DEVELOPMENT OF PISTON SEALS FOR DEEP OIL-WELL PROBES FOR HIGH PRESSURE AND TEMPERATURE MEDIUM

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Abstract

In this paper the results of a finalized phase of the present test programme are published. It is considered only a stage (of the research work on dynamic seals for oil well probe application) when reasonable solutions were found.

The results obtained are promising and already provide satisfactory solution for industrial problems. Compared to the requirements, the endurance test results proved much longer life at an acceptable friction loss.

During the possible next phases of the test programmes more design alternatives, seal materials, friction surface materials and qualities should be subjected to endurance test in order to select the most suitable alternatives of the seals and also to optimize them. (Where optimizing the seals means: to obtain, or develop, the most reliable alternatives which produce the possible longest life, belonging to leak proof operation, and smallest friction loss at the operation parameters.)

Friction characteristics tests should be carried out too for the selected design alternatives to obtain the required design aids for proper estimation of the expected friction forces at different working parameters.

Introduction

In the 3000 to 5000 m deep holes drilled for oil investigation the ambient earth temperature may reach $t_w = 180$ °C and the mud pressure may reach $p_w = 80$ MPa (800 bar). The leak proof operation of oil well probes is an indispensable requirement to produce safe geophysical measurements and to obtain this leak proof operation properly working (dynamic) seal systems must be applied.

A test rig was designed and produced, which simulates the probes' working conditions, to investigate dynamic seal systems of different materials and designs and to develop optimal seal systems for the probes.

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Following the production, the running in and the necessary calibrations a test programme was carried out. This paper presents a short description of the test rig and the test method as well as the results of the tests.

1. Need for the investigation

During investigations the oil well probes may be operated in a depth of 3000 to 5000 m where the working parameters, in the mud may reach $p_w = 80$ MPa pressure and $t_w = 180$ °C temperature.

For measurements the probe is lowered to starting position at the bottom of the drilled hole, and after adjusting the concerned measuring instruments the probe is dragged slowly up to the surface, while geophysical and hole-geometry measurements are carried out continuously.

For hole-geometry measurements three or more sensing arms are used. These arms work on lever principle where the pins of the arm-levers are joining to the probe's body. The outer end of each lever is following the surface of the drilled hole, while the probe is lifted. The moving inner end of the lever transfers the axial displacement, produced by surface change of the hole, to the measuring unit via a piston and an inductive transducer built in the probe. When examining the typical designs applied for the arm-lever and probe pin-joints the piston must be sealed as the mud might penetrate into the probe at the hinge points and as a prerequisite of the safe operation the probe must be leak-proof.

2. Test rig

The test rig (Fig. 1) consists of four principal parts:

I. Conditioning oven (1), to set the test temperature.

II. Rolling base frame (2) carrying the tester, to move the tester in mounting and test position which can be fixed in these positions.

III. The test unit's main parts are: the operating hydraulic cylinder (3) the test pressure multiplier (4), and the mounted test cylinder (5).

IV. Measuring system, to measure and record the operating (test) temperature, pressure and the friction force.

2.1. Test unit

It is fixed on the rolling base frame (Fig. 1). The interchangeable test tubes and pistons give the opportunity to apply:

— different hole (seal) diameters,

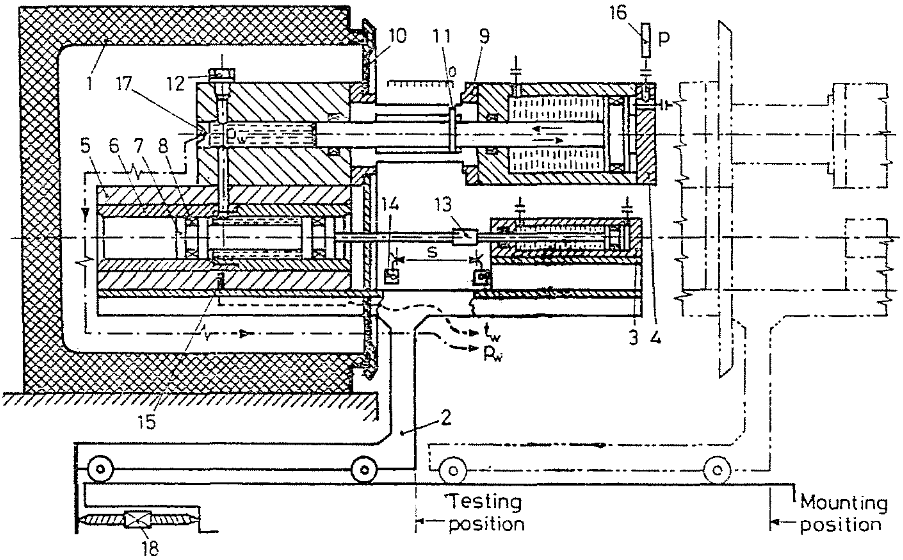

Fig. 1. Test rig

1. Conditioning oven; 2. Rolling base frame; 3. Operating hydraulic cylinder; 4. Pressure multiplier; 5. Test cylinder; 6. Test tube; 7. Test piston; 8. Seal specimen; 9. Adapter; 10. Heat protecting shield; 11. Leakage indicator; 12. Sealed bolts for filling; 13. Dynamometer; 14. Limit switch; 15. Thermocouple; 16. Manometer; 17. Membrane manometer

— different microgeometry (surface roughness) in the hole and
— different materials (having different physico-chemical characteristics) for the tube and the seal system members as well.

Test tube (cylinder) and seal specimens are shown in Fig. 2.

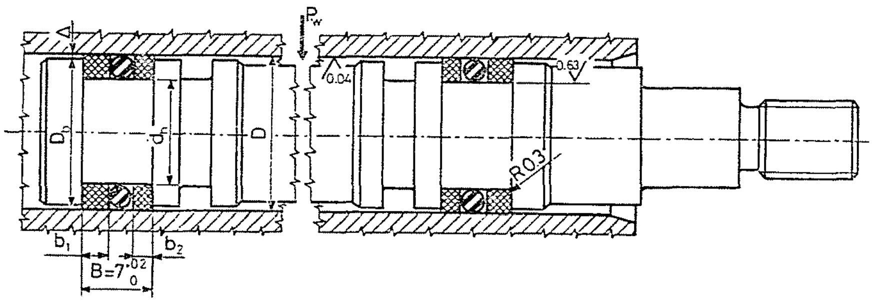

Fig. 2. Assembled test piston

2.2. Measuring system

The block diagram (Fig. 3) shows the measuring system which is suitable to measure and register the following quantities.

— Pressure is measured by strain gauge membrane dynamometer at the high pressure side of the pressure multiplier and the estimated error of the measurement is $\pm 3\%$.

— Friction force is measured by a strain gauge dynamometer on the piston rod having an estimated measurement accuracy within $\pm 3\%$ as well.

— Position registration: An inductive transducer is used to determine the seal position and the test piston speed.

— The above quantitites are recorded by a multichannel recorder.

— Setting and control of the test temperature: The test temperature is sensed by a thermo-couple and a thiristoric temperature regulator is controlled by the thermocouple signals within $\pm 3\%$ accuracy.

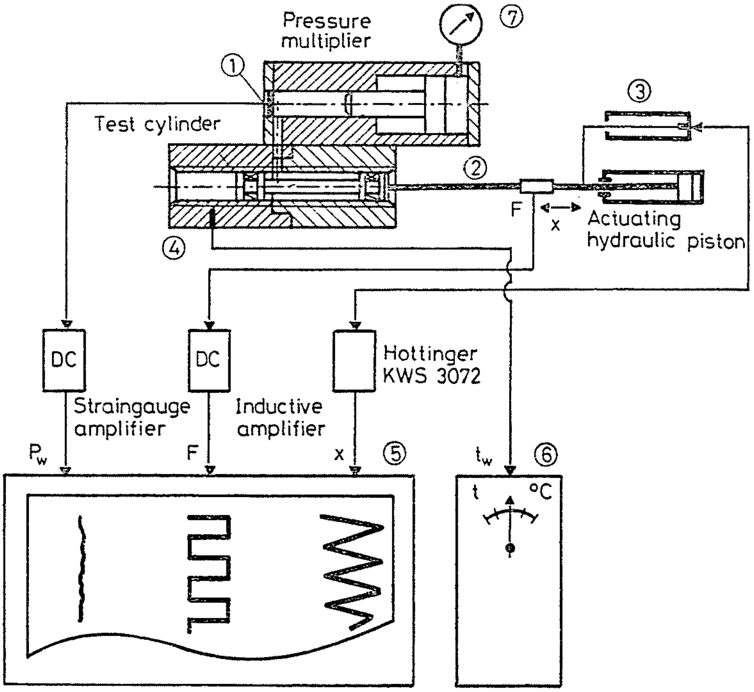

Fig. 3. Measuring system of the test rig

1. Membrane manometer; 2. Dynamometer; 3. Inductive transducer; 4. Thermocouple; 5. Multichannel recorder (compensation grapher)

