsizing of the actuating Diesel engine and with that a reduction in fuel consumption and emissions is the result.

For this purpose, the motor case needs to be drained and sufficient lubrication needs to be provided at the rotating kit and bearings. The concepts so far for draining standard motors suggest using an additional drain pump or floating the housing with gas which presses the oil out. These approaches need additional energy overhead and a complex system architecture. The presented new concept suggests using the extended motor housing as oil reservoir, working without any additional energy, so that the motor case is basically drained by gravity. This paper will prove the energetic benefit by using a dry case motor by this method. Due to high possible shaft speeds of bent axis motors, these are most suitable for a dry case operation. Based on efficiency comparison measurements, on component and complete driveline level, a corresponding motor is modeled and the potential in reduction of energetic and fuel consumption of typical mobile machines is analyzed.

2 New Drycase Concept for Bent-Axis Motors in Hydrostatic Drives

In the past, churning losses have been rarely investigated. The focus was more on developing high efficient rotating groups, like the 32° and 45° bent axis principle. This research and development tendency is probably due to the fact that the effort of draining the case can be likely higher than the improvement itself. The state-of-the-art for dry case operated motors can be summarized as follows.

- Patent /DE91/ suggests draining the oil out of the case with the help of a separate pump between the leakage connection and the oil reservoir.
- In /DE92/, a separate pump is installed in the inner case, which carries the oil from the leakage space in the oil reservoir.
- In /DE94/, the draining of the case is done by pressing a pressurized gas in the case from a separate pressure-gas-source. The oil will be consequently pressed out of the leakage line. A second method in /DE94/ is the usage of a vacuum pump which carries the oil out of the case with depression.

All three concepts lead basically to additional power consumption by the draining pumps. Supplementary, the individual draining pumps need non-neglectable installation space. Once the oil has been carried into the additional oil reservoir, due to turbulences and foaming, it takes a certain time for the gas to escape from the oil. For this amount of time, it will not be available for the hydraulic system and therefore the amount of hydraulic fluid needs to be larger than originally planned.

The new concept described in /DE09/ (see **Figure 1)** includes an oil reservoir, as extended motor housing, located directly underneath the rotating parts. The oil will not be carried out by a separate pump or additional gas pressure and therefore no turbulences appear. The oil flows due to gravitation into the attached or integrated reservoir, so less foaming will appear. Therefore, no pacification is necessary, and the charge pump for the low pressure line can be directly fed from this oil reservoir.

In consequence due to this direct feeding of the charge pump, no additional power loss for draining will appear. The bearing and kit lubrication, which is a relatively small amount of flow demand, can be e.g. realized through the synchron joint by a connection to the low pressure line.

