International Journal of Fluid Power 10 (2009) No. 1 pp. 37-46

EXPERIMENTAL INVESTIGATION OF FRICTION FORCE BETWEEN VANE TIP AND CAM-RING IN OIL VANE PUMPS

Mohamed El Ashmawy[1] and Hubertus Murrenhoff[2]

[1]*Suez Canal University, Engineering Science Department, Faculty of Petroleum and Mining Engineering, Suez, Egypt*
[2]*RWTH Aachen University, Institute for Fluid Power Drives and Controls, Steinbachstr. 53, 52074 Aachen, Germany*
arafat_696@yahoo.com, mh@ifas.rwth-aachen.de

Abstract

An experimental investigation was conducted to measure the friction forces between a vane tip and cam-ring in oil vane pumps. The incentive of this work is to study the effect of important parameters on the friction coefficient between a vane tip and cam-ring. Such parameters are relative speed between the vane tip and cam-ring, normal vane force, pressure difference between the two sides of the vane, and coating of the vane tip. A comparison was performed between five different (**P**hysical **V**apor **D**eposition) PVD-coated vane tips and the normal vane without coating. To satisfy such requirements a special test rig was designed and constructed in the lab. The results show that the effect of the vane force and the pressure difference between the two sides of the vane are very small compared to the relative speed. The coating material shows no significant effect on the friction force in vane pumps. Therefore the wear properties of the coating materials should be considered in future studies. For friction measurements in oil vane pumps a simple test rig design with no pressure difference between the two sides of the vane is proposed.

Keywords: vane pump, friction, PVD, hydraulic, power steering, coating

1 Introduction

The main advantages of vane pumps are low cost and compact design. Vane pumps are widely used in lower pressure industrial and mobile applications that require relatively low flow and pressure pulsations thus developing low noise levels, especially in automobile power steering systems.

Many parameters affecting the vane pump performance were theoretically and experimentally studied by others. The following parameters are selected from the previous studies:

- Vane tip radius of curvature

  Simulation results show that in spite of the pressure relief at the vane designed by the manufacturer high mechanical loads appear especially during the suction process (Ortwig, 1992). These loads can be reduced by an enlargement of the vane tip radius.

  A computer model was developed using a mixed lubrication model to investigate the tribological performance of the blade and liner interface in a transfer pump lubricated with diesel fuel (Sui, 1995). Results show that increasing the blade surface radius also greatly improves the film parameter and reduces the interfacial friction by as much as 80 %.

- Friction between the vane tip and cam contour and cam lift to vane thickness ratio ε

  The friction between a vane tip and cam contour was experimentally and theoretically investigated (Inaguma and Hibi, 2005), and the effect of friction on the efficiency of the vane pump was also described. These investigations were carried out for three different vane thicknesses. The method used in this study to measure the friction torque between vane tips and cam was by measuring the total torque of the vane pump and then subtracting other torques by means of theoretical relations. Pumps with the same ε (ratio of cam lift and vane thickness) have the same mechanical efficiency. A larger value of ε increases the efficiency of the vane pump. The limit of the vane strength governs the upper limit of ε.

  Experimental results show that the friction coefficient increases with increasing vane pressure (vane force in the radial direction) and decreases

This manuscript was received on 25 August 2008 and was accepted after revision for publication on 7 February 2009

with increasing relative speed between the vane tip and cam-ring (Faber, 2006).

- Cam-ring surface roughness and validity of using cylindrical cam rings

  Results indicated that the film thickness at the blade/liner interface in a transfer pump lubricated with diesel fuel is generally lower than the surface roughness level. Reducing the surface roughness has a significant effect on improving film thickness-to-surface roughness ratio (Sui, 1995).

  Friction torque arising from the friction between a cam contour and vane tip is significant (Inaguma and Hibi, 2007). First the coefficient of friction between the vane tip and cam-ring was measured by using cylindrical test rings with various values of surface roughness. Then the torque characteristics of vane pumps having cam-ring contours with various values of surface roughness were measured and their results compared with the results investigated using cylindrical test rings. As a result the friction torque was reduced by lessening the surface roughness of the cam contour, resulting in an improvement of mechanical efficiency. The coefficient of friction measured by using cylindrical test rings could be applied to the actual vane pumps.
- PVD-Coatings

  Coatings are applied using PVD (**P**hysical **V**apor **D**eposition). By increasing the surface pressure in an experiment it could be proven that the higher friction of most Zircon carbide ZrC-variations resulted from an oil film break down. Only the Zircon carbide with graded hardness profile ZrC_g-coating did not have an increase of friction at higher surface pressures (Bebber, 2002).

  PVD-coated pistons of an axial piston machine using ZrC_g as a coating material (Scharf and Murrenhoff, 2006) reveal limited wear within the first hours of operation and very low friction between piston and bushing. The experiments were performed on a single piston test bench.

In the present study the friction force between a vane tip and cam-ring is experimentally investigated. A cylindrical cam-ring and a rotor with two vanes and two gap seals are used. The friction torque between these two vane tips and cam-ring is directly measured via a torque sensor. The effect of some parameters on the friction coefficient is studied. These parameters are the relative speed between the vane tip and cam-ring, normal vane force, pressure difference between the two sides of the vane, and coating material of the vane tip. The essential distinction between this study and the previous studies is the study of the effect of two parameters on the friction coefficient between a vane tip and cam-ring. These parameters are vane tip coatings and pressure difference between the two sides of the vane. All parameters studied in this paper are measured for uncoated vane tips. In case of studying the effect of coating materials the uncoated vane measurements are considered as one of these materials.

2 Measuring Concept

The conventional way to measure friction torque, T_n, due to the friction between a vane tip and cam-ring is to measure the total pump torque, T, and then subtract the theoretical torque part, T_{th}, and the friction torque between pump shaft and oil seal and bearings, T_o, according to Eq. 1. T_o is independent of the pressure difference between the delivery and suction sides, P_d - P_s, while T_n is dependent on the pressure difference. The friction torque part, T_o, can be measured as the offset of the curve obtained between the total friction torque, $T_n + T_o$, and the pressure difference.

$$T_n + T_o = T - T_{th}$$
$$\text{where} \quad T_{th} = \frac{V_{th}}{2\pi}(P_d - p_s) \tag{1}$$

The disadvantage of this method is that a large torque needs to be measured, while the friction forces are comparatively small. This method is not suitable for comparing small differences of friction force, such as comparison between different PVD-coated vane tips. The concept presented in this paper is to measure the friction force between the vane tip and cam-ring directly; this will increase the accuracy of measurements.

