Compliant layer acetabular cups: friction testing of a range of materials and designs for a new generation of prosthesis that mimics the natural joint

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Abstract: Total joint replacements (TJRs) have a limited lifetime, but the introduction of components that exhibit good lubricating properties with low friction and low wear could extend the life of TJRs. A novel acetabular cup design using polyurethane (PU) as a compliant layer (to mimic the natural joint) has been developed. This study describes a series of friction tests that have been used to select the most appropriate material, optimize the design parameters, and fine-tune the manufacturing processes of these joints. To determine accurately the mode of lubrication under which these joints operate, a synthetic lubricant was used in all these tests. Friction tests were carried out to assess the lubrication of four PU bearing materials. Corethane 80A was the preferred material and was subjected to subsequent testing. Friction tests conducted on acetabular cups, manufactured using Corethane 80A articulating against standard, commercially available femoral heads, demonstrated friction factors approaching those for full-fluid-film lubrication with only approximately 1 per cent asperity contact. As the joint produces these low friction factors within less than half a walking cycle after prolonged periods of loading, start-up friction was not considered to be a critical factor. Cups performed well across the full range of femoral head sizes, but a number of samples manufactured with reduced radial clearances performed with higher than expected friction. This was caused by the femoral head being gripped around the equator by the low clearance cup. To avoid this, the cup design was modified by increasing the flare at the rim. In addition to this the radial clearance was increased. As the material is incompressible, a radial clearance of 0.08 mm was too small for a cup diameter of 32 mm. A clearance of between 0.10 and 0.25 mm produced a performance approaching full-fluid-film lubrication. This series of tests acted as a step towards the optimization of the design of these joints, which has now led to an *in vivo* ovine model.

Keywords: total joint replacement, polyurethane, friction, acetabular cup, lubrication

1 INTRODUCTION

The pain and discomfort caused by diseased or damaged joints can be relieved, to a certain extent, by total joint replacement (TJR). TJR is undoubtedly one of the most successful major surgical operations undertaken. In a study by Berry *et al.* [**1**] it was found that 77.5 per cent of Charnley total hip replacements survived for 25 years. However, the components that fail can cause significant suffering to the patients and are a drain on health care resources. There would be great benefit if TJRs could be designed to last longer and so reduce the need for revision procedures.

The overall objective of this work was to develop a new generation of artificial joints that would outperform currently available conventional prostheses. Conventional TJRs operate within the mixed lubrication regime [**2–5**] and contact between the joint surfaces results in wear debris [**6–13**]. This is associated with joint failure mechanisms such as wear

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particle induced osteolysis [**14–16**]. Lubrication theory predicts that, as an articulating joint moves from mixed lubrication to fluid-film lubrication [**17**, **18**], contact of the surface asperities reduces, leading to a reduction in the friction and wear of the joints and an increase in longevity.

Elastohydrodynamic lubrication (EHL) and squeeze-film action are the two main lubrication mechanisms in the natural joint [**19–23**]. During the stance phase EHL predominates, when pressure is generated in the lubricant by an entraining motion between the two joint surfaces. Squeeze-film action predominates at heel strike in the walking phase; the two surfaces move towards each other, squeezing the fluid out of the joint space. Both mechanisms are enhanced by elastic deformation in the joint surfaces.

The new design of artificial joint aims to mimic the natural synovial joint, which is a bearing lined with a layer of low modulus material operating with a film of lubricant completely separating the two articulating surfaces [**19–23**]. The artificial joint uses a low-modulus polyurethane (PU) load-bearing layer bonded to a higher modulus polyurethane backing to aid fixation. Such an arrangement should operate with extremely low friction and wear, extending component life.

This concept has been demonstrated in the past [**24–30**], but a number of technical issues remain, such as compliant layer modulus, joint diameter, and layer adhesion properties, that need to be resolved in order to develop a viable long-term soft layer prosthesis. This study looked at the friction and lubrication performance of compliant layer cups using different materials and, therefore, various layer compliances as well as different joint diameters and radial clearances. Other work [**31**] has assessed the performance of these joints with respect to the effects of head radius and layer thickness on the theoretical calculations of film thickness.

2 MATERIALS AND METHODS

2.1 Prostheses for friction testing

Medical grade thermoplastic PUs were chosen as the compliant layer materials as they have a track record as long-term implants [**32–36**] and their mechanical properties (tensile strength and tensile modulus) compare well with articular cartilage (see Table 1). A high modulus medical grade thermoplastic PU was used as the backing material (Corethane, Shore hardness 75D) with a lower modulus PU (Shore hardness 80–95 on the A-scale) used as the compliant layer [**38–41**]. For the component manufacturing process, a mould was prepared and the harder material was moulded in the appropriate cavity to produce the outer shell of the cup. The part was then de-moulded and a 4 mm hole was drilled through the pole. The modified part was then used as an insert to produce the acetabular cup with the hard outer and soft inner bearing. The modified shell was then loaded into the appropriate cavity and the softer material was injected in through the liner to produce the soft bearing layer. The cup was then de-moulded and inspected before being ready for use.

Table 1 Typical mechanical properties for common biomedical elastomers, compared with articular cartilage [**37**, **38**]

Material	Tensile strength (MPa)	Tensile modulus (MPa)
Articular cartilage	10–30	10–100
Polyurethane	20–60	10–100
Silicone rubber	3–12	3–6
Poly(olefin)	10–16	140–1600
Hydrogel	0.5–10	0.5–90

Tests were performed on 22, 28, 32, and 46 mm diameter compliant layer acetabular components articulating against similar diameter commercially available femoral heads (see Table 2). In addition to this, some tests were carried out with the 32 mm diameter acetabular component paired with a similar diameter chromium-plated ball bearing head (see Table 2).

2.2 Friction tests

A full theoretical analysis of the behaviour of compliant bearings would require a simultaneous solution of the Reynolds (fluid flow) and elasticity equations (deformation of the contact) in three dimensions, which is an extremely computationally intensive process. In order to make calculation more manageable, various simplified models have been proposed for EHL and micro-EHL (μ-EHL) [**42–44**]. As compliant bearings are viscoelastic materials that are likely to suffer creep under load [**45**], a series of investigations was performed to compare experimental values of friction factor with current EHL and μ-EHL theory.

