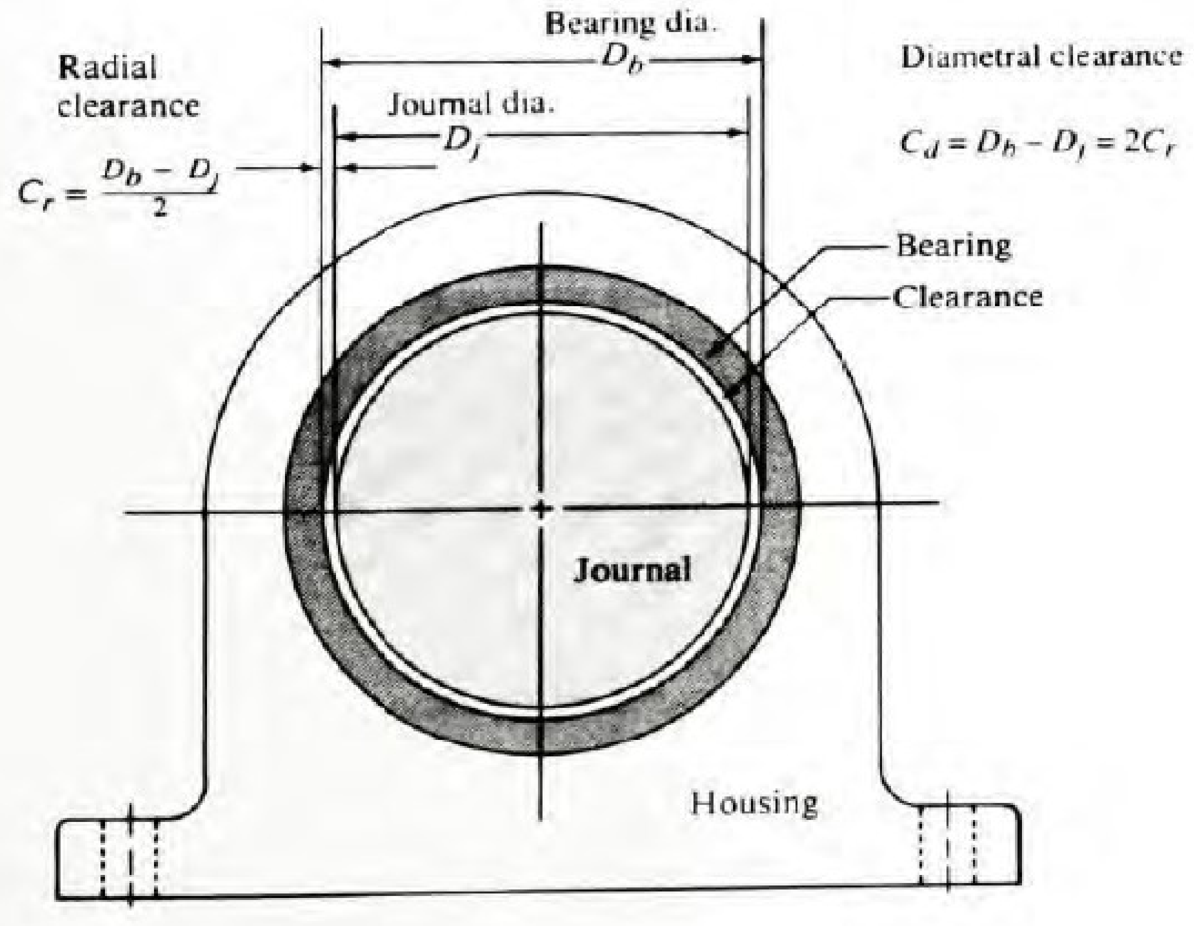
Machine Design –II (MEC -602)

- **CO1: Analyze the stress and strain on mechanical components.**
- **CO2: Demonstrate knowledge of basic machine elements used in machine design.**
- **Design machine elements to perform functions in order to obtain desired objectives under various operating conditions.**

Plain Surface Bearing Geometry



Plain Surface Bearing Short Description

Bearing Type	Description	Friction	Bearing Stiffness	Velocity	Life Span
Plain Bearing	Rubbing surfaces, usually with lubricant; some bearings use pumped lubrication and behave similarly to fluid bearings.	Depends on materials and construction e.g., PTFE has coefficient of friction ~0.05-0.35, depending upon fillers added	Good, provided wear is low, but some slack is normally present	Low to very high	Low to very high - depends upon application and lubrication

Lubrication Mechanisms

A given bearing system can operate with any of three types of lubrication:

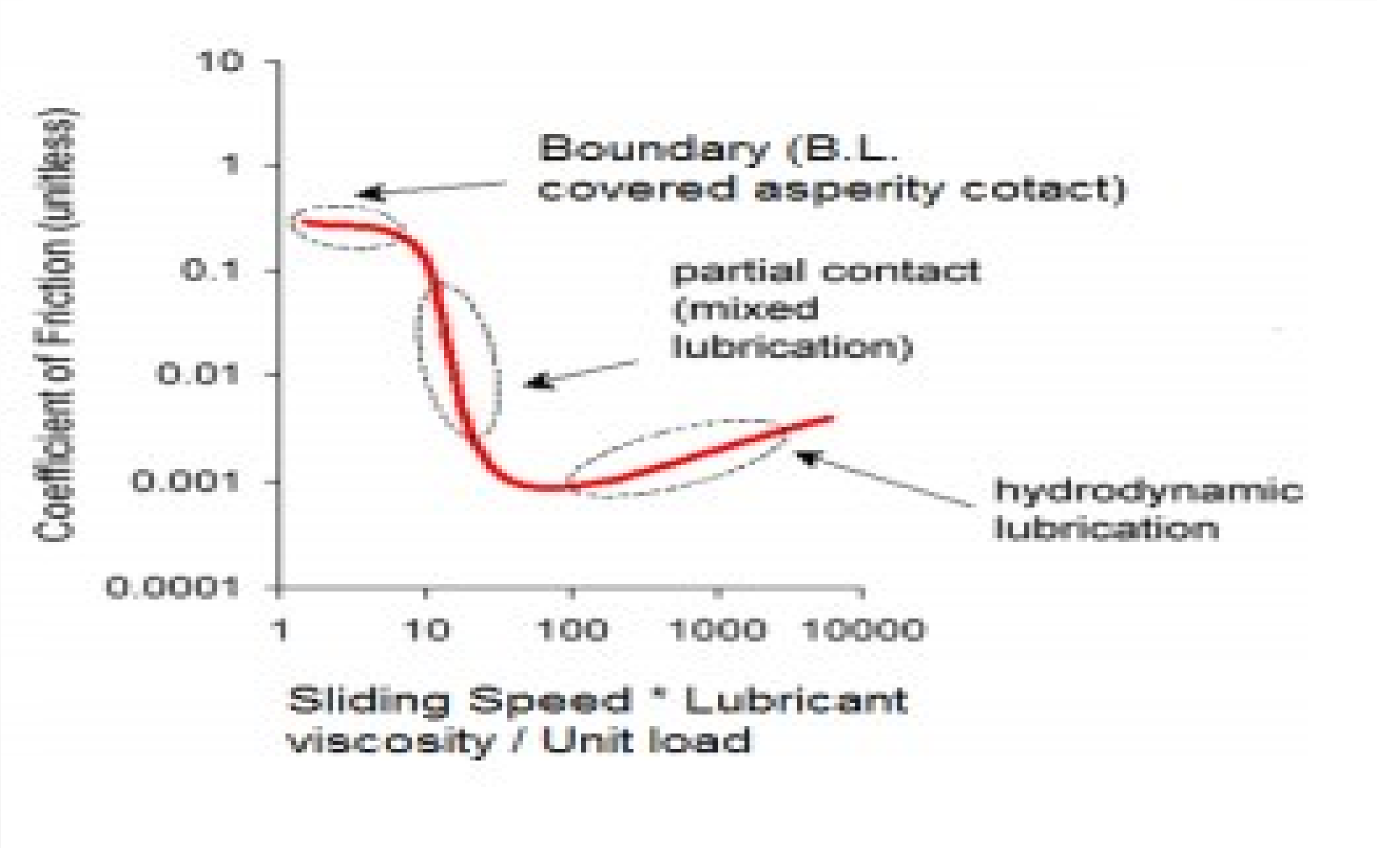
- ***Boundary lubrication:***

- ***Mixed-film lubrication:***

- ***Full-film lubrication:***

STRIBECK CURVE

Bearing performance and types of lubrication related to the bearing parameter, $\mu n/p$



The Bearing Design Task

Design of a plain surface bearing demands :

- Whether full-film hydrodynamic lubrication can be achieved with its advantages of low friction and long life. Or will the shaft operate in the bearing with boundary lubrication?
- What materials will the bearing and the journal be made from?
- What dimensions will be specified?
- What lubricant should be used?

Bearing Requirements

- Magnitude, direction, and degree of variation of the radial load.
- Magnitude and direction of the thrust load, if any.
- Rotational speed of the journal (shaft).
- Frequency of starts and stops, and duration of idle periods.
- Magnitude of the load when the system is stopped and when it is started.
- Life expectancy of the bearing system.
- Environment in which the bearing will operate.

Design Decisions

- Materials for bearing visa vis shaft
- Diameter & tolerances
- clearance
- R_a of Journal and Bearing
- Length of the bearing
- Method of manufacturing the bearing system
- Type of lubricant to be used and the means of supplying it
- Operating temperature of the bearing system and of the lubricant
- Method of maintaining the lubricant cleanliness and temperature

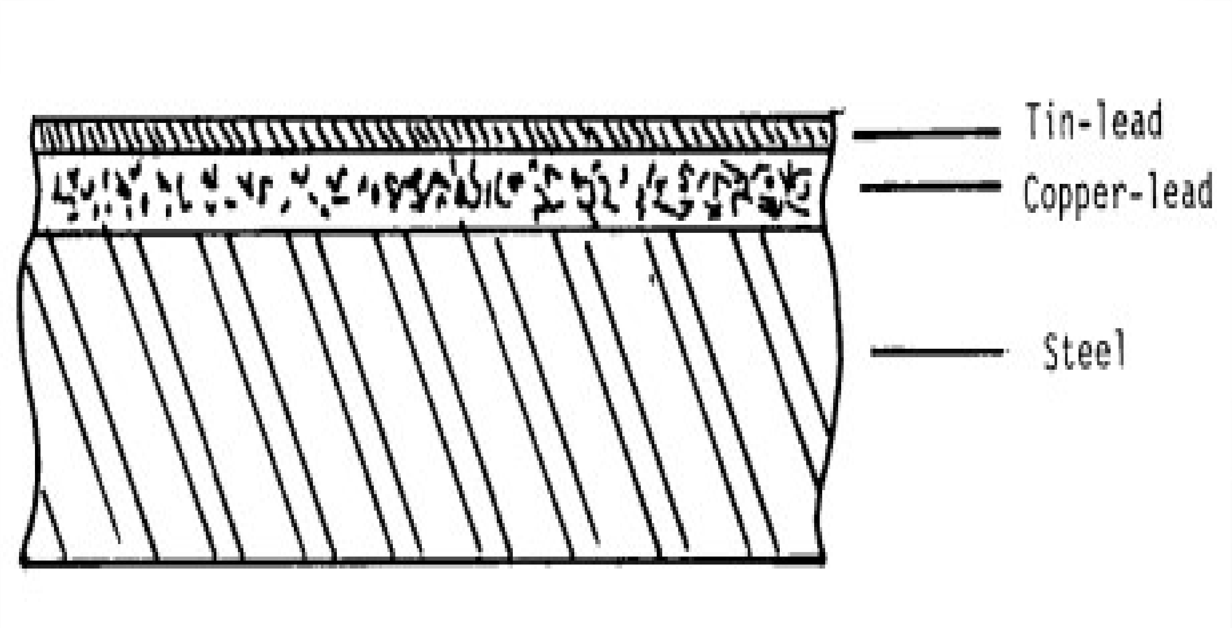
Analyses Required

- Type of lubrication: boundary, mixed-film, full-film
- Coefficient of friction
- Frictional power loss
- Minimum film thickness
- Thermal expansion
- Heat dissipation required and the means of accomplishing it
- Shaft stiffness and slope of the shaft in the bearing

