

Analysis of Run-in-Stage Wear Behavior and Contact Mechanics of Metal-on-Metal Hip Joint Bearings with Different Radial Clearances

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We systematically elucidate the effects of radial clearance on the wear behavior of hip joint prostheses bearings during run-in-stage. The results indicated that bearings with smaller radial clearances exhibited lower wear rate and less abrasive wear characterized by mild surface roughness and sphericity. The contact mechanics and lubrication regime of MOM bearings with different radial clearances were also analyzed, indicating that bearings with radial clearance of $\sim 20\ \mu\text{m}$ was more beneficial for small contact pressures even the enlarged clearance made the bearings operate under a full-fluid-to-mixed lubrication during wear. The agreement between the experimental results of wear rate (mm^3/Mc) and contact mechanics and lubrication analysis indicated that the radial clearance significantly affected the wear behavior of the MOM bearings. [doi:10.2320/matertrans.M2014440]

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1. Introduction

Compared to the ultrahigh molecular weight polyethylene (UMWPE) hip joints, metal-on-metal (MOM) bearings for hip prosthesis have attracted more attention over the last two decades.¹⁾ CoCr-MOM bearings exhibit extremely high survival rates over 5–10 years of service^{2,3)} owing to their excellent mechanical properties and extremely low wear rate,^{4–7)} and they are expected to serve for longer life than the present ones for younger, active patients. Despite of these encouraging results, recent reports also indicated that unexpected high failure rates for large head MOM bearings and debris detachment from the rubbing surfaces, which led to an alert from the Medicines and Healthcare products Regulatory Agency prompting an urgent review of all MOM articulations.⁸⁾ The debris enter the blood, serum, and urine of recipients, cause a high incidence of adverse reaction to metal debris (ARMD)^{9–11)} known as periprosthetic soft tissue reactions and systemic genotoxicity.^{12–14)} Some other complications with MOM bearings include loosening, joint dislocation and squeaky.^{15,16)} The contradiction between high-wear-resistant behavior and known risks of CoCrMo alloys appeal for further research to obtain MOM bearings with improved biocompatibility.

To achieve this aim, a variety of wear tests were conducted and their results have indicated that MOM bearings wear behavior could be affected by: (1) the materials (the use of ‘as cast’ or forged alloy with various heat treatments), (2) macro- and micro-geometry (difference in diameter and the clearance between mated components), (3) the resultant type and amount of lubrication, and, (4) the adoption of various hip joint simulators with various load and motion characteristics.^{15,17–25)} Regarding the geometry feature, researches have concentrated on the effects of head diameter and radial clearance between the head and cup.^{15,21,25)} Observations by

Rieker *et al.* indicated that longer run-in periods and greater run-in wear were associated with larger bearing clearances.²¹⁾ Brockett *et al.* also reported that increasing the clearance of large head bearings resulted in increased friction and incidence of squeaking.¹⁵⁾ The combined effects of the head diameter and clearance on the wear behavior of MOM bearings were investigated by Dowson *et al.*²⁵⁾ by a hip joint simulator, indicating that the run-in wear was more dependent on the diameter than the clearance. However, no optimum clearance design factors have been identified so far, and further systematic study is necessary on the MOM bearings wear behavior affected by clearance.

Many studies systematically analyzed the wear rates of MOM bearings during the whole wear (including run-in and steady state), but most of the wear took place during run-in.^{26,27)} As for the Co-Cr-MOM systems, it was observed that total volume wear during run-in stage is almost several times higher than the steady state one.²⁸⁾ In the present research, the clearance effect on MOM bearings wear behavior was focused on the run-in stage. However, this effect could not be distinctly expressed since MOM bearings experience only slight wear, since the CoCrMo alloys have superior tribological and mechanical properties. Therefore, Hanks-balanced salt solution (HBSS) with inert nature to CoCrMo alloys was chosen as lubricant rather than dilute bovine serum, to avoid any complex organic factor interference and artificial wear rate concerned by protein precipitation and degradation, or change of the boundary lubrication due to the temperature rise from friction.^{29–34)} One reminder is that all the results obtained in this HBSS lubricated experiment can be magnified for wear resistance estimation among bearings with various radial clearances, but without any clinical relevance because corresponding wear mechanism of the bearings alters in HBSS lubricated environment.

The MOM bearings are developed from material to finished products at one go without interruption, for an in-

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depth control for material microstructure of high-strength forged CoCrMo alloys and wear behavior research on MOM bearings made of these alloys from geometrical design aspect. The high precision of computer numerical control machine enables the production of MOM bearings with desired clearances ranging from several micrometers to about 100 micrometers. Detailed surface morphology and shape evolution were observed, on whose basis, the intrinsic relationships among the volume wear rate, contact mechanics, and lubrication evolution of MOM bearings with various radial clearances was analyzed, by both finite element method (FEM) and classical theory.

2. Models

The steady state dry contact between bearing surfaces was analyzed by ABAQUS (version 6.11-3, Dassault Systèmes Simulia Corp.). Figure 1(a) shows a CoCrMo alloy ball-in-socket model composed of a half cup and a half head with a 1.5 kN load ($w/2$, 2 time of a human weight) applied to the bottom surface of the head. The ϕ 40 mm heads paired with a series of ϕ 40.01–40.3 mm cups were aligned by “adjust only to remove overclosure”. Elastic modulus (E) of 230 GPa, Poisson’s ratio (ν) of 0.3, and isotropic and linear elastic properties were applied to the material. The geometry-based head and cup surfaces were defined as a contact pair; the former was considered the slave surface and the latter the master surface. “Surface to surface” and “small sliding” discretization was employed, and C3D8R (eight-node linear brick, reduced integration, hourglass control) elements were used to calculate the contact pressure between surfaces of the bearings. The mesh density was determined by mesh convergence, which indicated an appropriate head mesh density of $\sim 3 \times 10^4$, and a difference of only 0.07% in the maximum contact pressure when the mesh density increased to $\sim 5 \times 10^4$.