Stribeck analysis has been used widely to determine the lubrication regime in loaded contacts (a plot of the friction factor, f, against the Sommerfeld number, z). In this study, the friction factor was defined as [**3**]

$$f = \frac{T}{rL} \tag{1}$$

Table 2 Femoral components used in friction testing

Component	Material	Dimensions (mm)	Surface characteristics	
			R_a (μm)	R_q (μm)
Ball bearing	Cr-plated steel	31.74	0.067	0.100
Exeter* 32 mm	CoCrMo	31.92	0.008	0.010
Biolux† 32 mm	Al_2O_3	31.96	0.003	0.005
Exeter* 28 mm	CoCrMo	27.96	0.019	0.026
Exeter* 22 mm	CoCrMo	22.20	0.019	0.026

* Howmedica, UK.
† Centrepulse (Sulzer), Switzerland.

where T is the frictional torque between the bearing surfaces, r is the femoral head radius, and L is the load applied. The Sommerfeld number was defined as

$$z = \frac{\eta u r}{L} \tag{2}$$

where η is the viscosity of the lubricant and u is the entraining velocity of the bearing surfaces.

Figure 1 represents an idealized Stribeck plot. A falling trend indicates a mixed lubrication regime in which the load is carried by a combination of asperity contact and pressure developed in the lubricant, while a slightly rising trend with low overall friction indicates fluid-film lubrication with no asperity contact [**45**]. The transition from mixed to full-fluid-film lubrication is observed as the curve reaches a minimum, with typical friction factors below 0.01.

Friction tests were performed on the Durham friction simulator II [**4**, **45**]. The simulator works in a similar way to the Durham hip function simulator that has been described in detail elsewhere [**46**]. The friction simulator comprised a fixed main frame with an upper oscillating frame to provide the motion cycle. The hip prostheses were mounted inverted to the anatomical position, with the femoral component in the upper oscillating frame and the acetabular component below in a friction measuring carriage, which was itself mounted on externally pressurized bearings, providing both a 'self-centring' mechanism and a low-friction rotation axis in the sagittal plane. As frictional torque resulting from femoral component articulation tended to rotate the carriage, a Kistler piezoelectric force transducer, calibrated to measure frictional torque, was used to resist the rotation.

A vertical load was applied to the prosthesis using a servohydraulic system with closed-loop feedback control. A simple load profile comprising a low-load swing phase and a higher-load stance phase was used (similar to that found by English and Kilvington [**47**]), and the maximum and minimum forces exerted on the prostheses were 2000 and 50 N respectively. An initial warm-up test of 400 cycles was performed on each of the cups, to allow conditions in the joint to stabilize before taking any measurements. Sinusoidal motion was imposed on the femoral component in the flexion–extension plane with an amplitude of $\pm 24^\circ$ and a cycle frequency of 0.8 Hz. Load, frictional torque, and angular displacement were each measured 128 times per cycle.

Tests were performed with carboxymethyl cellulose (CMC) (BDH, UK) solution as the lubricant with

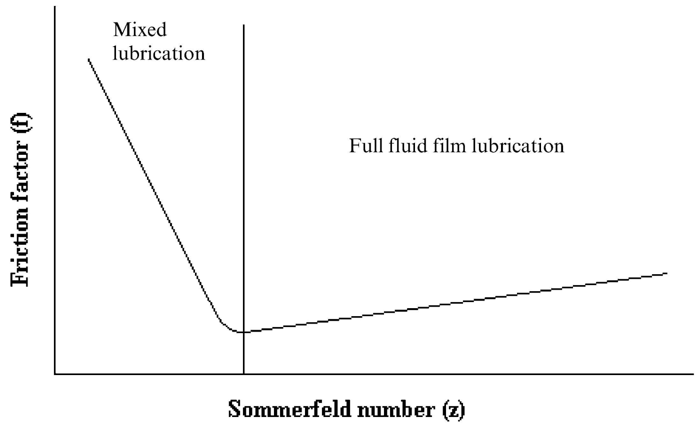

Fig. 1 Idealized Stribeck plot of lubrication mechanisms

a viscosity range of 0.001–0.150 Pa s. CMC fluids were used because of their similar rheological properties to synovial fluid [**48**]. Silicone fluids (Dow Corning 200 Fluid) were also used in some of the tests to enable tests to be performed with viscosities up to 29.25 Pa s.

The effect of layer modulus, creep, and radial clearance was assessed. Dry friction tests were also performed to determine the amount of asperity contact within these joints under different lubricated conditions. In addition to this, friction tests were done to determine the friction under start-up conditions. The objective of these tests was to investigate the design parameters that influence the tribological performance of compliant layer bearings.

For the tests investigating the effect of layer modulus, four PU materials [Corethane 80A (Corvita, Corporation), ChronoFlex AL-80A (CardioTech Int.), Pellethane 2363-80A (Dow Chemical), and CSIRO 85A (CSIRO Division of Chemicals and Polymers)] were tested against the Exeter head with CMC fluids used as the lubricant.

The tests to determine the amount of asperity contact occurring within lubricated conditions for these compliant layer joints used a C80A/C75D cup against both the chromium-plated ball-bearing head (tested in CMC fluids and silicone fluids) and the Exeter head (tested in CMC fluids alone). Dry friction tests were undertaken on four polyurethane bearings, the same bearings that were tested in the previous tests. These were coupled against Exeter and Biolux 32 mm femoral heads, at ambient temperature without lubricant. The maximum load was reduced to 1000 N to avoid overloading the friction transducer.

The frictional characteristics of the C80A/C75D acetabular cup against the Exeter head (32 mm bearing diameter) under start-up conditions were assessed. Conditioned and unconditioned cups were used in this test and were lubricated with distilled water. Unconditioned cups were as manufactured, while conditioned cups were immersed in Ringer's solution at 37 °C for several months before testing. The pre-load was set at 2000 N and was applied for 1, 2, 5, 10, and 20 minutes.