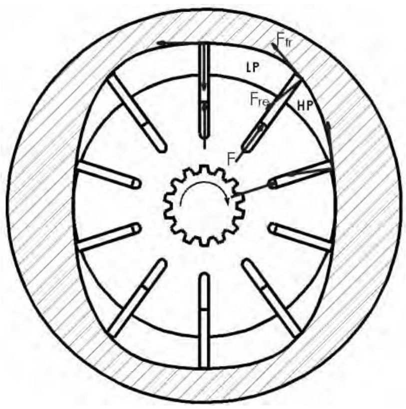


Fig. 1: *Typical vane pump arrangement*

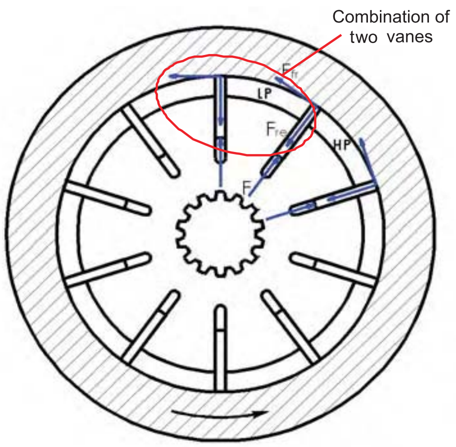


Fig. 2: *Simplified vane pump arrangement measuring the combination of two vanes*

As shown in Fig. 1, if one tries to measure the friction torque, the reaction forces of the cam-ring must be considered. These reaction forces have a component in the same direction as the friction force due to the oval shape of the cam-ring. The value of these reaction forces depends upon the vane force as well as the angle of rotation. Therefore the friction forces were measured according to a simplified arrangement as shown in Fig. 2. The objective is to simplify the oval shape of the cam-ring and to replace it with a cylindrical one. The same peripheral was used to achieve the same average relative velocity between the vane tip and cam-ring. Unfortunately a cylindrical cam-ring cannot create a pressure difference. An external pressure supply was installed to overcome this problem. The cam-ring is being turned and the rotor is fixed by means of a torque sensor. The torque sensor measures the total friction torque between the vane tip and cam-ring. The problem of using this concept is that the combination of two vanes is measured. If one vane rotates against high pressure, then the next vane will rotate against low pressure.

3 Test Rig Setup

3.1 Simplified Vane Pump Arrangement Allowing Single Vane Measurement

The concept is to measure the torque due to the total friction forces. To satisfy this principle the influence of the second vane has to be eliminated. Therefore a gap seal was used instead of the second vane. A relatively large gap should be achieved to avoid any mechanical contact between the rotor and the cam-ring. The gap surface area should be large enough to compensate the leakage due to the gap. This will cause a friction torque due to the oil shear forces within the gap. This torque depends on the speed of rotation and the pressure difference between the two sides of the gap. To keep this torque constant during the run, the speed of rotation and the pressure difference should be constant. Two vanes and two gap seals were used to balance the side forces and the torque as well, as shown in Fig. 3a. The torque sensor is connected to the rotor shaft from one side and to the ground from the other side. The cam-ring is rotated by an electrical motor. The speed of the cam-ring is controlled by a speed adjustable electric motor via a frequency converter.

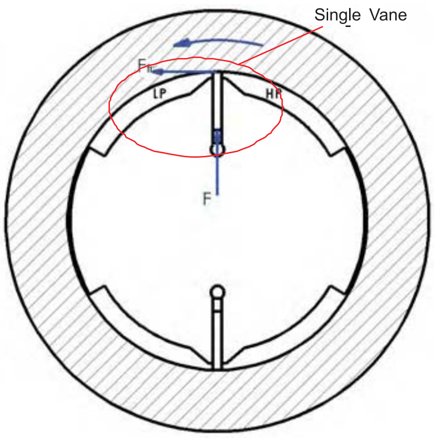


Fig. 3a: *Vane pump arrangement allowing single vane measurement*

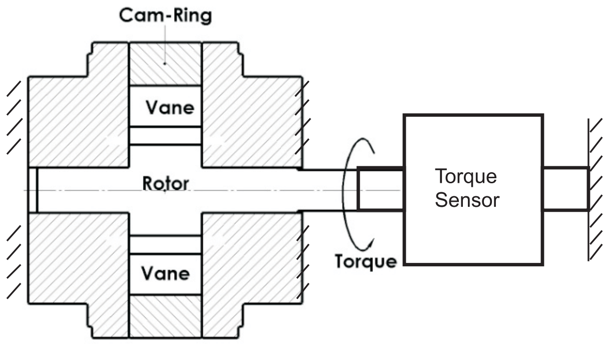


Fig. 3b: *Vane pump arrangement, side view*

To seal the chambers under pressure two side faces were constructed and attached to the ground as shown in Fig. 3b. Gap seals are also used between the faces and the cam-ring as well as to the rotor shaft.

To allow high speed rotations of the cam-ring a relatively large gap should be allowed, that will lead to a significant amount of external leakage. For recharging the associated losses an appropriate oil supply unit was used, and a leakage collecting and recirculating hydraulic circuit was also constructed, as shown in Fig. 4.

3.2 Hydraulic Circuit

The hydraulic circuit shown in Fig. 4 was designed to satisfy the pressure and flow requirements of the test section. The supply unit consists of a primary tank, primary pump, pressure relief valve, check valve, and 3/2-way valve. The high pressure side of the test section is controlled by a pressure control valve directly connected to the pressure supply unit. The low pressure side is connected back to the primary tank by an adjustable valve. By controlling the internal leakage flow, the pressure difference between the high and low pressure sides can be adjusted. The vane pressure is controlled by a pressure control valve that is connected to the primary supply unit. The external leakage caused by the gap seals between the cam-ring and the two faces is collected within the leakage tank mentioned before. This leakage is heavily mixed with air due to the high rotational speed of the cam-ring. The recirculation pump is used to transfer the leakage oil-air mixture to a bubble eliminator tank before the oil flows back to the primary tank.