2.3. Main parameters

— Test pressure for long time or endurance tests:
$p_w = 80$ MPa (800 bar)
— Proof test pressure and maximum pressure for short time testing:
$p_p = 100$ MPa (1000 bar)
— Test temperature:
$t_w = 190$ °C
— Maximum stroke length of the test piston:
$s = 50$ mm
— Test tube inner diameter. (Seal nominal outside diameter):
$D = 11$ to 20 mm
— Test piston speed range:
$v = 0$ to 15 mm/s ($v_{max} = 45$ mm/s).

3. Test parameters and registered values

The fixed values of the parameters during the tests and the registered values are:

Stroke length	$s = 45$ mm
Speeds of the test piston $v_c = 0.8$, 1.5, 4.5, 9, (20), (40); mm/s	
Test pressure	$p_w = 800$ bar
Test temperature	$t_w = 180$ °C

At fixed nominal parameter values the working pressure and the friction force change and the seal position are recorded by the recorder. The working temperature and the leakage indicator gauge position is read and fixed at given intervals.

4. Calculated values of the friction force

During the reciprocating motion of the test piston the dynamometer is sensing tension and compression forces produced by friction on the two seals.

As a consequence of the above mentioned factors the mean friction force values are calculated as follows:

— "Constant" friction force in the middle of the stroke:
$F_c = F_A/4$ and
— Friction force maximum, mostly at the *end* of the stroke:
$F_{max} = F_M/4$;

where F_A and F_M are the average and maximum friction force values, obtained from records.

5. Tested seals, design characteristics

Many variations of dynamic seal systems were considered. However, for the present phase of test programme the variations of elastomer 0-rings and plastomer backup rings were selected for endurance test (Fig. 4).

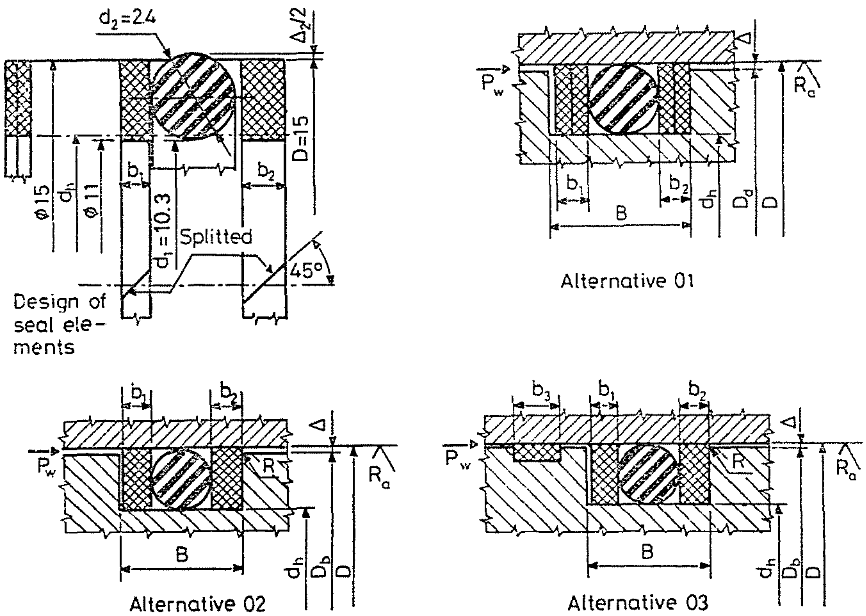

Fig. 4. Design alternatives

The tested 0-ring material was fluor rubber and the backup rings were made of glass fibre, bronze and coke reinforced PTFE (teflon) bar semiproducts. The average clearance between the test piston and the hole was $2\Delta=0.064$, 0.075 and 0.11 mm. The test tubes were made of heat treated alloyed stainless steel (Ko 16 N and Ko 13 N) and the measured surface roughness of the friction surface in the hole was $R_a=0.04$—0.08 μm in axial and $R_a=0.04$—0.01 μm in circumferential direction. The relative elastic interference value of the 0-rings was set, by the piston groove diameter (d_n) between 11 and 19%.