Radial clearance tests were performed on C80A/C75D cups coupled with the 32 mm diameter Exeter head. CMC fluids were used as the lubricant. A set of three cups (C1, C2, and C3) was manufactured with moderate radial clearances (0.12, 0.27, and 0.15 mm), while a second set (R1, R3, and R4) was prepared with slightly larger clearances (0.51, 0.54, and 0.24 mm) (see Table 3).

Further tests were carried out on the C-series cups to assess the effects of creep on the friction developed between the bearing surfaces. Three cups were moulded from each of the C1, C2, and C3 cores (32.346, 32.696, and 32.424 mm respectively). Replicas of the cups, using Provil MDC silicone rubber (Bayer, UK), were taken before testing, after soaking at 37 °C for 7 days in Ringer's solution and after loading at 2000 N for 7 days. One cup from each set was then allowed to rest for six months, re-soaked, and re-loaded with replicas taken at each stage. The replicas were measured on a coordinate measuring machine (CMM) (Howmedica, Limerick).

Finally, the effect of the femoral head radius was evaluated by testing 22, 28, 32, and 46 mm diameter femoral heads against C80A/C75D acetabular cups of the same nominal diameter.

3 RESULTS AND DISCUSSION

3.1 Effect of layer modulus

Earlier work demonstrated that optimal frictional performance was obtained from materials with a hardness of 4 N/mm^2 (equivalent to an elastic modulus of about 12 MPa) [**49**]. The objective of this series of experiments was to assess the effect of layer compliance using the new generation of bio-stable PUs. The joints generally performed poorly with mixed lubrication (friction factor in the range 0.002–0.06; see Fig. 2). The most compliant cup (ChronoFlex AL-80A) showed the best performance and in general the friction increased with increasing layer modulus. An increasing modulus resulted in a reduced lubricant film thickness with increasing asperity contact.

Table 3 Design parameters and minimum film thickness for cups used to assess conformity

Design parameters	C1	C2	C3	R1	R3	R4
Core diameter (mm)	32.35	32.70	32.42	33.00	32.70	32.42
Radius of acetabular component, R_2 (mm)	16.08	16.23	16.11	16.47	16.50	16.20
Thickness of compliant layer, h_t (mm)	1.92	1.77	1.90	1.54	1.76	1.80
Equivalent radius, R (m)	2.139	0.959	1.773	0.520	0.910	1.077
Circular contact radius, a (mm)	14.9	12.2	14.3	10.2	12.1	12.6
Minimum lubricant film thickness, h_{min} (μm) at a viscosity of 0.001 Pa s	0.113	0.077	0.104	0.055	0.075	0.082

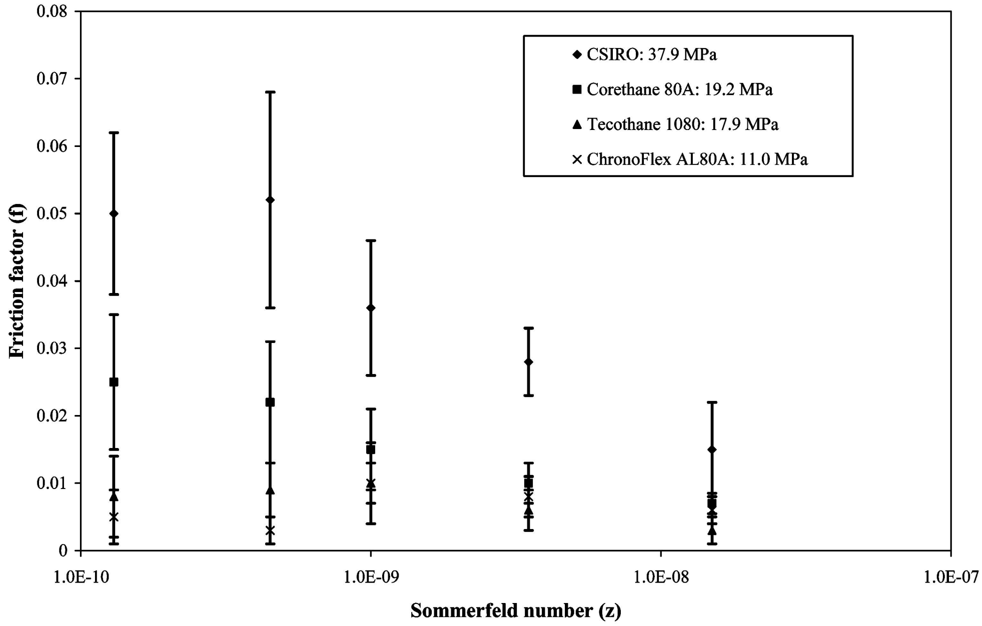

Fig. 2 The effect of layer compliance on the friction

However, compliant layer conformity appears to be a significant factor in these results. With the exception of the ChronoFlex sample, all of the combinations were highly conforming, having only small radial clearances (typically 0.08 mm). Although highly conforming contacts favour thick lubricating films, they will also be more sensitive to imperfections in the bearing form and to the effects of incompressibility of elastomers. Consequently, there appears to be a practical limit to increasing conformity, beyond which mixed lubrication tends to predominate. Therefore, this series of tests highlighted that the radial clearance is important for these joints. This will be discussed later.

3.2 Asperity contact

Using a Corethane (C80A/C75D) cup with a radial clearance of 0.17 mm for the lubricated test, the joints operated with extremely low friction with either head (Fig. 3) and the friction factors with the CMC were all below 0.01, well below the values measured for conventional prostheses [**5**, **30**, **46**]. The curve shows a minimum at a Sommerfeld number of around 10^{-9}, which equates to a viscosity of around 0.01 Pa s, the viscosity of synovial fluid [**48**]. A slightly falling Stribeck curve was observed over the lubricant range 0.001–0.01 Pa s, indicating mixed lubrication.