To compare the FEM results, the dry contact pressure of the bearings was also evaluated by the Hertz contact theory using an equivalent ball-on-plane semi-infinite model. The radius R of the equivalent ball is determined by:³⁵⁾

$$R = \frac{R_c R_h}{c}, \quad (1)$$

where R_c , R_h , and c are the radii of the cup and head, and the radial clearance, respectively. The contact radius r and maximum contact pressure $P_{\max}$ are given by:

$$r = \left(\frac{3wR}{E'} \right)^{1/3} \quad (2)$$

and

$$P_{\max} = \frac{3w}{2\pi R^2}, \quad (3)$$

respectively, where w is the load applied to the bearings, and the equivalent elastic modulus of the material E' can be simplified as:

$$\frac{1}{E'} = \frac{1 - \nu^2}{E}. \quad (4)$$

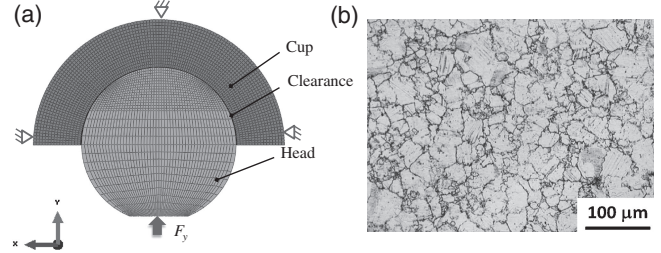


Fig. 1 A ball-in-socket configuration of MOM bearings for contact simulation, indicating a mesh refinement where surfaces contact with each other; (b) Optical image of the initial microstructure of CCMN alloy and cup surface prior to wear tests.

3. Material and Methods

3.1 Material characterization

The material was forged Co-27.66Cr-5.5Mo-0.13N alloy (CCMN, mass%).³⁶⁾ The alloy ingot was prepared by a vacuum induction melting process and subsequent homogenization at 1523 K for 18 h. This was followed by hot forging at 1273 K until a reduction rate of $\sim 45\%$, and then water quenching.

Figure 1(b) shows the initial microstructure that for bearing manufacturing, by using optical microscopy (Olympus BH2-UMA). The result shows equiaxed fine grains with a mean grain size of $\sim 20 \mu\text{m}$. A lot of annealing twins was also observed in the grains.

3.2 Wear tests

3.2.1 Shape and surface characterization

“Large diameter” femoral heads ϕ 40 mm and acetabular cups were manufactured, for a reduced risk of dislocation and increased motion range.^{20,37)} Each of the components was finished by precision polishing in strict control to guarantee same initial geometrical factors. Shape and surface topography estimation was evaluated by radius, sphericity and center-line-average surface roughness (Ra).

The radii and sphericities were measured by a coordinate-measuring machine (RVA600, Tokyo Seimitsu) on the superior hemisphere, where dominant contact occurs, at 17 different locations. The radial clearance was calculated by simply subtracting the head radius from the cup radius. Four bearings with different initial radial clearances were used in the tests.

The roughnesses of the samples was measured by a traditional dimensional contacting profilometer (Surfcorder DSF500, Kosaka), equipped with a diamond stylus tip of radius $5 \mu\text{m}$, and a digital filter assigned according to JIS01/ISO97. Roughness measurements were performed before and after wear test at a trace speed of 0.2 mm/s , with a vertical resolution of 7.5 nm . The cut-off used was 0.8 mm with a length for each measurement of 7.5 mm . The cups and heads were measured at hemisphere in the orthogonal 2 directions at 4 locations over each sample surface³⁸⁾ and 12 measurements were taken, from the geometric centre to the edge, 90° apart with each other. The composite roughness Ra of the bearings was calculated by:³⁹⁾

$$Ra = \sqrt{Ra_c^2 + Ra_h^2}, \quad (5)$$

where Ra_c and Ra_h are the roughness of the cup and head, respectively.

The detailed measurement locations for local sphericity and roughness measuring on the contact hemisphere, are shown in Fig. 2.

3.2.2 Wear test design

Prior to the wear tests, all the samples were ultrasonically cleaned (Sonocleaner 200D, Kaijo) using detergent, distilled water, and ethanol subsequently, and then kept in a vacuum drying oven (DP33, Yamato) for at least 24 h.

To replicate the physiological walking conditions, the wear test was conducted in a universal hip joint simulator (8870 series testing system, Instron Corp., Tokyo, Japan) at $37 \pm 2^\circ\text{C}$ under a HBSS-lubricated environment (NaCl 8.0, KCl 0.4, CaCl_2 0.14, NaHCO_3 0.35, $\text{Na}_2\text{HPO}_4 \cdot 2\text{H}_2\text{O}$ 0.06, KH_2PO_4 0.06, $\text{MgSO}_4 \cdot 2\text{H}_2\text{O}$ 0.2 g/L). The simulator configuration is shown in Fig. 3(a), in which the cup was mounted in an anatomical position above the head⁴⁰ at an inclination of 30° . A variable load ranging between 0.3 and 3 kN, and triaxial rocking motions (TRM) consisting of flexion-extension (flex/ext, 43° range), abduction-adduction (ab/ad, 11° range), and in-out pelvic rotation (in/out, 12° range) at frequencies of 1 Hz⁴¹ were applied on the bearings (Fig. 3(b)).