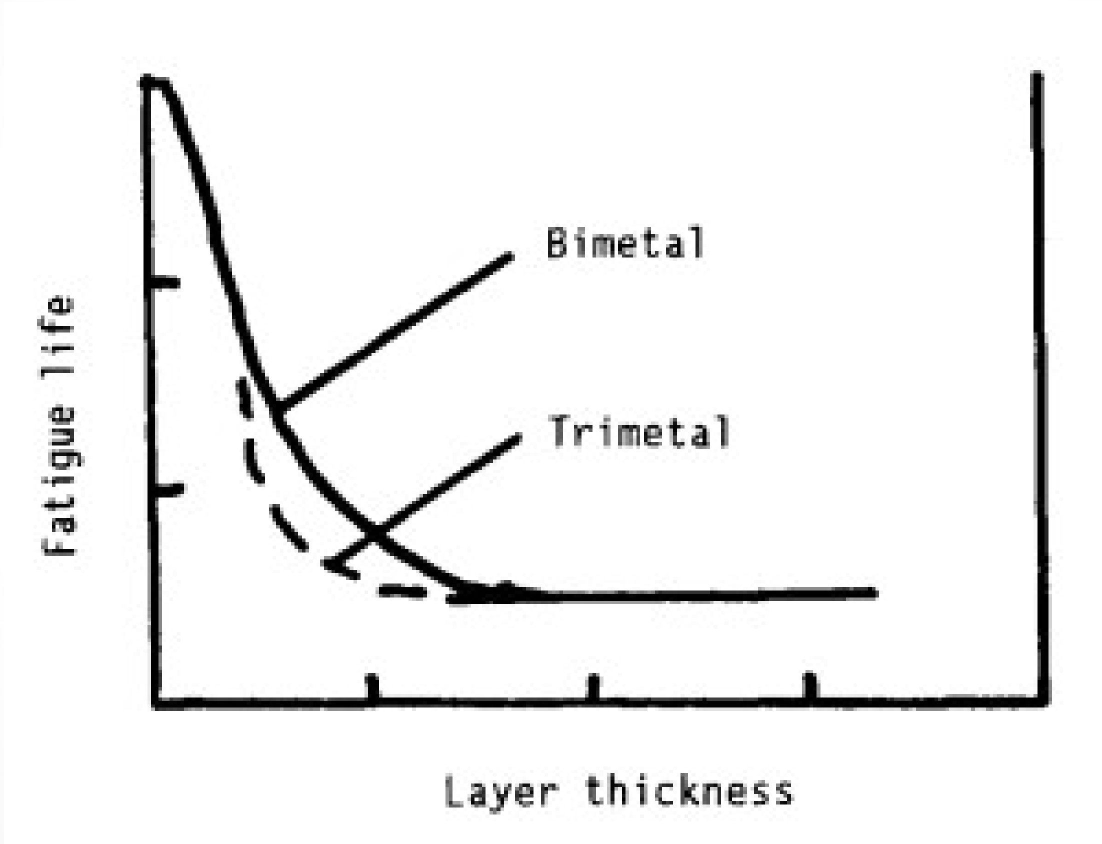
BEARING MATERIALS

- Bronze (88% copper & 12% tin)
- Babbitt
- Aluminum
- Zinc
- Porous metals: Sintered from powders of bronze, iron and aluminium; some mixed with lead or copper.
- Plastics (nylon, TFE, PTFE, phenolic, polycarbonate, filled polyimide)

Diagram of the multilayer soft metal bearing



Fatigue life as a function of babbitt layer thickness



Design requirements

- *Compressive Strength*
- *Fatigue Strength*
- *Embeddability*
- *Compatibility*
- *Corrosion resistance*
- *Conformability*
- *Wettability*

Properties Continued…

- *High thermal conductivity*
- *Low thermal expansion*
- *Low coefficient of friction*
- *Relative hardness*
- *Availability*
- *Cost*
- *Elasticity*
- *Availability*

Commonly used Bearing Materials & their Properties

Material		Load Capacity	Maximum Operating Temperature	Compatibility	Conformability	Embedability	Corrosion Resistance	Fatigue Strength
Tin Based Babbit		5.5 to 10.3	149^{0}C	1	1	1	5	5
Lead Based Babbit		5.5 to 8.3	149^{0}C	1	1	3	5	5
Copper Lead		10.3 to 17.2	177^{0}C	2	2	5	3	3
Lead Bronze		20.7 to 27.6	232^{0}C	3	4	4	2	2

1 = Highest Range
5 = Lowest Range

Applications of different materials of plain bearing

- **Cast Bronze:** It possesses a good combination of properties for such uses as pumps, machinery, and appliances.
- **Babbitt:** Because of their softness, babbitts have outstanding embeddability and resistance to seizure, and are often applied as liners in steel or cast iron housings.
- **Aluminum:** With the highest strength of the commonly used bearing materials, aluminum is used in severe applications in engines, pumps, and aircraft.

Continued….

- **Zinc:** Used during operation on steel journals, a thin film of the softer zinc material is transferred to the steel to protect it from wear and damage. It performs well in most atmospheric conditions except for continuously wet environments and exposure to seawater.
- **Porous Metals:** Such bearings are particularly good for slow-speed, reciprocating, or oscillating motions.

GRAPHALLOY….A Special Bearing Material

GRAPHALLOY, a graphite/metal alloy, is formed from molten metal, graphite and carbon; it is a uniform, solid, self-lubricating, bushing and bearing material. From this material we manufacture a unique, self-lubricating bearing solution, offering superior performance in hundreds of applications.

GRAPHALLOY material is suited for submerged or high temperature bearings or bushings - applications where oil, grease and plastics fail.

Features & Benefits of GRAPHALLOY

Features	Benefits
Self-Lubricating	Requires no grease or oil. Permits continuous operation and eliminates downtime.
Hot	Works at high temperatures where oil-based lubricants burn off or oxidize and plastics fail. Operating temperatures to 1,500º F / 800º C (in non-oxidizing atmospheres). Will not gum or seize.
Cold	Does not congeal or solidify at low temperatures or cryogenic conditions. Maintains self-lubrication.

Features & Benefits Continued…..

Dry	Works without lubrication. Survives run dry applications. Eliminates galling or seizing in hot and dry conditions. Will not attract dust.
Wet	Operates submerged. Will not swell or wash out. Withstands a wide variety of hostile fluids.
Dimensionally stable	Does not cold flow or deform under pressure. Maintains its size and shape.
Chemically Resistant	Insoluble in most industrial liquids. Works in acids, alkalies, hydrocarbons, water, and liquid gases.

Features & Benefits Continued….

Low Coefficient of Friction	Constant, low coefficient of friction. Not just a surface layer, solid throughout.
Linear Motion	Maintains lubrication during linear motion. Lubrication is not drawn out and dust is not pulled in.
Current Conducting	Eliminates sparks and static. Conducts well.
FDA Accepted	Accepted for food services. No lubrication to drip or cleanup. Easily steam cleaned.

GRAPHALLOY High Temperature Bearings

A graphite/metal alloy, is ideally suited to applications where temperatures are too high to permit the use of oil or other lubricants. GRAPHALLOY High Temperature Bearings and Bushings will not soften at extreme temperatures or extrude under load. Many grades are suitable for temperatures to 750°F (400°C) in air. In addition, special grades provide service up to 1000°F (535°C) and higher in non-oxidizing atmospheres.

Applications of Graphalloy Bearings

1. GRAPHALLOY High Temperature Bearings Solve Problems in Bakery Ovens.

Applications continued...

2. High Temperature Bearings Deliver Savings of More than $10,000 per year in Pulp and Paper Industry.

Applications Continued..

3. Glass Manufacturer Saves Over $20,000 per year with GRAPHALLOY High Temperature Bearings.

GRAPHALLOY
Bearing Solutions for Food Applications

Food processing solutions for Ovens, Dryers, Roasters, Mixers, Freezers, Sterilizers

- Graphalloy bearings are key ingredients for minimal maintenance and uninterrupted production in food contact equipment.
- Self-lubricating; works without external lubrication
- Operate in extreme temperatures (-450° F to +750° F/-240° C to +400° C)
- Avoid loss of lubricant in steam/pressure washes
- Contain no oil or grease
- Can be immersed in liquids without contaminating food production
- Operate submerged in low viscosity and corrosive liquids

Wear in Bearings

- Wear of the bearings can be calculated as
  K= W/FVT
- K Wear factor (in^3 min / lb ft hr)
- Expected life time, **T** is **T= t/KPV**
- **W** = volume of material lost, in
- **T** = hours of running time
- **F** = load
- **K** = wear factor
- **t** = thickness of wear

Design of Boundary Lubricated Bearings

The factors to be considered when selecting materials for bearings and specifying the design details include the following:

- *Coefficient of friction:* Both static and dynamic conditions should be considered.
- *Load capacity, p:* Radial load divided by the projected area of the bearing (Pa).
- *Speed of operation, v:* The relative speed between the moving and stationary components, m/s.
- *Temperature at operating conditions.*
- *Wear limitations.*
- *Producibility:* Machining, molding, fastening, assembly, and service.

PV Factor

The product pv between the load capacity(p) and speed of operation(v) is an important performance parameter for bearing design when boundary lubrication occurs.

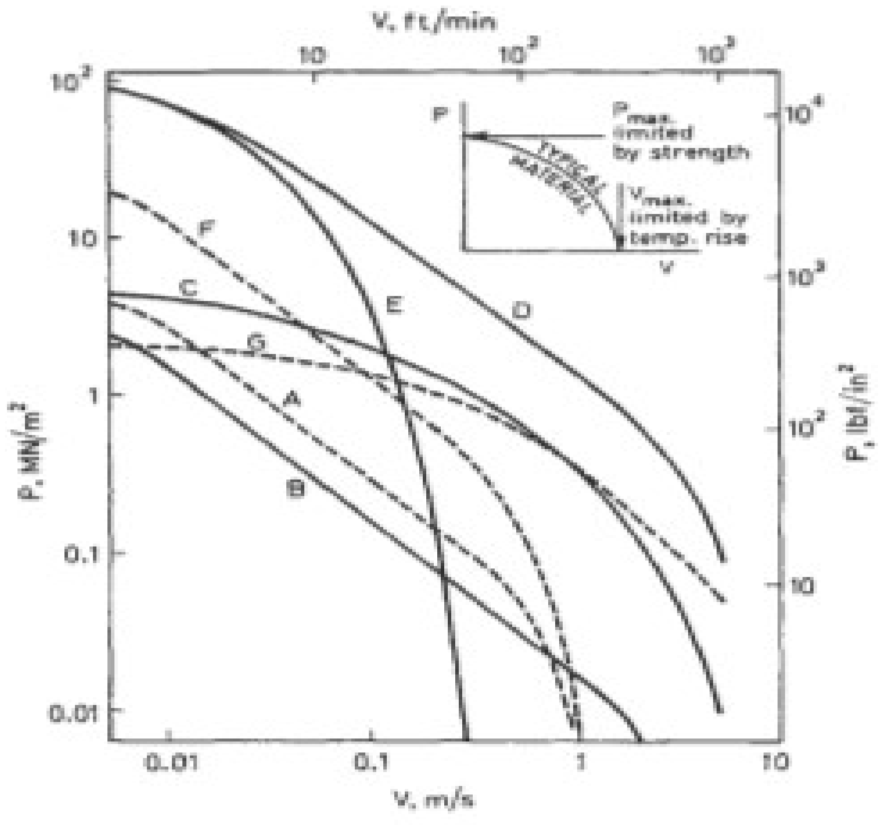
Therefore, pv can be thought of as the rate of energy input to the bearing per unit of projected area of the bearing if the coefficient of friction is 1.0.

- In SI units,

$$pv = kPa.m/s$$

PV chart

- *A* Thermoplastics
- *B* PTFE
- *C* PTFE + fillers
- *D* Porous bronze + PTFE + Pb
- *E* PTFE-glass weave + thermoset
- *F* Reinforced thermoset + MoS_2
- *G* Thermoset/carbon-graphite + PTFE



Operating Temperature

Most plastics are limited to approximately 200°F (93°C). However, PTFE can operate at 500°F (260°C). Babbitt is limited to 300°F (150°C), while tin-bronze and aluminum can operate at 500°F (260°C). A major advantage of carbon-graphite bearings is their ability to operate at up to 750°F (400°C).