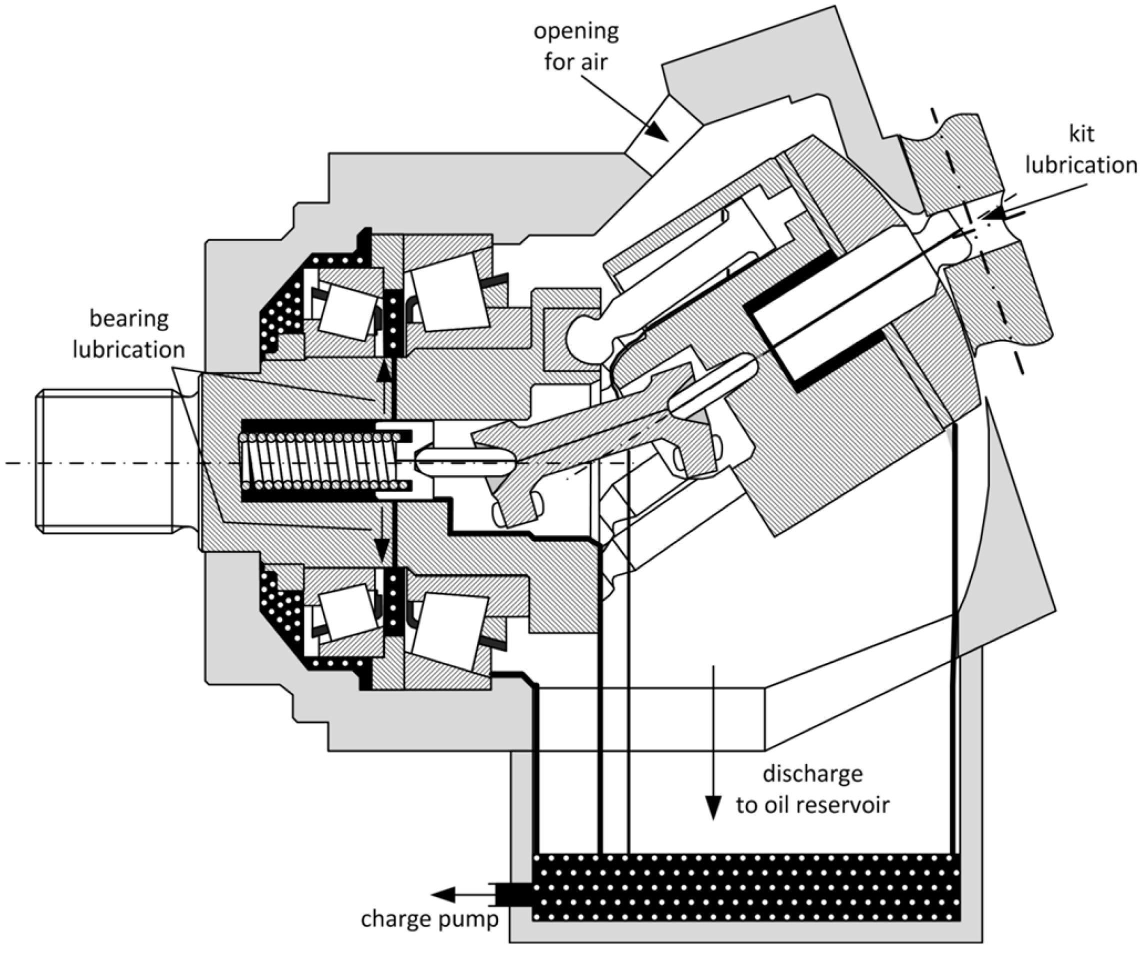


Figure 1: New concept to dry the case of a bent-axis unit

3 Efficiency Measurement of Full and Dry Case Motor Operation

3.1 High pressure calibration of flow meters

ISO 4409 defines basically to measure the volumetric flow in high pressure for an efficiency measurement of hydrostatic units. A low pressure measurement of flow and calculation to the high pressure flow is not possible with dry case units, because the kit leakage cannot be measured reliably. I.e. here, a measurement of the volumetric flow in the high pressure line is essential. Therefore a calibration of suitable flow meters under high pressure has to be done, because generally flow meters are not calibrated in high pressure, what often leads to a noticeable systematic deviation with increasing pressure /Rah10/.

Helical rotor flow meters are expected to have least leakage and fluid resistance in this operation situation compared to other principles. Therefore, two identical helical rotor flow meters (new designed VSE RS400) with a special solid design were analyzed in series connection, while one was in low and the other one in high pressure. It was proven that even under high pressure, this flow meter measures the correct amount of flow, based on the given and ISO4409 defined accuracy (class A), and therefore no correction curve is required for the following efficiency measurements.

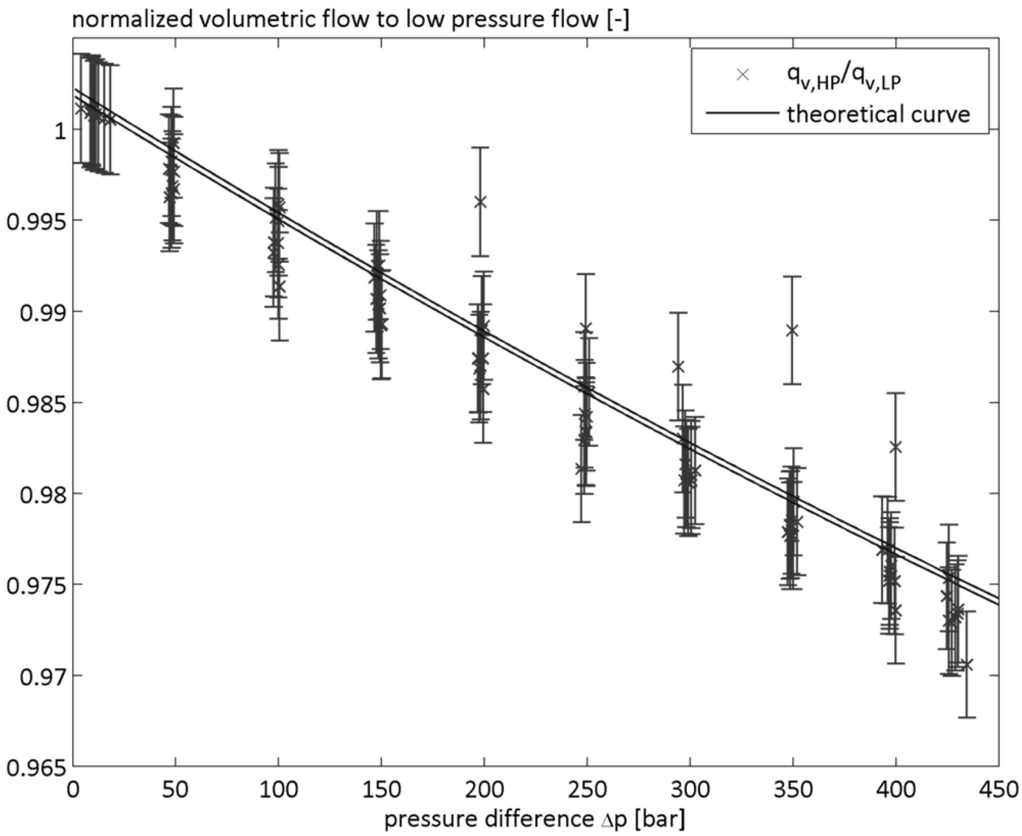


Figure 2: Normalized volumetric flow and theoretical normalized density curve

Figure 2 depicts the evaluation method: the normalized volumetric flow measurements in high pressure versus the flow in low pressure at 11 cSt. The theoretical curve, calculated with the density properties, is displayed as well with the estimated error in fittings. No systematic deviation with pressure or temperature is detected. The outliers only occurred at low flow rates, and with that their real uncertainty is insignificant due to the fact that the measurement uncertainty refers to the measured value.

3.2 Efficiency measurements with full and dry case operation

The efficiency test is done with two 45° bent axis motors with a maximal displacement of 233 ccm, while one was running in motoring and the other one in pumping mode (see /Rah10/, “combined mode setup”). The full and dry case measurements are done on the same test stand, and with the same measurement equipment. The most demanding task is to hold the steady-state conditions constant and equal for full and dry case. Especially, equivalent temperature settings in the lubrication gaps are required for a direct comparison, as well as an equal pressure level for the analyses at the corresponding operation condition.