The wear tests were subjected to 1 million cycles (Mc), which is equivalent to $\sim$ half a year of walking of a normal human being,⁴² standing for a run-in stage of wear.^{19,27} The evolution of the progressive weight loss was carried out by using a balance (AUW 300, Shimadzu), at the interrupt of every 0.1 Mc between 0 and 0.5 Mc, and every 0.25 Mc between 0.5 and 1.0 Mc during the test. Worn surface morphologies of the bearings were examined by scanning

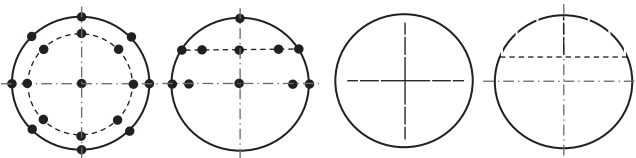


Fig. 2 Local sphericity and roughness measuring on the contact hemisphere, in which 17 points and 12 trajectories for sphericity and roughness measurements, respectively.

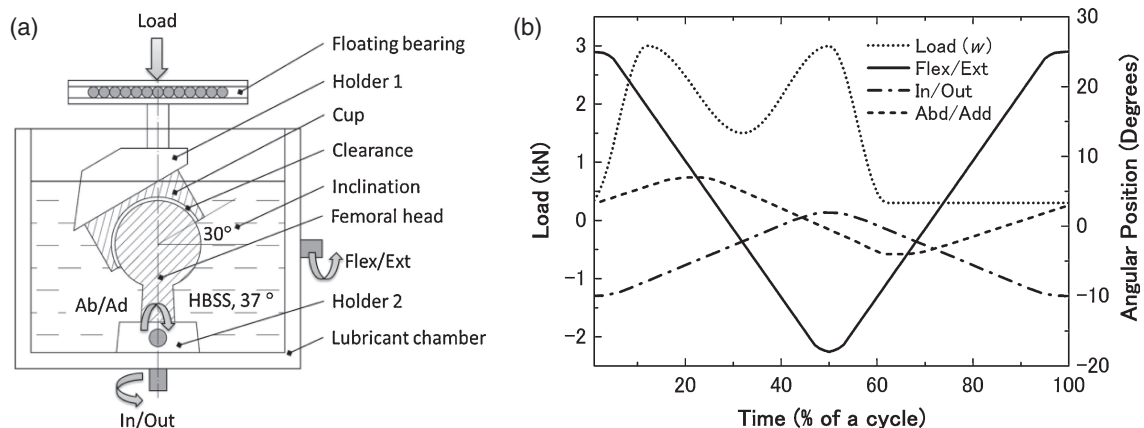


Fig. 3 (a) Configuration of the hip joint simulator produced by INSTRON, (b) Variation in (a) load and (b) angles for normal walking conditions during gait cycle based on ISO 14242-1.

electron microscopy (S-3400N Scanning Electron Microscope, Hitachi) and surface roughness analysis.

4. Results

4.1 Sample characterization

Table 1 shows detailed shape and surface characterization of the bearings before and after wear test, including radius, radial clearance, composite roughness (in flex/ext, ab/ad directions and the average values), and composite sphericity.

The results show that the initial radial clearances of the bearings were 19, 38, 106, and $149 \mu\text{m}$, and hereafter denoted by c_1 , c_2 , c_3 , and c_4 , respectively. The initial surface roughness satisfied the requirement for that of total hip prostheses made of metallic material according to ISO 7206-2.^{36,38} No significant difference was observed for the initial surface roughness (in different directions or averaged as a whole), and sphericity of the bearings by the Student's t-test (assuming unequal variances; two-tail, $P < 0.05$).

After wear test, radial clearance increased by increase of the cups. Oppositely, heads changed little in radius and sphericities, especially for bearings of small clearances, and heads of c_1 and c_2 showed excellent conforming surfaces after wear. All the bearings have deteriorative surface roughness and the increased Ra , especially in flex/ext direction. The composite Ra of c_1 and c_2 increased approximately 2 fold and that of c_3 and c_4 3 fold, respectively, indicating that surface irregularities have increased significantly after 1 Mc wear test.

4.2 Surface morphology

After 1.0 Mc wear tests, abrasive wear characterized by scuffing was observed on the bearing surfaces. Figure 4(a) shows typical SEM micrograph of new surfaces on cup apex, superior hemisphere. By contrast, the worn surface topographies at the same location on cup apex, for c_1 – c_4 , are shown in Fig. 4(b) to (e), respectively, in which the dominant wear direction in flex/ext is indicated by gray arrows. Longitudinal scuffing, varying in different extents, were observed on the worn surfaces angled with the flex/ext direction as a result of composited triaxial motions. Indistinct scratches and a few plowing grooves can be observed on the

Table 1 Initial and worn shape and surface characterization of the bearings.

