Heavy Movable Structures, Inc.

SEVENTH BIENNIAL SYMPOSIUM

November 4 - 6, 1998

Grosvenor Resort
Walt Disney World Village
Lake Buena Vista, Florida

"Design of Optimized Seals for Leak Free Hydraulic Cylinders for Movable Bridges"

by

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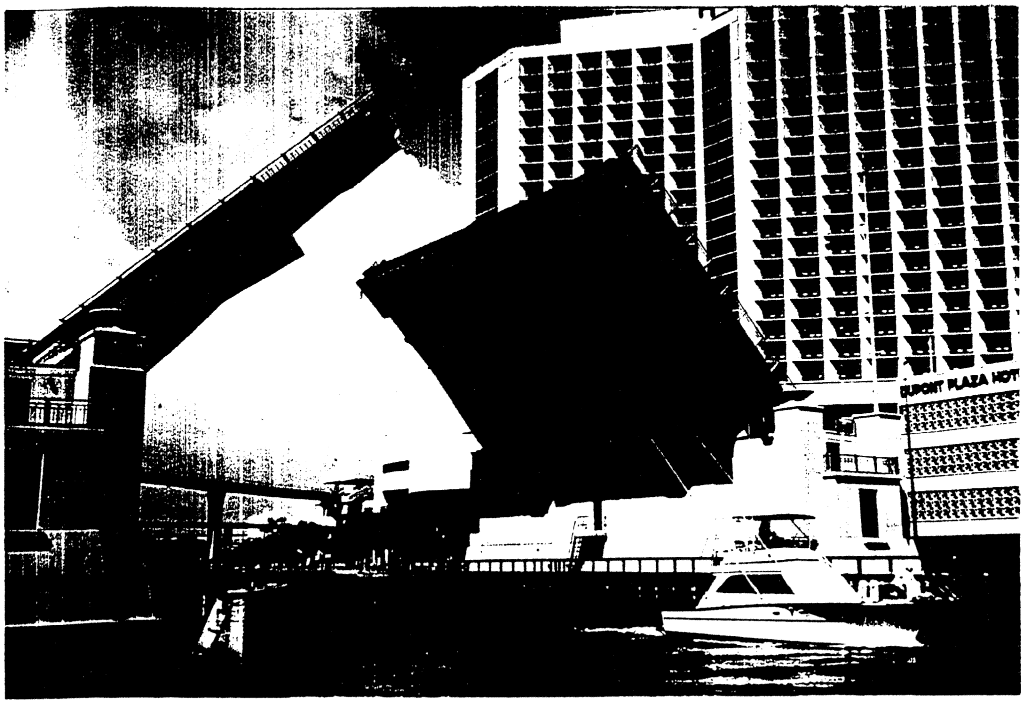

Design of Optimized Seals for Leak Free Hydraulic Cylinders

HMS Heavy Movable Structures, Inc.
7th Biennial Symposium, Orlando FL
November 4, 5 and 6 1998

Presented by:

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July 1998

HYF 10003/3

Design of optimized seals for leak free hydraulic cylinders

Title:

Research on the total sealing concept of reciprocating seals running against ceramic and chromium plated rods for hydraulic cylinders.

Abstract

The paper deals with the total sealing concept of reciprocating rod seals. The fundamental basis of this approach is the design of seals based upon understanding of the phenomena in the sealing contact. As a first step the surface profile of the rod is described in detail with a modern non-contacting measuring method. This is done for a rod with conventional chromium plating and for a rod which has a new technology "thermo-sprayed" ceramic layer for high corrosion resistance. Secondly material characteristics of different seal materials are established in tests such as stress/strain relations, dynamic elastic modulus and wear- and friction values. Basic sealing criteria are determined under static conditions at different temperatures.
Finite element computer analysis is used to determine the state of stress and deformation of the seal concept. Finally, for a verification of the scientific development work with reality, a test unit has been built which is capable of testing the possible seal solutions under different operating conditions.

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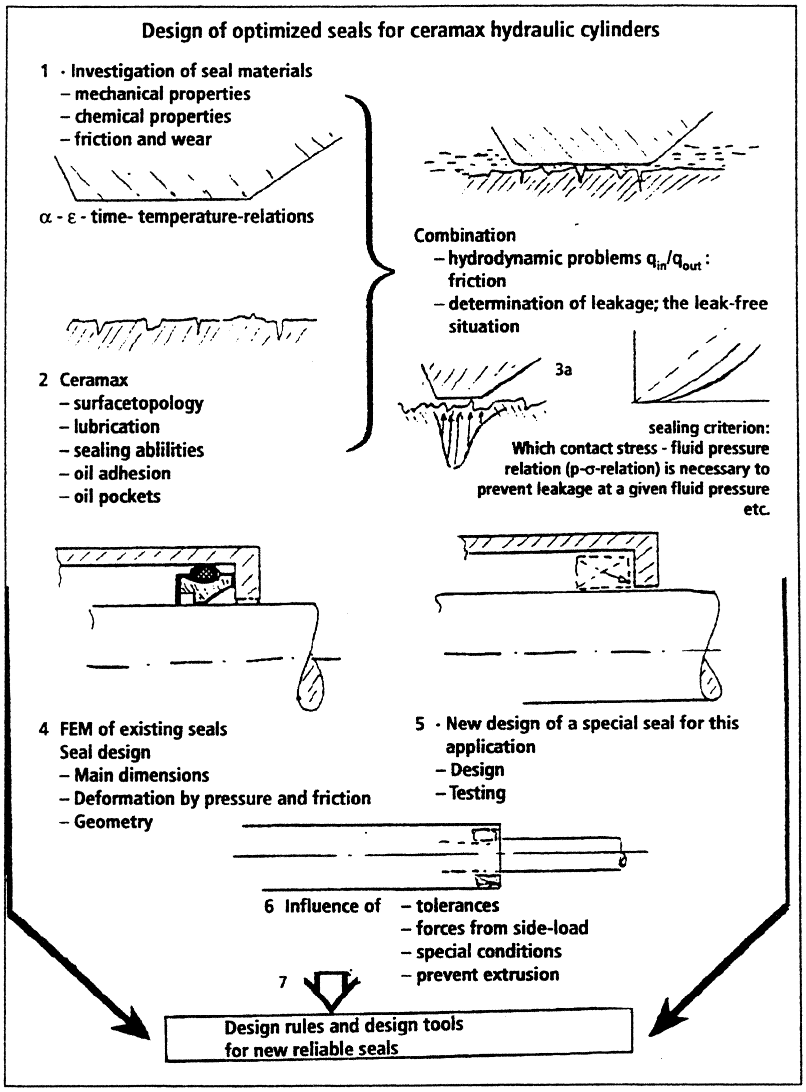

fig 1: Design of optimized seals for ceramax hydraulic cylinders

Introduction

Seals are critical parts in modern hydraulic cylinders as well as in other rotating or reciprocating hydraulic/mechanical equipment. Nowadays it is necessary to develop a 100% leak free hydraulic cylinder. On the other hand friction must be as low as possible to reduce wear and loss of energy. A leak free cylinder with minimum wear and friction means that the seal is running on an extremely thin oil film (in- and outstroke). This means that more care should be taken to prevent "stick-slip" phenomena.

