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AN INVESTIGATION OF SCUFFING FAILURE IN ANGULAR CONTACT BALL-BEARINGS

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Summary

There is little agreement in the literature on what scuffing is, what causes it, or what appearance is. From a strictly mechanical point of view, the estimation of bearing's scuffing risk supposes to settle the interdependence between the Kinematics, dynamic, lubrication and friction phenomena realised in bearing's rolling contacts subjected to an important sliding.

Our research focused on a specified angular contact ball-bearing realised an experimental and theoretical modelisation both on discs test specimen and ball bearings.

Some scuffing tests have been carried out on the high-speed twin-disk machine available at LMC facilities. Another series of scuffing tests were applied on the 7206 angular contact ball bearing included in an original test rig. An energetically criterion to estimate scuffing limits has been proposed.

The proposed scuffing criterion is sufficiently effective and reliable for designers and producers in order to offer a covering tool related to scuffing risk in angular contact ball bearings.

***Keywords**: Scuffing, Angular contact ball bearing, Energetically criterion, Test device.*

1. INTRODUCTION

There is little agreement in the literature on what scuffing is, what causes it, or what appearance is.

This has resulted in significant confusion because various researchers have point out very different physical processes into the general heading of scuffing.

Scuffing is a complex phenomenon of severe adhesive wear generated under particular combinations including contact pressure, lubrication, speed and friction.

Scuffing is described as failure of lubricant films, desorption of active chemical species, destruction of oxides, large plastic deformation of the surface, unstable growth of contact conjunctions, accumulation of wear debris and bulk subsurface failure.

There are hydrodynamic, temperature and chemical approaches of scuffing phenomenon [1-4].

Scuffing involves the sudden collapse of the lubricant film and is generally regarded as resulting from thermal phenomena. At this temperature, the breakdown film with local welding or adhesion of the contacting surfaces can appear. As result of local adhesion, the friction increases and causes a still higher temperature, then the process can become catastrophic.

Scuffing can take place either at low sliding velocity and high normal pressure, by softening and melting a thick surface layer of the contacting bodies, or at high sliding velocities and low normal pressure, by a process of abrasion and adhesion. Also, occasionally abnormal operation causes premature bearing failure, that is usually micro-scuffing and/or scuffing [5].

Therefore, details of the circumstances and the way in which this type of distress occur is still the subject of debate. Among the multiple models, none is sufficiently effective and reliable.

The diversity of running conditions and, implicitly, of their typical influences on scuffing failure in bearings rolling contacts is a limiting factor for researcher's options to a global study.

From a strictly mechanical point of view, to estimate the bearing's scuffing risk means to settle the interdependence between the Kinematics, dynamic, lubrication and friction phenomena realised in bearing's rolling contacts subjected to an important sliding.