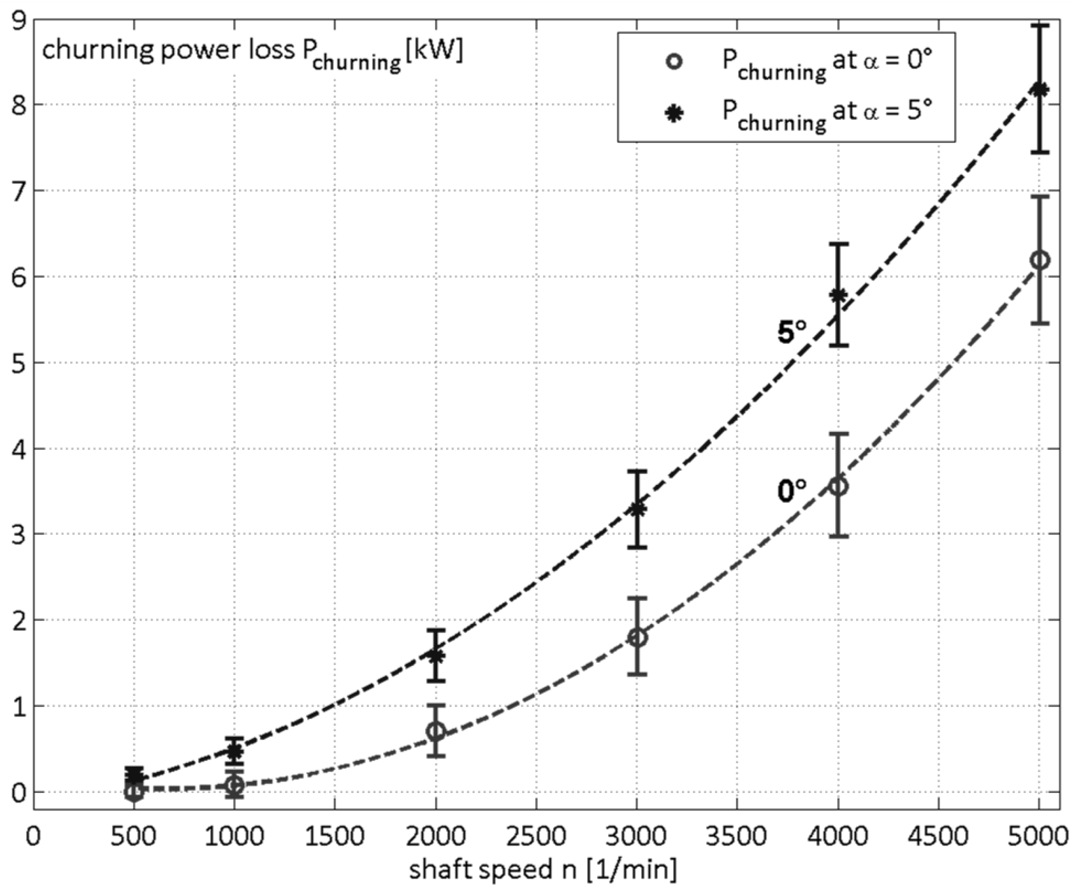


Figure 3: Measured power loss difference full vs dry case at 30 cSt, 45° bent-axis motor with max. 233 ccm