The aim of the work, presented here is to develop a new design strategy for hydraulic cylinder seals. Special attention is paid to cylinders with a ceramic-coated rod. This cylinder type for heavy duty conditions shows a steady growing field of applications in all sectors of engineering world wide. The initial development of cylinder rods protected with a "thermo-sprayed" ceramic non-conducting layer, is because of its much better protection against corrosion in comparison with conventional chromium plated rods. A second important reason lies in the fact that ceramic plated rods are much better protected against any mechanical damage from external influences leading to a big improvement in total lifetime of a cylinder. The possibility to have a cylinder integrated measuring system protected against any

Design of optimized seals for leak free hydraulic cylinders

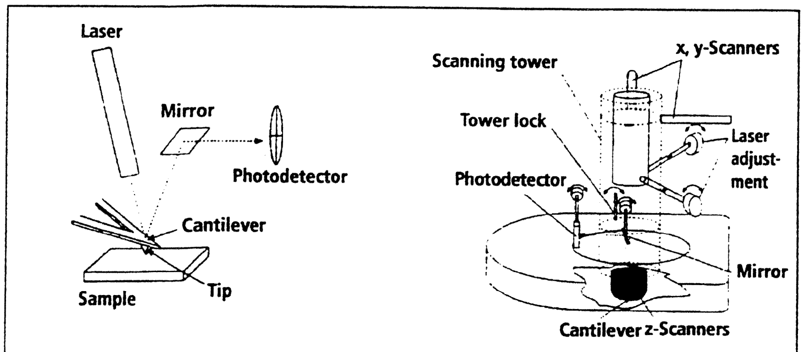

fig. 2: AFM measuring equipment

Average table CERAMAX :

– **Roughness:**

Ra:	201 nm	biggest valley	Ra:	528 nm	without valley	Ra: 183 nm
Rms:	292 nm		Rms:	632 nm		Rms: 257 nm
Mrng:	2869 nm		Mrng:	2529 nm		Mrng: 2332 nm

– **Area Measurement:** Projected area: 10000 µm_
Surface area: 10262 µm_

– **Fractal:**

lake z-Threshold:	50 µm_	2 lakes	4 µm_	12 lakes
(default)	25 µm_	5 lakes	1 µm_	20 lakes

– **Peak/Valley Analysis:**

Orientation	0° (hor.)	0°	90° (ver.)	90°
Peak Height	0 nm	30 nm	30 nm	0 nm
Mean Peak Spacing	2.92 µm	4.70 µm	3.39 µm	1.98 µm
Mean Peak Angle	2.01 °	2.57 °	4.03 °	3.19 °

– **Line-Measurement/Analysis 2D (through biggest valley):**

Ra: 382 nm	bigges valley	hor. side-angle	21 °
Rp: 457 nm		ver. side-angle	24 °
Rpm: 367 nm		depth valley	2256 nm
Rt: 2416 nm		hor. width valley	16 µm
Rtm: 978 nm		ver. width valley	17 µm

– **Bearing Ratio figure:**

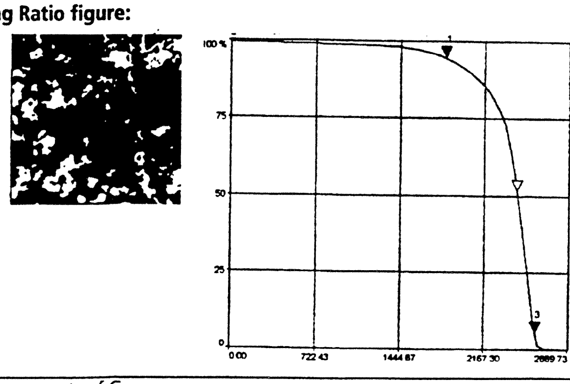

fig. 3: Measurements of Ceramax

external influence is the third reason why this ceramic coating has been developed. Finally, it has to be said that, a chromium layer is also not a very environment friendly production method. Because of the different surface conditions in a ceramic rod cylinder design, a special seal concept is necessary in order to reach optimized service conditions.

In recent decades scientific research on elastomeric contact seals has led to an enormous increase of knowledge on the fundamental mechanisms and the important parameters for seal operation. Inspite of these research activities the working mechanism of elastomeric seals is not yet well understood in all details.

The present fundamental knowledge has in general not yet led to important changes in the practical approach towards seal design and application.

In this approach an integration was made of fundamental research and development work on seals by the Eindhoven University of Technology and the industrial development work together with longtime field experience of Hydraudyne Cylinders B.V. Boxtel, Holland.

The different steps carried out, are presented in figure 1 in a schematic way.

The fundamental basis of the approach is the design of seals based on an understanding of the phenomena in the sealing contact, so on the combination of "know why" and "know how". With these tools, new seals are designed and tested under service conditions.

As it is presented in figure 1 an important design parameter is the polymer or polymer compound to be used for the seal. Mechanical properties under service conditions and friction/ wear behaviour when running against the counter face material, need to be known.

The second influencing factor is the surface topology of the rod coating. This has a direct influence on the micro deformation and on the lubricating and sealing conditions in the sealing contact.

The combination of seal material properties, surface topology, fluid parameters as well as geometry and service conditions is responsible for the fluid film formation in the contact and so for friction and leakage. The sealing criterion, defining the minimum necessary contact stress to prevent leakage for a given fluid pressure to be sealed, is important in this situation.