#	R_c (mm)	Radial clearance (μm)	Ra_c (flex) (mm)	Ra (flex) (mm)	Ra_c (ab) (mm)	Ra (ab) (mm)	Ra_c (mm)	Ra (mm)	$Sphericity_c$	Composite sphericity (mm)
	R_h (mm)		Ra_h (flex) (mm)		Ra_h (ab) (mm)		Ra_h (mm)		$Sphericity_h$	
Initial at 0.0 Mc	c_1	19	0.014 ± 0.003	0.024 ± 0.004	0.015 ± 0.002	0.023 ± 0.004	0.014 ± 0.002	0.023 ± 0.004	0.005	0.014
			0.019 ± 0.006		0.018 ± 0.005		0.018 ± 0.006		0.013	
	c_2	38	0.020 ± 0.004	0.026 ± 0.003	0.018 ± 0.002	0.025 ± 0.001	0.019 ± 0.003	0.025 ± 0.002	0.005	0.010
			0.017 ± 0.001		0.017 ± 0.001		0.017 ± 0.001		0.009	
	c_3	106	0.020 ± 0.004	0.026 ± 0.003	0.017 ± 0.001	0.023 ± 0.001	0.018 ± 0.003	0.024 ± 0.002	0.005	0.008
			0.016 ± 0.001		0.016 ± 0.001		0.016 ± 0.001		0.006	
	c_4	149	0.021 ± 0.003	0.027 ± 0.002	0.022 ± 0.002	0.028 ± 0.002	0.021 ± 0.002	0.028 ± 0.002	0.009	0.009
			0.017 ± 0.001		0.018 ± 0.001		0.018 ± 0.001		0.009	
Worn at 1.0 Mc	c_1	27	0.036 ± 0.008	0.061 ± 0.020	0.021 ± 0.013	0.043 ± 0.005	0.028 ± 0.013	0.052 ± 0.017	0.019	0.020
			0.048 ± 0.021		0.036 ± 0.009		0.042 ± 0.016		0.007	
	c_2	47	0.033 ± 0.021	0.053 ± 0.016	0.027 ± 0.010	0.051 ± 0.014	0.030 ± 0.016	0.052 ± 0.014	0.025	0.025
			0.037 ± 0.014		0.044 ± 0.011		0.040 ± 0.013		0.005	
	c_3	110	0.045 ± 0.027	0.083 ± 0.029	0.055 ± 0.017	0.075 ± 0.023	0.050 ± 0.022	0.079 ± 0.026	0.062	0.067
			0.064 ± 0.033		0.048 ± 0.027		0.056 ± 0.030		0.025	
	c_4	149	0.050 ± 0.005	0.109 ± 0.054	0.057 ± 0.033	0.070 ± 0.028	0.053 ± 0.023	0.089 ± 0.046	0.062	0.067
			0.090 ± 0.067		0.035 ± 0.010		0.063 ± 0.054		0.025	

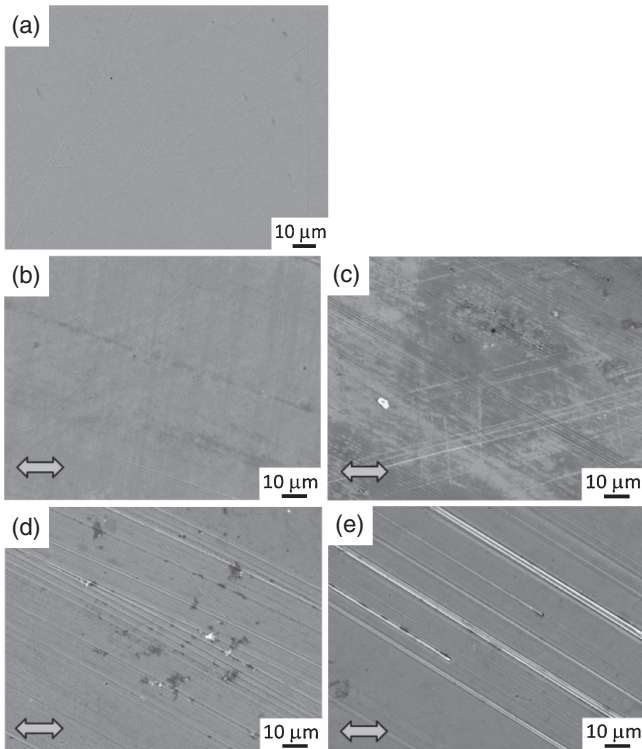


Fig. 4 (a) the initial SEM micrograph of new cup apex surfaces; typical SEM micrograph of worn cup apex surfaces of (b) c_1 , (c) c_2 , (d) c_3 , (e) c_4 ; gray arrows indicate the flex/ext sliding direction.

worn surfaces of c_1 ; whereas the radial clearance gets larger, wider and deeper plowing grooves due to plastic deformation were observed. Especially for c_3 and c_4 , obvious built-up edges generated at both sides of the grooves, and they tended to overlay each other or to be partially knocked off by the repeated contact and rubbing of the wear process, resulting in

the formation of debris.⁴³⁾ From the above results, we deduce that the abrasive wear of MOM bearings with larger radial clearances is more pronounced in the wear test under HBSS lubricated environment.

4.3 Volume loss and wear rate

Figure 5(a) shows the total volume loss V of the bearings with different radial clearances as a function of cyclic number. All the bearings were characterized by a linear increase in the volume loss with cyclic number. No obvious “stable” stage was observed throughout the 1.0 Mc wear tests. Volume loss was affected by radial clearance as the former feature increased gradually with the latter feature from c_1 to c_4 . Table 2 shows the volume wear rate k calculated from the volume loss, and the results revealed that the volume wear rate for c_1 was $\sim 55\%$ lower than that for c_4 . Furthermore, the volume wear rate of the cup was greater than that of the head for a given radial clearance.

Figure 5(b) shows the linear increase of the radial clearance with the number of cycles by regression analysis. The result in Fig. 5(b) and Table 1 indicate that smaller clearances tended to enlarge faster than the larger ones; the radial clearance increments Δc for c_1 and c_4 were 6.5 and 2.7, respectively. As respect to shape, heads showed excellent conforming surfaces during wear than cups. Figure 6(a), (b) shows that all cups and the heads of c_3 and c_4 showed increasing sphericities, whereas heads of c_1 and c_2 showed decreasing sphericities. Except for the less sliding distance, the self-repairing phenomenon may be another reason for the relative less volume wear rate of the heads. Since radial clearance affect sphericity and surface roughness, more or less, bearings with different clearances operated with different contact and lubrication status and had different wear behaviors.

