Is the wear factor in total joint replacements dependent on the nominal contact stress in ultra-high molecular weight polyethylene contacts?

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Abstract: The exact dependence of wear factor on contact stress, load and apparent contact area is much disputed in the literature. This study attempts to solve this dispute. Pin-on-plate studies of ultra high molecular weight polyethylene against stainless steel were conducted under different combinations of load (33–250 N), nominal stress (0.56–12.73 MPa) and face diameter, as well as two tests where both stress and load were kept constant, while the diameter was changed. For these tests the centre of the pin face was bored out to create four different average pin diameters with similar face areas. Diameter and load were found to have no significant effect on the wear factor, while the wear factor decreased with increasing contact stress according to the relation $K = 2 \times 10^{-6}\sigma^{-0.84}$.

Keywords: ultra high molecular weight polyethylene wear, contact stress, load, pin-on-plate set-up

NOTATION

K	wear factor (mm^3/N m)
L	load (N)
R_a	surface roughness parameter (μm)
UHMWPE	ultra high molecular weight polyethylene
V	wear volume (mm^3)
x	sliding distance (m)

1 INTRODUCTION

1.1 Wear

The simple model of wear, according to the Lancaster equation [**1**] states that $V = KLx$, where V is the wear volume (mm^3), L is the load (N), x is the sliding distance (m) and K is the wear factor (mm^3/N m). This indicates that the wear volume is dependent on the load and the sliding distance for any given configuration. The nominal contact stress across the interface does not appear. This is because the real contact area increases with increasing load, while the apparent contact area remains constant [**2**]. The Lancaster equation is a simplified form of the Archard wear equation [**3**], which included a term for the hardness. In this case, the hardness term has been incorporated into the wear factor. Archard assumed Hertzian contact theory for a flat non-deformable surface in contact with a nominally flat deformable surface with spherical asperities evenly distributed in depth.

1.2 Effect of stress and load on wear factor

Although the Lancaster equation suggests that the wear volume is dependent only on the particular combination of materials, the load and the distance slid, the situation may not be quite so simple. The literature is divided as to the exact effect of load and stress on the wear rate of ultra high molecular weight polyethylene (UHMWPE). This is made more difficult by the different operating conditions in each study, including differences in apparatus and lubricants.

Barbour *et al.* [**4**], using pin-on-plate apparatus, indicated that the wear factor decreases with increasing nominal contact stress. Wang *et al.* [**5**] showed the effect of maximum contact stress on the wear of UHMWPE using artificial hips, by altering the radial clearance. These findings agreed with the Barbour *et al* study but contradicted previous work [**6**, **7**].

Rose *et al.* [**6**] and Rostoker and Galante [**7**] both found an exponential increase in wear with increasing load and increasing contact stress respectively. The results in these studies were presented as mass loss per unit sliding distance, and penetration depth per unit sliding distance respectively rather than wear factor.

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When the results were converted to a wear factor, this relationship was found to be no longer the case. Rose *et al.* showed very little variation in the wear factor except for a large increase at the very highest stress, while Rostoker and Galante showed lower wear factors for stresses around 10 MPa, but higher wear factors both above and below this.

Sathasivam *et al.* [**8**] found that the average mass loss on the pins at first increased and then decreased with decreasing nominal stress. This trend does not change when converted to a wear factor since all pins in the study were subjected to the same loads.

More recently, Mazzucco and Spector [**9**] concluded that the wear factor for UHMWPE pins against CoCrMo plates was not dependent on the load nor on the contact stress. Instead they found it to be dependent on the apparent contact area. However, their results did show a decrease in wear factor with increasing contact stress.

Archard's work appears to be applicable to all the references cited above, in which UHMWPE is tested against a metal. This paper attempts to solve the dispute within the literature concerning the exact relationship between the magnitude of the wear factor and the load applied to the system.

2 APPARATUS

The four-station pin-on-plate machine had independently generated reciprocating and rotational motion. Figure 1 shows the set-up of the rig. The plates reciprocated at 1 Hz with a stroke length of 25 mm, while the pins rotated at 1 Hz around their central vertical axis. Each of the pins had a separate motor to provide rotation. The addition of rotation to the reciprocation motion has been found to give wear rates more comparable with clinically observed wear rates, and to give worn surfaces with similar wear patterns and defects to those seen in clinically retrieved samples. This has been seen in pin-on-plate [**10**, **11**], pin-on-disc [**12**] and simulator [**13**] studies.