In order to design a seal geometry including the sealing lip(s) and the overall cross section, FEM-simulations are used and an optimization of the design can take place. Those calculation results however are worthless if these are not based on the real material properties. The real stress-strain-time-temperature behaviour under service conditions must be used. The main stress situations in the seal are non-uniform three-dimensional states of compression, whereas other points in the cross section show combinations of 3 D - compression and tension. As a result FEM - calculations using material results from one-dimensional tension results are of no value. At least compression results must be used. A new technique for non-uniform 3 D - compression testing of seal materials was developed. First tests were started with this equipment.

After production of prototype seals, extensive testing takes place using a specially designed test facility for hydraulic cylinders seals and where variation of all important parameters can be carried out.

Seal Contact Phenomena Surface Topography:

Characterization of chromium and Ceramax surfaces

For investigation an AFM (Atomic Force Microscopy) force sensor, figure 2, has been used. This will measure at the surface the deflection of a cantilever. A tip is mounted on the cantilever such that, when the cantilever moves, the light beam from a small laser moves across the face of a four section photo detector. The amount of motion of the cantilever can then be calculated from the difference in light intensity on the sectors.

Hooke's Law gives the relation-ship between the cantilever's motion, x, and the force required to generate the motion, F.

$$F = -k \cdot x$$

The control of the SPM (Scanning Probe Microscope) sensor over extremely small distances is made possible by the use of piezoelectric ceramics. A feedback electronic circuit controls the positioning mechanism. SPM instruments magnify in three dimensions, the x, y and z axes with a resolution of several nanometres which is dependent of the probe tip geometry.

Processing of data is possible with the Image Display & Analysis of the Topometrix software. All data about statistics and the geometry can be viewed and analysed such as:

Average table CHROMIUM:

- Roughness:

Ra:	51 nm	biggest	Ra:	52 nm	without	Ra:	47 nm
Rms:	65 nm	valley	Rms:	64 nm	valley	Rms:	59 nm
Mrng:	450 nm		Mrng:	337 nm		Mrng:	429 nm

– Area Measurement:

Projected area:	10000 µm_
Surface area:	10060 µm_

– Fractal:

lake z-Threshold:	50 µm_	6 lakes	4 µm_	18 lakes
	25 µm_	7 lakes	1 µm_	34 lakes

– Peak/Valley Analysis:

Orientation	0° (hor.)	0°	90° (ver.)	90°
Peak Height	0 nm	30 nm	30 nm	0 nm
Mean Peak Spacing	2.31 µm	5.62 µm	12.87 µm	1.56 µm
Mean Peak Angle	0.78 °	0.98 °	1.84 °	1.17 °

– Lines-Measurement/Analysis 2D (through biggest valley):

at work-	Ra:	51 nm	in biggest	Ra:	22 nm
direction	Rp:	134 nm	valley	Rp:	73 nm
	Rpm:	92 nm		Rpm:	42 nm
	Rt:	338 nm		Rt:	124 nm
	Rtm:	212 nm		Rtm:	70 nm

– biggest valley at workdirection of the seal

Side-angle	6 °
Biggest valley	219 nm
Width valley	5 µm

– Bearing Ration figure :

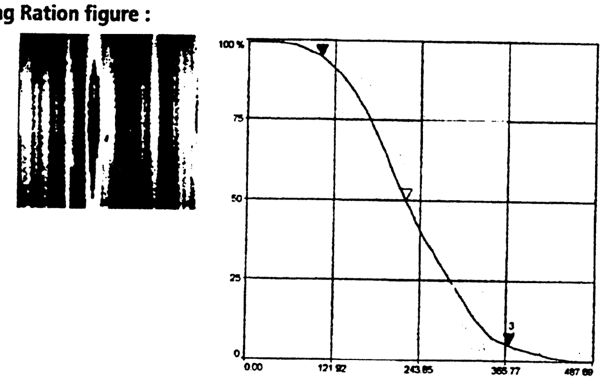

fig. 4: Measurements of chromium

- Roughness of the surface (Ra, Rms, Max. Range), analysed with or without a chosen area.
- Measurement of the projected surface and the real surface.
- Fractal analysis gives a view of a chosen section of the surface divided in above and beyond that level (z-threshold).
- Bearing ratio is important to determine the real contact at the surface of a counterpart.

Examples of measured results are presented in figure 3 and 4.