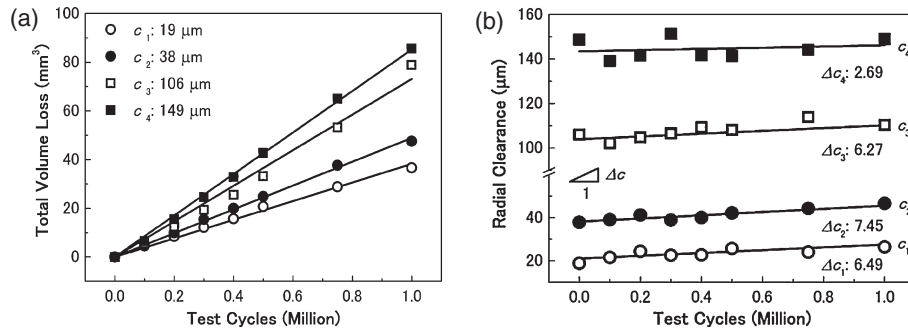


Fig. 5 (a) total volume loss (mm³) for c_1 – c_4 during 1.0 Mc wear test; (b) Evolution of radial clearances for c_1 – c_4 during 1.0 Mc wear test.

Table 2 Wear rate (mm³/Mc) for c_1 – c_4 .

Radial clearance (μm)		Cup wear rate (mm ³ /Mc)	<i>R</i> ²	Head wear rate (mm ³ /Mc)	<i>R</i> ²	Total wear rate (mm ³ /Mc)	<i>R</i> ²
<i>c</i> ₁	19	25.81	0.9951	10.65	0.9962	36.46	0.9959
<i>c</i> ₂	38	26.69	0.9959	21.65	0.9991	48.34	0.9981
<i>c</i> ₃	106	46.48	0.9926	31.45	0.9867	77.93	0.9905
<i>c</i> ₄	149	46.94	0.9994	40.36	0.9977	87.30	0.9992

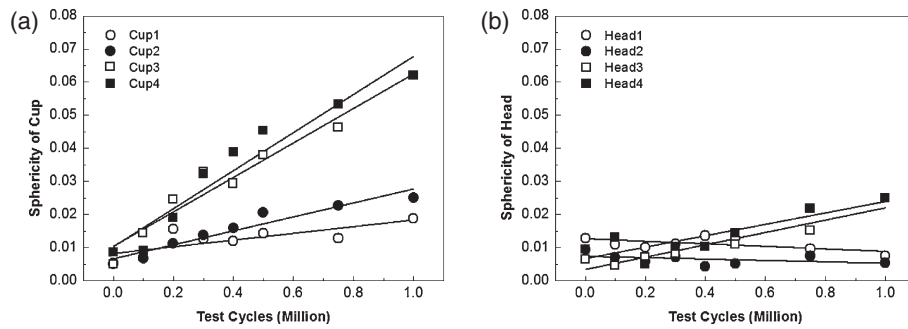


Fig. 6 Spherities of (a) cups and (b) heads for c_1 – c_4 during 1.0 Mc wear test.

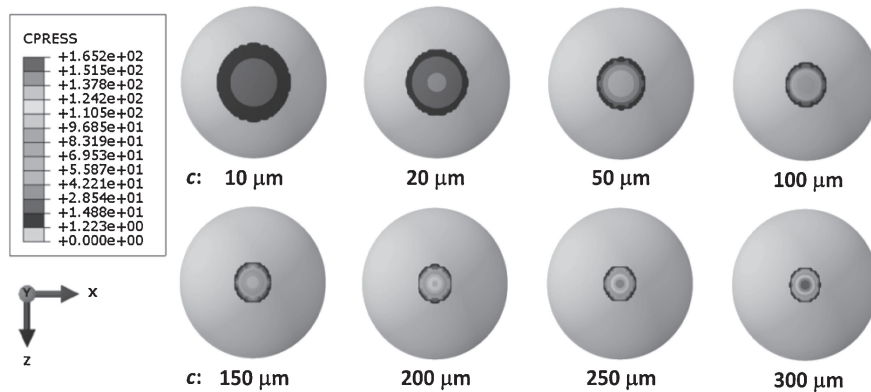


Fig. 7 Contour plots of dry contact pressure (MPa) and apparent contact area at the f 40mm bearing surface of femoral heads with different radial clearances.

4.4 Contact analysis

Figure 7 shows contour plots of the dry contact pressure distribution on the surfaces of the bearings with different radial clearances. A comparison of the plots reveals typical hard-hard contacts, which axisymmetrically distribute in the circumferential direction, and gradually decrease from the contact center in the radial direction. The latter feature in cross section of the samples is shown in Fig. 8, in which an

obvious decrease in the apparent contact area and increase in the maximum contact pressure with increasing radial clearance can be observed.

Figure 9(a) is a comparison of the maximum contact pressures and apparent contact areas of the bearings obtained by ABAQUS (ABA) and the Hertz contact theory (Hertz), respectively. It was observed that the results were in close agreement. The values of the maximum deviation ε of the

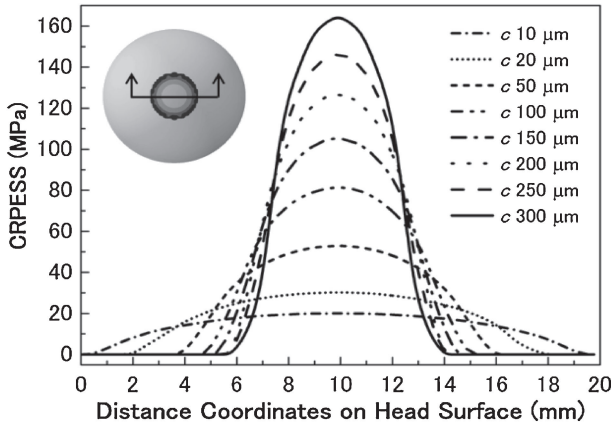


Fig. 8 Comparison of predicted maximum contact pressures in radial direction of MOM bearings with different radial clearances.