Under mixed lubrication, friction is a simple sum of the friction caused by asperity contact and lubrication film shearing. Therefore, the level of asperity contact can be estimated from the equation below [**3**]

$$\mu_{\text{mixed}} = \left(\frac{\%\ \text{contact}}{100}\right)\mu_{\text{dry}} + \left(\frac{1-\%\ \text{contact}}{100}\right)\mu_{\text{EHL}} \tag{3}$$

Dry friction tests were performed on acetabular cups made from four different PU bearing materials. Extremely high dry friction factors (see Table 4), in the range 0.8–1.0, were recorded, indicating that dry contact in compliant layer joints would produce high shear forces at the bearing surface with consequent rapid wear, structural failure in the cup, or failure at the fixation interface. In contrast, well-lubricated compliant bearings have operated with very low friction, typically <0.01.

For both compliant and conventional bearings, the EHL theoretical fluid-film friction (assume a viscosity of 0.01 Pa s) is less than 0.001 [**50**]. Since the measured friction factors in compliant bearings are less

Table 4 Friction factors measured for compliant layer cups without lubricant

	Dry friction factor			
	Exeter head*		Biolux head†	
Bearing material	Mean	SD	Mean	SD
ChronoFlex	0.98	0.08	0.95	0.12
Tecothane	0.77	0.30	0.93	0.17
Corethane	1.01	0.18	1.02	0.13
CSIRO	1.07	0.12	0.83	0.17

*Howmedica, UK.
†Centrepulse (Sulzer), Switzerland.

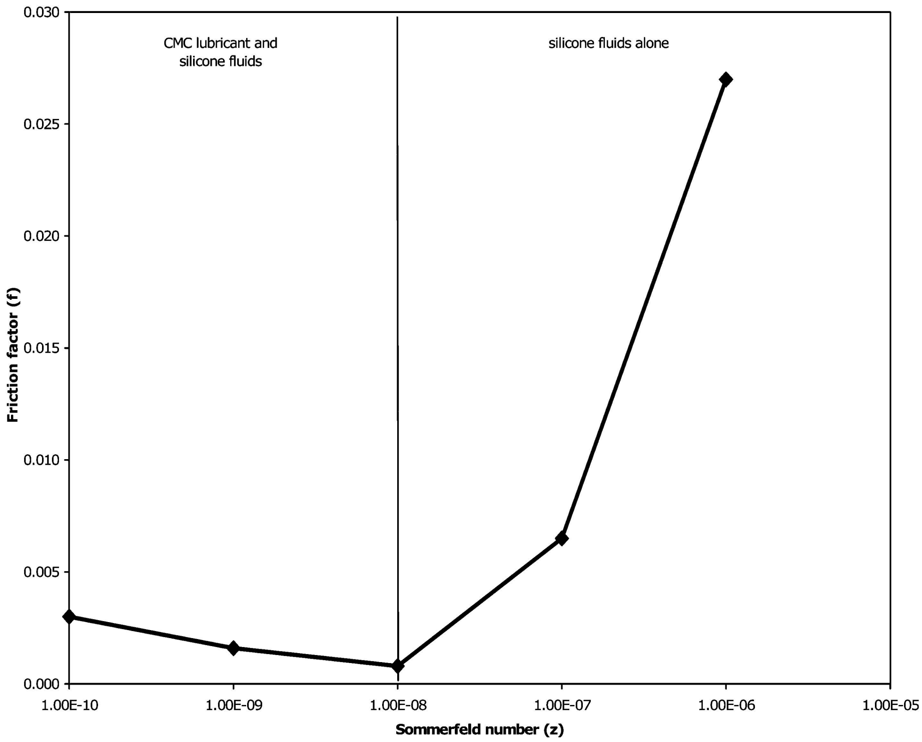

Fig. 3 Stribeck analysis using CMC and silicone lubricants

than 0.01, less than 1 per cent of asperity contact is estimated. In contrast, as the dry friction factor is around 0.1 in conventional joints, the asperity contact is estimated at around 50 per cent [**38**].

3.3 Start-up friction

In dry conditions, the friction between metal-on-PU joints is an order of magnitude greater than that of conventional metal-on-UHMWPE (ultra-high molecular weight polyethylene) systems [**4**, **38**]. If the joint were to function for any length of time under such conditions, the interface and the fixation would quickly become damaged and this might lead to failure. One important area of investigation examined the possibility that, after a period of inactivity, fluid could take a finite length of time to be drawn through the joint space and thus represent a potential source of damage. Therefore start-up friction is thought to possibly be a concern [**27**, **28**].

The level of friction at the start of a cycle, when the joint had been rested for various time periods under load, was measured, with particular attention given to the amount of time taken to develop full-fluid-film lubrication following restoration of the motion. The start-up friction factor plotted against pre-load time is given in Fig. 4.

The level of friction observed in Fig. 4 and the subsequent frictional torques (circa 15 N m for a 20 minute pre-load time for the unconditioned cup) were still below those required to cause failure at the fixation interface [**51**]. However, the torque measured by Andersson *et al.* [**51**] was the torque that was required to remove a well-fixed acetabular component by a single application. The high start-up frictional torques developed by the compliant layer joints could not be tolerated if it were to hold for any length of time. However, when the joint was articulating, the frictional torque reduced rapidly (see Fig. 5). As relative motion started, the frictional torque was high, but it very quickly reduced to a very low steady state value within one cycle. The results clearly demonstrate that the joint was functioning under full-fluid-film lubrication in less than half a walking cycle and consequently the start-up friction should not be an issue in compliant bearings.

3.4 Effect of radial clearance

EHL theory predicts that bearing conformity is a critical factor in achieving fluid-film lubrication in compliant bearings. The tests within this study to assess the effect of layer modulus also indicated that bearing conformity is a critical factor. To test the theory, various radial clearances of compliant layer joint were tested.