Wear Factor

The wear factor **K** is measured under fixed conditions with the material loaded as a thrust washer. When equilibrium is reached, wear is measured as a volumetric loss of material as a function of time. The load and the velocity affect wear, thus **K** is defined as

$$K = W/FVT$$

Where, W= Wear, volume of material lost

F= Applied load

V= Linear velocity

T= Time

Note that the K factor and the above equation may not be used to predict actual wear in the application. The K factor only allows the comparison of relative wear among alternative materials

Design Procedure for BL Plain Surface Bearings

Given Information: Radial load on the bearing, *F* (N); speed of rotation, *n* (rpm); nominal minimum shaft diameter, D_{min} (in or mm)

Objectives of the design process: To specify the nominal diameter and length of the bearing and a material that will have a safe value *of pv*.

•Step 1: Specify a trial diameter, *D,* for the journal and the bearing.

•Step 2: Specify a ratio of bearing length to diameter, *L/D,* typically in the range of 0.5 to 2.0. For non-lubricated (dry-rubbing) or oil-impregnated porous bearings, *L/D* = 1 is recommended. For carbon-graphite bearings, *L/D* = 1.5 is recommended.

- Step 3: Compute *L = D(L/D) = nominal length of the bearing.*
- Step 4: Specify a convenient value for *L.*
- Step 5: Compute the bearing pressure (N/m2 =Pa):

  $p = F/LD$
- Step 6: Compute the linear speed of the journal surface:

  In SI metric units: $V = \pi Dn/(60\,000)$ m/s
- Step 7: Compute *pv (Pa.m/s).*
- Step 8: Multiply *2(pV) to obtain a design value for pv.*
- Step 9: Specify a material with a rated value *of pv equal to or greater* than the design value.

Complete the design of the bearing system considering diametrical clearance, lubricant selection, lubricant supply, surface finish specification, thermal control, and mounting considerations.

Example Problem :A bearing is to be designed to carry a radial load of 667N from a shaft having a minimum acceptable diameter of 38mm and rotating at 500 rpm. Design the bearing to operate under boundary-lubrication conditions.

Solution:

- *Step l.* Trial diameter: Let $D = D_{min}$ = 38mm = 0.038 m
- *Steps 2-4.* Let $L/D = 1$, Then $L = D$= 0.038m
- *Step 5.* Bearing pressure: $p = F/LD$ = (667N)/(0.038m)(0.038m)

  = 461911.36 N/m2
- *Step 6.* Journal speed:

  $V = \pi Dn/(60\ 000)$ m/s = π(0.038)(500)/60000 = 9.94×10^{-4} m/s
- *Step 7.* pv factor:

  pv= (461911.36)(9.94×10^{-4}) = 459.139Pa-m/s
- *Step 8.* Design value *of pv* = 2(459.139) = 918.279 Pa-m/s

Problem Continued…

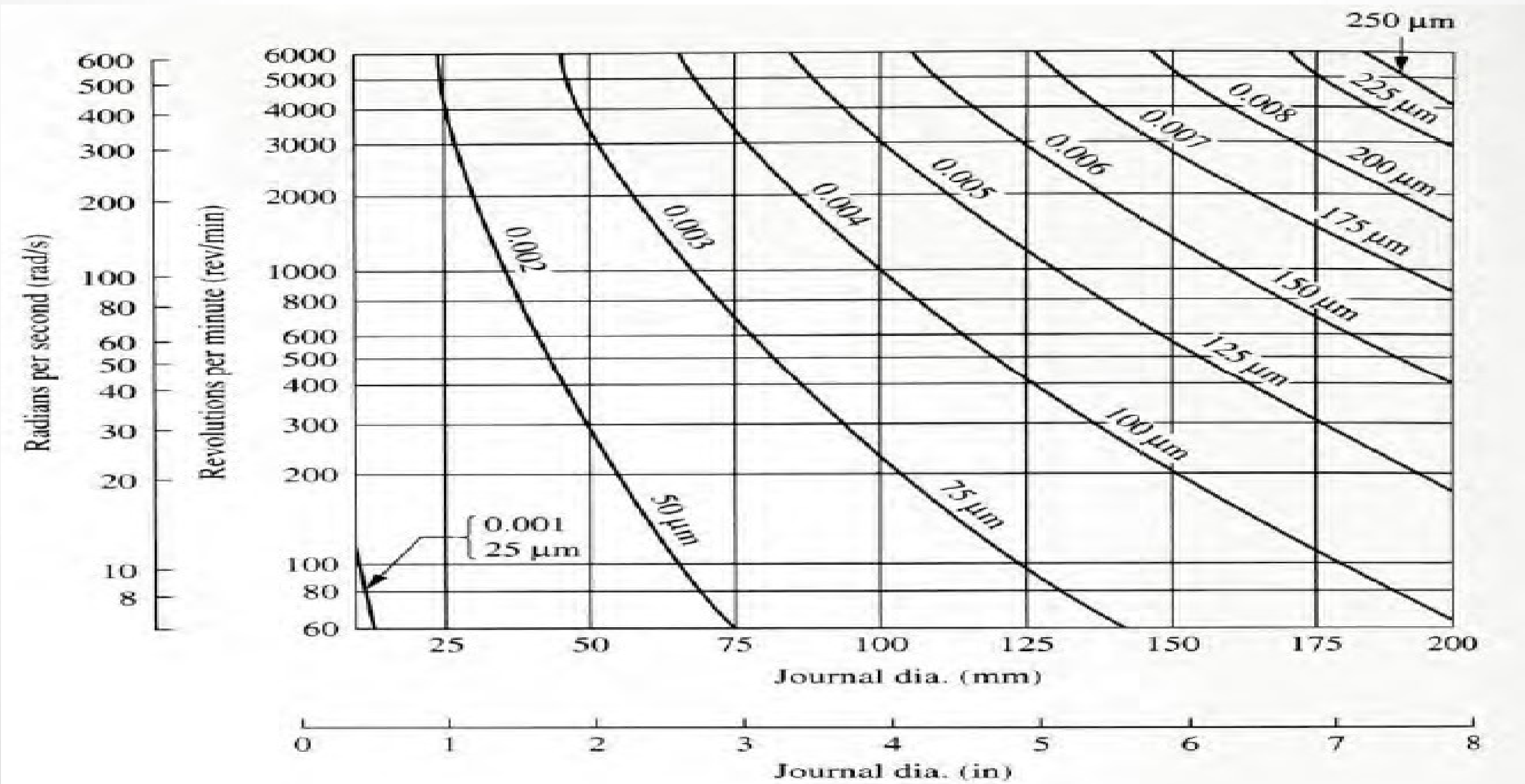
- *Step* 9. From Table below, we could use a bearing made from high tin babbit having a rated value *of pv* of 1050kPa-m/s.

Material	pV psi-fpm	pV kPa-m/s	
Vespel® SP-21 polyimide	300 000	10 500	Trademark of DuPont Co.
Manganese bronze (C86200)	150 000	5250	Also called SAE 430A
Aluminum bronze (C95200)	125 000	4375	Also called SAE 68A
Leaded tin bronze (C93200)	75 000	2625	Also called SAE 660
KU dry lubricant bearing	51 000	1785	See note 1
Porous bronze/oil impregnated	50 000	1750	
Babbitt: high tin content (89%)	30 000	1050	
Rulon® PTFE: M-liner	25 000	875	Metal backed
Rulon® PTFE: FCJ	20 000	700	Oscillatory and linear motion
Babbitt: low tin content (10%)	18 000	630	
Graphite/Metallized	15 000	525	Graphite Metallizing Corp.
Rulon® PTFE: 641	10 000	350	Food and drug applications (see note 2)
Rulon® PTFE: J	7500	263	Filled PTFE
Polyurethane: UHMW	4000	140	Ultra high molecular weight
Nylon® 101	3000	105	Trademark of DuPont Co.

Problem Continued…

- *Steps 10-11*. Nominal diametrical clearance:

From Figure below, we can recommend a minimum C_d = 0.002 in based on D = 38mm and n = 500 rpm. Other design details are dependent on the details of the system into which the bearing will be placed.



Properties of Dry Rubbing Bearing Materials

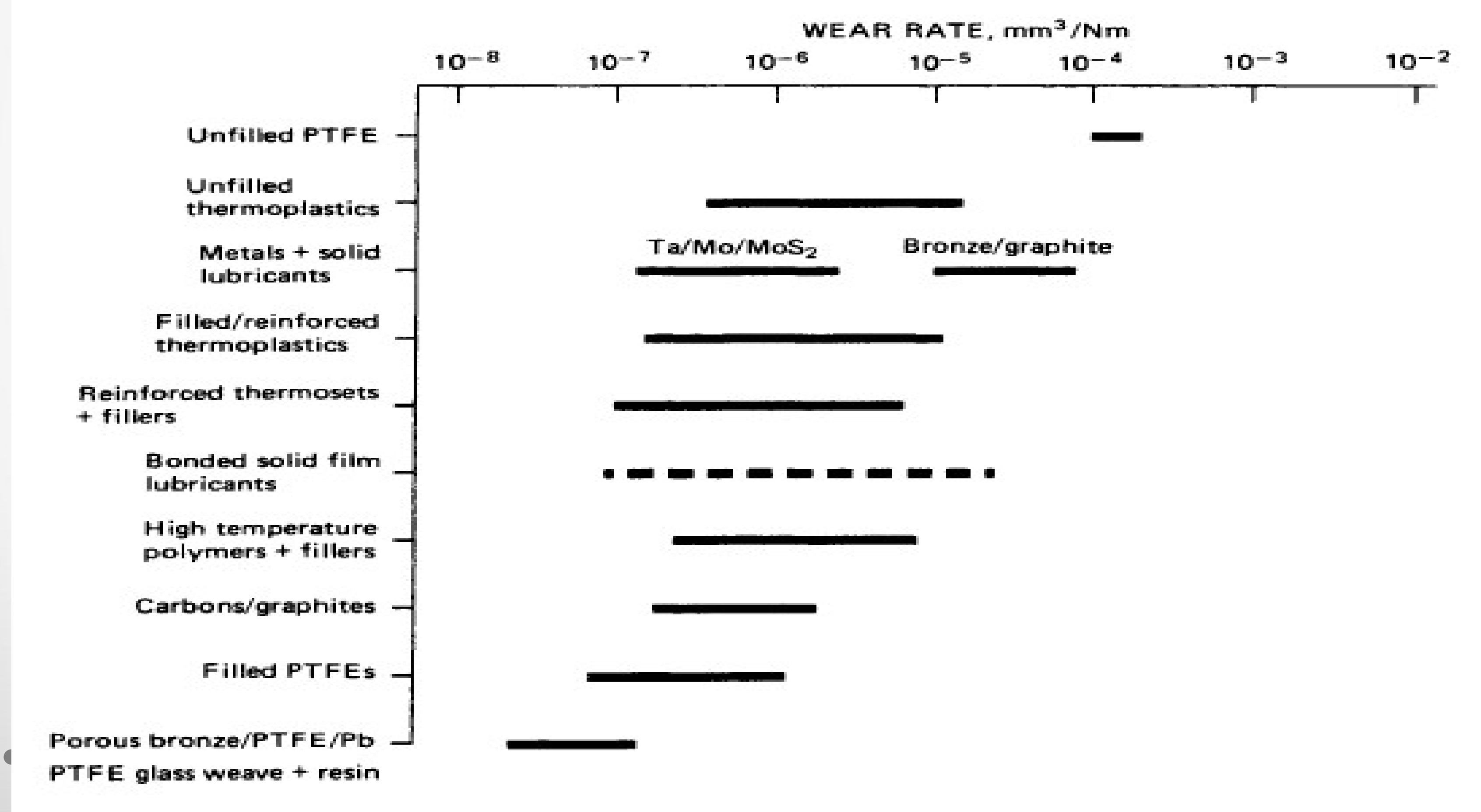
Type	*Examples*	*Max. static load*		*Max. service temp.*	*Coeff. exp.*	*Heat conductivity*		*Special features*
		MN/m^2	10^3 lbf/in^2	°C	10^{6}/°C	W/m°C	Btu/ft h °F	
Thermoplastics	Nylon, acetal, UHMWPE	10	1.5	100	100	0.24	0.14	Inexpensive
Thermoplastics +fillers	Above + MoS_2, PTFE, glass, graphite, etc.	15–20	2–3	150	60–100	0.24	0.14	Solid lubricants reduce friction
PTFE + fillers	Glass, bronze, mica, carbon, metals	2–7	0.3–1	250	60–100	0.25–0.5	0.15–0.3	Very low friction
High temperature polymers (+fillers)	Polyimides polyamide-imide PEEK	30–80	4.5–12	250	20–50	0.3–0.7	0.2–0.4	Relatively expensive