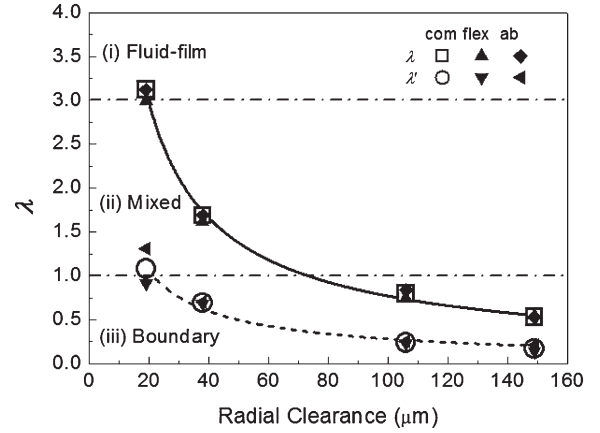


Fig. 10 Evolution of lubrication regime for c_1 – c_4 at the initial and final wear test.

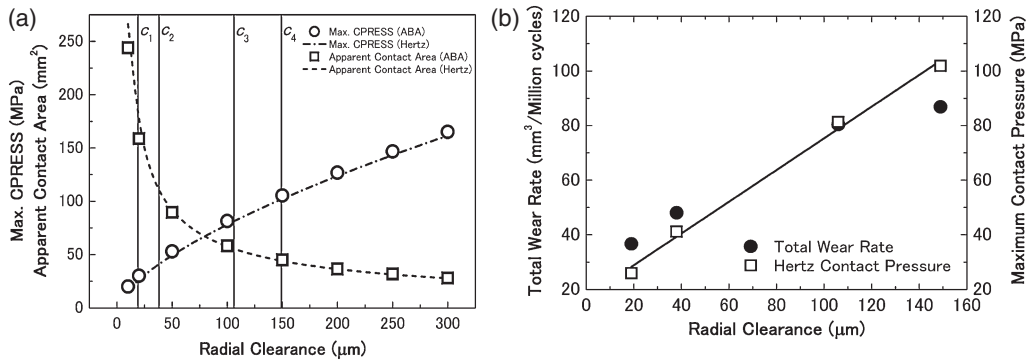


Fig. 9 (a) Comparison of predicted maximum contact pressure and apparent contact area by ABAQUS and Hertz contact theory for MOM bearings with different radial clearances; (b) Comparison of total wear rates and predicted maximum contact pressures (Hertz contact theory) for c_1 – c_4 .

contact pressures and areas obtained by the two methods were $\sim 15.7\%$ and 8.4% , respectively, for bearings with a radial clearance of $10\ \mu\text{m}$. The total wear rates of the bearings after the 1.0Mc wear tests are shown as a function of the radial clearance in Fig. 9(b), together with the contact pressure predicted by the Hertz contact theory. Both the volume wear rate and the contact pressure were observed to vary linearly with the radial clearance.

4.5 Film thickness

The lubrication regime of the bearings can be divided into (i) fluid-film lubrication, as $\lambda \geq 3$, (ii) mixed lubrication, as $1 < \lambda < 3$, and (iii) boundary lubrication, as $\lambda \leq 1$, where the dimensionless parameter λ is defined as the ratio of the minimum film thickness to the composite roughness Ra :²⁷⁾

$$\lambda = h_{\min}/Ra. \quad (6)$$

The lubrication is a transient phenomenon as the load varies with the cyclic time, thus the film thickness would also vary between the steady state values determined from the highest and lowest load in the gait cycle. A simplified transient elastohydrodynamic lubrication model of central film thickness, h_c , calculation was proposed by Chan,⁴⁴⁾ Hamrock³³⁾ and Medley *et al.*:⁴⁵⁾

$$h_c = 5.083 \frac{(\eta u)^{0.660} R^{0.767}}{(E')^{0.447} w^{0.213}} \quad (7)$$

Table 3 Mechanical properties of the material, geometrical parameters of the bearings, and experimental conditions for lubrication calculation.

E (GPa)	u	R_h (mm)	h (Pa s)	w (rad/s)	w (N)
230	0.3	20	0.001	2	1365

where η is the viscosity of the lubricant (Pa s), w is the applied load (N), constant entrainment velocity u , is approximated with the only angular velocity ω (rad/s) in the flex/ext direction:

$$u = \omega R/2 \quad (8)$$

Only pure squeeze action between flat, rigid and parallel surfaces was considered in the model, assuming one surface is an axially-oriented disc of contact radius and the other is a plane of infinite extent. As the difference between h_c of the cyclic steady model and h_c calculated by eq. (7) with the average load over the gait cycle was estimated to be less than 3% ,^{44,46)} the average load of 1365 N ($\sim 2\text{ BW}$ body weight) was used in the present model for h_c prediction. The HBSS was assumed to behave as an isoviscous, Newtonian fluid, with similar viscosity of water, $\sim 0.001\text{ Pa s}$ at 37°C and 0.1 MPa .^{45,47,48)} The parameters used for the calculation are summarized in Table 3.

Figure 10 shows the evolution of the lubrication conditions for c_1 – c_4 at the initial and final wear test, which could be revealed by the λ and λ' values. It was found that, in the initial