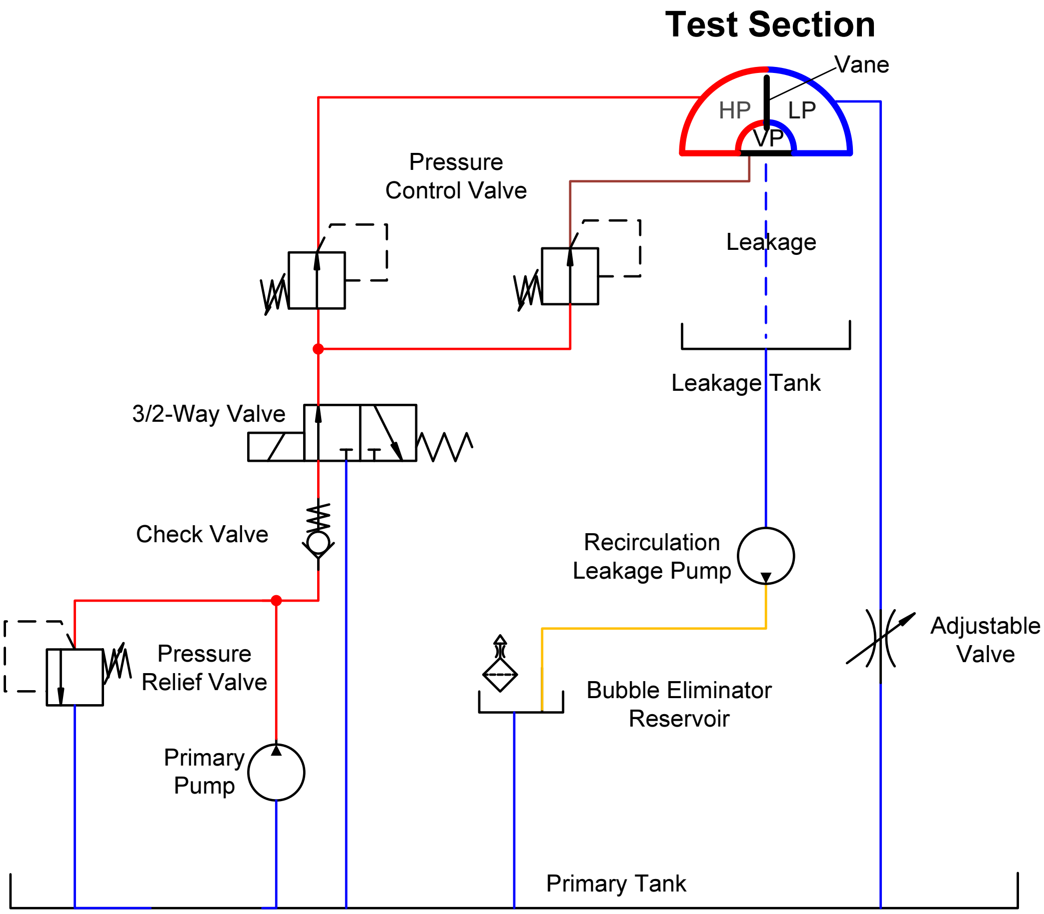


Fig. 4: *Hydraulic circuit*

4 Experimental Results and Discussion

4.1 Concept of Measuring and Error Analysis

All measurements were performed for the oil type HLP46, at an operating temperature of 40 °C.

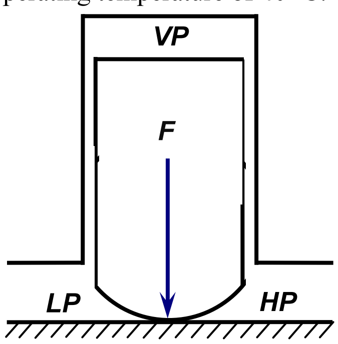


Fig. 5: *Vane hydrostatic balance*

The normal vane force, F, is calculated according to Eq. 2. The pressure distribution curve under the vane tip is not linear at the condition of relative speed between vane tip and came-ring. This curve depends on the speed of rotation according to the Reynolds equation. The normal vane force value is based only on the hydrostatic pressure balance at the condition of no relative speed between the vane tip and cam-ring as a reference value as shown in Fig. 5.

$$F = VA\left(VP - \frac{HP + LP}{2}\right) \tag{2}$$

The offset at the origin of Fig. 6 is due to friction within the rotor bearings and shear torques within the gap seals between the rotor and cam-ring and the two side faces. The shear torque within the gap has two components. The first component is due to the pressure difference between the two sides of the vane, and the second component is due to the relative movement due to the cam-ring rotation. These two torque components are in the same direction if the vane moves against high pressure, while they are in opposite directions if the vane moves against low pressure. The friction due to rotor bearings is always opposite the rotating direction. This means that the offset of the measured curve at the origin can be positive or negative according to the frictional torque balance. The slope of the curve of Fig. 6 indicates the friction coefficient, λ, as defined in Eq. 3. The solid line indicates the curve fitting of the experimental measurements after subtracting the offset value to show the friction force without disturbances. A speed of 2000 rpm was selected as an example. The friction coefficient, λ, will be used as a comparison parameter.

$$\lambda = \frac{F_{fr}}{F} \tag{3}$$

Error analysis (repeatability) for the measured data was performed at 2000 rpm rotational speed and 0 bar pressure difference. The same curve was measured four times under the same conditions and the error was determined as shown in Fig. 6. The friction factor coefficient, λ, varies by ±6.6 % (or simply ±7 %) for the four curves, which can be considered as the error margin of λ. The maximum error of the data is 10.4 % and was at lower values of friction force around 2.6 N. The minimum error is 2.8 % and is at higher values of friction force around 9.7 N. The repeatability analysis shows that the error for each point is almost a constant (around 0.27 N).

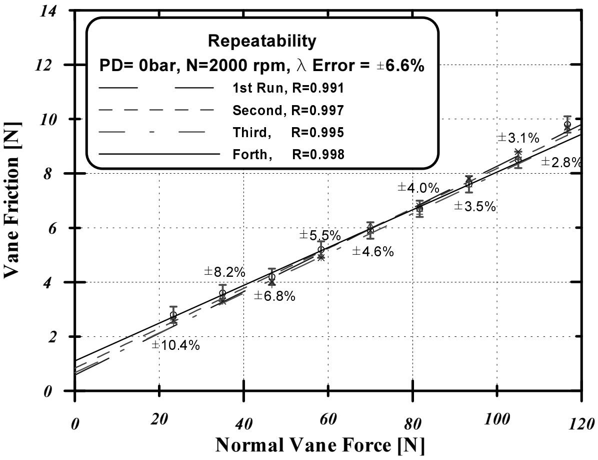


Fig. 6: *Relationship between vane friction and normal vane force showing repeatability*

Figure 7 shows the error behaviour for the test section measurements. According to this error behaviour the smallest value of the measured friction force should not be less than 2.6 N; that makes the maximum error 10.4 %.