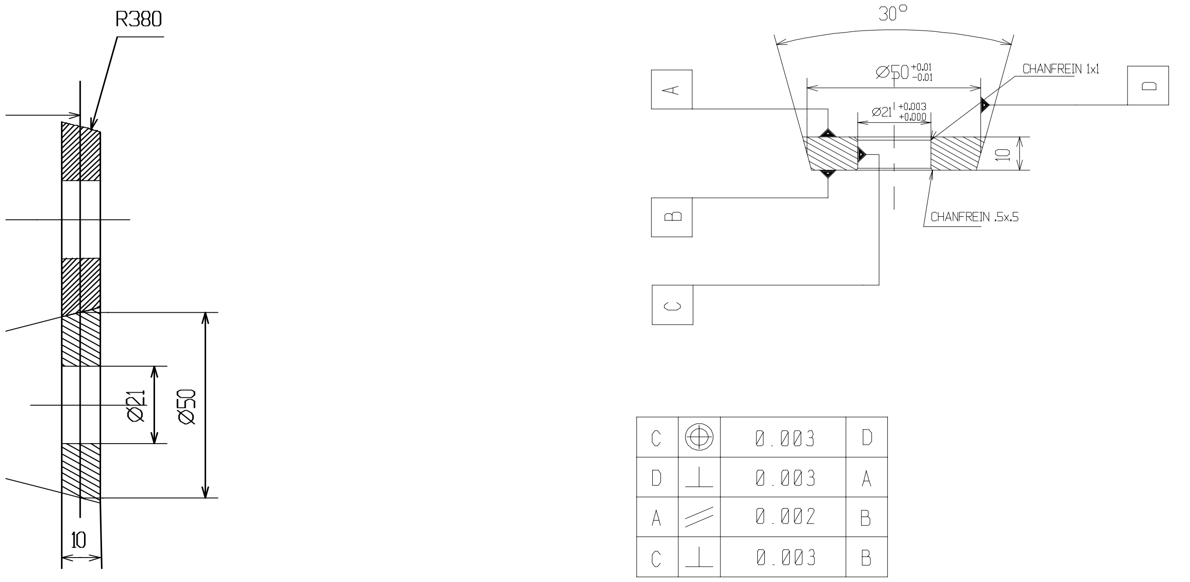


Fig.1. Disk shapes in simulative testing

This paper continues our research [6], [7], focused on a specified angular contact ball-bearing (7206C). It has been established an experimental and theoretical modelisation both on discs test specimen and ball-bearings. It has been resulted an algorithm and computer code to estimate scuffing limits in angular contact ball bearings.

2. EXPERIMENTAL APPROACH

Scuffing in angular contact ball bearing was experimentally studied also on specimens that on specified ball bearing (7206C).

2.1 Investigation on specimens

Some scuffing tests have been carried out on the high-speed twin-disk machine described in [5], available at LMC-INSA Lyon facilities.

For this simulative tribotesting, the simulation criteria reefer to material, geometrical characteristics and the lubrication regime of the practical system, which is 7206 angular ball-bearing.

2.1.1 Operating conditions

♦ The contacting disks, presented in figure 1, were arched performed in order to reproduce the ball-inner ring contact from 7206 ball bearing (the ellipticity factor k = 8.964).

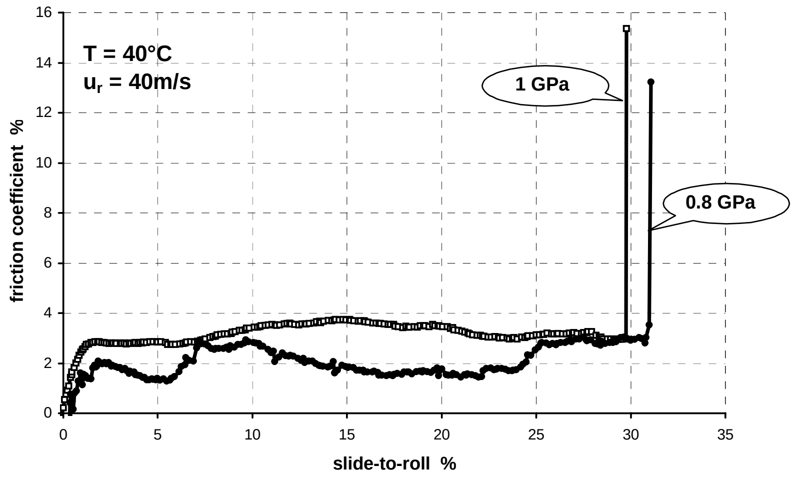
♦ The lubricant used in these tests is a mineral oil, ELF 154 NS. Its rheological behavior is estimated from experimental traction curves obtained on the twin-disk machine [7].

♦ In the experimental procedure the scuffing limit is reached by increasing progressively the sliding speed, other operating conditions such normal load and the rolling speed being kept constant. The evidence of disk scuffing is a sudden increase of friction force, which stops automatically the machine at an earlier stage of damage process.

♦ Scuffing limits for mean rolling speed of 40m/s, three maximum hertzian pressures (0.8, 1.0 and 1.2 GPa) and three oil feed temperatures (40, 70 and 100 ^{0}C) have been studied on the twin-disk machine.

♦ A total of 16 tests have been carried out.

Figure 2 presents some results on scuffing limits.