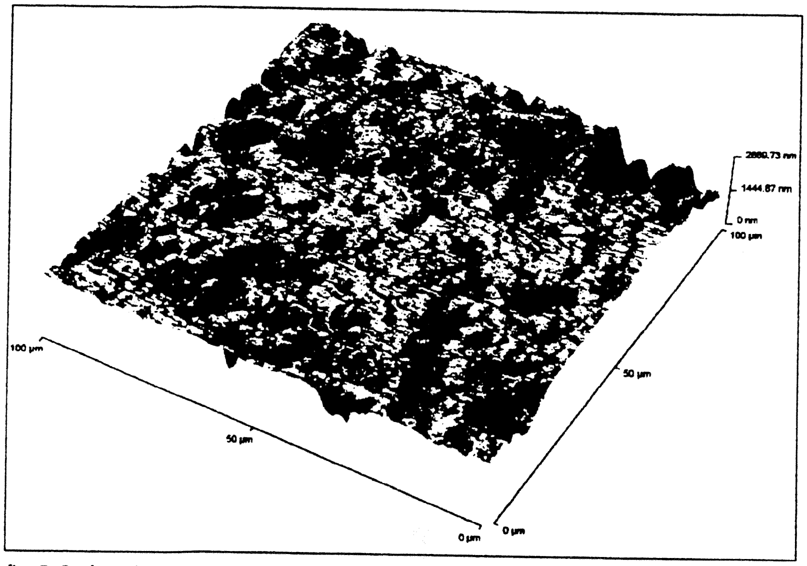

fig. 5: Surface characteristics of Ceramax

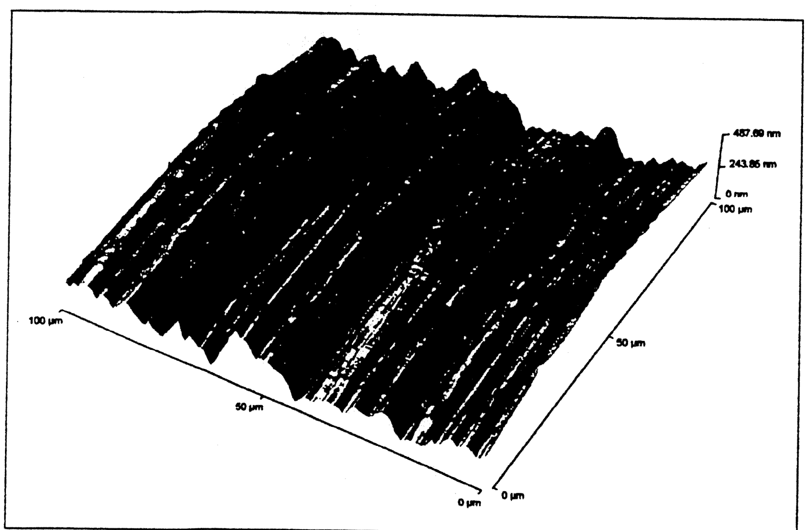

fig. 6: Surface characteristics of chromium

Representative 3 D-views and measured values of Ceramax and chromium are shown in figure 5 and 6 (see Appendix).
A comparison of results for Ceramax and chromium gives the following conclusions:

- Chromium shows a regular wave surface pattern.
- Ceramax shows a random surface profile.
- The space between two following peaks is bigger at the surface of chromium, but the angle of that peak is smaller than of the peak at the Ceramax surface (at mean measurements).
- The surface area of Ceramax is much bigger than for chromium (also the randomized character is influencing the surface area).
- Chromium has not much height-difference compared to Ceramax.
  The biggest valley of Ceramax has (relative) high measure-values and has almost the same value of width and length compared to a 'line'-valley of a chromium surface.
- The angle of this Ceramax valley is almost four times bigger compared to a chromium surface.

The behaviour of seals against Ceramax will be different compared to chromium.

Hydrodynamic phenomena in the sealing contact

In fact the rod-seal-contact is an elastohydrodynamic (EHD) contact. On a macro-scale, hydrodynamic effects are combined with the influences of elastic deformations of the seal-surface.
Aspects of the EHD-phenomena for reciprocating seals were studied by Kanters (1990), where special seal geometries were studied under conditions of relatively high speed.
Normal operating conditions in seal contacts as studied here show a lower speed and a different seal-lip geometry.
For the seals studied here, a general shape is presented in figure 7, from Kuiken (1996) who studied especially the sealing behaviour and seal design for shock absorbers.

On a macroscopic scale first, the contact problem can be treated as an EHD-contact with smooth surfaces where the inverse hydrodynamic lubrication theory after Blok (1963) is used to solve the film and pressure problem.
On a microscopic scale the surface topography has to be taken into account as it is also presented in figure 7.

When solving the macroscopic problem as described by Kuiken (1996), pressure profiles under the lip are determined. Examples of the pressure profiles, depending of the direction of rod speed are presented in figure 8.

Those profiles are calculated for a typical non-symmetric sealing lip, also according to the schematic drawing of figure 7 and resulting in a high pressure gradient at the oil side and a low

pressure gradient at the air side of the seal. Details of the calculation are presented in Kuiken (1996) and Van Dijnsen (1995).
A non-symmetrical seal geometry results in a non-symmetrical pressure profile under the sealing lip, and is essential for a well functioning seal.
Under normal operating conditions of a hydraulic cylinder, with a surface roughness as presented before, there will not be a full lubricant film separating both surfaces. As a result a regime of mixed or boundary lubrication will arise and the friction force is higher than for the theoretically determined full film lubrication conditions for smooth surfaces.
For the real contact situation a new hydrodynamic model is being established where hydrodynamic effects are present on a micro scale as presented by the roughness profile giving micro tilting areas with each a certain length and width and a tilting angle. Here, especially, statistical information from surface topology must give a part of the input information on the total number of micro tilting pads and their influence on the load bearing capacities.
For the determination of friction in the sealing contact the same topology information is used. In this regime of mixed or boundary lubrication the total friction is a result of three different friction components each based on a physical effect between the moving surfaces:

$$F_{tot} = F_{adh} + F_{visc} + F_{hyst}$$

where

F adh	is the friction based on adhesive forces between seal and rod material
F visc	is the viscous friction component
F hyst	is friction resulting from hysteresis in the seal material due to micro deformations in the visco elastic seal material

Here calculations show that the viscous component is small compared to the other two effects so for this contact situation the following equation holds:

$$F_{tot} = F_{adh} + F_{hyst}$$

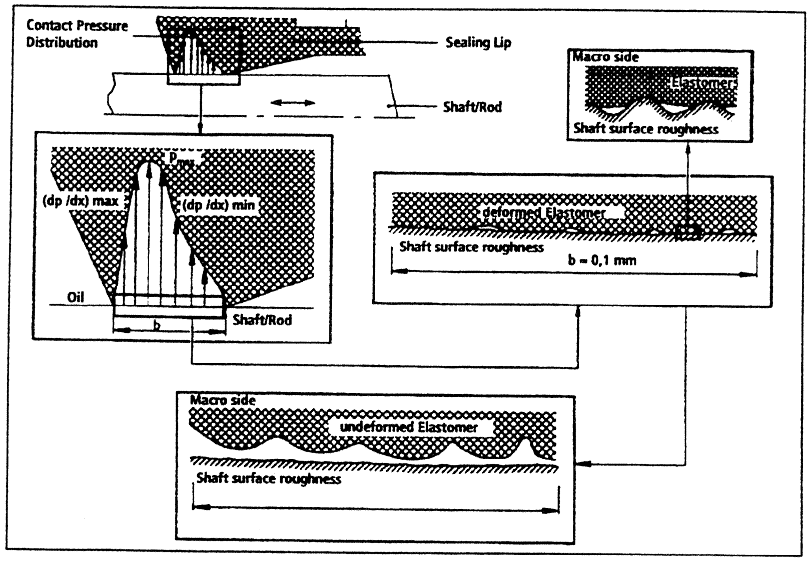

fig. 7: General shape of the seal-rod contoct.