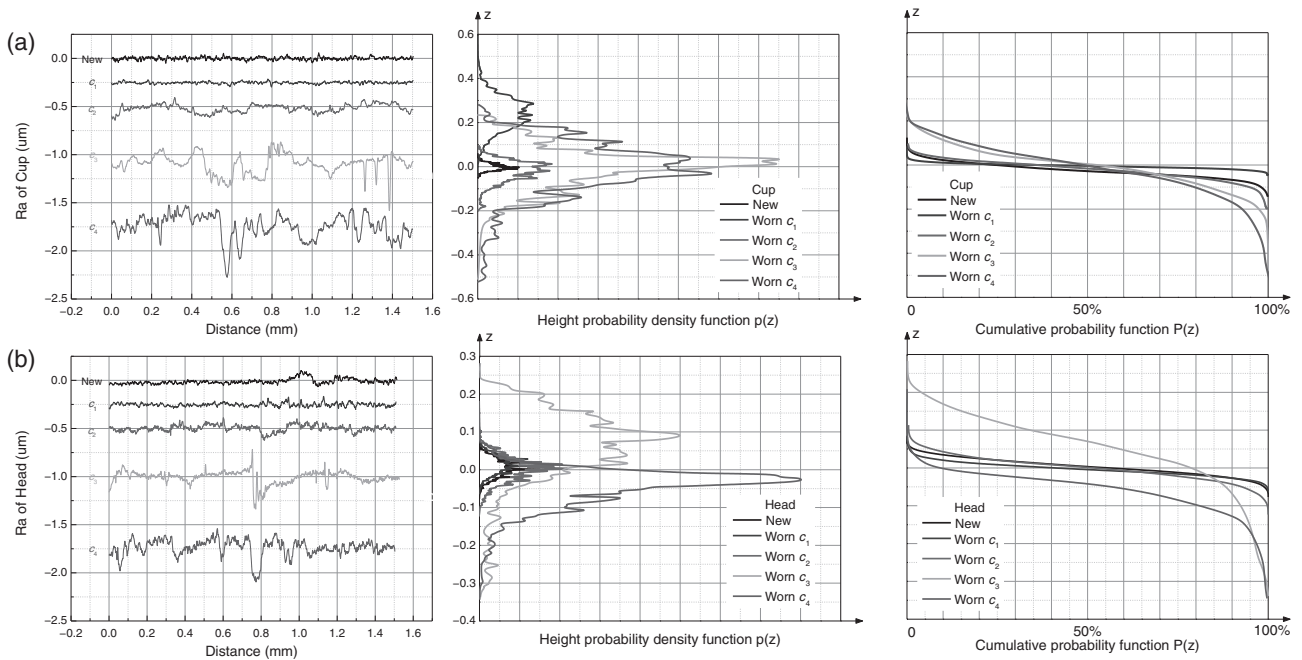


Fig. 11 Height probability density function and the Abbott-Firestone curve, obtained in the same position on the surface of the (a) femoral heads and (b) cups.

stage of the wear tests c_1 operated under fluid-film lubrication, c_2 under mixed lubrication, whereas c_3 and c_4 under boundary lubrication. During the wear tests, c_1 and c_2 evolved into mixed and boundary lubrication, respectively; c_3 and c_4 kept the boundary lubrication. However, λ' for the bearings decreased ~ 3 fold as wear processed, with a similar increase in composite Ra we speculate that deteriorated surface roughness led to lubrication deterioration predominantly.

5. Discussion

The results showed that the radial clearance could affect contact pressure and the lubrication, thus affect the wear behavior of MOM bearings.

Theoretically, a head and cup with identical radii would mate without clearance and function with maximum contact area and minimum contact pressure.⁴⁹⁾ Jagatia, Mak, and Meng^{35,50,51)} similarly observed that a small clearance reduced the contact pressures of MOM, COC, and COM bearings, respectively, similar to the results in Fig. 7 and Fig. 8. The predicted apparent contact area A of MOM bearings contracted by 88.6%, and the correspondingly maximum contact pressure increased by 87.9% when the radial clearances of MOM bearings increase from 10 to 300 μm .

In the actual test, the real contact area is also related to surface roughness,⁵²⁾ assuming that the initial micro-asperity contacts are elastic^{53–55)}

$$A_1 \propto n^{\frac{1}{3}} \cdot \left(\frac{w}{E'} \right)^{\frac{2}{3}}, \quad (9)$$

where n is the number of contacting asperities. The volume loss V is related to both the real contact area and the sliding distance by⁵⁶⁾

$$V = KA_1s, \quad (10)$$

where K is the proportionality constant.

The height probability density function and the Abbott-Firestone curve, which is usually applied in surface damage distribution for ground surfaces of bearings,³⁸⁾ were obtained on the apex surfaces of the femoral head and cup in Fig. 11(a) and (b), respectively. The curves of worn bearings with different radial clearances, are compared with one typical curve before wear, were superimposed for comparison. These figures show that cup or head before wear, the surfaces exhibit symmetrical Gaussian height distribution, with low Ra and P-V. The surfaces after wear, most of which are negatively skewed, increased obviously in Ra and P-V in the z direction, especially for bearings with larger radial clearances. That indicates asperities on worn surfaces with larger clearance, are more distinctly protuberant from the reference plane, contact with each other to lead to decreasing real contact area and increase probabilities of detachment of debris from the surface. This is very consistent with the results shown in Fig. 9(b), where a similar trend can be observed in the calculated contact pressure and experimental volume wear rate. The regression analysis revealed that both the total wear rate and the maximum contact pressure of the bearings increased with increasing radial clearance (the values of R^2 were 0.9436 and 0.9924, respectively), although the contact pressure increased 1.5 times faster than the total wear rate.