6. Evaluation of the tests

The circumstances and results for six endurance tests are summarized in Table 1. The tested seal design alternatives (Fig. 4.) were: the given two 0-ring products and different backup rings, having the outside, thrusting, backup ring thickness from $b_2=1.5$ to 2.5 mm. The test fluids were emulsion and water.

Table 1

Test number	Seal design: alternative	Seal design: denomination	Test fluid	Cycle number N [cycle] from	Cycle number N [cycle] to	At 9 mm/s "Constant" friction force F_e [N] from	At 9 mm/s "Constant" friction force F_e [N] to	Seal life, appearance of leakage
1	2	3	4	5	6	7	8	9
01	01	Outside backup ring $b_2=1{,}5$ mm fibreglass reinforced PTFE+SIMRIT 0-ring 83 FKM 592+inside backup ring SPA $10{,}3\times2{,}4-1{,}2$	emulsion, cutting oil+water	0	2080	60*	30*	Without leakage indication, the seal burst out at $N_1=2080$ cycle.
02	02	Outside and inside backup ring $b_1=b_2=1{,}5$ mm fibreglass reinforced PTFE+0-ring 83 FKM 592	emulsion, cutting oil+water	0	1300	60*	80*	Without leakage indication, burst out at $N_1=1300$ *cycles.*
03	01	Identical with test No. 01	water	0	700	125	75	Without leakage indication, the seal burst out at $N_1=1300$ *cycles.*
				780	1300	(180) 125	85	
04	03	Outside backup ring $b_2=2{,}5$, the inside backup ring $b_1=$ $=1{,}5$ mm fibreglass reinforced PTFE+0-ring 83 FKM 592	water	0	1430	(100) 137	(152) 247	Some leakage started at about $N_1=2750$ cycle and the seal burst out at $N_3=2920$ *cycles.*
				1440	2920	160	210	
05	03	Identical with test No. 04	water	0	1635	(98) 122	216	Some leakage started at $N_1=3920$ cycle, restarted at $N_2=3402$ and significant leakage started at $N_3=4420$ cycles.
				1640	3400	(142) 215	260	
				3402	4420	(147) 180	200	
06	03	$b_1=b_2=2$ mm backup rings, fibreglass reinforced PTFE+NATIONAL 0-ring Vi 90/23	water	0	1310	(74) 93	167	Without any leakage indication the seal burst out at $N_1=2315$ cycles.
				1320	2250	(132) 167	220	
				2254	2315	167	240	

Notes: * Measured at different speeds
* Not characteristic values

Regarding friction forces, the measured values in emulsion were somewhat smaller than in water (see test No. 01. and 03.). Increased thickness of the backup ring and the application of guiding ring (alternative 03) increased the friction force. Considering the "constant" friction force values measured (see Table 1), the calculated coefficient of friction is $\mu_e = 1.3$ to $4.2 \cdot 10^{-3}$ for emulsion and $\mu_w = 3.2$ to $11 \cdot 10^{-3}$ for water test fluid. The coefficient of friction was calculated from the formula of $\mu = F_c/F_n$, where $F_n \cong p_w \cdot A$ [1] [2] and $A \cong D\pi\ (d_2 + b_1 + b_2)$.

In case of increased backup ring thickness the seal life was longer. For $b_2 = 1.5$ mm outside, thrusting, backup ring the life was between $N_1 = 1300$ to 2080 cycles, equal to $S_1 = 120$ to 190 m frictional travels, and for $b_2 = 2$ and 2.5 mm the life was found between $N_1 = 2000$ to 3920 cycles, equal to $S_1 = 190$ to 300 m travels.

According to experience the failure of the seals is caused by wear. Following a certain operation time the outside backup rings weakened somewhere on the circumference, mostly at the split, and burst out when the static strength limit of the ring was reached.

Probably this is the reason why the appearance of leakage and the burst out (failure) of the seals mostly coincide.

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