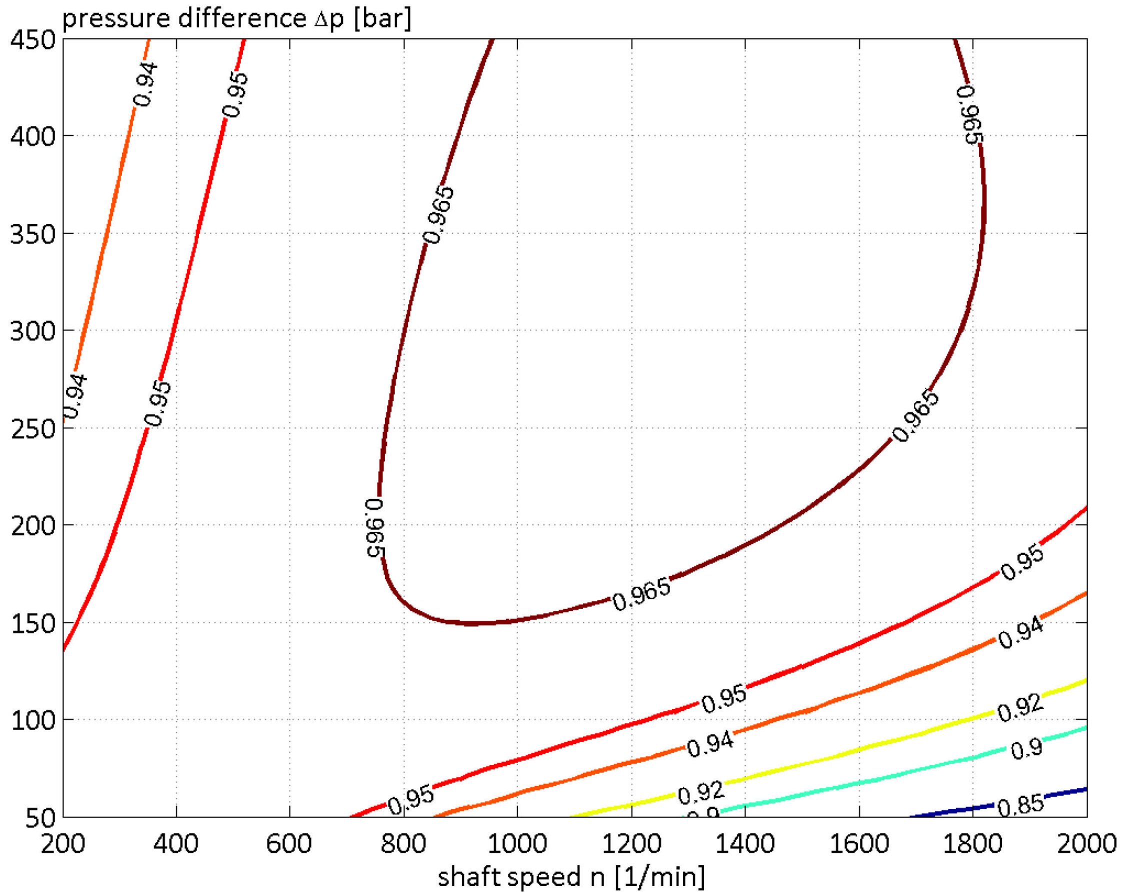


Figure 4: Mussel plot for 45° 233 ccm motor at 100% and 11 cSt, dry case

After the evaluation of the measurements with full and dry case unit operation, it was proven that churning losses are a pure torque loss and dependent upon:

- speed and
- displacement,
- temperature (viscosity), as well as,
- rotating kit geometry and form of surrounding housing.

Following equation applies to calculate the churning power loss for motoring and pumping, respectively:

$$P^{M}_{churning} = 2\pi n \left(M_{dry} - M_{full}\right) \qquad P^{P}_{churning} = 2\pi n \left(M_{full} - M_{dry}\right) \tag{1}$$

The elimination of churning losses leads to less input torque for the pump and on the other hand gains the output torque of the motor.

Figure 3 depicts this measured power loss difference between full and dry case, over speed for two displacement settings. A nonlinear dependency upon speed can be concluded. In **Figure 4** the resulting excellent overall efficiency (highest efficiency of conventional series products) for a 45° bent axis motor with 233cm³ at 100% displacement is shown for dry case operation. The efficiency of full and dry case operation is here at 100% quite similar due to lower speeds.

4 Churning Loss Modelling

At first, the full case loss model is built with the multi-dimensional least-squares fit to the measurement data by the POLYMOD method, see /Rah10/. This polynomial is a function of derived displacement volume, shaft speed and pressure difference. The dry case consideration is achieved by charging a designed churning loss model against the full case torque loss model with:

$$M_{s,dry} = M_{s,full} - M_{churning} \tag{2}$$

In /Jan97/, churning losses are described with the help of the fluid resistance and Newton's shear stress. This approach is taken as basis for churning loss modeling for bent axis motors here, see **Figure 5**. In order to regard the displacement dependency, a term was added to this physical model to display the dependency on displacement and correlation to speed as well, see Equation 3. Here, not only the drag resistance of the pistons and resulting cylinder block sheer stress are analyzed, but the shaft and synchron joint are taken under consideration, too.

$$M_{churning} = C_1 D_1\, n \,+\, C_2\, n^2\, (1 + \sin\alpha) \tag{3}$$

$$C_1 = 4\,\pi^2\,\eta_{dy} \qquad C_2 = 2\,z\,c_d\,\rho\,\pi^2 A_p\,r_p^{\,3} \qquad D_1 = \left[\frac{{r_{cb}}^3\,l_{cb}}{r_h - r_{cb}} + \frac{{r_{sj}}^3\,l_{sj}}{r_p/2} + \frac{{r_s}^3\,l_s}{r_{hs} - r_s}\right] \tag{4}$$

For the power loss due to churning follows consequently:

$$P_{churning} = 2\,\pi\,n\,M_{churning} \tag{5}$$

The first quantity in Equation 3 is based upon sheer stress and dependent on the shaft speed with the first order. The coefficients are influenced by the dynamic viscosity and parts dimensions of cylinder block, synchron joint and shaft. The second quantity in this equation has the highest influences on churning losses with a second order dependency upon shaft speed. /Jan97/ defines the drag coefficient with 0.1. Due to the cycled assembly of the pistons, the resistance of one piston is smaller. Here, this drag coefficient of 0.1 could not be verified. Instead, a c_d value of 0.575 had to be taken into account, which is an empirical value from the comparison of the model to all differences in torque measurements from the efficiency testing. The simplified physical model compared to other modeling approaches showed the best measurement correlation, respecting the measurement and operating condition adjustment uncertainty, and with that the most consistency over the whole operation range.