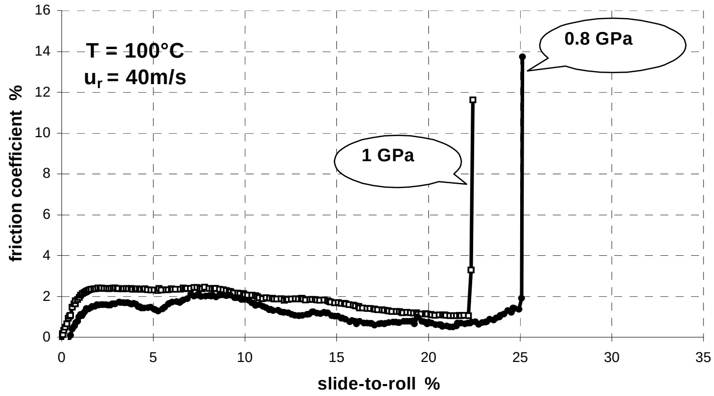


Fig 2. Scuffing limits on specimens

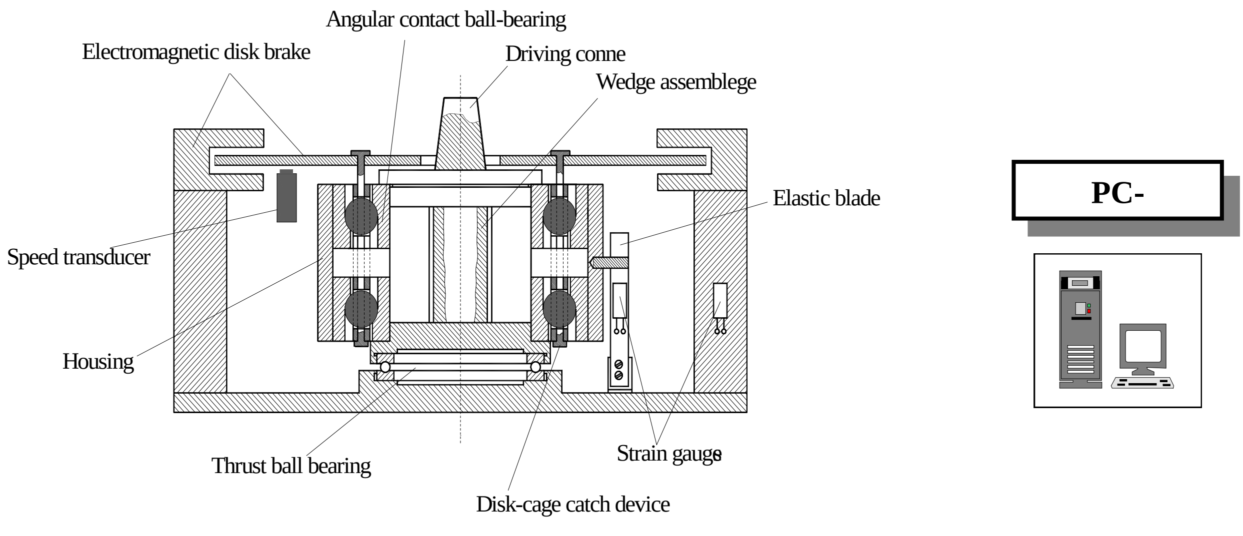


Fig.3 Axial section through the scuffing test rig

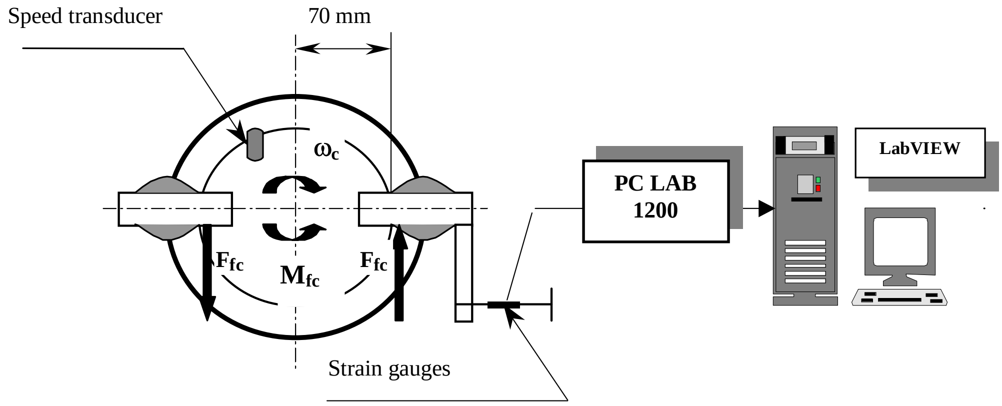


Fig.4 Electromagnetic brake from the scuffing test rig

2.1.2 Observations

♦ Scuffing sliding speed has critical value up to 9 m/s (for testing specified conditions and a complete EHL regime).