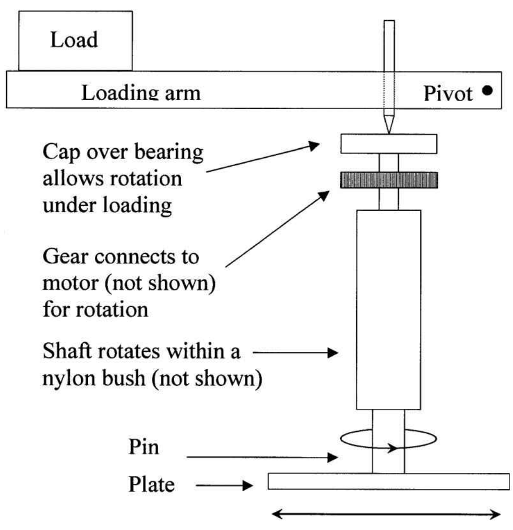

Fig. 1 Schematic diagram of pin-on-plate set-up

The pins were each independently statically loaded by placing masses at various lengths along four loading arms (Fig. 1). Bovine serum diluted to 30 per cent, with 0.2 per cent sodium azide added to retard bacterial degradation, lubricated the pin–plate interface (15.6 g/l protein concentration).

The number of cycles was measured by a non-contacting Hall-effect probe. The temperature was kept at a constant 37 °C. A level sensor was attached to one side of the bath to control the amount of lubricant in the bath. An unloaded control pin was also used to correct for the amount of fluid uptake. This was in the bath to the same depth as the test pins but did not articulate.

3 MATERIALS AND METHODS

3.1 Materials

3.1.1 Stainless steel

Stainless steel plates of dimensions 48 mm × 24 mm × 3 mm were machined from bar stock supplied by RS Components. The stainless steel was 316 highly corrosion resistant material. While the British Standard refers to an initial $R_a < 50$ nm, industry produces joints with surface roughnesses that are much lower than this. The initial R_a values of the plates used were between 5.25 and 13.30 nm, which is comparable with the values produced on artificial joints by manufacturers.

3.1.2 Ultra high molecular weight polyethylene

The pins were machined from a rod of gamma-irradiated UHMWPE (0.254–0.297 MRad). All pins were machined with the same orientation within the bar to prevent any possible directional effects.

All pin designs and test conditions are detailed in Table 1 and had a connector of 5 mm diameter to fit into the pin holder. For tests B and C, the pins were required to have similar face areas, while differing in radius. As such the centres of pins 2 to 4 were machined out to a depth of 2 mm, causing the pin's face to be an annulus for those pins. For test F, a pin of face diameter 5 mm was used. However, owing to the large loads applied during this test, a pin with a larger midsection was manufactured to decrease any buckling or bending of the pin under load.

One pin was tested under each set of conditions, except in the case of tests D and E which were conducted under identical conditions. The results from these two tests agreed well but were presented as individual data points in accordance with the other tests. This gave an overall indication of the trend for each of the factors investigated.

Table 1 Conditions and pin design for each test

Test ID	Test pair	Load (N)	Contact stress (MPa)	Pin type	Pin name
A	1	40	5.66	Tapered; initial face diameter, 3 mm	Leeds tapered
	2	40	2.04	Cylinder; diameter, 5 mm	Durham
	3	40	0.63	Cylinder; diameter, 9 mm	ASTM
	4	40	0.56	Cylinder; diameter, 9.5 mm	Leeds flat
B	1	40	1.46	Cylinder; diameter, 5.9 mm	Pin 1
	2	40	1.47	External diameter, 3 mm; internal diameter, 6.6 mm	Annulus face pin 2
	3	40	1.47	External diameter, 6 mm; internal diameter, 8.4 mm	Annulus face pin 3
	4	40	1.50	External diameter, 8 mm; internal diameter, 9.9 mm	Annulus face pin 4
C	1	70	2.56	Cylinder; diameter, 5.9 mm	Pin 1
	2	70	2.58	External diameter, 3 mm; internal diameter, 6.6 mm	Annulus face pin 2
	3	70	2.58	External diameter, 6 mm; internal diameter, 8.4 mm	Annulus face pin 3
	4	70	2.62	External diameter, 8 mm; internal diameter, 9.9 mm	Annulus face pin 4
D and E	1	33	1.68	Cylinder; diameter, 5 mm	Durham
	2	61	3.11	Cylinder; diameter, 5 mm	Durham
	3	79	4.02	Cylinder; diameter, 5 mm	Durham
	4	49	2.50	Cylinder; diameter, 5 mm	Durham
F	1	180	9.17	Face diameter, 5 mm	Durham larger midsection
	2	250	12.73	Face diameter, 5 mm	Durham larger midsection
	3	90	4.58	Face diameter, 5 mm	Durham larger midsection
	4	120	6.11	Face diameter, 5 mm	Durham larger midsection