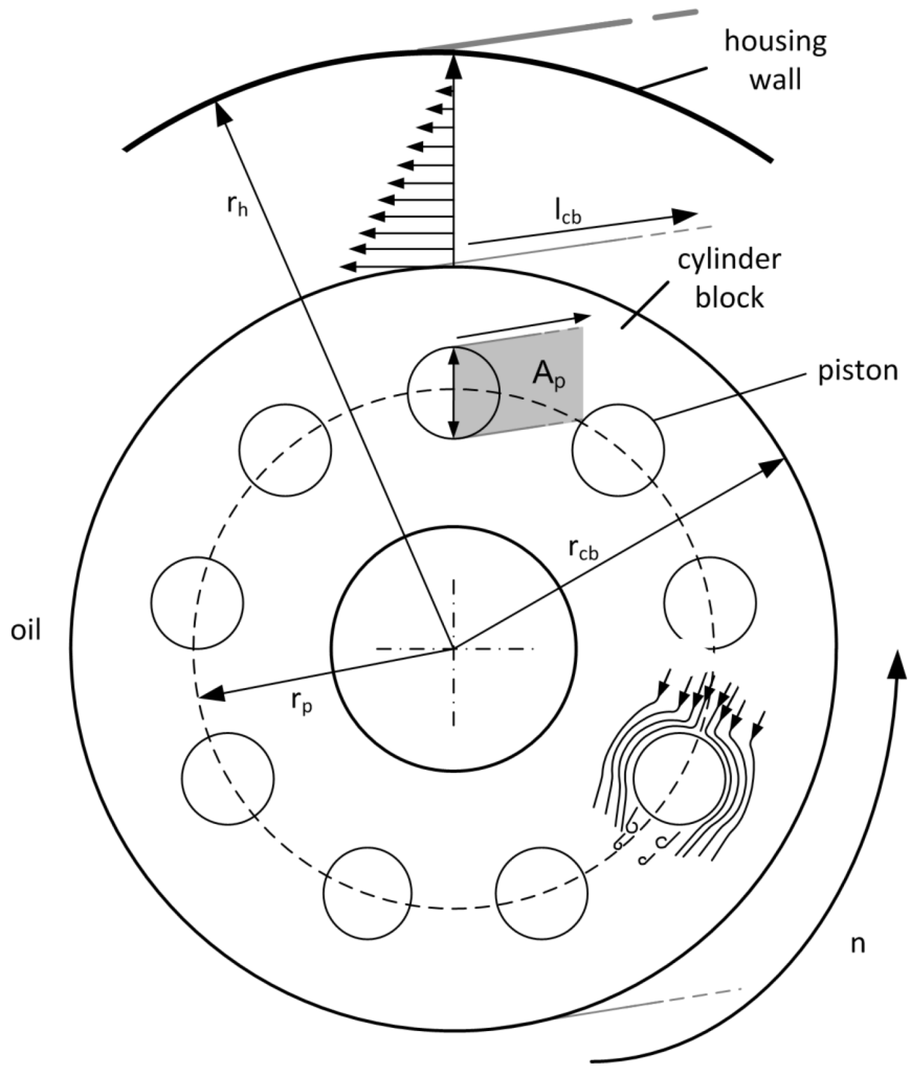


Figure 5: Churning loss consideration

The churning loss calculation with the physical model (in [kW]) in contour lines can be found in **Figure 6** with comparison to measurement points, all according to Equation 1. The stars (*) mark the measurement results, and the given value is the calculated churning power loss $P_{churning}$. The solid lines represent the physical model of $P_{churning}$ according to Equation 3. Already at this point, it is obvious that the shaft speed has to be at least 2000 rpm for a clear difference between full and dry case. The drawback of model accuracy to measurement data, by using a simple physical modeling approach, increased the model consistency significantly, which has the major advantage of being a filter to the measurements. Note that this model also offers the advantage of a somehow easy transfer to other unit sizes. This churning loss model has been verified with CFD simulations in some conditions.

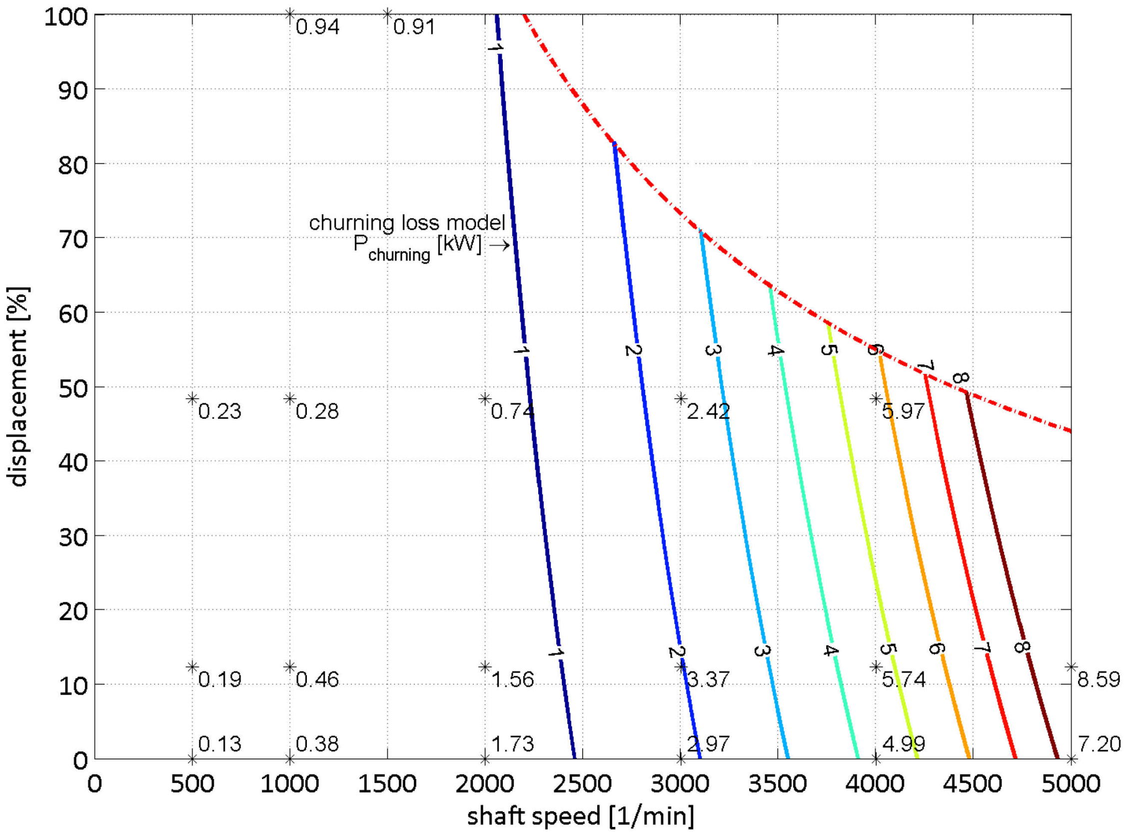


Figure 6: Calculated churning losses compared with measurements, at 30 cSt (* measurements [kW], – physical model [kW])