In the tests performed on the C-series cups, there was no perceptible difference in the Stribeck curves

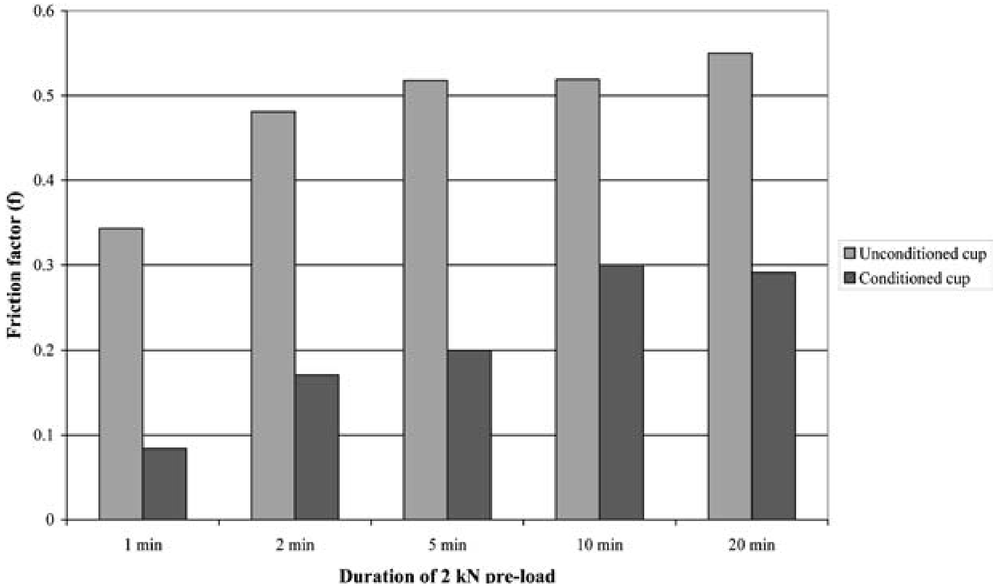

Fig. 4 Start-up friction factor against pre-load

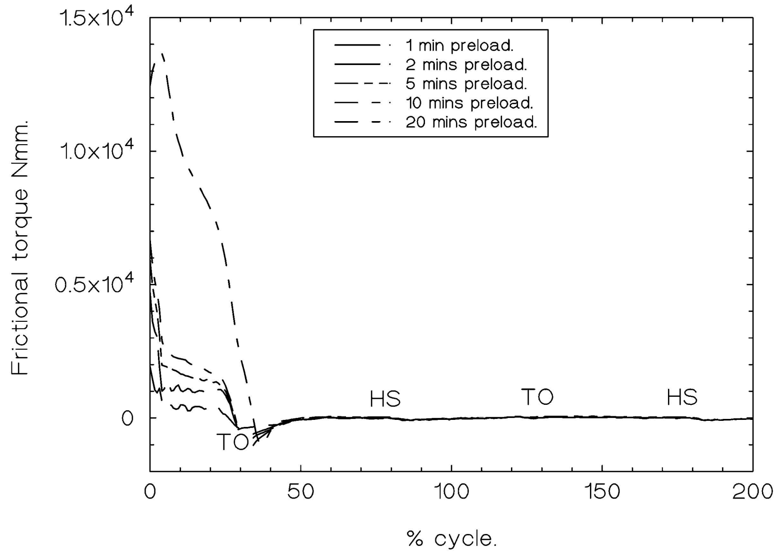

Fig. 5 Frictional torque measured for two articulation cycles

of the different cups, indicating that in this clearance range, conformity is not critical (see Fig. 6).

In the R-series (Fig. 7), the degree of conformity appeared to be more important. In general the R-series did not perform as well as the C-series, with the largest clearance giving the lowest friction factors. However, a less-than-optimal manufacturing regime caused the larger clearance R-series cups to have 100 μm lobes (subsurface) on their bearing surfaces (measured by Rank Taylor Hobson using a Talyrond 73, illustrated in Fig. 8). The friction results for the R-series of cups seem to be more influenced by the presence of these lobes than clearance *per se*. When the contact area was small (largest clearances R1 and R3) and predominantly confined to the pole of the cup, interaction with the lobes was minimized. In contrast, if the area of contact extended up the cup sides, there would be greater interaction with the lobes' high spots. Clearly bearing surface sphericity is an important design and manufacturing consideration. Small amounts of out-of-roundness appear to be best accommodated by lower conformity bearings and the R1 results (friction factors for R1 in the range 0.003–0.007 compared with friction factors in the range 0.005–0.016 measured for R4) suggest that large clearance cups with significant out-of-roundness can still perform with low friction.