Properties Continued…

Type	*Examples*			*Max. static load*			*Max. service temp.*	*Coeff. exp.*
Thermosets + fillers	Phenolics, epoxies + asbestos, textiles, PTFE	30–50	4.5–7.5	175	10–80	0.4	0.25	Reinforcing fibres improve strength
Carbon–graphite	Varying graphite content; may contain resin	1–3	0.15–0.45	500	1.5–4	10–50	6–30	Chemically inert
Carbon–metal	With Cu, Ag, Sb, Sn, Pb	3–5	0.45–0.75	350	4–5	15–30	9–18	Strength increased
Metal–solid lubricant	Bronze–graphite -MoS_2; Ag–PTFE	30–70	4.5–10	250–500	10–20	50–100	30–60	High temperature capability
Special non-machinable products	Porous bronze/PTFE/Pb	350	50	275	20	42	24	Need to be considered at the design stage
	PTFE/glass weave + resin	700	100	250	12	0.24	0.14	
	Thermoset + PTFE surface	50	7.5	150	10	0.3	0.2	
	Metal + filled PTFE liner	7	1	275	100	0.3	0.2	

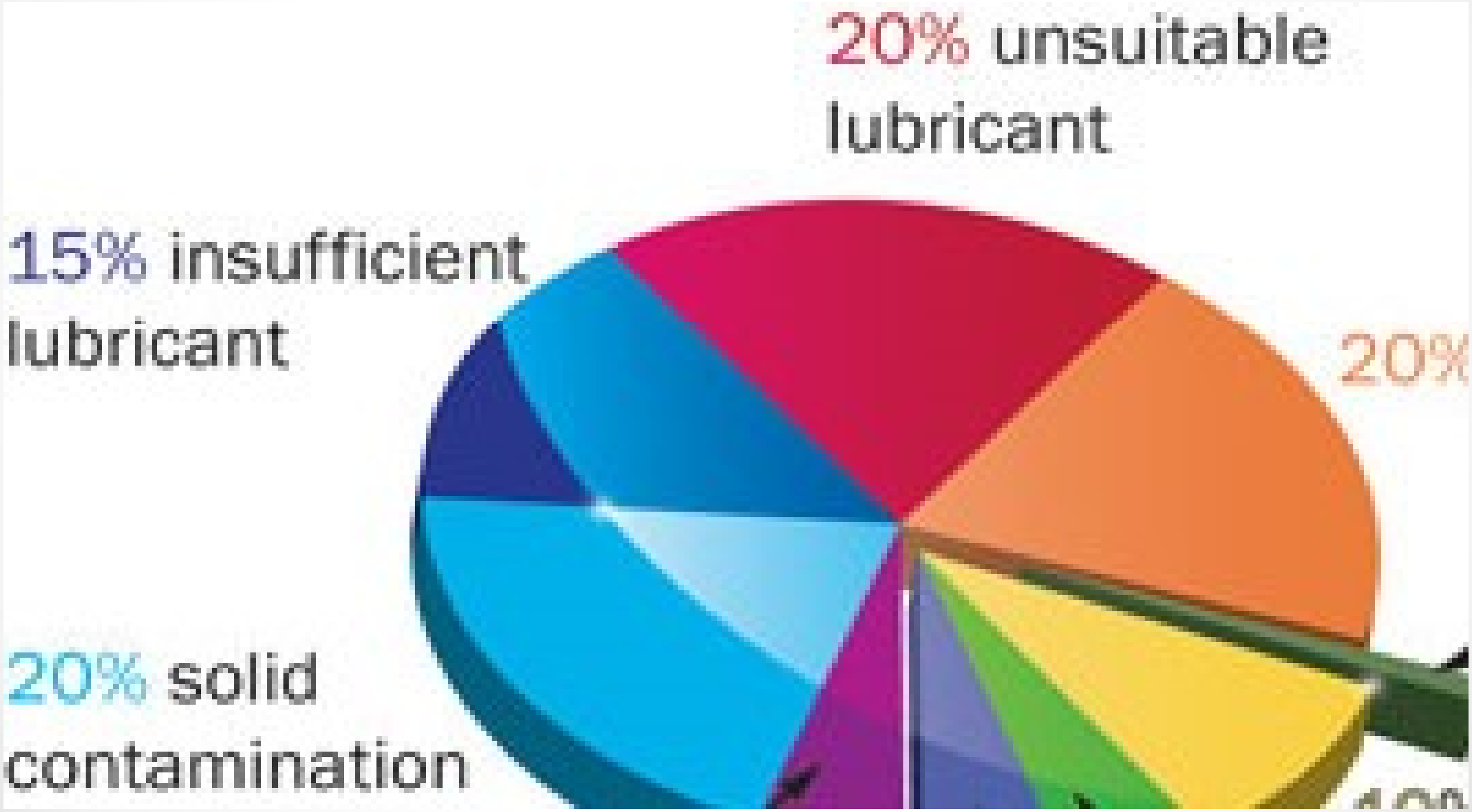
wear rates of dry bearing material groups

Light loads and low speeds (frictional heating negligible) against smooth (0.15 µm Ra) mild steel.