5 Verification at Driveline Test Stand

The main objective of the measurements at the driveline test stand was to verify the component efficiency measurements and the generated churning loss model under realistic conditions for a complete hydrostatic driveline. In addition, first investigations on tank designs were carried out to find a compromise of minimal tank volume and sufficient air separation, which is fundamental for a later implementation in mobile working machines. The following driveline test setup was used: a 89 ccm axial piston pump is driven by a Diesel engine at a constant speed of 1800 rpm. The pump itself is connected in a closed circuit to the above utilized 45° bent-axis motor, a 233 ccm unit adjusted to an angle of 5°, which was operated in full as well as in dry case, respectively. The driveline can be loaded continuously with torque, utilizing a secondary controlled hydrostatic unit. Speed-torque measuring hubs are used to identify the in- and output power. In contrast to Figure 1, the tank in dry case operation is located externally beneath the motor housing in this setup. Only the kit and bearing lubrication as well as negligible volumetric losses flow from the housing into the tank. The charge pump is mainly fed from the pump leakage and circuit flushing. The volume flow rate from the tank has only a share of about 15 % of the total charge flow. **Figure**

7 shows a simplified version of the hydraulic circuit diagram for only one direction of load and rotation, as well as the structure of the test setup.

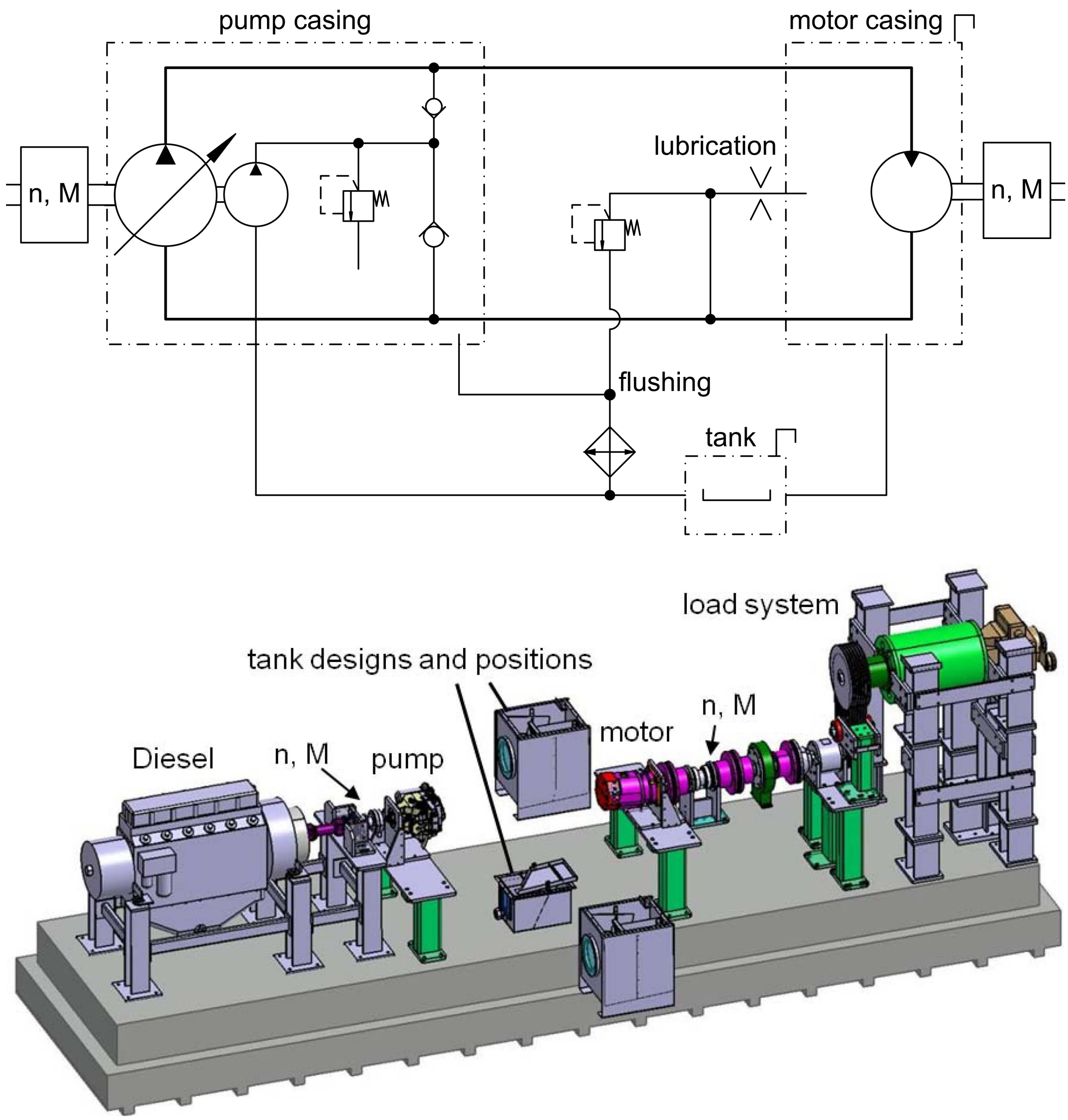


Figure 7: Implemented hydraulic circuit at driveline test stand

By measuring the mechanical in- and output power in full as well as in dry case operation, the total power loss under both operating conditions could be determined. The difference in both power losses corresponds to the churning losses in the motor housing. **Figure 8** shows the measured churning power for two different motor loads. For reliable results, all conditions and the measurement system were equal in both operating modes (except the tank position). The results <u>verify</u> the component efficiency measurements according to Figure 3 and the load independent churning loss model.