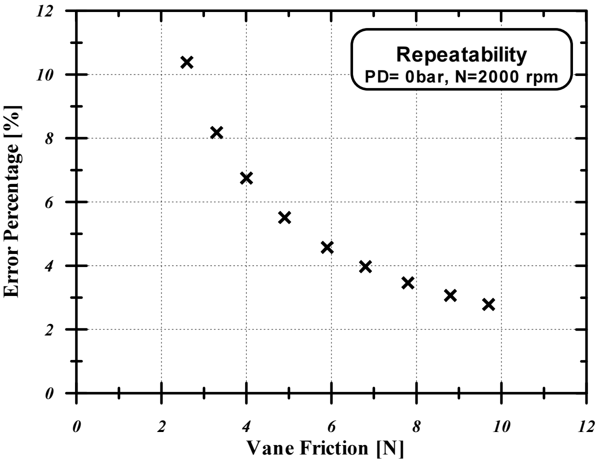


Fig. 7: *Error behaviour of the measurements*

4.2 Effect of Relative Speed on Friction Coefficient

Each value of the friction coefficient, λ, was measured as the slope of the curve as shown in Fig. 6. Figure 8 shows that the relative speed between the vane tip and cam-ring has a strong effect on the friction force. Increasing the speed of rotation decreases the friction force between the vane tip and cam-ring. The speed of rotation should not be less than 2000 rpm (4.8 m/s) relative speed to limit surface contact reducing friction and wear. A comparison with (Faber, 2006) results shows a good agreement between those results and the findings obtained here. The shift in the results is due to the different temperature used. Since 80 °C oil temperature is used by (Faber, 2006), the friction coefficient proposed is lower than for oil at 40 °C due to a decrease in oil viscosity. The results obtained by (Inaguma and Hibi, 2005) show a slightly different trend.

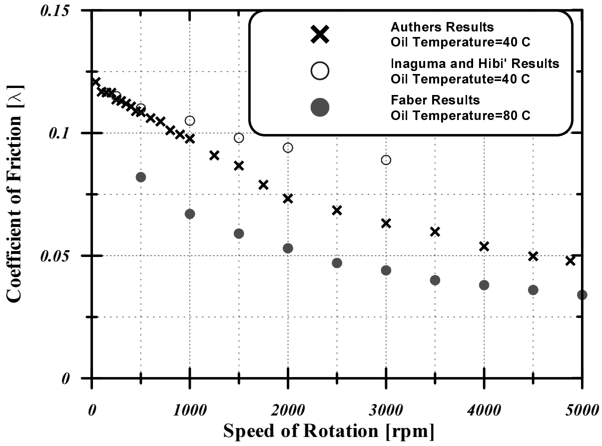


Fig. 8: *Coefficient of friction, λ, vs. speed of rotation*

4.3 Effect of Vane Force on Friction Coefficient

Figure 9 shows that the effect of normal vane force on the coefficient of friction is very small.

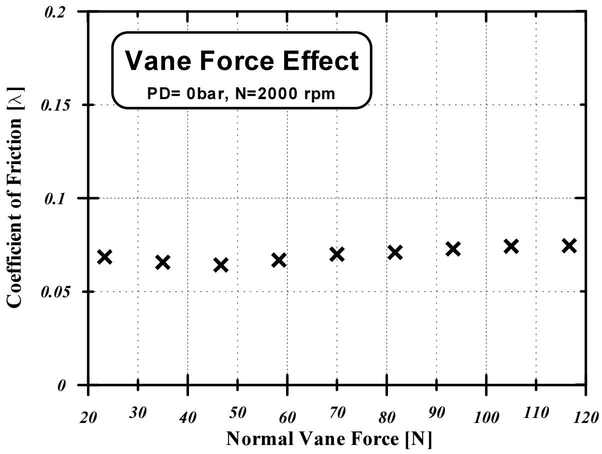


Fig. 9: *Relationship between coefficient of friction, λ, and normal vane force*

Figure 10 shows the effect of under vane pressure on the coefficient of friction (Faber, 2006). The under vane pressure indicates the normal vane force. The results were taken at an oil temperature of 80 °C. The trend of the results at 500 rpm shows almost no effect of normal vane force on the coefficient of friction. At 2500 rpm, there is a slight increase in the coefficient of friction with a rise of the normal vane force. The results at 5000 rpm show an unclear trend.

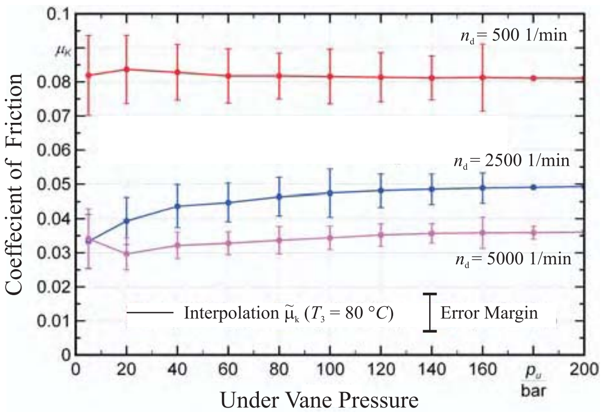


Fig. 10: *Relationship between coefficient of friction, λ, and under vane pressure (Pu) (Faber, 2006)*

4.4 Effect of Pressure Difference between the Two Vane Sides on the Friction Coefficient

Figure 11 shows the effect of the pressure difference on the friction coefficient. The trend of pressure difference measurements in Fig. 11 is not clear. The majority of the measured points are almost the same except two points at -10 bar and -20 bar which gives an indication that the effect of pressure difference on the friction coefficient is small. The positive sign of the pressure difference indicates that the vane moves against high pressure, while the negative pressure difference indicates the opposite.