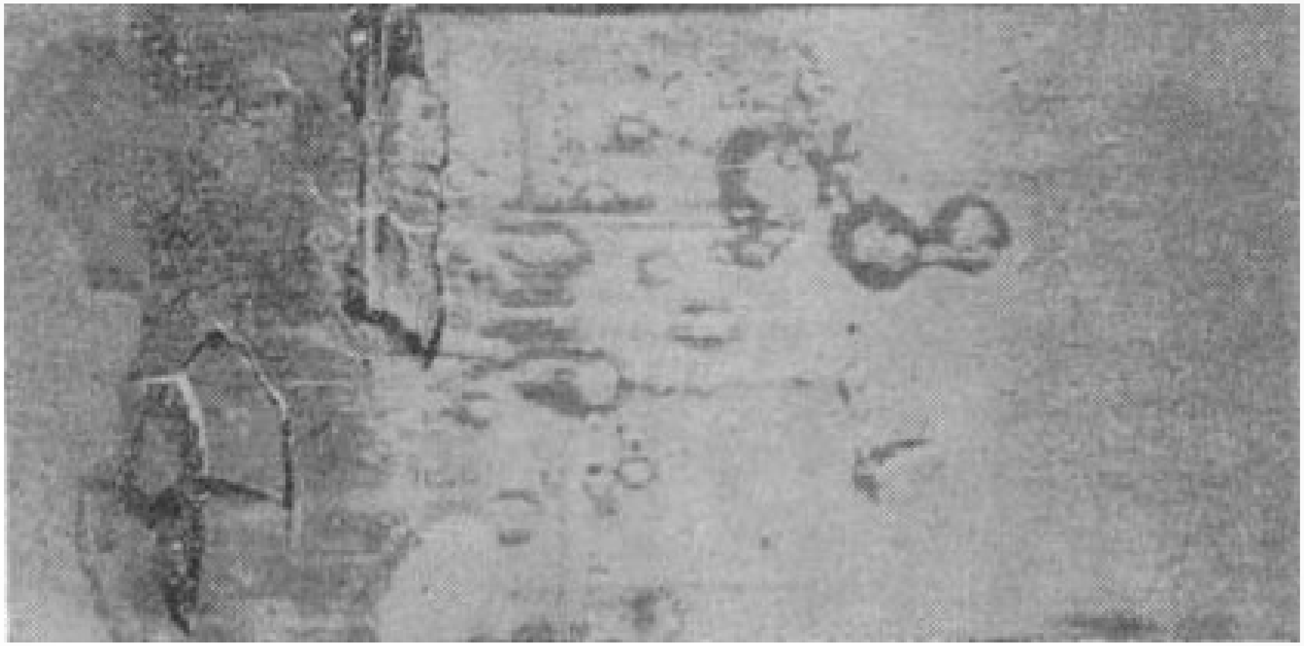


PLAIN SURFACE BEARING FALIURES

BEARING FAILURES

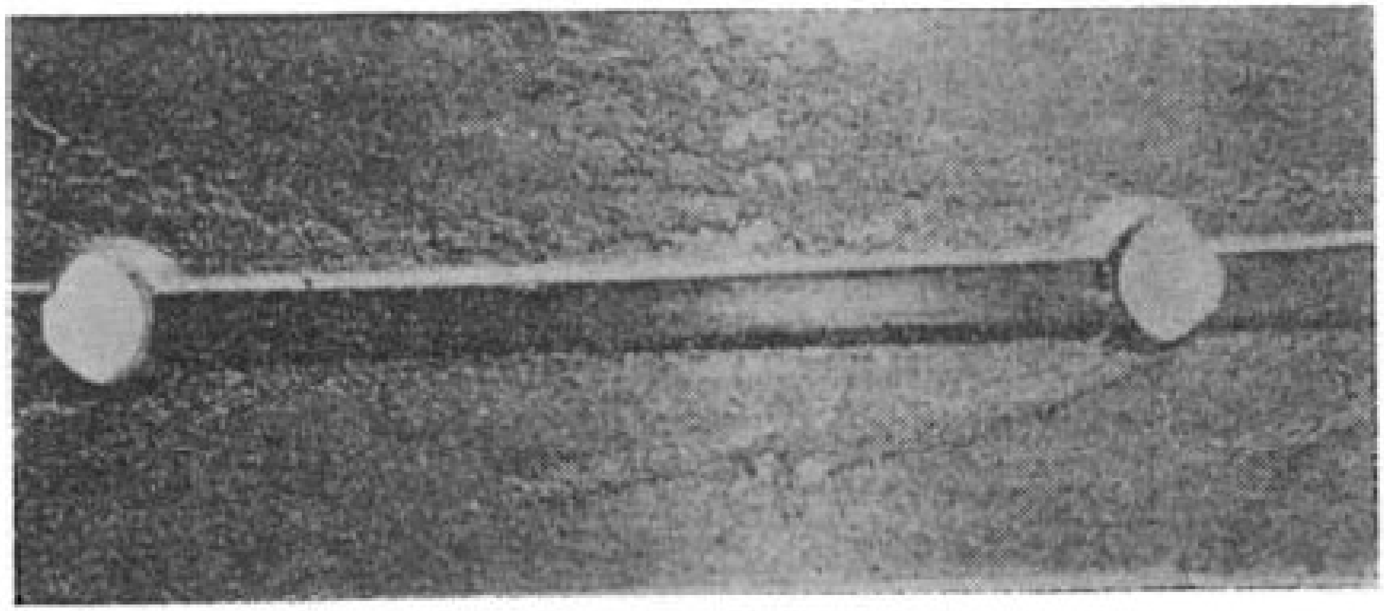


Foreign Matter



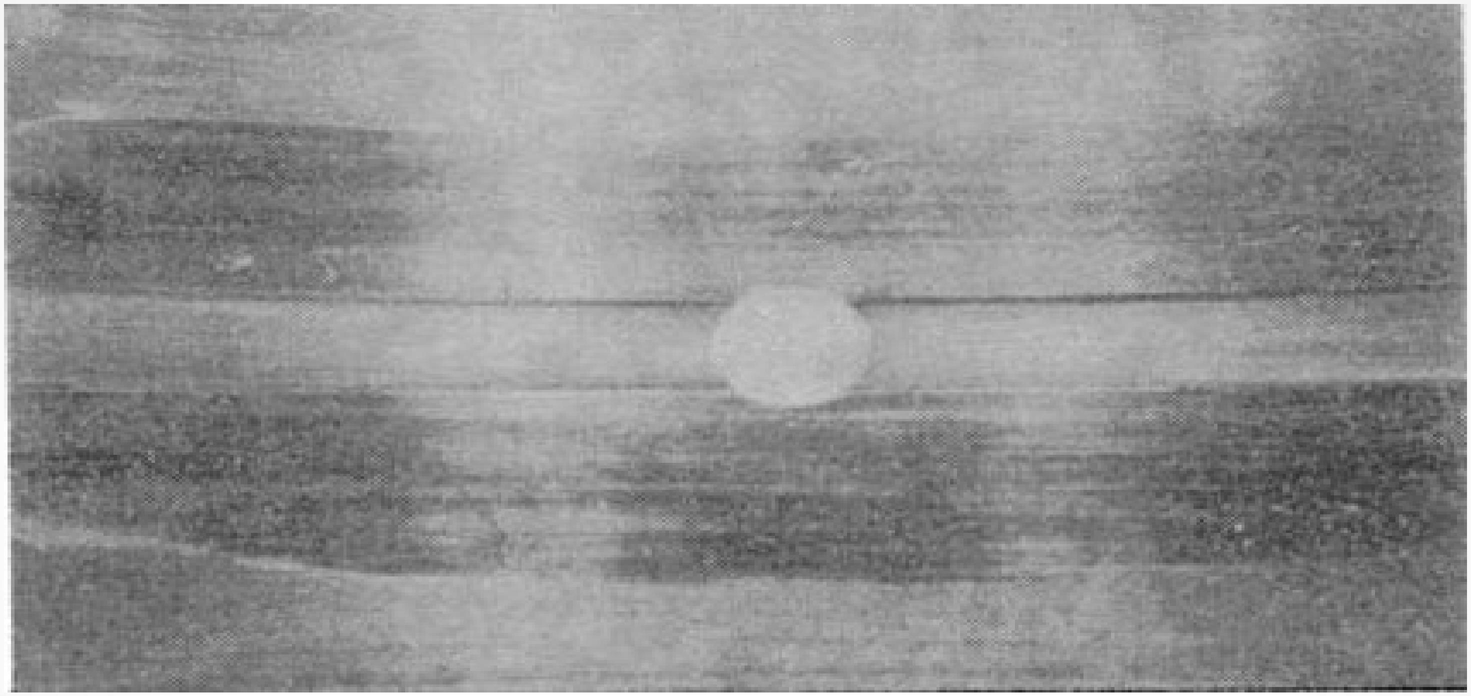
- **Causes:** Dirt particles in lubricant exceeding the minimum oil film thickness.

Foreign matter



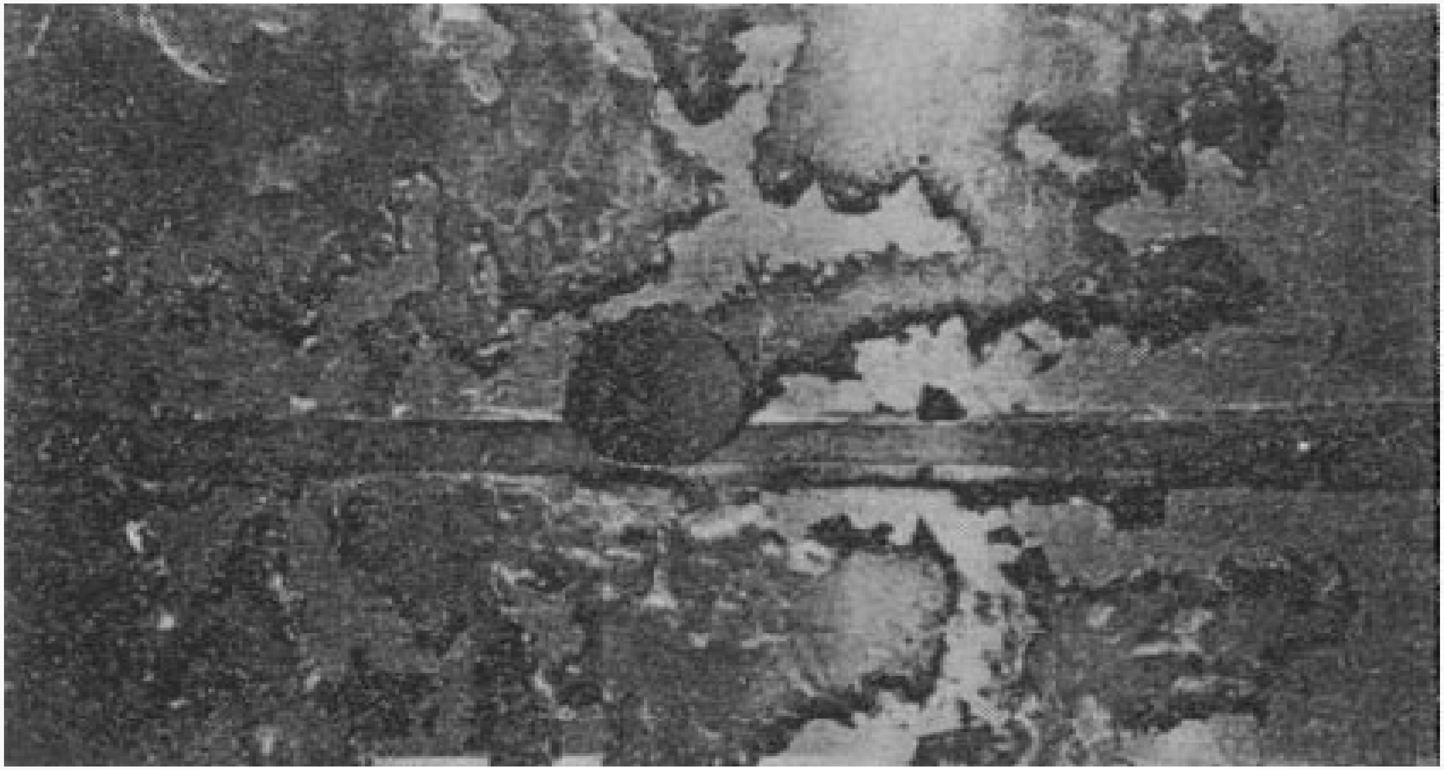
- Causes: Contamination of lubricant by excessive amounts of dirt particularly non-metallic particles which can roll between the surfaces.
-

Wiping

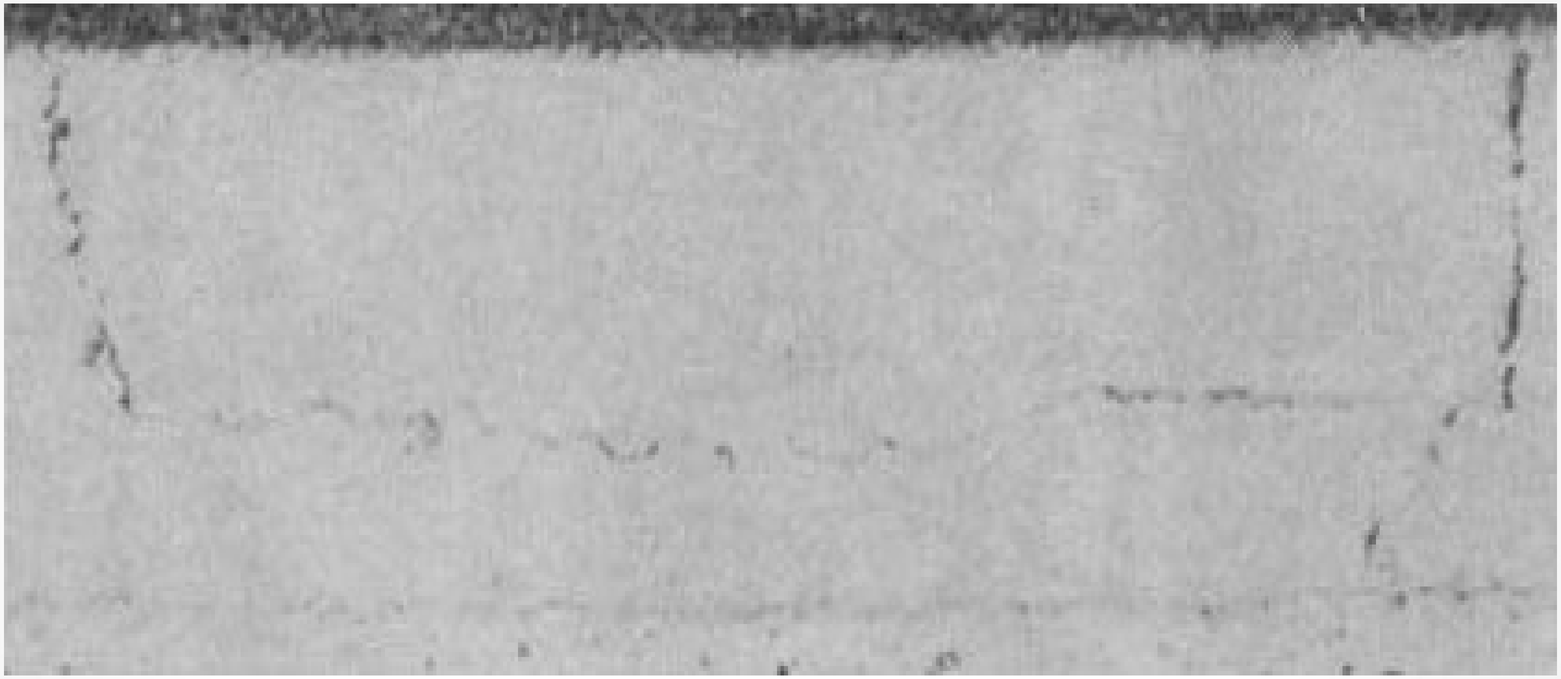


- **Causes: Inadequate clearance, overheating, insufficient oil supply, excessive load, or operation with a non-cylindrical journal.**