3.2 Methods

3.2.1 Cleaning and weighing protocols

The pins and plates were cleaned according to the protocols in the Appendix. Each component was weighed four times and the mean and standard deviation of the readings was found. The pin mass change was adjusted for fluid uptake using the mass of the control pin.

3.2.2 Experimental procedure

After approximately 250 000 cycles the experiment was stopped and the pins and plates removed, cleaned and weighed. The mass loss measured was converted to a volume loss using the density of UHMWPE (953 $\mu g/mm^3$) and the density of stainless steel (7.85 mg/mm^3) for the pins and plates respectively. The wear factors were found by using the Lancaster equation.

The lubricant was discarded and replaced with fresh serum each time that the machine was stopped to enable the pins and plates to be weighed. Each test was stopped after a minimum of 1.5×10^6 cycles of steady state wear, and all wearing-in data were disregarded. The plates were analysed on the Zygo NewView 100 non-contact profilometer approximately each 0.5×10^6 cycles and both pins and plates were analysed post-testing.

3.2.3 Effect of the rotational element of motion on the sliding distance

Because of the rotational motion, paths taken by different points on the pin surface differed according to position [**11**]. Since the rotation and reciprocation frequencies were both 1 Hz, the path lengths of different points on the perimeter also differ. Figure 2 [**11**] shows the paths taken by points which began at different positions on the circumference of the pin. Clearly not all points traversed the same path or have the same path length.

Scholes [**14**] developed a computer program to find the actual sliding distance of pins undergoing the particular motion used in the Durham machines. For the purposes of the present study, Scholes' program was modified to take into account the different pin designs. This is a numerical integration over the surface of the pin for small increments of angles and radii.

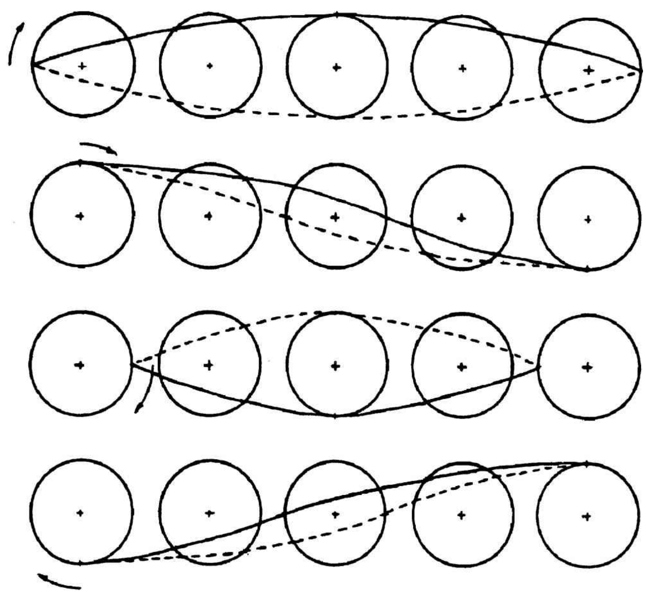

Fig. 2 The paths taken by various points along the circumference of the pin during 1 cycle [**11**]

Table 2 Percentage increase in sliding distance due to rotational element of motion for all pins used in this study

	Increase in sliding distance due to rotation (%)
Leeds tapered	0.67
Durham and test F	1.69
ASTM	4.82
Leeds flat	5.30
Tests B and C: pin 1	2.27
Tests B and C: pin 2	4.51
Tests B and C: pin 3	9.04
Tests B and C: pin 4	13.22

As the number of points taken into account was increased, the output values settled to the values given in Table 2. The percentage increase found for each pin was used to adjust the values of the sliding distance, and it was always these adjusted values that were used for the sliding distance rather than the reciprocation distance.

4 RESULTS

4.1 The effect of pin radius on wear factor

As can be seen in Fig. 3, the wear factor was not affected by the radius of the pin when both the load and the stress remained constant.