♦ Increasing contact pressure and also, oil temperature increase, leads to decreasing the scuffing limit for sliding speed.

2.2 Investigation on ball bearing

Scuffing tests were performed on 7206 angular contact ball bearing included in an original test, presented in figure 3. The test rig is adapted on 4-ball machine from our laboratory.

The operating speed on the test rig was moderate (max. 2800 rpm), so the ball-inner ring sliding speed too. As the bearing scuffing risk is deeply correlated to high sliding speed (typical to high-speed ball bearings) we realised these conditions by the cage's braking.

During the running process, an electro-magnetic brake (figure 4) fixed on the cages of bearings system, through disk-cage device, realized the cage’s braking.

The speed transducers quantify the braked cage speed ω_c and the braking torque, implicitly the force F_c between ball and cage. With these two known parameters, the correlation between experimental and theoretical investigations can be settled.

The test rig also allows measuring the friction torque by means of strain gauges. The sudden increase of the friction force (as in experimental tests on disks) indicates the scuffing occurrence.

A general view of the test rig presents figure 5.

Scuffing limits obtained presents figure 6.

2.2.1 Operating conditions

♦ Tested bearings were subjected to pure axial loading.

♦ A Raynger MX4 infrared thermometer has measured average bearing temperature.

♦ An electric resistance measurement device from our laboratory facilities quantified the oil film thickness. It has been imposed a poor lubrication, in order to realise a boundary regime for scuffing occurrence.

♦ The operating conditions and the lubricant were the same for the scuffing tests on the bearing test rig as on specimens, except the mean rolling speed.

2.2.2. Observations

♦ Owing to the cage’s braking, the ball-race sliding speeds were substantially increased (~ 3 m/s) and the scuffing appeared.

♦ The boundary lubrication regime potentates the scuffing risk

3. THEORETICAL SCUFFING APPROACH

The scuffing approaches analysis and our original experiments on bearing's contacts have been detached the idea that, any scuffing mechanism considered, there is an energetically unbalance in the rolling contact. This unbalance generates disruptions in lubrication conditions and the scuffing risk appears.

So, we consider the most adequate model to estimate the scuffing limits an energetically one.

Therefore, according to lubrication regime, the amount of heat is related to the oil film parameter, λ, and can be estimated as follows:

for complete EHL regime ($\lambda \geq 3$):

$$q_{fluid} = F_l \cdot \dot{\gamma} \qquad (1)$$

where:

q_{fluid} the contact energy dissipation

$\dot{\gamma} = V / h_c$ lubricant shear rate

$F_l = \int \tau dA = \mu \cdot p$ shear friction force

h_c central film thickness.

p average contact pressure

V ball-race contact sliding speed.

This is measurable for specimens experimental simulation, but for angular contact ball bearings tests it must be calculated.

(1) for boundary lubrication regime (λ = 1…3):

$$q_{mixt} = \mu_{ech} \cdot p \cdot V \qquad (2)$$

where q_{mixt} represents the energy dissipated as consequence of the two friction components combined action (fluid and solid).