Fatigue

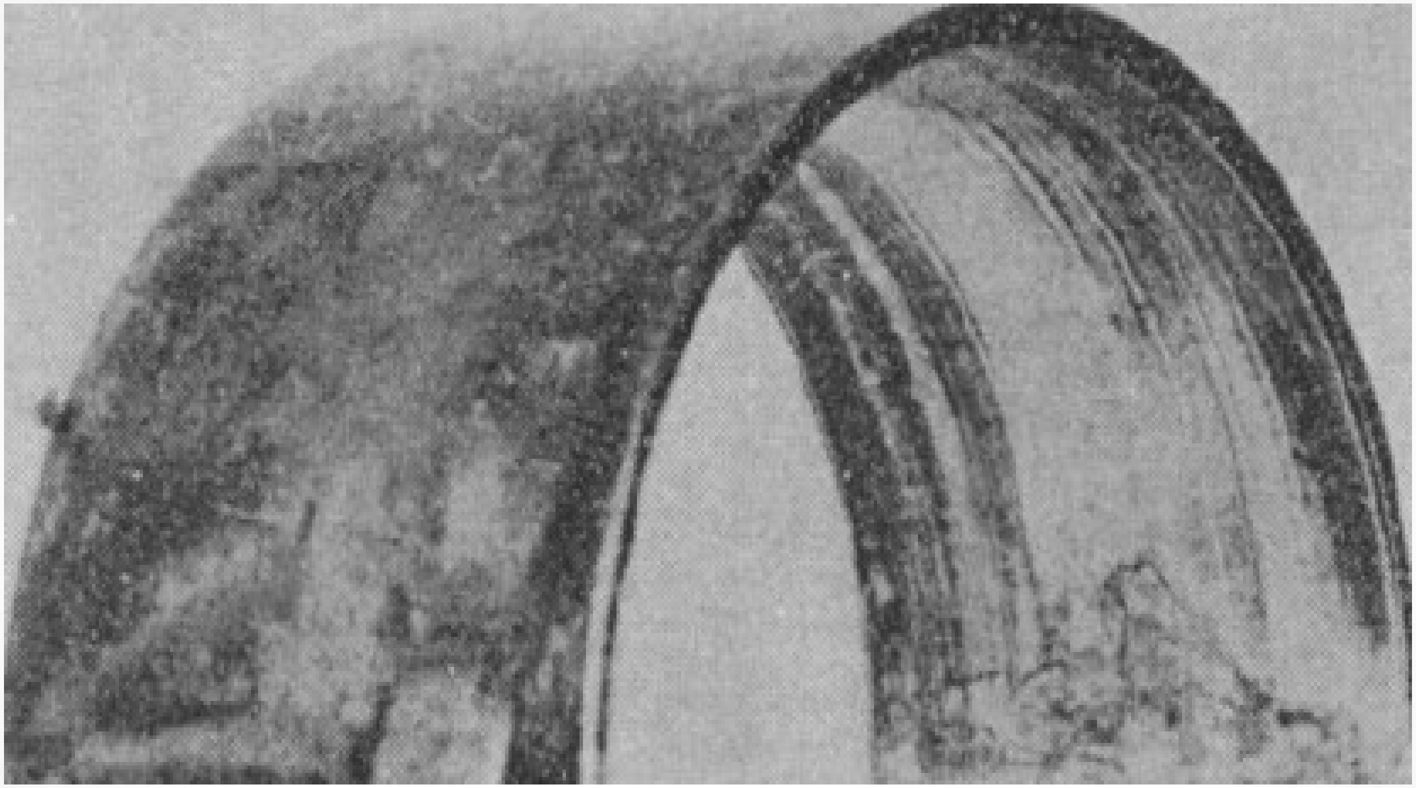


Fatigue



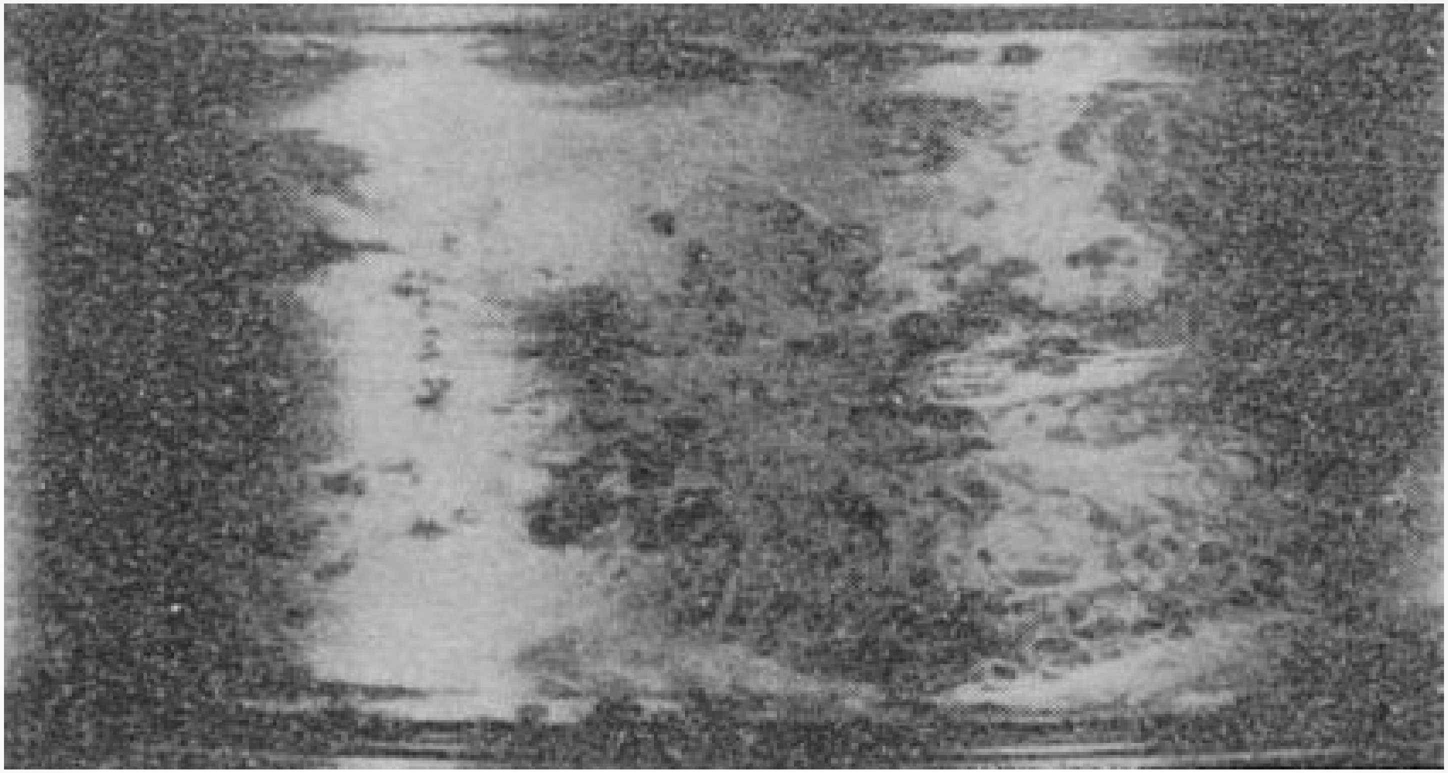
- Causes: Excessive dynamic loading which exceeds the fatigue strength at the operating temperature.

Excessive interference



- Causes: Excessive interference fit or stagger at joint faces during assembly.
-

Fretting



- Causes: Inadequate interference fit; flimsy housing design; peirmtting small sliding movements between surfaces under operating loads.
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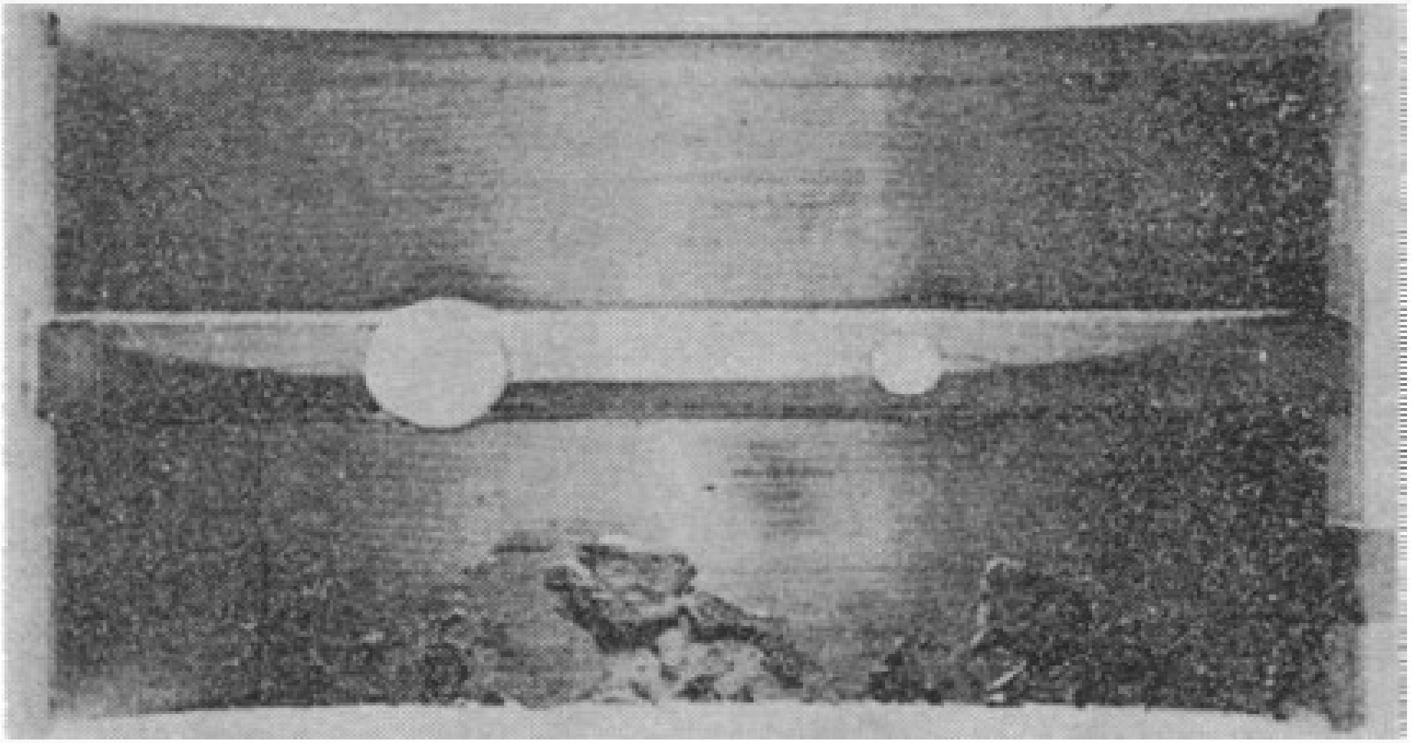
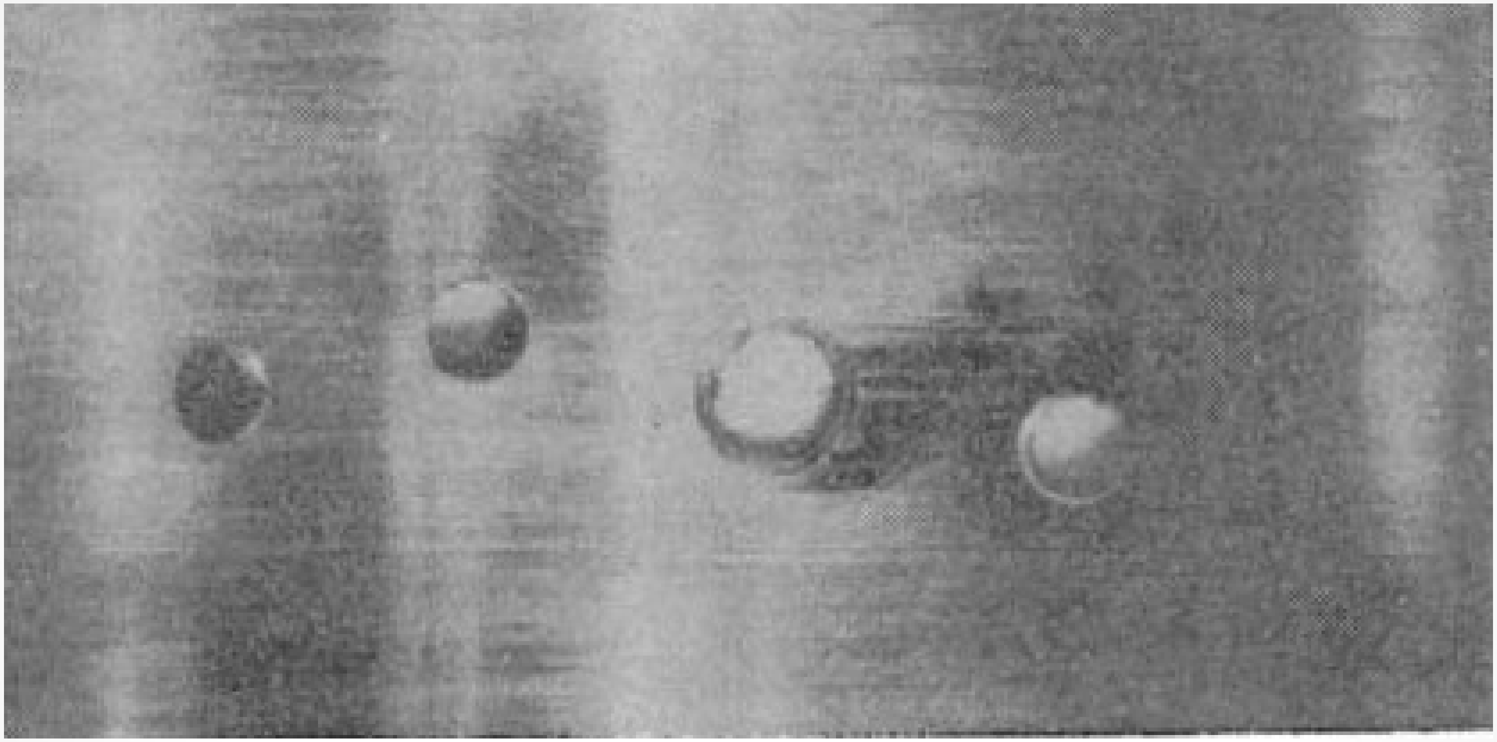
-