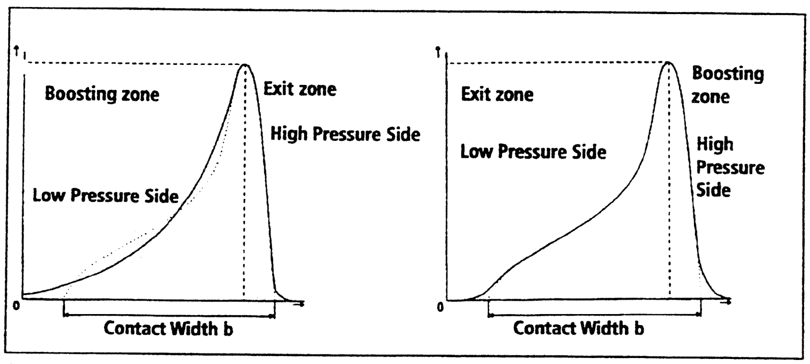

fig. 8: Elastohydrodynamic pressure profiles under the sealing lip for both sides of high- and low pressure (Van Dijnsen, 1995)

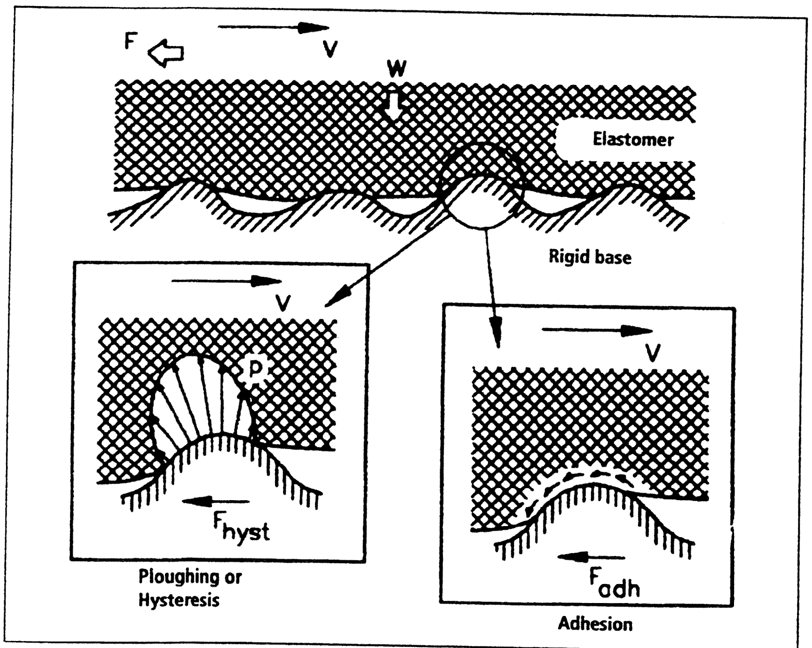

fig. 9: Principle components of elasomeric friction (Moore 1972)

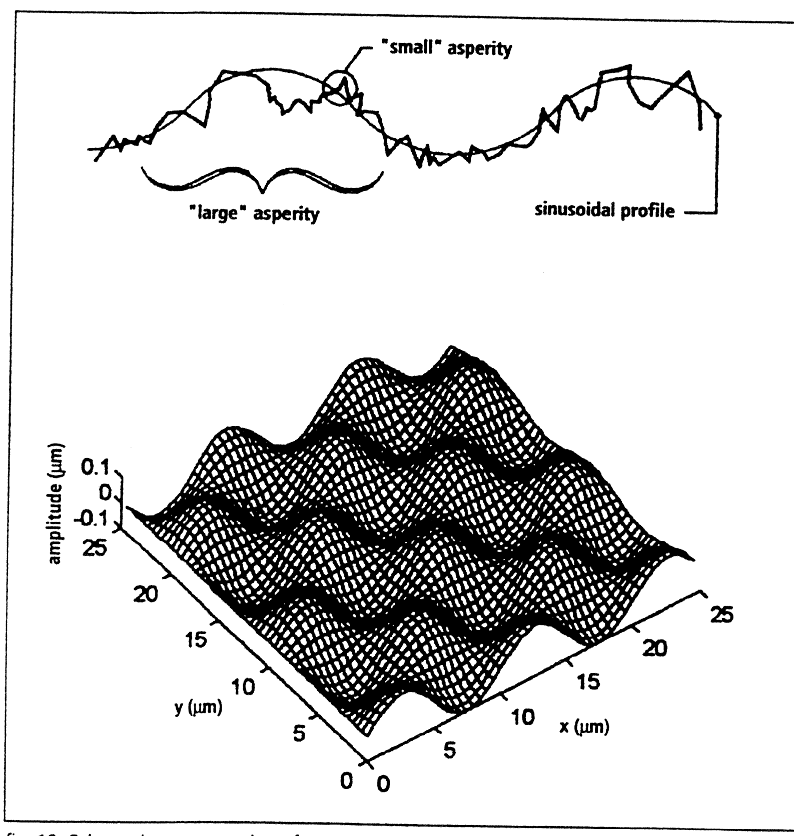

fig. 10: Schematic representation of an actual and the model sinusoidal roughness profile in 2 and 3 dimensional shape (Van Klooster, 1996)

The principle diagram belonging to this situation was presented by Moore (1972), figure 9.

Now for the rod-seal combination the surface roughness is modeled into a model surface. An example is presented in figure 10.
On the other hand the seal material is presented as a multi-Maxwell model where numerical data are available from DMTA testing of the seal material.
The combination of the surface profile and material model gives the dynamic model for the hysteresis part in the friction from the rod-seal contact as presented in figure 11.
Results of both friction components from adhesion and hysteresis are strongly depending on service conditions of the hydraulic cylinder but here a tool is present to give numerical information on both effects.

Sealing - criteria

In order to establish a sealing-criteria under static conditions, a test unit as shown in figure 12 was built.
Here the relationship is tested between the minimum necessary contact stress between the seal material and the rod material, to prevent leakage for given levels of fluid pressure.

The test was done with 3 seal materials i.c. PUR, PTFE/C and UHMWPE at 2 different temperatures i.c. 303 K and 383 K. The load F was applied with "dead weight".

. Major conclusions from the tests are:
- All 3 materials start to leak when the oil pressure is equal to the contact pressure.
- The effect of temperature changes is lowest for PUR material.

Mechanical material characteristics

Seals under normal operating condition are always under "all-side" pressure. This means that basic material characteristics also under these conditions should be established. For that reason it is not preferable to use data from stress/strain curves from tension tests, as mentioned in the introduction.

Because 3D-compressive testing is starting up at this moment, results from uni-directional compressive tests can be used with better results as from the tension tests.