The experimental friction factors were slightly

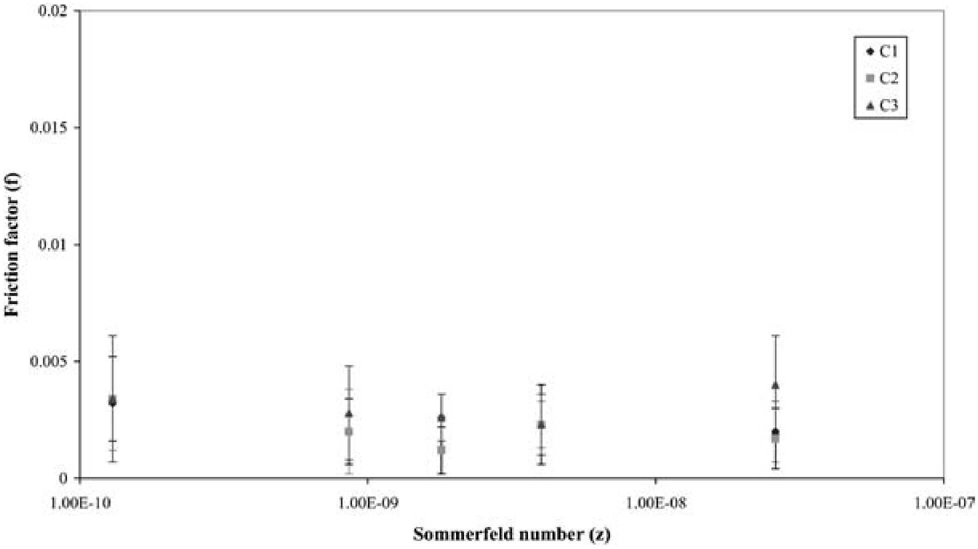

Fig. 6 Stribeck analysis for the C-series cups

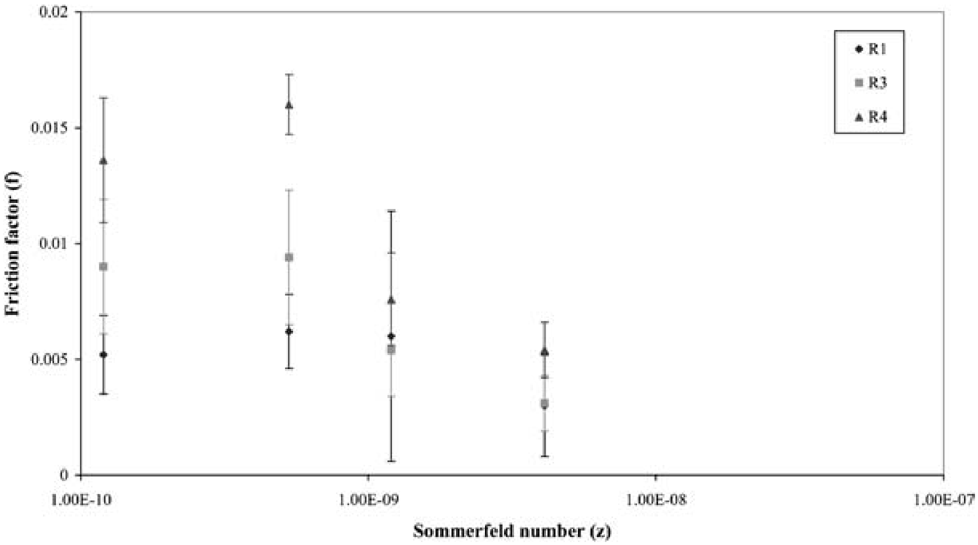

Fig. 7 Stribeck analysis for the R-series cups

higher than theory predicted for the C-series cups. As the lobes in the R-series resulted in a degree of bearing contact, no theoretical comparison was made.

While EHL theory is a useful tool in component design, it cannot accommodate manufacturing inaccuracies, nor does it take account of the inevitable creep, which can lead to pinching at the femoral head, with consequent lubrication starvation and increased friction. Design recommendations should also be drawn from empirical evidence. A number of cups with inadequate radial clearances performed with higher-than-expected friction, caused by creep-induced femoral head pinching, with subsequent lubrication starvation [**45**].

As the method of cup fixation could affect the degree of creep, a series of tests was carried out to monitor the level of creep that might be expected with the typically modular design of insert found in non-cemented, metal-backed acetabular cups. In these tests, the modular compliant layer insert (ABG, Howmedica, Inc.) had a minimum space (0.4 mm) between the insert and the shell [**45**]. The gap allowed the material to creep, with consequent femoral head pinching, lubricant starvation, and increased friction. A comparison of cemented versus modular metal-backed fixations, each subjected to a load of 2 kN at 37 °C for 7 days, showed that a penetration range of 0.2–0.3 mm was well tolerated by cemented cup designs, provided an adequate radial

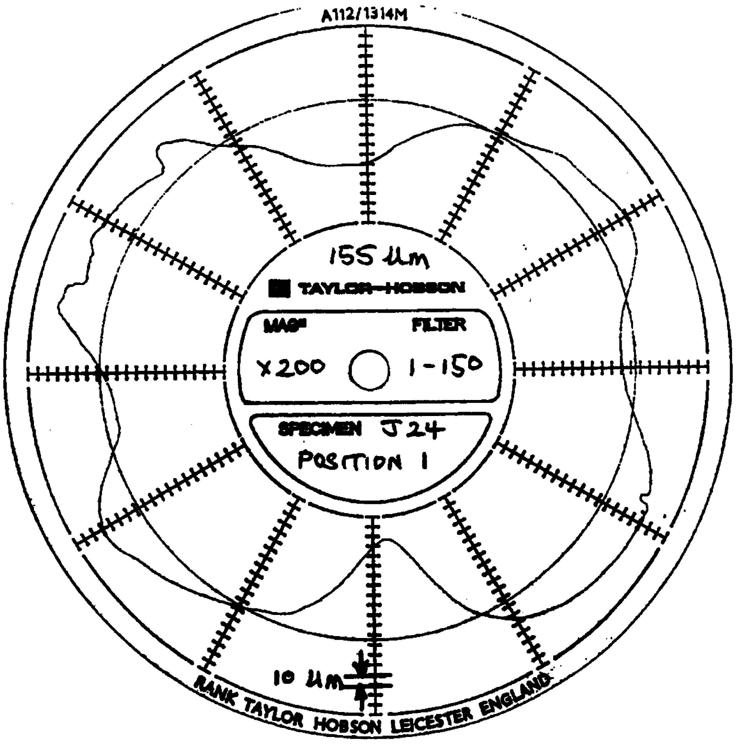

Fig. 8 Talyrond measurement for lobed cups

clearance (0.10–0.20 mm) was present, but when a modular snap-fit approach was used the metal shell allowed further form changes.

In order to develop a more complete understanding of the way that clearance and creep affect friction in compliant cups, nine compliant layer cups were tested in the Durham friction simulator in a range of CMC fluids and the level of both plastic and elastic deformation assessed [**52**]. All cups showed a slight decrease in internal diameter after loading as cups deformed to fit the femoral heads (representative measurements are given in Table 5). The deformation due to loading largely recovered after the resting period. In these tests, the creep deformation due to short-term loading was measured. It would, however, be interesting to determine the effects on creep of longer-term loading, as this may be more indicative of the long-term behaviour of these joints.

The friction factors measured for all of the cups were very low before loading (less than 0.01). For the C1 and C2 cups (the C1 cups are shown in Fig. 9) there was only a slight increase in friction after loading (friction factor of 0.02 or less). This slight increase remained after the rest period. The C3 cups, however, showed higher friction (friction factor in the range 0.03–0.07) after loading. This was because the C3 cups had a tendancy to pinch the femoral head, causing lubricant starvation and increased friction. After resting for 6 months, the friction factor for this cup had returned to values close to its initial value. This is shown in Fig. 10.