Misalignment



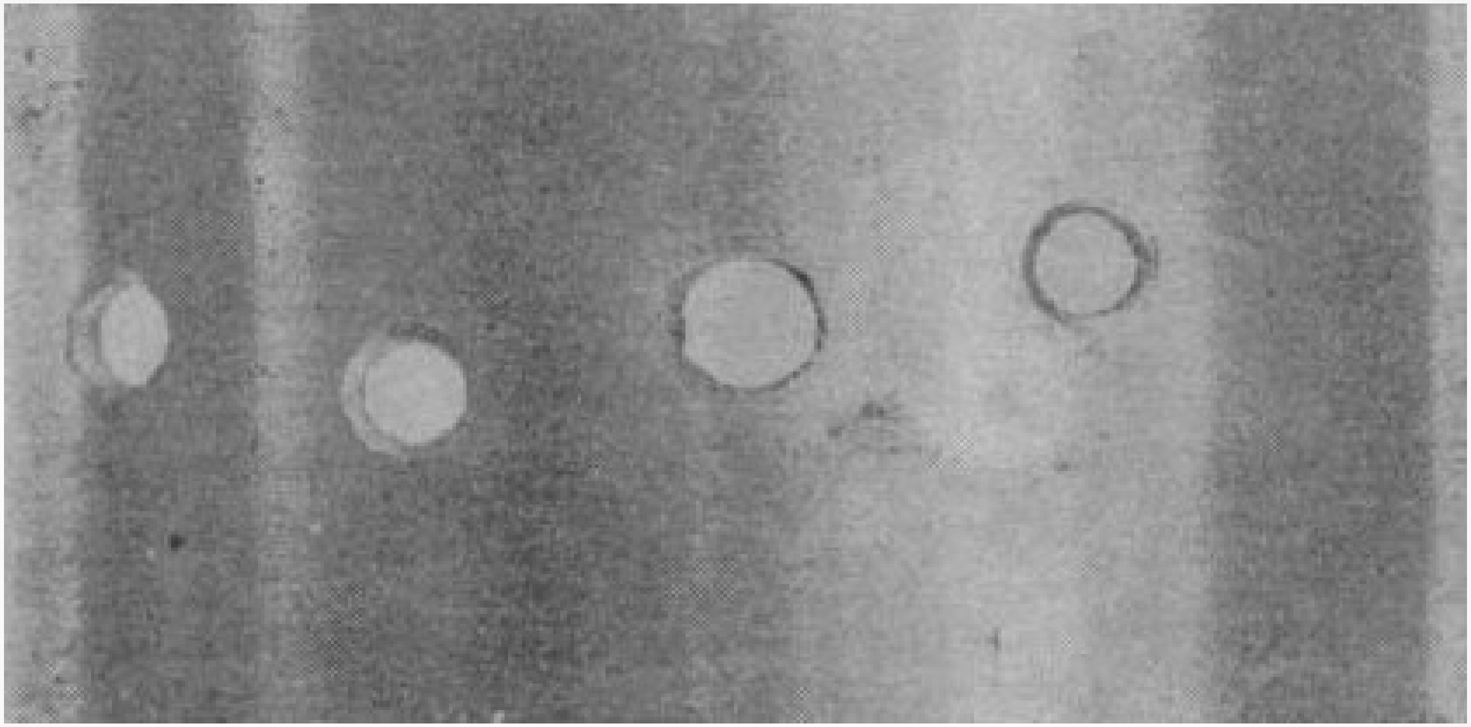
- Causes: Misalignment of bearing housings on assembly, or journal deflection under load.
-

Dirty Assembly



- Causes : **Entrapment of large particles of dirt (e.g. swarf), between bearing and housing, causing distortion of the shell, impairment of heat transfer and reduction of clearance**

Dirty Assembly

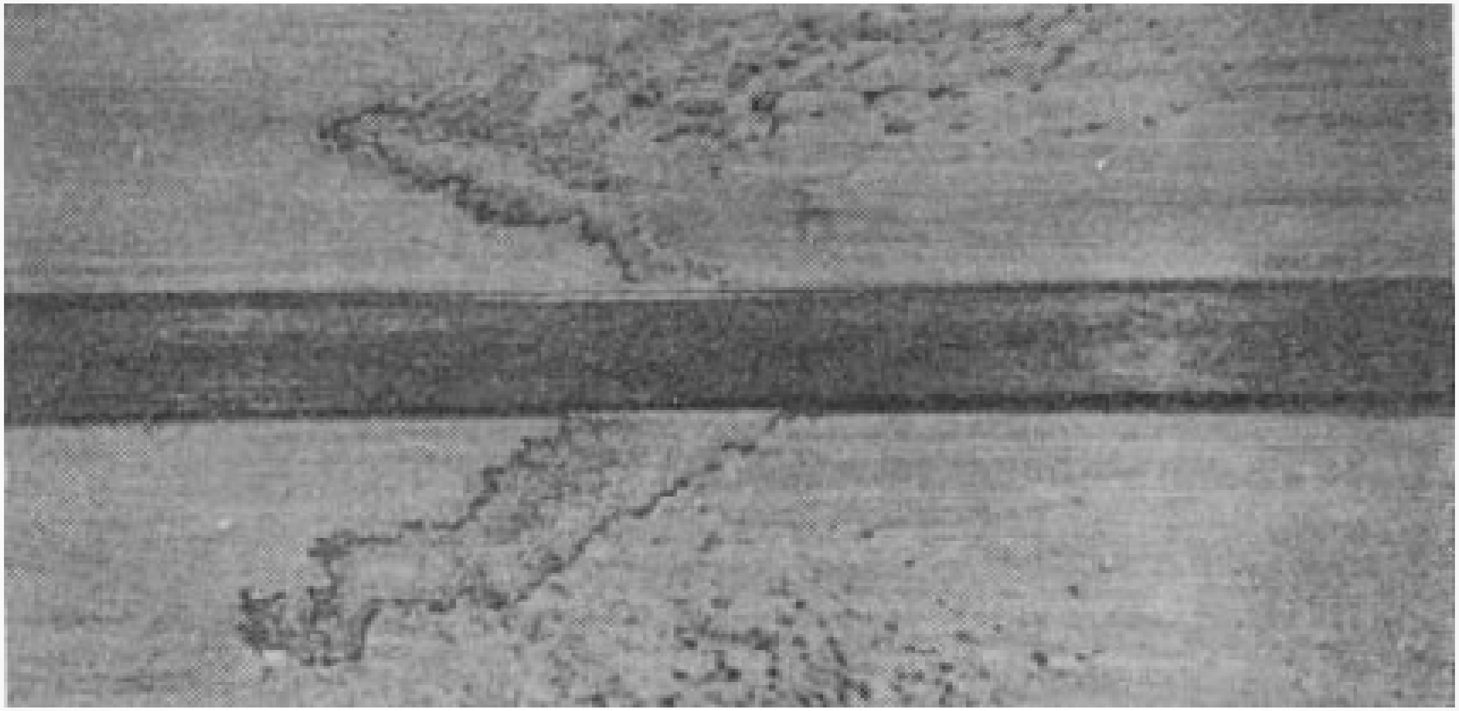


- **Causes:** Entrapment of dirt particles between bearing and housing. Bore of bearing is shown in previous column illustrating local overheating due to distortion of shell, causing reduction of clearance and impaired heat
- transfer.

Cavitation Erosion

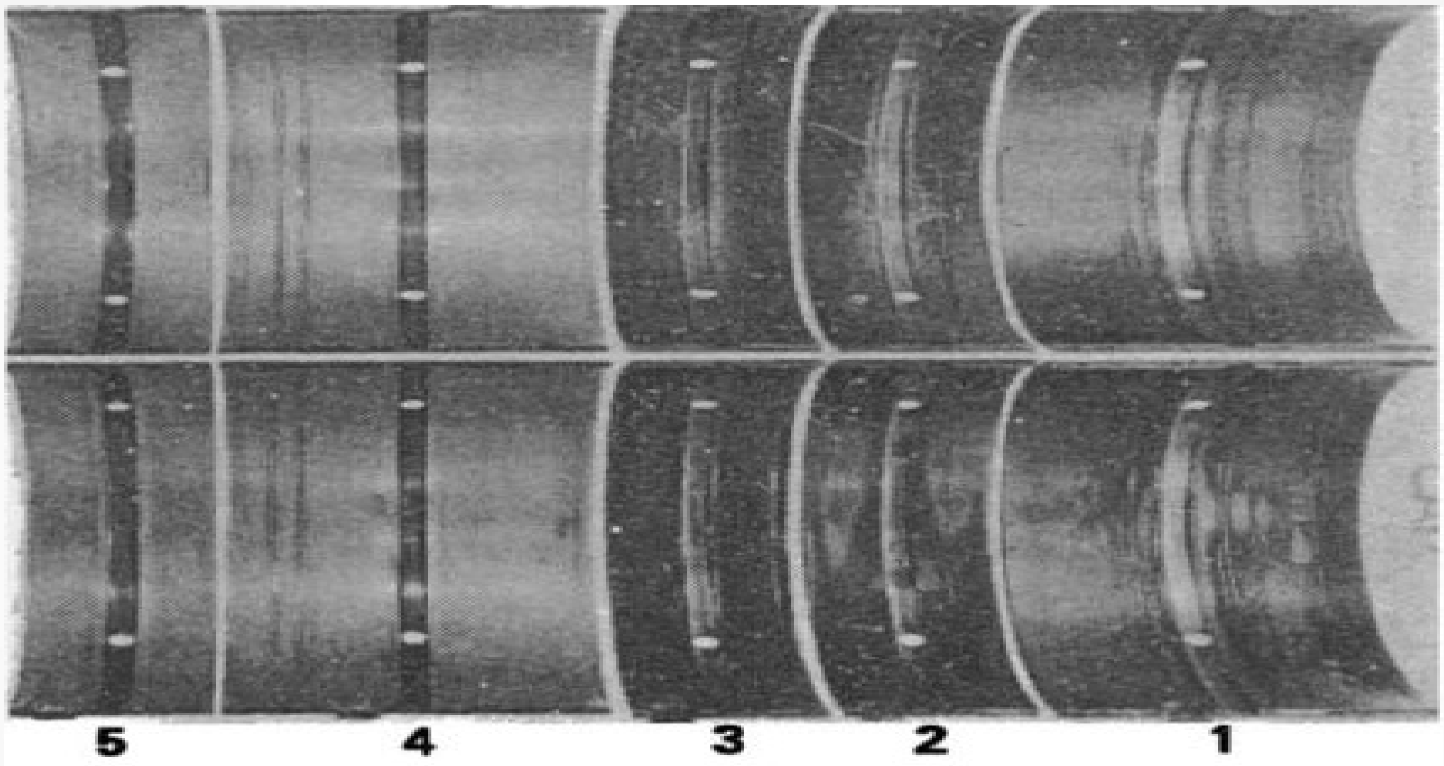
- Causes: Changes of pressure in oil film associated with interrupted flow.

Dsicharge cavitation erosion



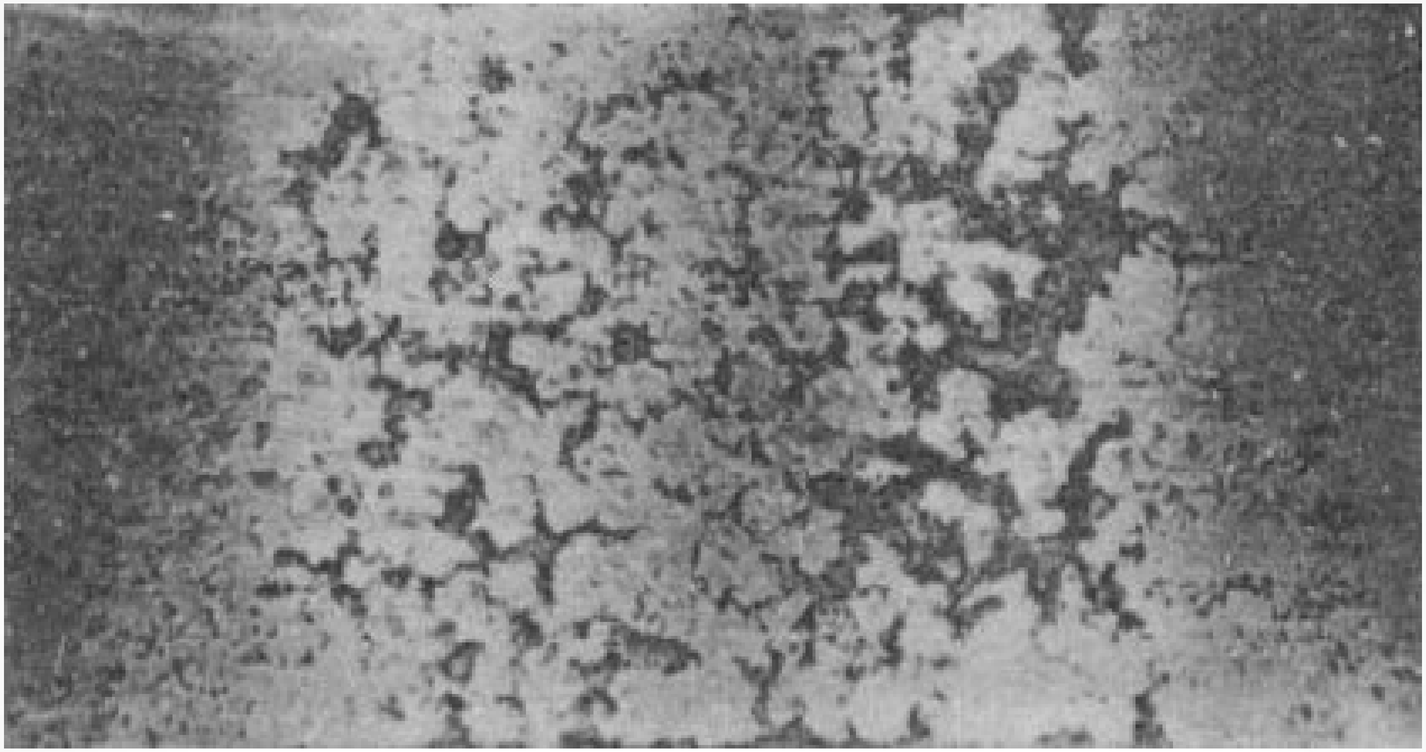
- Causes: Rapid advance and retreat of journal in clearance during cycle. It is usually associated with the operation of a centrally grooved bearing at an excessive operating clearance.