For the investigation of the dynamic seal behaviour the test method "Dynamic mechanical temperature analysis" (DMTA) is used where the dynamic elastic modules can be measured. The test piece (ø 2,2 x 0,6 x 30 mm) is excited by a subscribed force. Both forces and displacements are accurately recorded. Next to this also the "visco-elastic" behaviour can be established as a time-difference between the exciting force and the resulting displacement.

Three major conclusions can be drawn from the tests done with PTFE/C and UHMWPE material.
- The initial dynamic elastic modulus for PTFE/C is higher.
- Both PTFE/C and UHMWPE show a minor visco-elastic behaviour.
- In case of UHMWPE there is a bigger difference between dynamic elastic modules at low (293 K) and high temperature (353 K).

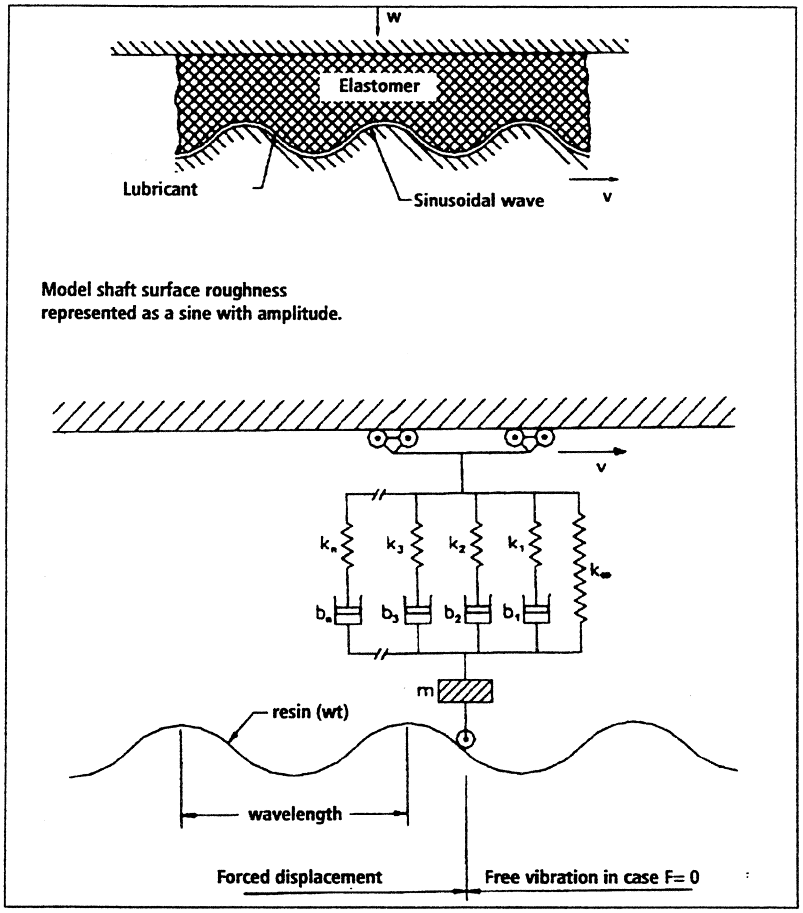

fig. 11: Seal contact model and hysteresis friction model (Van Klooster 1996)