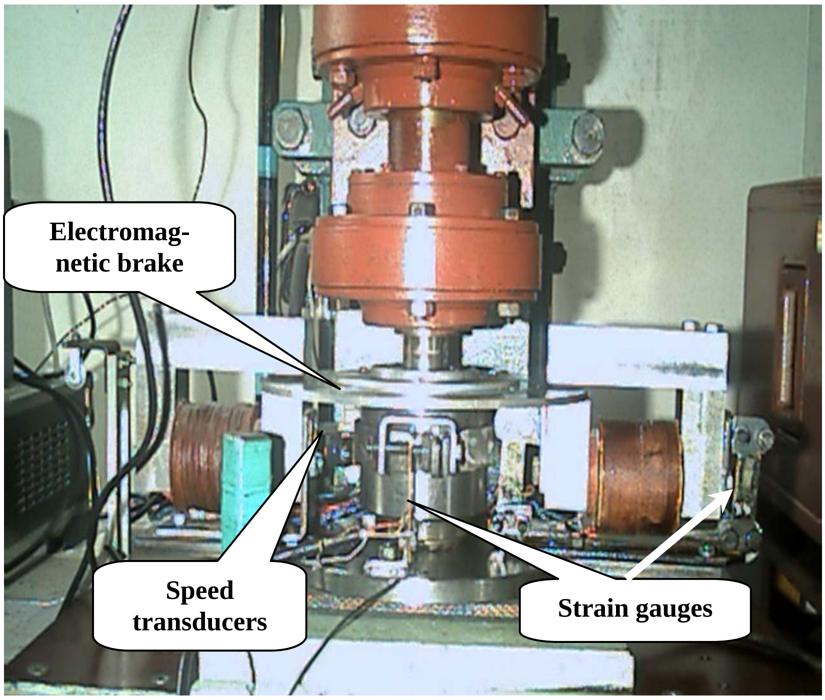


Fig.5. Scuffing test rig

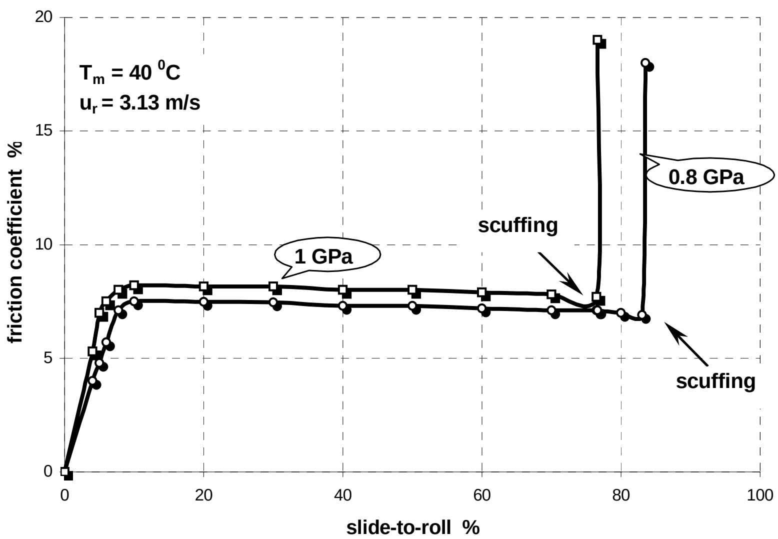


Fig.6.Scuffing limits in angular contact ball

(2) for direct contact (λ< 1):

$$q_a = \mu_a \cdot p \cdot V \quad (3)$$

where q_a represents the energy dissipated on the asperities, when there is no lubrication.

The sliding friction is related to the degree of film formation or portion of load supported by the asperities contact. So, the model consider:

♦ μ - traction coefficient on ball-race contact for EHL regime, determined as follows:

$$\mu = (A + B\cdot\xi)\cdot\exp(- C\cdot\xi) + D \quad (4)$$

This equation simulates the traction behaviour, taking account both of slide-to-roll ratio ξ and the lubricant rheological characteristics derived from traction curves experimentally determined using a twin-disk machine. The coefficients of the model, A, B, C, D are depending on operating conditions and are also determined from the experimental investigations.

A mineral oil (ELF 154 NS) was used as lubricant. Its rheological parameters (shear modulus G and shear stress τ_0), included in the general model for estimating scuffing risk, were predicted, pressure and temperature dependent, as described in [2].

♦ μ_{ech} -friction coefficient on ball-race contact for boundary lubrication regime, considering both the fluid friction (F_l) and solid friction on the asperities (F_{asp}) under the normal load Q:

$$\mu_{ech} = \frac{F_l + F_{asp}}{Q} \quad (5)$$

♦ μ_a - friction coefficient on ball-race contact asperities, determined by a methodology described in [8], with B_0 and C_0 experimentally estimated:

$$\mu_a = \frac{F_{asp}}{Q} = 0.18 \cdot \exp (- B_0 \cdot \lambda^{Co}) \quad (6)$$

It can be concluded that the following relation expresses the energy dissipated:

$$q = \begin{cases} \mu \cdot p \cdot V / h_c & \lambda \geq 3 \\ \mu_{ech} \cdot p \cdot V & 1 \leq \lambda \prec 3 \\ \mu_a \cdot p \cdot V & \lambda \prec 1 \end{cases} \quad \textbf{(7)}$$

Therefore, a dependence between contact pressure, sliding speed and critical level of energy dissipated can be settle for any operating conditions in a ball bearing which contact geometry and lubricant properties are known.