Taking into account all the other tests, regardless of experimental conditions, this became more evident. The external radius of each pin was plotted against the wear factor. Figure 4 shows that there was no clear correlation between the pin radius and the wear on the pin. The results for the 2.5 mm radius were particularly indicative since they spanned almost the entire range of the wear factors seen.

4.2 The effect of load on wear factor

As can be seen from Fig. 5, the magnitude of the wear factor is not greatly affected by the applied normal load for a constant nominal contact stress. This is as expected since the load is used in the calculation of the wear factor. This implies that some other factor affects the wear factor more significantly than just the load applied.

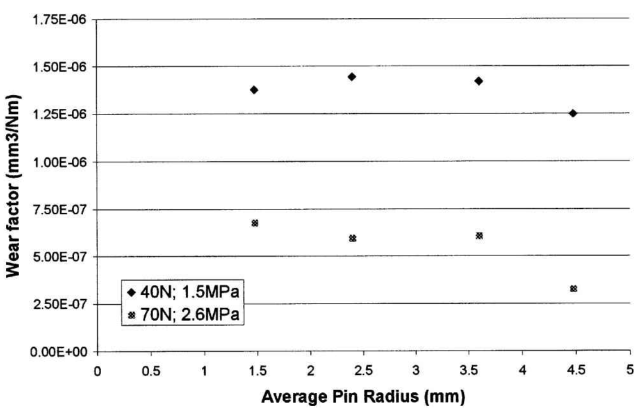

Fig. 3 Dependence of wear factor on average pin radius for two conditions of constant stress: 1.5 and 2.6 MPa

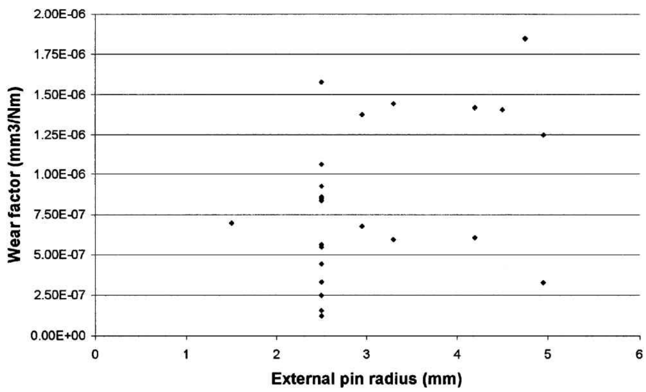

Fig. 4 The effect of external pin radius on wear factor for all test conditions

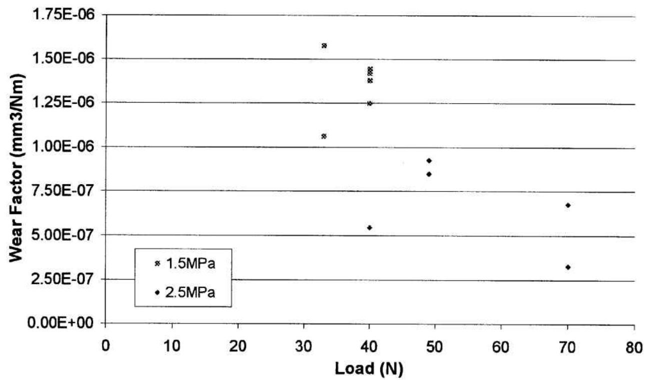

Fig. 5 The effect of normal load on wear factor for two nominal contact stress values: 1.5 and 2.5 MPa

4.3 The effect of nominal contact stress on wear factor

From the results in Fig. 3, we can see that, at higher loads and stresses, lower wear factors are achieved. This difference is found to be statistically significant ($p < 0.05$); a closer analysis including results from all test conditions follows.

Including all data at 40 N the effect of the stress on the pin wear can be seen, and this is shown in Fig. 6. A decrease in wear factor is noted with increasing stress.

Figure 7 shows all data regardless of the experimental conditions. The remaining data were found largely to overlap the 40 N data and are thus considered as one data set. A power-law fit to these data gives the relation $K = 2 \times 10^{-6}\sigma^{-0.84}$. This means that at low stresses the magnitude of the wear factor changes more rapidly as the stress increases than it does at higher stresses.