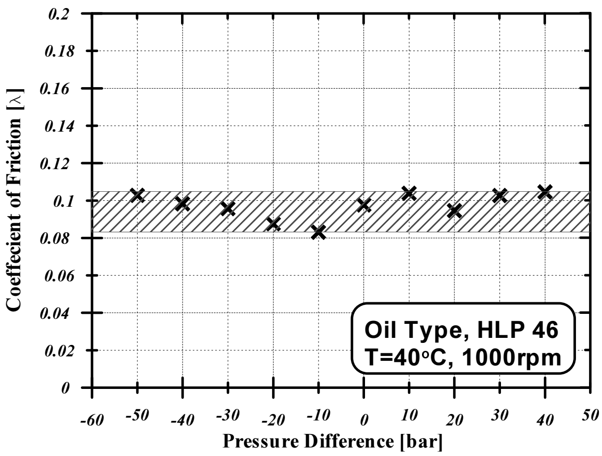


Fig. 11: *Relationship between coefficient of friction, λ, and pressure difference between the two vane sides*

4.5 Effect of PVD-Coatings on Friction Coefficient

4.5.1 PVD-Coating Materials

Five different PVD-coatings are used, Me1, Me2, Zr1, Zr2 and ZrCg. They were provided by the project partner based on process capability and capacity. Table 1 shows the five material details. The incentive of this study is to discover the best PVD-coating material regarding friction between PVD-coated vanes and cam-ring. PVD-coatings based on carbon system with materials (Zirconium, Chromium and Titanium) were used. Zr1, Zr2 and ZrCg are based on Zr. The coating system ZrCg has a graded hardness profile with a maximum hardness in the centre of the coating, as shown in Fig. 12. Zr1, Zr2 and ZrCg were produced by sputtering of two Zirconium targets at a temperature of 160 °C. The Zr coatings were synthesized by varying process parameters reactive gas flow, substrate bias voltage and pulse duty and pretreatment.

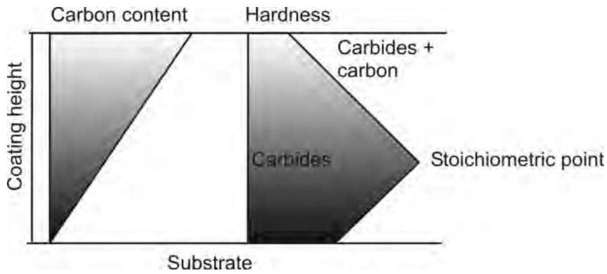


Fig. 12: *Schematic diagram of carbon and hardness distribution of ZrCg-coating*

Figure 13 shows the nanocomposite of Me-coatings. The coating system Me1 and Me2 consist of varying Ti- and Cr- rates. Figure 14 shows the nanostructure of the PVD-coatings with base material.

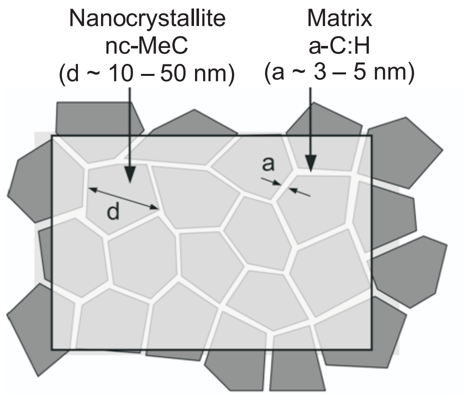


Fig. 13: *Schematic diagram of nc-MeC nanocomposite construction [Source: EPF, Lausanne]*

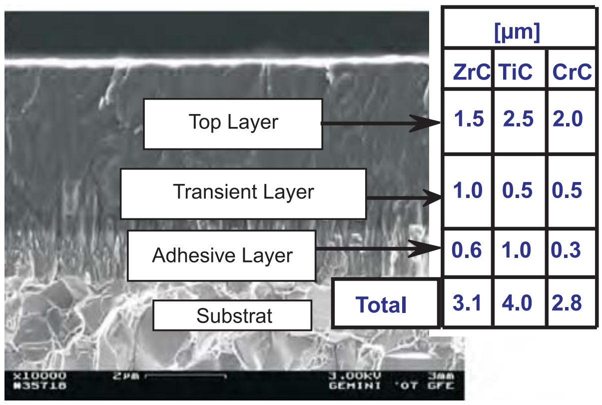


Fig. 14: *Structure of PVD-coatings with nanocomposite [Source: IOT, RWTH Aachen]*

Table 1: PVD-Coating materials

	Carbide base	Hardness [GPa]	E-Mod [GPa]	Thickness [µm]
ZrCg	Zirconium	16.5	172.6	4.0
Zr1	Zirconium	12.9	120.5	3.5
Zr2	Zirconium	19.4	147.6	3.3
Me1	Titanium	16.27	128.7	4.0
Me2	Chromium	20.23	163.5	2.8

Steel S3-3-2 (60-66 HRC) material was used for uncoated vanes, while 100Cr6 (58-63 HRC) material was used for uncoated rings.

4.5.2 Experimental Results

Figure 15 shows the effect of PVD-coated vanes on the friction at various rotational speeds, 1000, 2000, 3000 and 4000 rpm respectively in the case of no pressure difference between the two vane sides. The error margin of the uncoated vane is taken as a reference margin for the coated-vanes results (hatched area). The analysis of the friction behaviour as shown in Fig. 15 indicates that there is no significant effect of the coating material.

Figure 16 shows the average curve calculated from the results of Fig. 15. All average points lays within the error margin of the uncoated vane which confirms that the effect of coating materials is not significant.

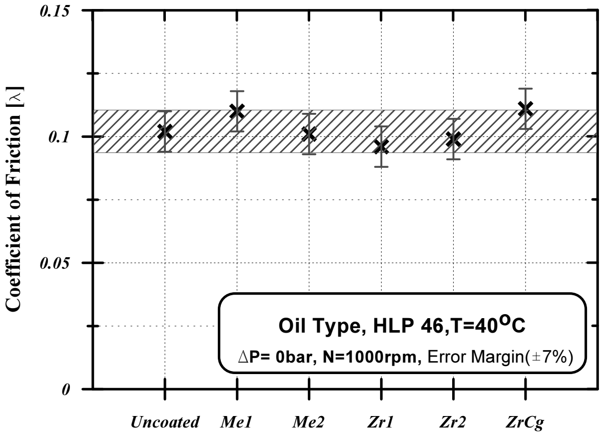


Fig. 15a: *Effect of coating material on the coefficient of friction, λ, at 0 bar ΔP and 1000 rpm*