The initial modular cup design was modified by increasing the flare at the rim, which obviated the

Table 5 CMM measurements of replicas taken from creep and re-creep samples

Cup type*	Manufactured	Post-soaking	Post-loading and friction test	Post-rest and re-soak	Post-reloading	Post-friction test
C1 ϕ	32.005	32.006	31.901	Not measured	32.051	31.881
Δ	0.0589	0.0667	0.0357		0.1489	0.1327
C2 ϕ	32.218	32.226	32.034	32.165	32.051	32.149
Δ	0.0957	0.0687	0.0700	0.0721	0.1175	0.2277
C3 ϕ	32.082	32.014	31.803	31.990	31.874	31.956
Δ	0.0335	0.0102	0.0500	0.1106	0.1521	0.1107

*ϕ = average diameter of each replica; Δ = deviation from average.

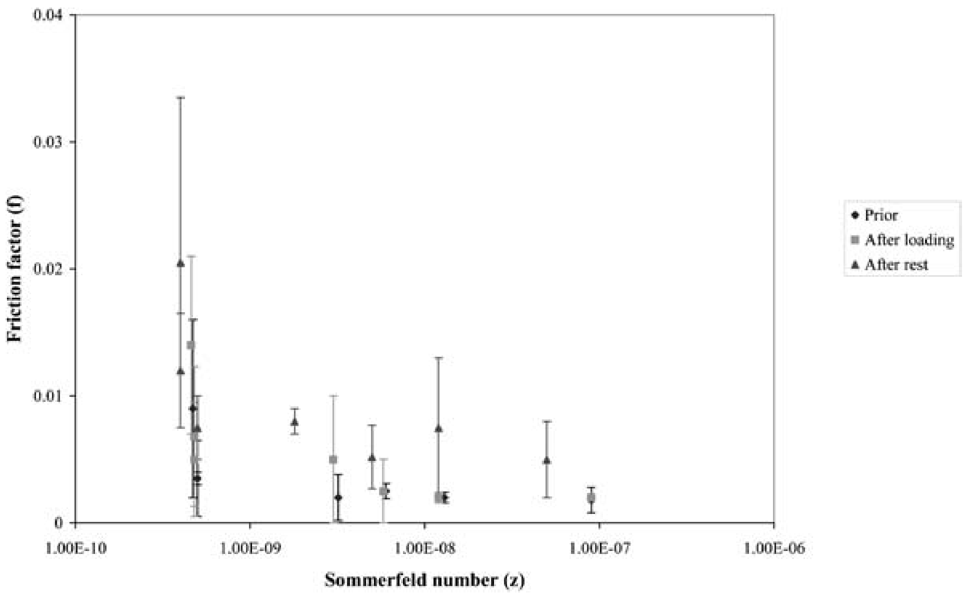

Fig. 9 Stribeck analysis for C1 cups in creep tests

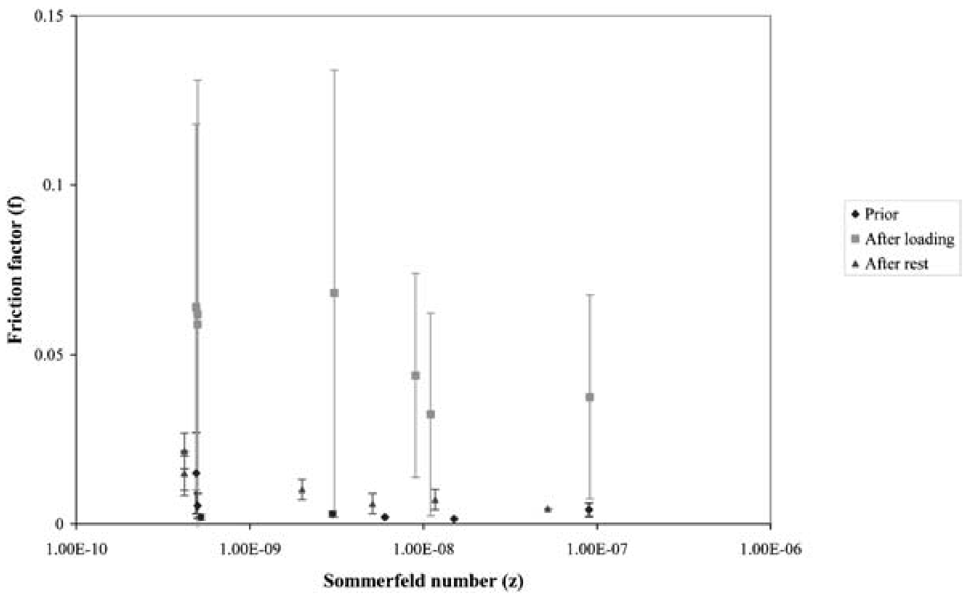

Fig. 10 Stribeck analysis for C3 cups in creep tests

problem of pinching. The present evidence shows that the clearance between the cup and head is not critical, so long as it is not so small that significant contact occurs. For 32 mm cups a radial clearance of 0.08 mm was too small, while a clearance of between 0.10 and 0.25 mm performed well. Significant out-of-roundness can be better tolerated by large clearance bearings.

3.5 Effect of femoral head size

In conventional total hip replacements, femoral heads are normally in the diameter size range of 22–32 mm. In the absence of fluid-film lubrication the smaller diameter prostheses exhibit lower wear volumes and lower frictional torques, while the larger prostheses have reduced penetration rates and lowered contact stresses. However, when considering fluid-film lubrication, entraining velocity, which is directly related to head size, has been found to be an important factor in determining film thickness [**17**]. In hip simulator wear studies, large-diameter (46 mm) compliant polyurethane resurfacing hips were found to produce very low steady-state wear, suggesting an effective fluid-film lubrication regime [**53**].