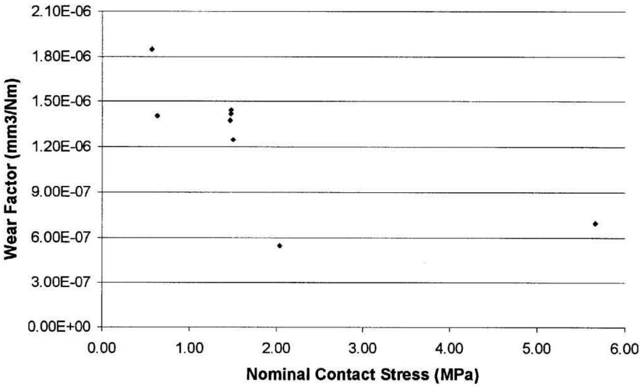

Fig. 6 The dependence of wear factor on nominal contact stress at a constant load of 40 N

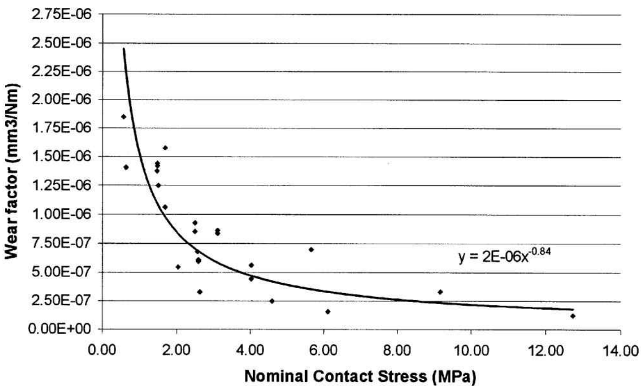

Fig. 7 The dependence of wear factor on nominal contact stress

4.4 Surface study

4.4.1 Plates

All but two plates showed a significant ($p < 0.05$) increase in surface roughness between the start and end of the test. Final mean surface roughnesses were in the range 8.81–90.0 nm. Multidirectional scratching was seen on the wear track. No correlation was found between the final roughness for each plate and the conditions of testing ($R^2 = 0.1$ for both load and contact stress). This indicates that, although the roughness of the plates increased significantly, the increase was not directly related to either the load or the stress under which the test was conducted, nor was the final roughness dependent on the initial roughness. In addition, no correlation was found between the wear factor of the polyethylene and either the initial or the final roughness value of the plate ($R^2 = 0.2$ and 0.0 respectively).

4.4.2 Pins

During the course of testing, the machining marks visible on the pin faces were removed, leaving a more polished surface. The mean final R_a was in the range 221–803 nm. No correlation was found between the final Ra and either the load or the stress under which the pins were tested ($R^2 = 0.4$ and 0.2 respectively).

For all solid cross-section pins, a nipple became visible at the centre of the pin (Fig. 8). Some smaller features were seen on the micrographs both at the edges and towards the centre of the pin faces. Multidirectional scratching was noticed in some areas on many pins.

5 DISCUSSION

5.1 General comments

The wear factors found by Barbour *et al.* [**4**] were lower than those found in this study by a factor of 100 while those found by Rose *et al.* [**6**] were higher by a similar order of magnitude. The values found in the Durham Laboratories are, however, comparable with those found clinically. From Dowson and Wallbridge [**15**] the relationship between volume of wear and clinical wear factor is seen to be $k_{\text{clinical}} = \text{volume}/2.376NWr$ (neglecting creep). For a volumetric loss of 50 mm^3/year on a Charnley joint of 22 mm diameter, implanted in a 75 kg (750 N) person, this yields a wear factor of 2.55×10^{-6} mm^3/N m. This figure was confirmed in a study of over 200 explanted acetabular components [**16**]. Therefore, while the literature contains a wide range of reported wear factors, those from the current work do have clinical relevance.