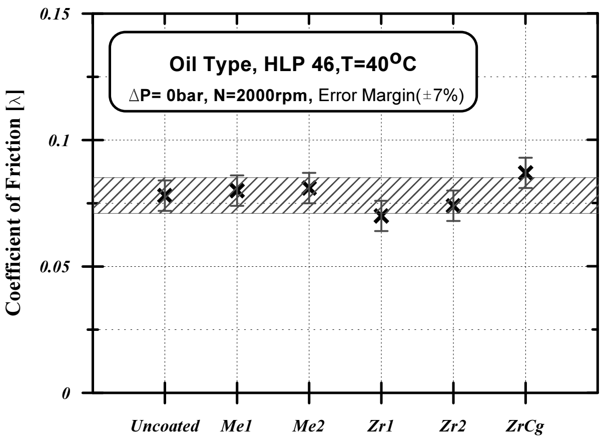


Fig. 15b: *Effect of coating material on the coefficient of friction, λ, at 0 bar ΔP & 2000 rpm*

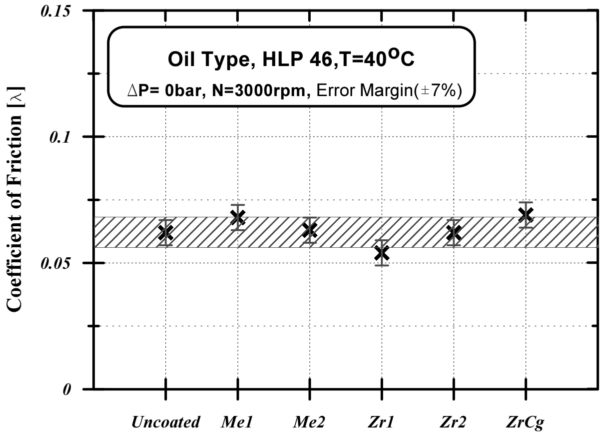


Fig. 15c: *Effect of coating material on the coefficient of friction, λ, at 0 bar ΔP & 3000 rpm*

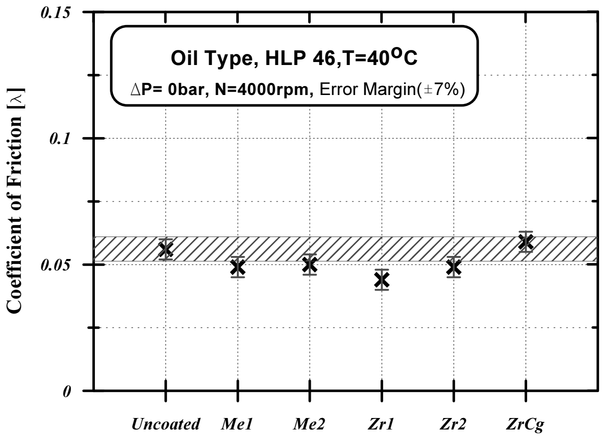


Fig. 15d: *Effect of coating material on the coefficient of friction, λ, at 0 bar ΔP & 4000 rpm*

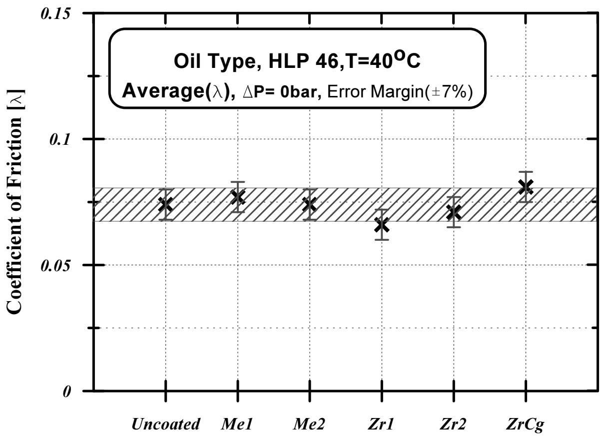


Fig. 16: *Average effect of coating material on the coefficient of friction, λ, at 0 bar ΔP*

The results obtained for PVD-coated pistons of an axial piston machine (Scharf and Murrenhoff, 2006) show that using ZrCg coating material reduces friction between the piston and bushing. This distinction could be explained that the line contact between the vane tip and cam-ring and the curvature of the vane tip helps to bring significant amount of lubrication film between sliding parts thus almost avoiding contact.

5 Recommendations

The following points are recommended for future work:

- It is not necessary to consider the pressure difference between the two sides of the vane. That will simplify the test setup and at the same time the measurements will not be much different from the results shown here.
- To investigate the effect of coating materials on the vane friction the test should be carried out without lubrication just like water or air fluid medium.
- The coating materials should be studied from the wearing point of view.
- Because the suction part shows the maximum vane friction, it is feasible to concentrate future studies on reducing vane friction within the suction part.

6 Conclusions

In the presented study an experimental investigation was conducted to measure the friction between a vane tip and cam-ring within oil vane pumps. HLP46 oil type at 40 °C was used. A comparison was made between five different hard and smooth PVD-coating materials and the standard vane without coating which can be considered as the state of the art. The friction was measured directly to increase the accuracy of the measurements.

The following conclusions were obtained:

- Relative speed between vane tip and cam-ring should be higher than 4.8 m/s, around 2000 rpm for reduced friction.
- The effect of the normal vane force on the friction coefficient is very small.
- The effect of the pressure difference between the two sides of the vane on the friction coefficient is very small compared to the effect of the relative speed between the vane tip and cam-ring.
- Different coating materials show no significant effect on the friction within oil vane pumps due to good lubrication conditions of the sliding parts. This is different from the piston bushing situation in a swash plate piston unit.

Nomenclature

λ	Friction coefficient	[-]
F	Normal vane force	[N]
F_{fr}	Friction vane force	[N]
F_{re}	Reaction vane force	[N]
HP	High pressure	[bar]
LP	Low pressure	[bar]
VP	Vane pressure	[bar]
ΔP	Pressure difference between vane sides	[bar]
VA	Vane cross sectional area	[m²]
T	Total pump torque	[Nm]
T_{th}	Theoretical pump torque	[Nm]
T_n	Friction torque between vane tip and cam-ring	[Nm]
T_o	Friction torque due to oil seals and bearings	[Nm]
V_{th}	Pump displacement	[m³]
P_d	Delivery pressure	[bar]
P_s	Suction pressure	[bar]

References

Bebber, D. van and **Murrenhoff, H.** 2002. Metal/carbon layers (ZrCg and HfCg) to reduce wear and friction in hydraulic components. *3rd International Fluid Power Conference*, Aachen, Germany, Vol. 2, pp. 443-455.