Cups were manufactured to accommodate 22, 28, 32, and 46 mm (46 mm is a novel design that mimics the natural physiological acetabulum [**54**]) femoral heads and their Stribeck curves are given in Fig. 11. Theory predicts that the larger heads would be expected to perform with lower friction, but in practice all of the cups performed effectively with the 22, 28, and 32 mm diameter joints producing similar low friction factors. The larger diameter joint produced the highest friction, but this friction was still considerably lower than that produced by conventional joints [**46**] and is similar to that found by other workers for compliant layers [**30**, **45**, **49**].

3.6 Friction testing lubricant

All of the friction tests reported in this study used CMC fluids as the lubricant. Although this lubricant has similar rheological properties to synovial fluid, i.e. it is shear thinning, it is a synthetic lubricant that does not contain any proteins that are present in natural, synovial fluid. The presence of these proteins is likely to have an effect on the friction factors produced by these joints, as has been seen with other material combinations [**46**]. The protein adsorption will affect the friction produced (as protein–protein rubbing causes higher friction than lubricant shearing but lower friction than metal-on-polyurethane contact), but it will not affect the mode of lubrication under which these joints are acting. As this study was a comparative analysis to determine the effects of material compliance, head diameter, and radial clearance, CMC fluids were chosen as the lubricant. Further studies have taken place to measure and compare the friction produced by compliant layer joints using both bovine serum as the lubricant and CMC fluids. This will be reported in a future publication.

3.7 A comparison of theory with experiment

The experimental results generally showed a poor correlation with theoretical predictions, which assume full-fluid-film lubrication [**37**, **55**]. Three main sources of error are suggested:

1. Manufacturing inaccuracies such as out-of-roundness, which alters the equivalent radius of the cup, making EHL film thickness estimates inappropriate.
2. Possible film breakdown leading to load sharing by asperity contact. Bearing surface topography will influence both fluid-film formation and asperity load carrying, and as such is a particularly important parameter.
3. Bearing surface creep under load. No matter how accurately the bearing is characterized before testing, as soon as it is loaded it will creep, changing the effective equivalent radius and layer thickness.

4 CONCLUSIONS

While EHL theory provides a good first step in the design process, empirical evidence is required to identify design and manufacturing parameters. By measuring the frictional resistance at the articulating

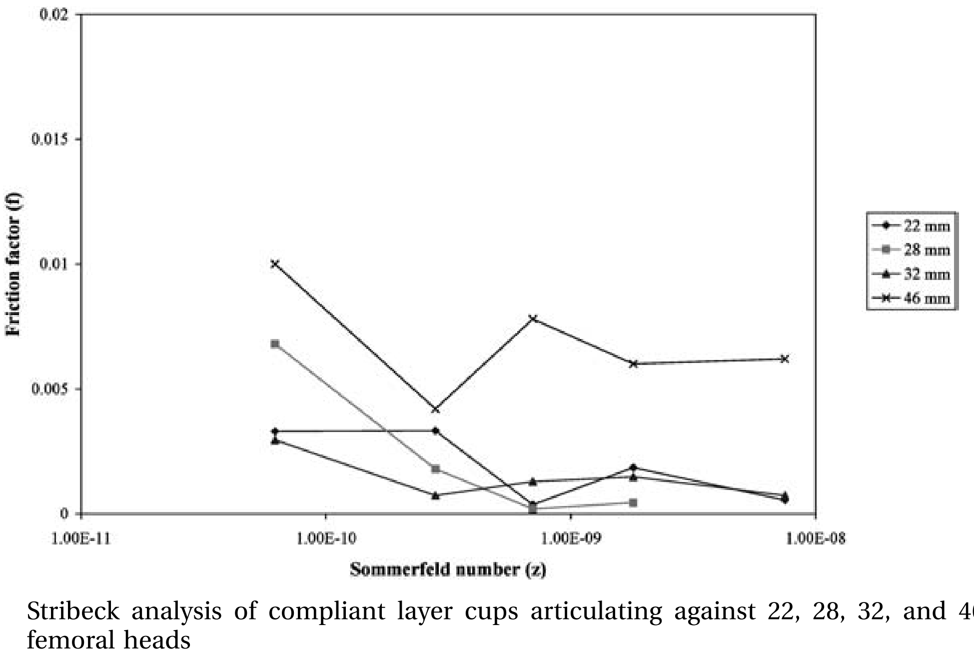

Fig. 11 Stribeck analysis of compliant layer cups articulating against 22, 28, 32, and 46 mm femoral heads

surface, the presence of a complete lubricating film of testing fluid was confirmed in the best performing samples. The design of the joint appears not to be particularly critical, with fluid-film lubrication observed over a wide range of conformity, head size, and layer modulus. This is important, as the bearing geometry will change with time due to creep. However, radial clearances below 0.1 mm and out-of-roundness should be avoided. The joints that performed in a less-than-optimal way provided important design guidance by pointing the way to a flared-form cup that avoids creep-induced pinching.

An excellent bond between the layer and backing is fundamental to the success of these joints if they are to perform well for several decades, and the development of new bonding technology [**56**] has ensured very good performance over many loading cycles. The performance of medical grade polyurethane in a bioenvironment has been questioned in the past, but the new generation of 'biostable' polyurethanes used here [**32**] offer significant improvements. Corethane was chosen as the best material for the acetabular cup because of its performance in these tests and its biostability [**57**]. There have been many developments in the field of TJRs, some with doubtful benefits in terms of performance and value for money [**58**]; however, the authors are encouraged by these results obtained to date.

Following this work, a scaled-down acetabular cup suitable for sheep, based on Howmedica's *Exeter Hip System*, was developed to test the design in a physiological environment and provide further practical evaluations of its effectiveness [**57**, **59**].

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APPENDIX

Notation

f	friction factor
L	load (N)
r	femoral head radius (mm)
T	frictional torque (N mm)
u	entraining velocity (mm/s)
z	Sommerfeld number
η	viscosity (Pa s)
μ_{dry}	coefficient of friction – dry
μ_{EHL}	coefficient of friction – full-fluid-film lubrication
μ_{mixed}	coefficient of friction – mixed lubrication