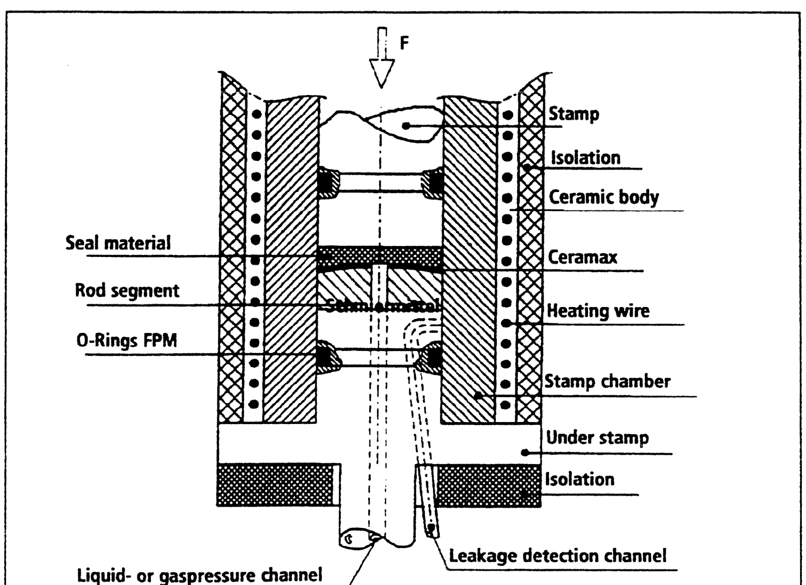

fig. 12: Tes equipment for static sealing criteria

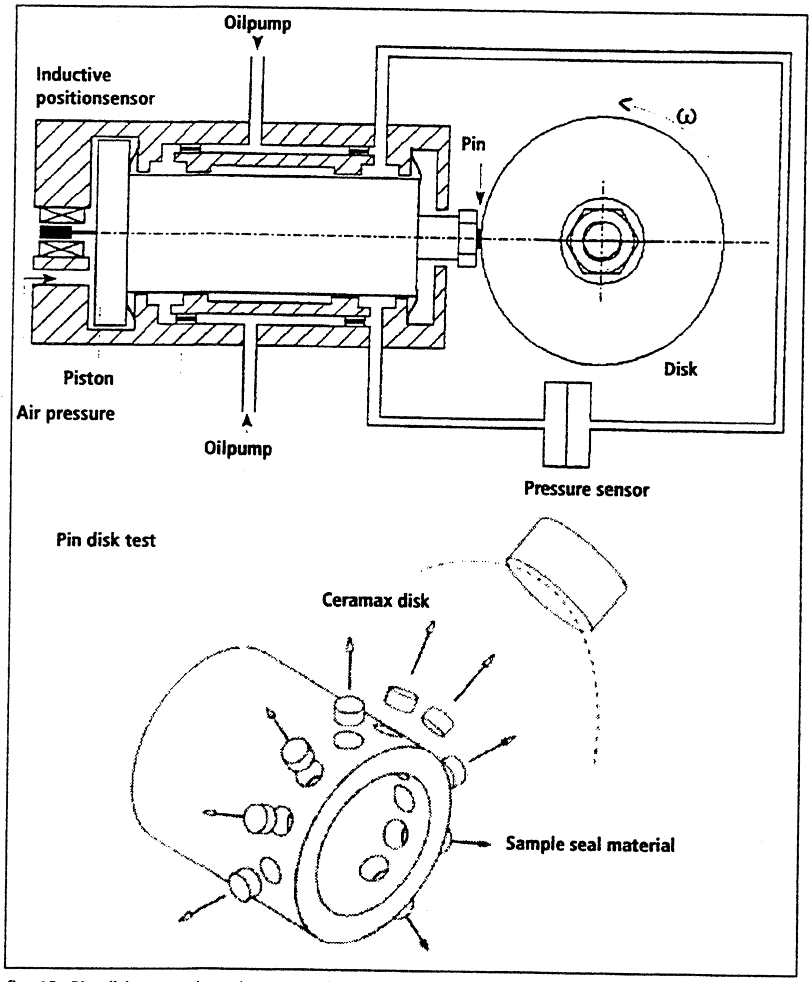

fig. 13: Pin-disk test unit and test samples from seal materials

Friction and wear measurements

With the commonly used pin-disk method, PTFE/C and UHMWPE seal materials were tested versus chromium and Ceramax.

This was done with 2 speeds i.c. 0,1 and 1 m/s and 2 contact stresses i.c. 10 and 32 N/mm^2.

These conditions are representative for normal cylinder operation.

In order to give a constant contact stress distribution the shape of the test pieces was adapted to the disk, figure 13.

Temperature was kept low and constant at 295 K with cooling / lubrication of oil.

From the tests the following important points can be concluded:

- Under all circumstances UHMWPE shows lower friction- and wear values.
- Especially for UHMWPE there is a 3 up to 5 times difference between static and dynamic friction coefficient.
- Wear characteristics of UHMWPE are comparable for running against chromium or Ceramax.

Test Unit for Measurement at Hydraulic Seals

Description of the test unit:

There are four goals set for the test unit (figure 14).

- measurement of data to verify the theoretical hydrodynamic model i.c. leakage, friction force and pressure-breakdown across seals.
- establish a 'Stribeck curve' (figure 15).
- investigation of possible 'stick-slip' phenomena.
- overall testing of new seal types.

The test unit should be capable of handling operating conditions which are representative for the major amount of applications where Hydraudyne uses their cylinders in the field.
(Speed range: 2 up to 200 mm/s; pressure: 1 to 300 bar ; temperature: 30 to 80 °C).
Research on elastomeric contact seals for reciprocating piston rods has revealed that operation takes place in the boundary/mixed lubrication region (Kanters (1990) and Van Dijnsen (1995)). To get good and consistent measurements it is necessary to separate all parameters that are involved in the total problem. External influences outside the sealing area should be avoided.
In order to compare sealing systems, and to save test time, two different rod configurations (e.g. roughness, chromium vs. Ceramax) and two seal chambers are put in series. To investigate two piston-rods (in series) it is important to get the test unit acting along one centerline (concentric). Very close tolerances on production parts are necessary, because the force transducers are fixed without any flexibility or coupling body. Tolerances over the geometry of the whole concept are very close to the best achievable values. The whole construction is pre-stressed so the necessary high stiffness is repeatable and under control, this to avoid any dynamic influences (of the construction itself or of the environment) on the measurements. The stiffness of the test unit is also important to avoid buckling and control frequencies of the test unit. The two piston rods (ø 180 mm) in series are moving with a maximum stroke of 500 mm (in- and out movement, results in a stroke of 1000 mm). The test unit frame is in vertical position. The rod will not have any side loads of gravity or anything else to prevent any additional friction on the

bearings and the seals. At the bottom of the test chamber it is not possible to use another seal configuration to prevent leakage because of its influence (static as well as dynamic and possible stick-slip) on the other seal configuration at the top. Therefore a hydrostatic bearing has been used which will have almost no friction. The test rods are driven by a separate cylinder at the bottom of the test unit.
When there is a need to look after the effects of stick-slip the test unit can be re-assembled half the size and using only one test chamber, so dynamic influence (a double mass-spring-system) of the other seal-configuration is excluded. (The maximum sample frequency is 150 Hz).

Characteristics of the test unit and data acquisition system:

To read all data from the test unit a data acquisition and control software of WorkBench PC (STI) has been used together with a 16 bit data acquisition card (MINI 16; 8 analog input-channels). Friction force of the seal against the counterpart Ceramax will be the main point to focus for these measurements. To get these friction force measurements, load-cells (force transducers) are used.
Two channels have been used for force transducers. The precision GTM force transducers, with 50 kN and 20 kN range, have a total error of 0.4 % of reading in the force-range 1 % - 100 % * Fnom (otherwise 0.8 %). The transducers work according to the bending-ring-system patented for GTM. Two channels are selected for pressure transducers with a range of 0 - 300 bar (with piezo-elements for dynamic pressure under "static" circumstances). Two channels are chosen for temperature-measurements (range of 30 - 80 °C) measured by Pt100 elements with an accuracy of 0,1 °C at 0 °C excluding the error of the resistance (20 kΩ, 0,1 %) on the data acquisition card and the card itself. One channel has been used for the position of the piston-rod in the test unit. The position is measured by an analogue Balluff linear transducer (0 - 10 V, stroke of 500 mm resolution ≤ 2μm). This signal will be used to control the whole movement by a HNC 100 (Mannesmann Rexroth) control unit (position-controlled) and two servo-valves (Mannesmann Rexroth) one for low velocity (0 - 40 mm/s) and a proportional valve controlled by a servo valve for high velocity (40 - 230 mm/s). The control parameters of the HNC 100 can be changed on