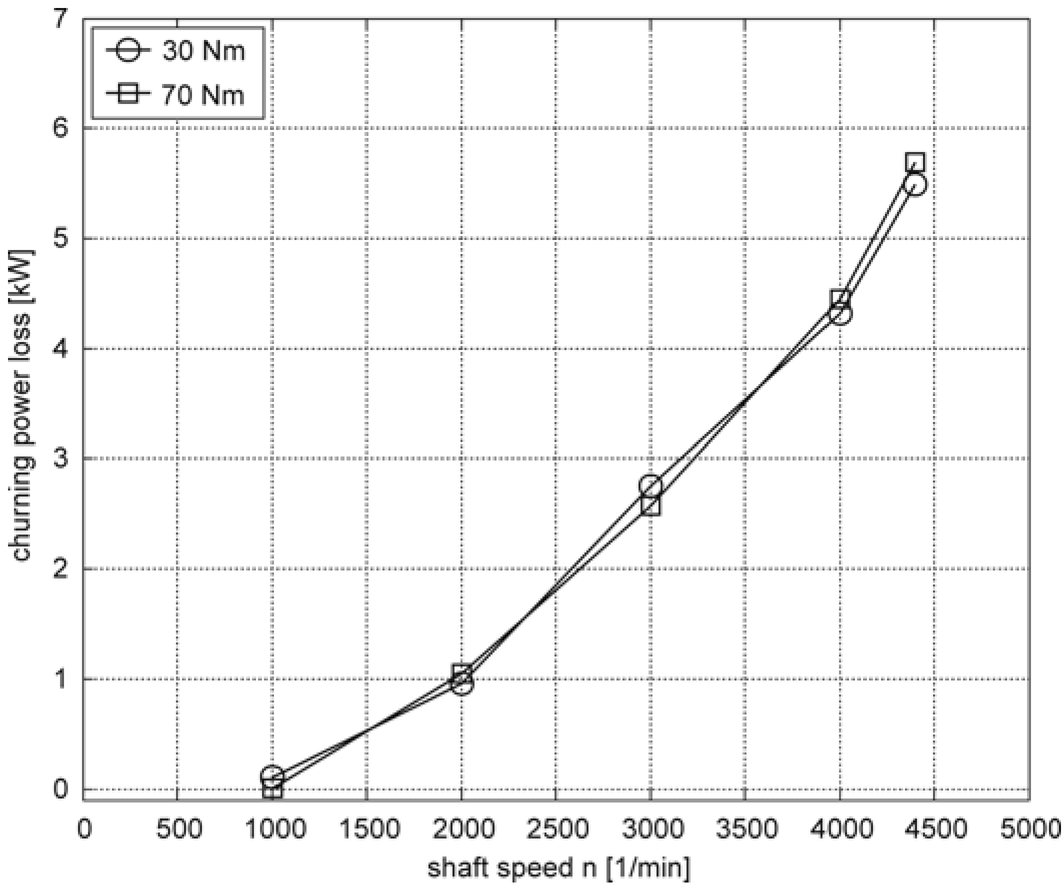
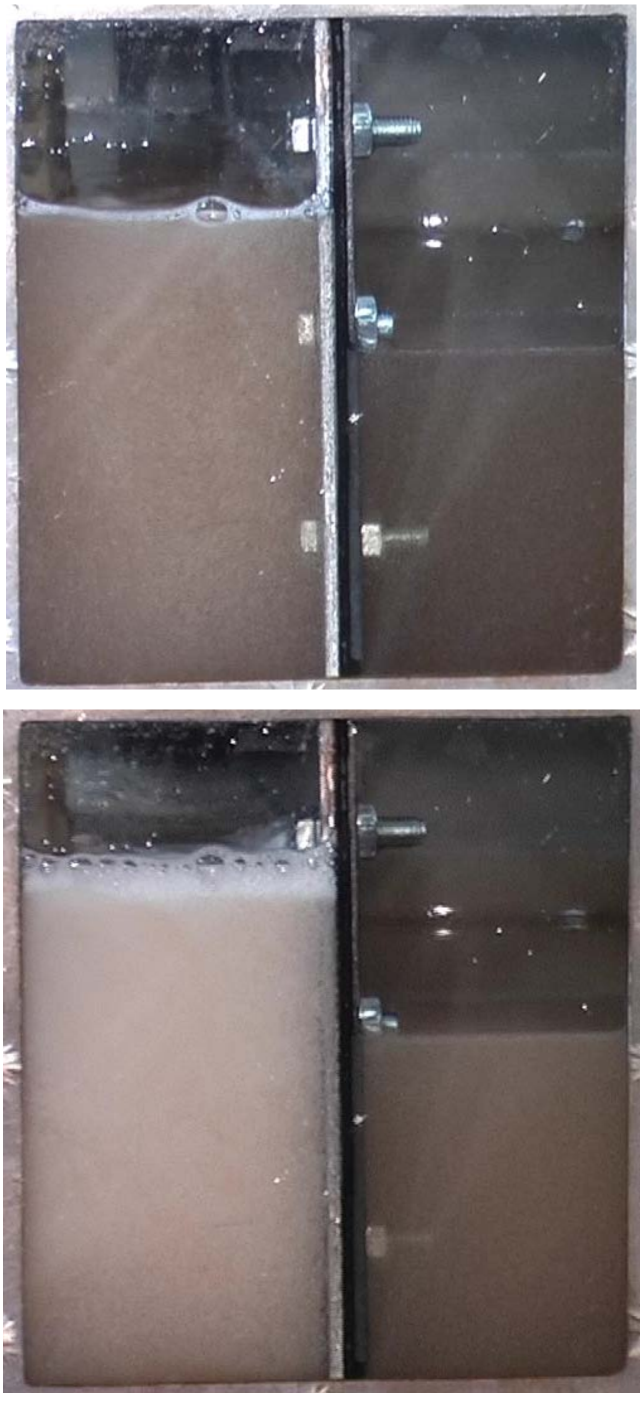