5.2 The effect of pin radius on wear factor

The radius of the pin does not affect the wear factor, when all other conditions are kept the same. In these tests the nominal face areas of each pin are very similar, allowing both load and stress to be kept constant. As such, it is expected that the actual contact areas will be very similar for all designs of pin [**2**], and subsequently there is very little difference in the contact of the surfaces on a microscopic scale. This was expected but Lloyd [**17**]

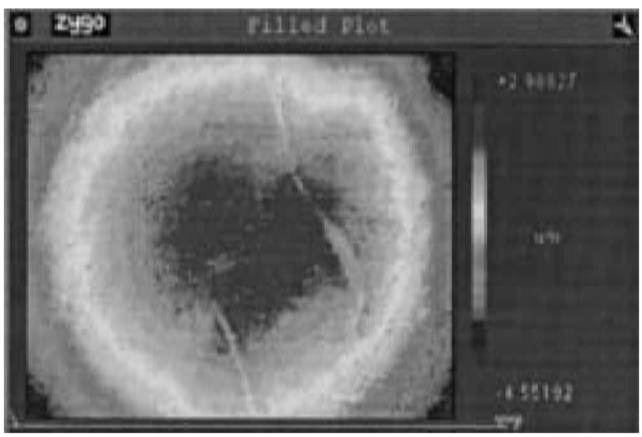

Fig. 8 Central nipple on pin 1 of test B, tested under 40 N load and 1.46 MPa stress

in his thesis suggested that there was a link between the wear factor and the radius.

5.3 The effect of applied normal load on wear factor

The wear factor does not seem to be affected by the normal load. This is in agreement with the Archard equation [3] and the Lancaster equation [1], such that the wear factor is a constant of proportionality between the volume loss and the product of the load and distance slid for a particular material combination. The Lancaster equation is used to calculate the wear factor and, as such, the load is not expected to have an effect on the wear factor.

5.4 The effect of nominal contact stress on wear factor

A decrease in wear factor is seen with increasing nominal contact stress. The relation is found to fit a power law such that $K = 2 \times 10^{-6} \sigma^{-0.84}$. This indicates that the magnitude of the wear factor reduces more as the contact stress increases at low stresses than it does at high stresses. This is consistent with the work of Barbour *et al.* [4], but not with the studies of Rostoker and Galante [7] or of Rose *et al.* [6]. Rostoker and Galante found an exponential increase in penetration depth per unit sliding distance with increasing contact pressure, while Rose *et al.* reported an increasing trend in mass loss with increasing load. This is also contrary to the findings of Mazzucco and Spector [9].

5.5 Surface study

The polymeric components became smoother as the concentric machining marks were removed during the test. At the centre of each of the pins with a circular face geometry, a nipple became visible. This has been noted in the past by other researchers [18, 19]. No correlation was found between the final surface roughness of the pin faces and the conditions of testing (load and stress). Saikko *et al.* [20] found a power-law relation with R_a raised to a power less than one. This would suggest a maximum of a twofold variation in wear factor over the range of roughnesses seen in this study. Even this variation would not obscure the results found herein, where the wear factors vary by an order of magnitude.

6 CONCLUSIONS

The wear factor of UHMWPE when tested against stainless steel increases as the nominal contact stress increases according to a power law $K = 2 \times 10^{-6} \sigma^{-0.84}$. This indicates that there is a high dependence on contact stress for low stresses, but at higher stresses the wear factor is not affected to the same degree. The radius of the pin and the load do not affect the wear factor at all, as expected.

During the testing, the concentric machining marks were removed from the surface of the pins early in the testing, leaving a more polished appearance. A nipple became visible at the centre of the pins. Multidirectional markings were noted on the pin surface.

The plates became significantly rougher during the course of testing, although this increase in roughness was not found to correlate with the testing conditions or the initial roughness of the plate. Neither the initial nor the final roughness value of the plate was found to correlate with the wear factor of the polyethylene component tested against them ($R^2 = 0.2$ and 0.0 respectively).

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APPENDIX

Cleaning protocols

(a) Stainless steel plates

The plates are cleaned using the following protocol:

1. Rinse with tap water to remove bulk contaminants.
2. Immerse in a solution of 1% Neutracon and place in an ultrasonic bath for 10 min at 37 °C.
3. Rinse in distilled water.
4. Dry with a lint-free wipe.
5. Wipe with acetone and a lint-free wipe.

(b) UHMWPE pins

The pins were cleaned according to the following protocol, closely following the ASTM recommendation (ASTM F732-00, part A6):

1. Rinse with tap water to remove bulk contaminants.
2. Immerse in a 1% solution of Neutracon and place in an ultrasonic bath for 15 min at 37 °C.
3. Rinse in a stream of distilled water.
4. Immerse in distilled water and place in an ultrasonic bath for 5 min at 37 °C.
5. Dry with a lint free tissue.
6. Immerse in acetone for 3 min.
7. Dry with a lint-free tissue.
8. Allow to dry in a biological flow cabinet at room temperature for 30 min.