Cho, I. S., **Oh, S. H.**, **Song, K. K.** and **Jung, J. Y.** 2006. The lubrication characteristics of the vane tip under pressure boundary condition of oil hydraulic vane pump. *Journal of Mechanical Science and Technology*, 20 (10), pp. 1716-1721.

Faber, Ingo. 2006. *Theoretische und experimentelle Untersuchung der Flügelkopfreibung in einer Flügelzellenpumpe*. Dissertation. Bochum University, Germany.

Inaguma, Y. and **Hibi, A.** 2005. Vane pump theory for mechanical efficiency. *Proceedings of the Institution of Mechanical Engineers Part C-Journal of Mechanical Engineering Science*, 219 (11), 1269-1278.

Inaguma, Y. and **Hibi, A.** 2007. Reduction of friction torque in vane pump by smoothing cam ring surface. *Proceedings of the Institution of Mechanical Engineers Part C-Journal of Mechanical Engineering Science*, 221 (5), pp. 527-534.

Scharf, S. and **Murrenhoff, H.** 2006. Wear and friction of ZrCg-coated pistons of axial piston pumps. *International Journal of Fluid Power*, Vol. 7, No. 3, pp. 13-20.

Sui, P. C. 1995. Prediction of film thickness and friction at a rotary pump blade and liner interface. *American Society of mechanical Engineering (ASME)*, Vol 72, ASME, New York, pp. 115-122.

Ortwig, H. 1992. Theoretical analysis of physical load parameters in the tribological system, vane/cam-ring/hydraulic oil of a vane pump shown. Pt. I: Load calculations of the vane pump shown. *Tribologie und Schmierungstechnik*, Hannover, 39 (1992) 4, pp. 219-226.

Ortwig, H. 1992. Theoretical analysis of physical load parameters in the tribological system vane/cam-ring/hydraulic oil of a vane pump shown. Pt. II: Calculation of the lubricated contact parameters. *Tribologie und Schmierungstechnik*, Hannover, 39 (1992) 3, pp. 144-151.

Ortwig, H. 1992. Pressure and temperature measurements in tribosystem blade/lifting/ring/hydraulic oil of a vane cell pump. *Tribologie und Schmierungstechnik*, Hannover, 39 (1992) 6, pp. 339-347.

Appendix 1

Dimensions of the vane and the ring

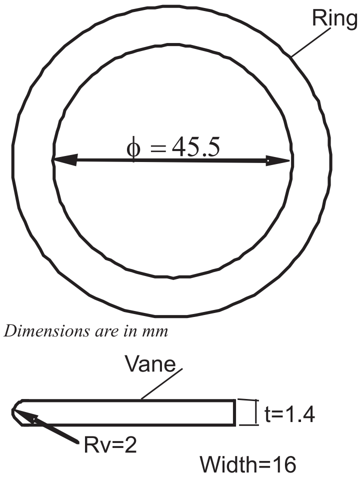


Fig. 1A1: *Dimensions of the vane and the ring used in the presented investigation*

Appendix 2

Samples of the curves used to predict the friction coefficient

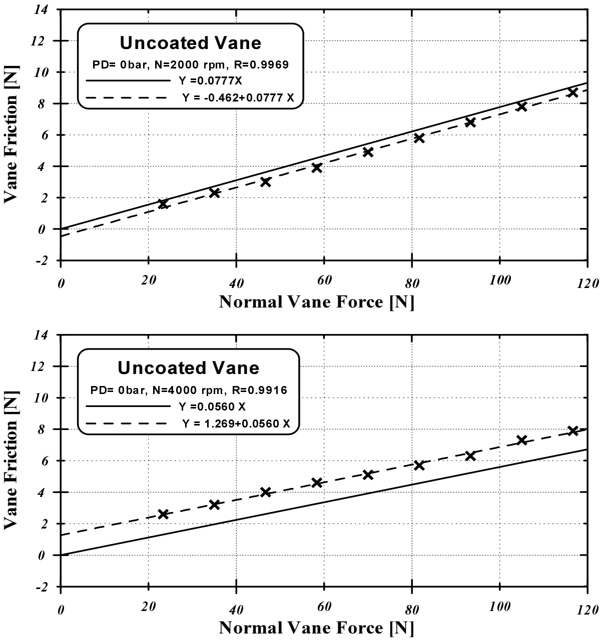
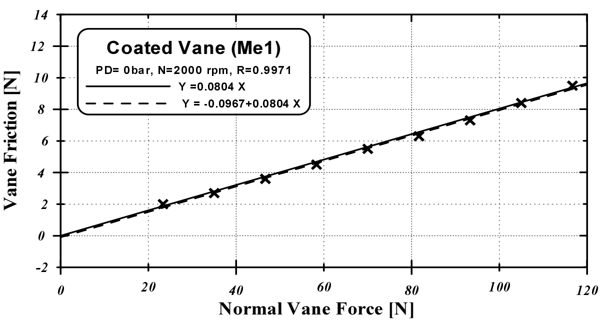


Fig. 1A2: *Sample of curves used to predict λ in the case of uncoated vane*



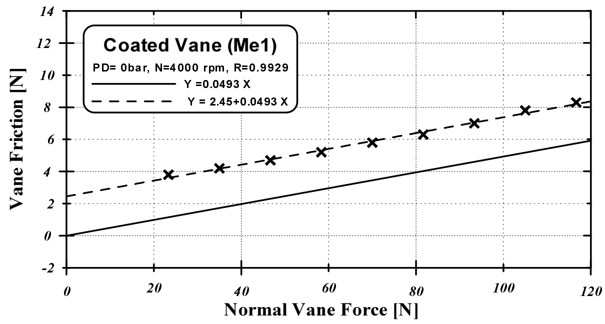


Fig. 2A2: *Sample of curves used to predict λ in the case of vane coated by Me1 material*