Figure 8: Measured power loss difference full vs dry case at 30 cSt and 5° motor angle, at driveline test stand (89 ccm pump, 233 ccm motor)

For a future application of dry-case units with extended motor housing (as in Figure 1), the oil volume has to be reduced compared to the external tank used for the verification. However, the foaming and the air content of the oil have to be on an acceptable level. Thus, the next step was to find a tank design that satisfies these boundary conditions. High oil temperature enhances the air separation and high speeds of the motor lead to increased foaming. Therefore, the critical operating condition is at high speeds and low oil temperature, and the tank was consequently optimized by tests under these conditions. Several measures taken inside of the tank and at the inlet contribute to enhance the air separation, e.g. the use of guide plates and screens with different mesh sizes as well as diffusors that ameliorate the homogeneity of the oil flow. The influence of individual measures and combinations of measures on foaming and air separation was documented and compared. As a result of these iterative investigations, a tank design was found that comprises a certain combination of measures and leads to significant lower air content at the suction tube.

Figure 9 shows the difference in oil condition at the inlet and at the suction pipe for a certain small tank configuration as an example for two motor speeds. The oil flows from the inlet at the left side around the guide plate in between to the suction pipe that is placed right to the guide plate. Screens are used to separate the air from the oil in the tank. In addition, nested diffusors and screens are installed at the inlet of the tank. The condition of the oil at the inlet and at the suction pipe is indicated by the oil color on the left and on the right side of the picture respectively. High air content leads to a bright color. The screens in the tank reduce the air content as shown in Figure 9, where the oil is darker at the suction pipe compared to the area of the inlet.



800/min

4100/min

Figure 9: Oil conditions at small tank, inlet (left) and outlet (right), 30 cSt

6 Energetic Potential for Mobile Working Machines

The wheel loader analysis, utilizing the above derived churning loss model, is based on a standardized duty cycle from the VDMA driveline project /Jäh06/. The hydrostatic driveline is again considered by a 89 ccm pump and a 233 ccm bent axis motor, driven by a 80 kW Diesel engine. During duty cycle operation, the dry case benefit of this system is not too high (1.4%), but in driving mode, especially at the maximal vehicle velocity, the potential saving is significant (15.4%), see **Table 1**. Considering a combined operation of 70 % working in long Y-cycle truck loading and 30 % driving at maximal velocity, the fuel saving potential sums up to 5 %.

Another analysis for a crawler, considering a 115 ccm pump and a 233 ccm motor in a dual-path transmission driven by a Diesel engine of 140 kW, showed different and less results. Because the hydraulic motor is not working at high shaft speeds, especially in ground moving, the potential due to lack of churning losses is minor. This shows that a dry case motor implementation in a drivetrain is worth more, the more often the vehicle works with high motor shaft speeds. Considering an application that only needs its maximal Diesel power for driving at the highest traveling velocity, a dry case motor is a suitable possibility for downsizing the combustion engine.

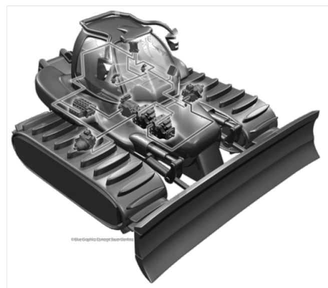
Fuel saving: wheel loader  80 kW, 89 ccm pump, 233 ccm motor		Fuel saving: crawler  140 kW, 115 ccm pump, 233 ccm motor	
long Y-cycle (see /Jäh06/):	1.4 %	ground moving:	0 %
max. speed:	15.4 %	road driving:	6.3 %
combined operation (70/30):	**5.0 %**	combined operation (70/30):	**1.6 %**

Table 1: Fuel saving potential for wheel loader and crawler by dry case motor

7 Conclusions

It has been proven by different measurements that generally upon speeds of 2000 rpm, a dry case motor operation leads to a noticeable energy saving in the driveline. In this area, a dry case operation like in the presented concept is another option for high efficient hydrostatic units to improve the loss behavior furthermore. More investigations on churning losses with different housings and inlets are currently done in the VDMA project "Increasing Efficiency in the Partial Load Area" at the IFAS, RWTH Aachen.

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Nomenclature

P	power	kW	n	shaft speed	rpm
$P_{churning}$	churning power	kW	α	displacement	deg
M	torque	Nm	c_d	drag coefficient	-
η_{dy}	dynamic viscosity	Pa·s	ρ	density	kg/m³
z	number of pistons	-	l	length	m
A	reference area	m²	r	radius	m