Regarding the lubrication, a small radial clearance in the range of 10–70 μm is beneficial for an initial thick initial film, and it produces a larger value of λ for bearings with the common machining accuracy.³⁸⁾ It can be clearly observed from Fig. 10 that the initial lubrication regimes of the bearings in the present study were different; those with radial clearances between 18 and 70 μm (for c_1 and c_2) operated under full-fluid and mixed lubrication, whereas those with radial clearances larger than 70 μm (for c_3 and c_4) operated under boundary lubrication. However, it was also observed that smaller clearances increased more rapidly during wear, and the increasing rate for c_1 is 2.4 times that for c_4

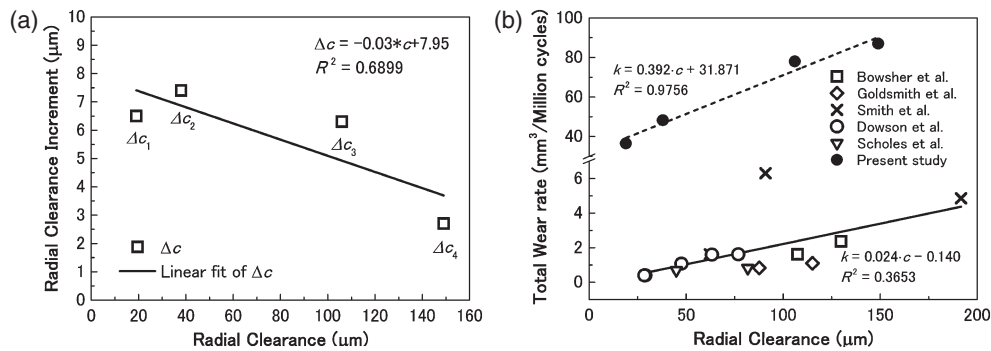


Fig. 12 (a) Radial clearance increase for c_1 – c_4 at 1.0 Mc wear test; (b) Comparison of wear rates of MOM bearings with different equivalent radii in bovine serum and HBSS.

(Fig. 5(b)). Figure 12(a) shows the different radial clearances increment Δc of bearings at 1.0 Mc wear test by regression analysis, following the function of:

$$\Delta c = -0.03c + 7.95. \quad (11)$$

Consequently, all the HBSS-lubricated bearings operated under boundary lubrication and enlarged clearance at the end of the 1.0 Mc test. Bearings designed with proper clearance operated under a full-fluid lubrication might develop into some status else, just as the calculated result for c_1 . Assuming general increment of radial clearance of MOM bearings is in accordance with eq. (11), an initial radial clearance of $10 \mu\text{m}$ would be theoretically possible for a full-fluid lubrication regime at 1 Mc wear, as the predicted minimum thickness and value of λ were $0.076 \mu\text{m}$ and 3.03 , respectively. However, an extremely small clearance would create mating problems such as equatorial contact, which would lead to high torque, clamping, and loosening of the bearings.^{43,57,58} So, the predictions indicate that for the run-in stage of the *in vitro* wear tests, most of the large-head MOM bearings with small clearances operated under mixed-to-boundary lubrication in HBSS rather than with full-film lubrication. Among the bearings tested in HBSS in the present study, those with clearances of $\sim 20 \mu\text{m}$ were observed to have the best wear resistance, although they did not really operate under full-fluid lubrication. Till date, manufacturing accuracy limitations have made it difficult to determine the exact threshold between equatorial contact and uniform contact.

The results of the present study showed that the volume wear rates of the HBSS-lubricated MOM bearings were strongly affected by the radial clearance. Coincidentally, previous studies by Dowson *et al.* also revealed that *in vitro* MOM bearings with small and well controlled clearances were characterized by lower contact pressure and wear rates in hip joint simulator wear tests.^{19,20,25,58,59} Radial clearance-volume wear rate function of Fig. 12(b), in which all the bearings were normalized to $\phi 40 \text{ mm}$ diameters, shows a similar linear increase of wear rate with increased radial clearance. The wear rates of the HBSS-lubricated bearings are much higher than those of the bovine serum-lubricated bearings, even for the bearings with same radial clearance. So the results obtained in HBSS lubricant can't be clinical relevance because the absence of protein and lack of viscosity of HBSS. However, it is notable that certain point of total wear rate deviates from the fitting line considerably for bovine-serum-lubricated wear test, as protein precipitation

and degradation cause an unstable fluctuation for the wear behavior of MOM bearings. Pure clearance effect on bearing wear behavior can be revealed better in HBSS lubricated environment by neglecting some complex lubricant environment described as low-viscosity, or more protein-contained periprosthetic synovia of osteoarthritis patients.⁶⁰ More wear tests are required for the further studies lasted longer for the stable stage observation.

6. Conclusions

We analyzed the dry contact mechanics and lubrication regimes of HBSS-lubricated MOM bearings with different radial clearances to evaluate their wear behavior. The volume wear rate and contact pressure were found to increase with the radial clearance of the bearings. Analyses of the lubrication regime using the Hamrock and Dowson formula revealed that all the MOM bearings operated under boundary lubrication over 1.0 Mc wear tests. However, bearings with a radial clearance of $20 \mu\text{m}$ exhibited less abrasive wear and lower volume wear rate, which was due to their lower contact pressure and better lubrication. The observed volume wear rates and contact mechanics indicated that the radial clearance significantly affected the contact and lubrication and that a small clearance improved the wear resistance of lubricated MOM bearings.

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Nomenclature

- A : apparent contact area (mm²)
 A_i : real contact area (mm²)
 c : radial clearance (μm)
 Δc : radial clearance increment (μm)
 E : elastic modulus (MPa)
 E' : equivalent elastic modulus (MPa)
 η : viscosity of lubricant (Pa s)
 h_c : central film thickness (nm)
 K : proportionality constant
 k : volume wear rate (mm³/Mc)
 k_c : volume wear rate of cup (mm³/Mc)
 k_h : volume wear rate of head (mm³/Mc)
 λ : dimensionless ratio
 n : number of contact asperities
 $P_{\max}$: maximum contact pressure (MPa)
 r : contact radius (mm)
 R : equivalent radius for the ball-on-plane model (mm)
 Ra : composite roughness (nm)
 Ra_c : arithmetic average roughness of cup (nm)
 Ra_h : arithmetic average roughness of head (nm)
 R_c : radius of cup (mm)
 R_h : radius of femoral head (mm)
 s : sliding distance (mm)
 u : entraining velocity (mm/s)
 ν : Poisson's ratio
 V : total volume loss (mm³)
 ω : angular velocity in Flex/Ext direction (rad/s)
 w : applied load (N)