4. RESULTS

Figure 7 illustrates the comparison between scuffing limits obtained on both tests (specimens and ball bearings), using the scuffing criterion on experimental results.

These results lead to establishment of unique scuffing limits of $1.5\cdot10^8$ $W\cdot m^{-2}$ (in terms of energy dissipation) in a given ball bearing (7206C), for imposed running conditions.

Interpolation of energy dissipated values offers the following expression for the scuffing limits in angular contact ball ***bearing 7206:***

$$p \cdot V^{0.83} \cdot \mu^{0.94} = 1.5\cdot10^8 \quad (8)$$

or

$$\mu \cdot p \cdot V^{0.8} = \text{constant} \quad (9)$$

This relation reflects the contact behavior concerning scuffing limits in a given ball bearing for any operating conditions.

Figure 8 expresses the scuffing limits calculated by means of the relation (9).

5. CONCLUSIONS

⇒ Scuffing limits are settled both on specimens operating to high speed and complete EHL regime (λ > 3), and on ball bearings subjected to moderate running speed and gross sliding in boundary lubrication regime ($\lambda \approx 2$)

⇒ Increasing contact pressure and also, oil temperature increase, leads to decreasing the scuffing limit for sliding speed.

⇒ Sliding speeds enough high to initiate scuffing phenomenon were obtained by cage's braking, despite moderate rolling speed performed on the test rig.

⇒ Corroboration between all these experimental results allows a scuffing criterion to estimate this type of damage in angular contact ball bearing (7206C). Scuffing areas corresponds to scuffing criterion $\mu \cdot p \cdot V^{0.8}$

⇒ Sliding speed and contact pressure with scuffing risk were found coexisting in an anthagonic dependence, in good agreement with literature [9].

⇒ The scuffing risk is hardly tied of the possible transition in bearing’s running from EHL regime to boundary lubrication. The sliding friction is related to the degree of film formation or portion of load supported by the asperities contact.

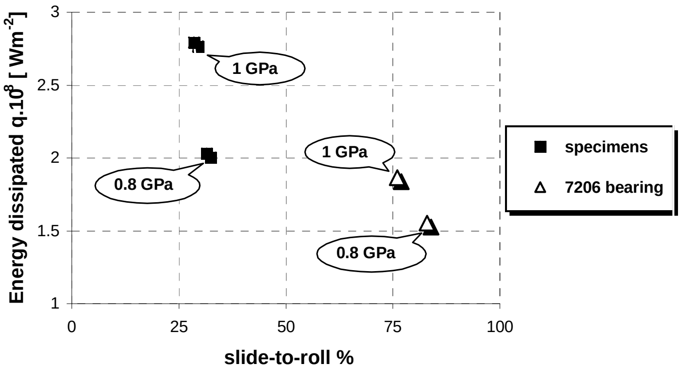


Fig.7 Critical energy dissipation related to scuffing (T_{oil} = 40 ^{0}C)

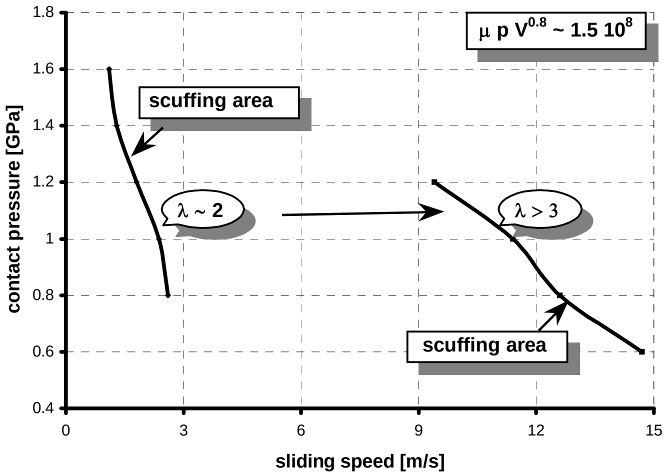


Fig.8 Scuffing limits in angular contact ball bearing 7206

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