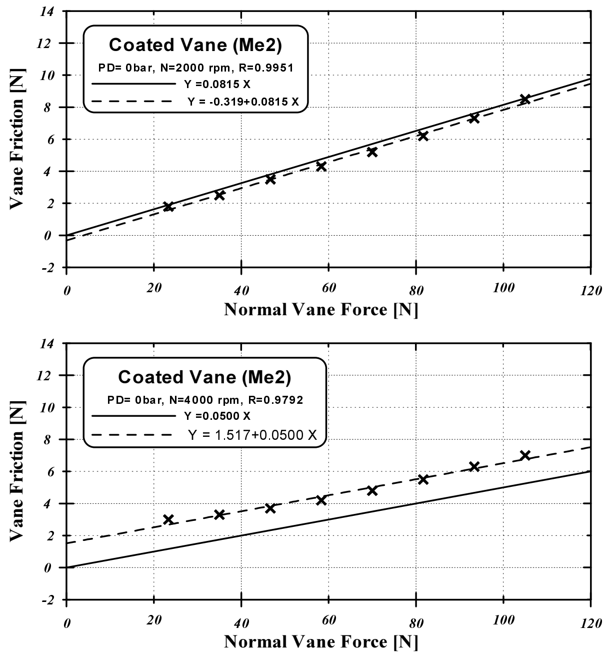


Fig. 3A2: *Sample of curves used to predict λ in the case of vane coated by Me2 material*

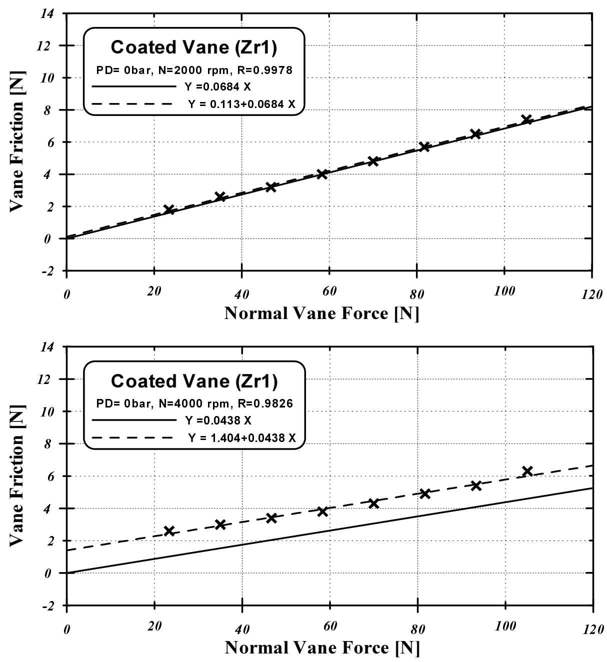


Fig. 4A2: *Sample of curves used to predict λ in the case of vane coated by Zr1 material*

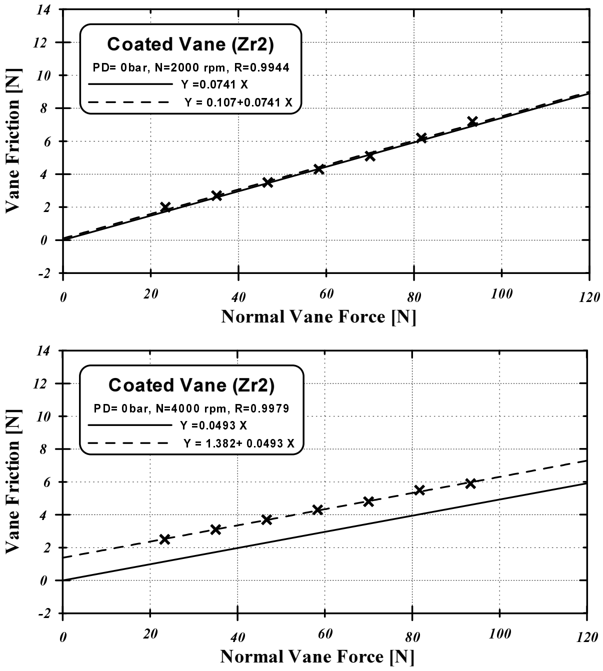


Fig. 5A2: *Sample of curves used to predict λ in the case of vane coated by Zr2 material*

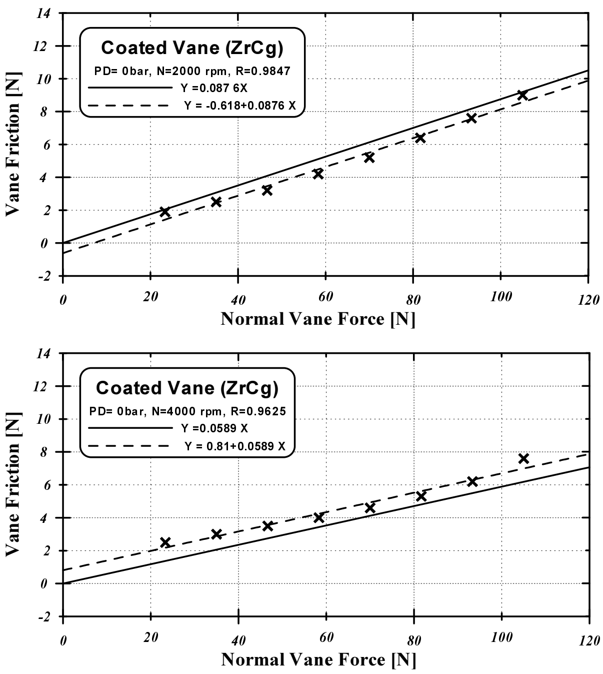


Fig. 6A2: *Sample of curves used to predict λ in the case of vane coated by ZrCg material*

Mohamed El Ashmawy
Born in Cairo, Egypt in 1969. He acquired his Master's degree in Mechanical Engineering in 2001, at Mansoura University, Egypt, with a thesis title: Improvements in BWR Cores Using Intra-Fuel-Bundle Jet Pumps. From 2006 to 2008, he was a scholarship holder as a Ph.D. student at the Institute for Fluid Power Drives and Controls (IFAS) at RWTH Aachen University, Germany. Now he is a teaching assistant at Engineering Science Department, Faculty of Petroleum and Mining Engineering, Suez Canal University, Egypt.

Hubertus Murrenhoff
Born August 13th, 1953 he is Director of the Institute for Fluid Power Drives and Controls (IFAS) at RWTH Aachen University, Germany. Main research interests cover hydraulics and pneumatics including components, systems, controls, simulation programs and the applications of fluid power in mobile and stationary equipment.