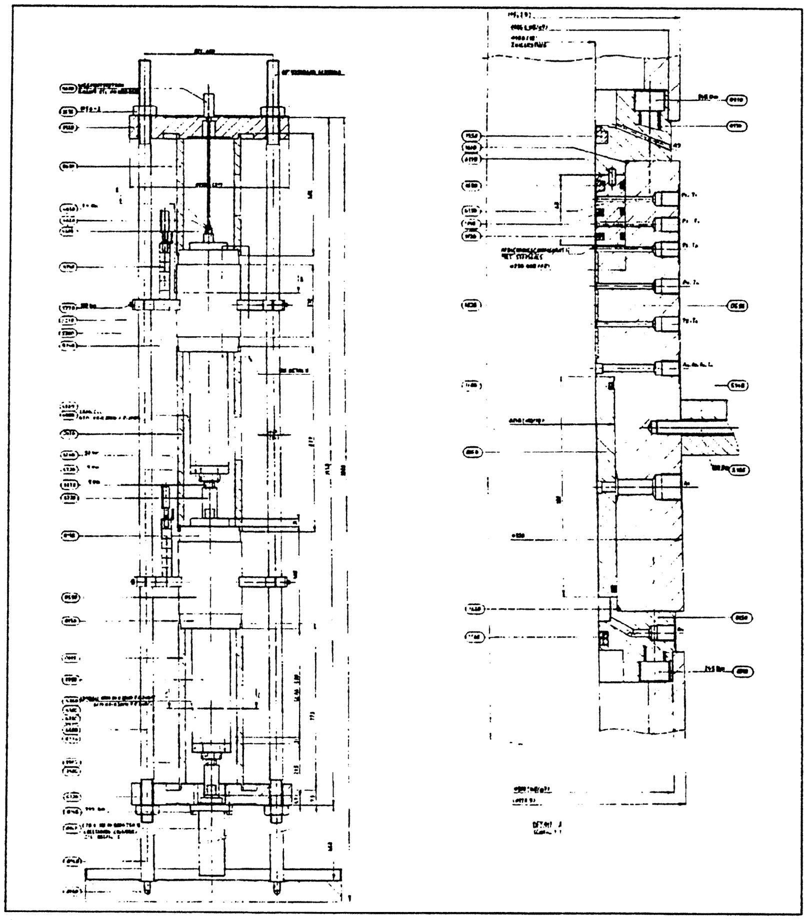

fig. 14: Test unit Hydraudyne Cylinders B. V.

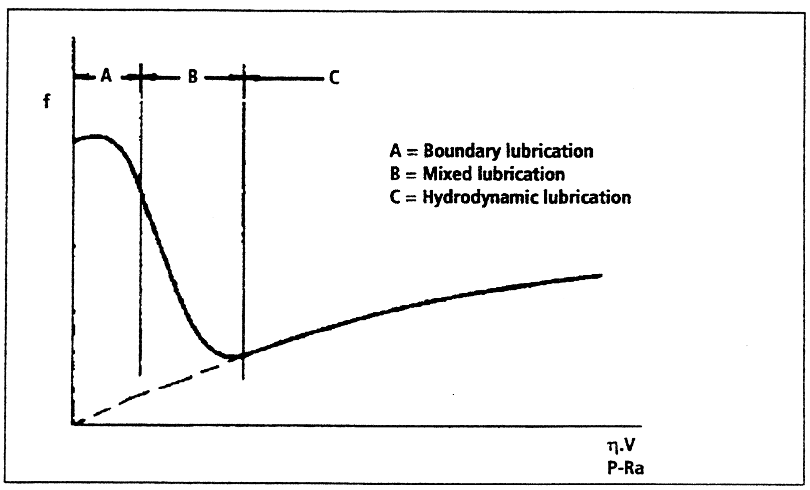

fig. 15: Stribeck Curve

line with a special program and will be active at the next stroke of the test unit.

The velocity path of the test unit is trapezoid and only when the velocity is stable (constant), the measurements of all channels will be taken (controlled by WorkBench-frame of the test unit) automatically and saved to a file. The system pressure just behind the seal configuration can be adjusted at any oil pressure. Three valves are connected with the test unit and a switch-cupboard which has six different programs to use special combinations of oil pressure. (Different system pressures at in- and outstroke of the piston-rod are possible). Also there is the possibility to adjust the amount of total strokes automatically. Force transducers, temperature gauges, position and oil pressure transducers are calibrated. The test unit will have friction by its own, like the scrapers, the drive-cylinder, the bearing strips in combination with oil and the hydrostatic bearing against Ceramax. Therefore the unit must be calibrated without seals to detect all friction forces on every velocity and at pressure (Δ P) 0 bar. This will give the offset of the test unit so when the real measurements have been taken this offset is subtracted from the rough-data.

'DATA' is used and designed as a process-program in MATLAB to analyse the total data and convert them into real values. All calibrations are built into this automatically menu-process-program and details of statistics (for example filtering, mean values, correlation value, polyfit etc.) can be presented in data, diagrams and figures.

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Conclusion

An extensive seals research and development program was set up, using a combination of fundamental research and long time field experiments in order to design optimized seals both in configuration and material selection for Ceramax and chromium hydraulic cylinders.

All important aspects of the seal-rod interaction are studied and combined.

This new approach, where the hydraulic cylinder manufacturer is involved both in designing the cylinder and the total sealing concept, gives an important opportunity to come to a real optimized and well balanced combination of rod and seal, necessary for the best performance in service.

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Nomenclature

h	dynamic viscosity of oil	[Pa.s]
s	stress component	[Pa]
e	nominal strain	[%]
qin	flow per unit of with going inside	[m^2/s]
qout	flow per unit of with going outside	[m^2/s]
p	oil pressure	[bar]
Ra	arithmetic average roughness height	[µm]
F	force	[N]
x	cantilever's motion	[nm]
k	spring stiffness	[N/m]
v	velocity	[mm/s]
b	contact width	[mm]
(dp/dx) max	maximum contact pressure gradient	[N/mm^2]
(dp/dx) min	minimum contact pressure gradient	[N/mm^2]
Pmax	maximum contact pressure	[Pa]
Ptop	maximum contact pressure	[Pa]
b	damper constant	[Ns/m]
m	mass	[kg]
t	time	[s]
r	wave amplitude	[µm]
w	angular frequency	[rad/s]
W	load per unit of width	[N/m]
f	friction coefficient	[-]
T	temperature	[°C]
Fnom	nominal force	[N]
y	coordinate in axial direction	[µm]
z	coordinate in axial direction	[µm]

Abbreviations

FEM	Finite Element Method
AFM	Atomic Force Microscopy
SPM	Scanning Probe Microscopy
EHD	Elasto-Hydro-Dynamic
DMTA	Dynamic Mechanical Temperature Analysis
3 D	three dimensional

Materials

PUR	PolyUrethane	-40/110°C
PTFE/C	PolyTetraFluoroEthylene/graphite 25%	-200/260°C
UHMWPE	UltraHighMolecularWeightPolyEthylene	-270/90°C