Cavitation Erosion



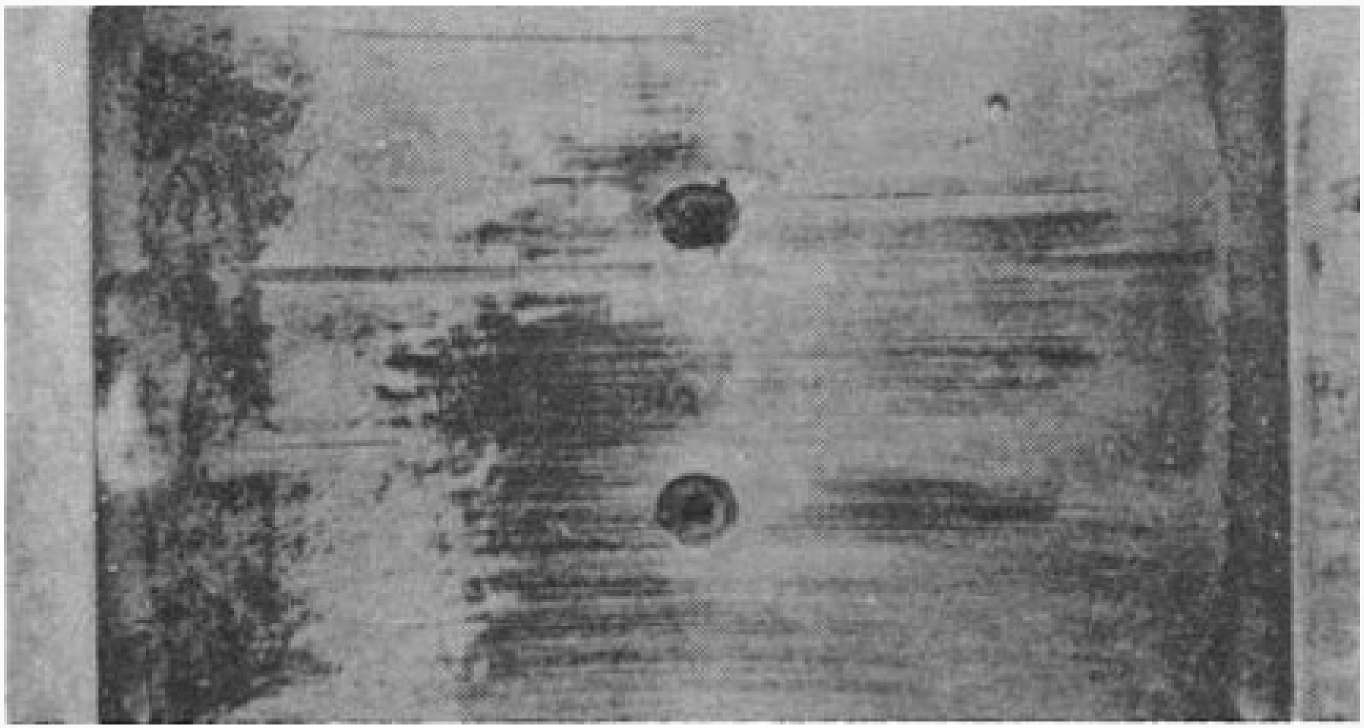
- Causes: Impact fatigue caused by collapse of vapour bubbles in oil film due to rapid pressure changes. Softer overlay (Nos 1, 2 and 3 bearings) attacked. Harder aluminium -20% tin (Nos 4 and 5 bearings) not attacked under these particular conditions.

Corrosion

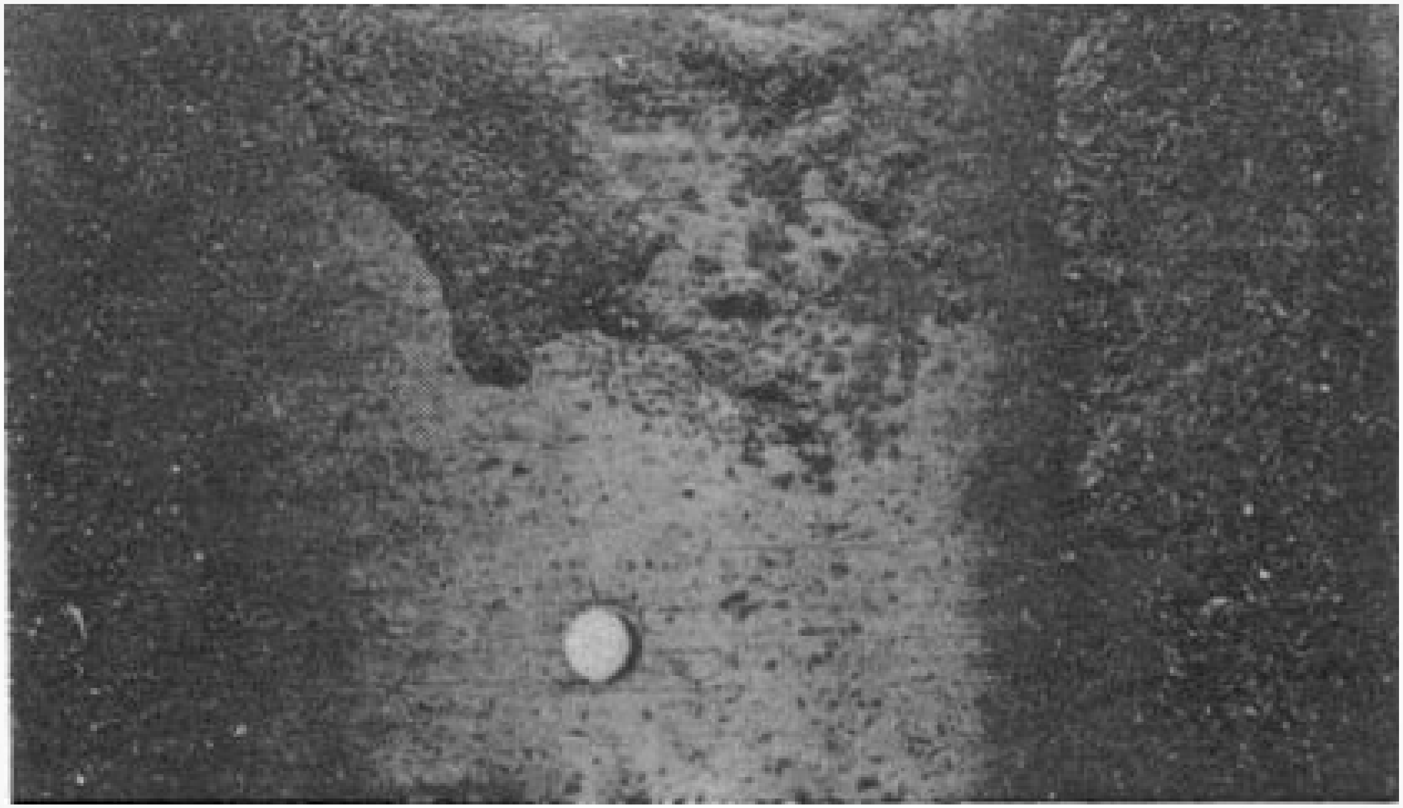


- Causes: Formation of organic acids by oxidation of lubricating oil in service. Consult oil suppliers; investigate possible coolant leakage into oil.

Tin dioxide Corrosion

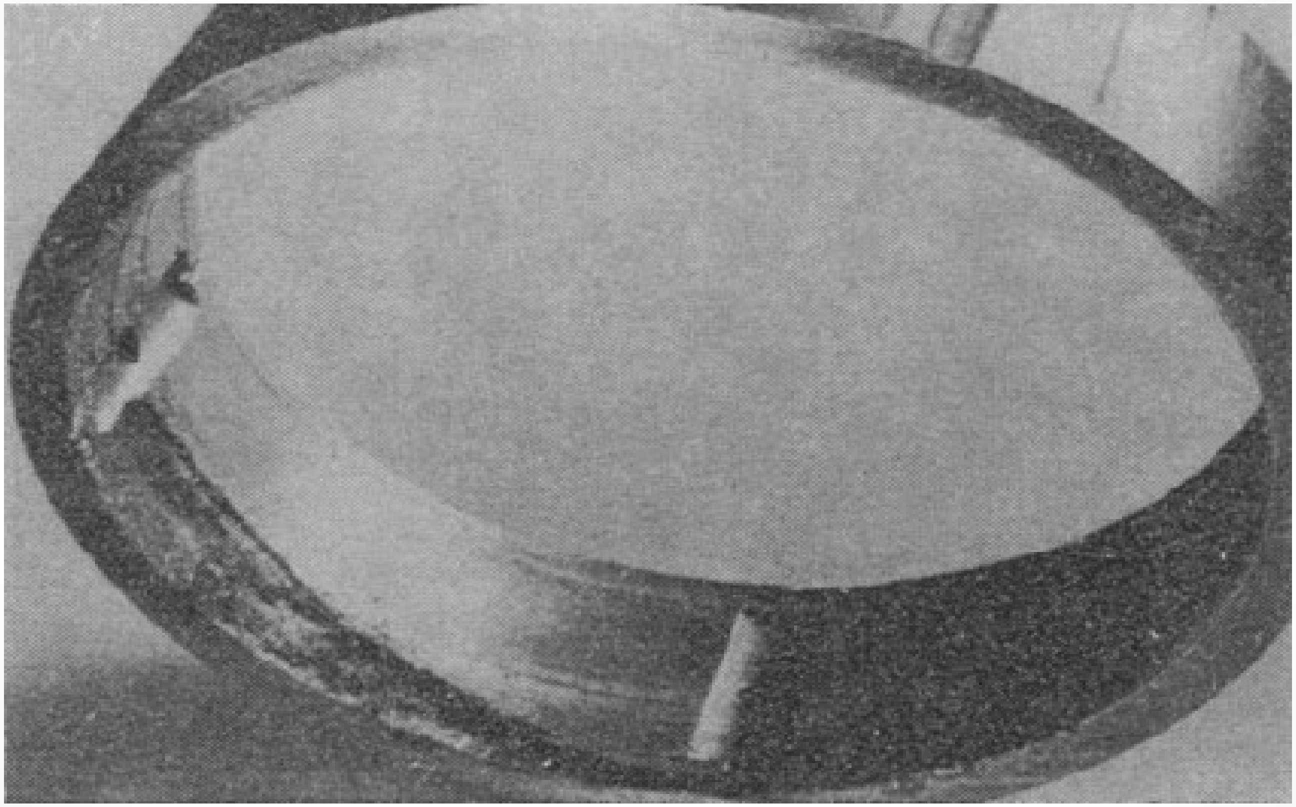


Sulphur corrosion

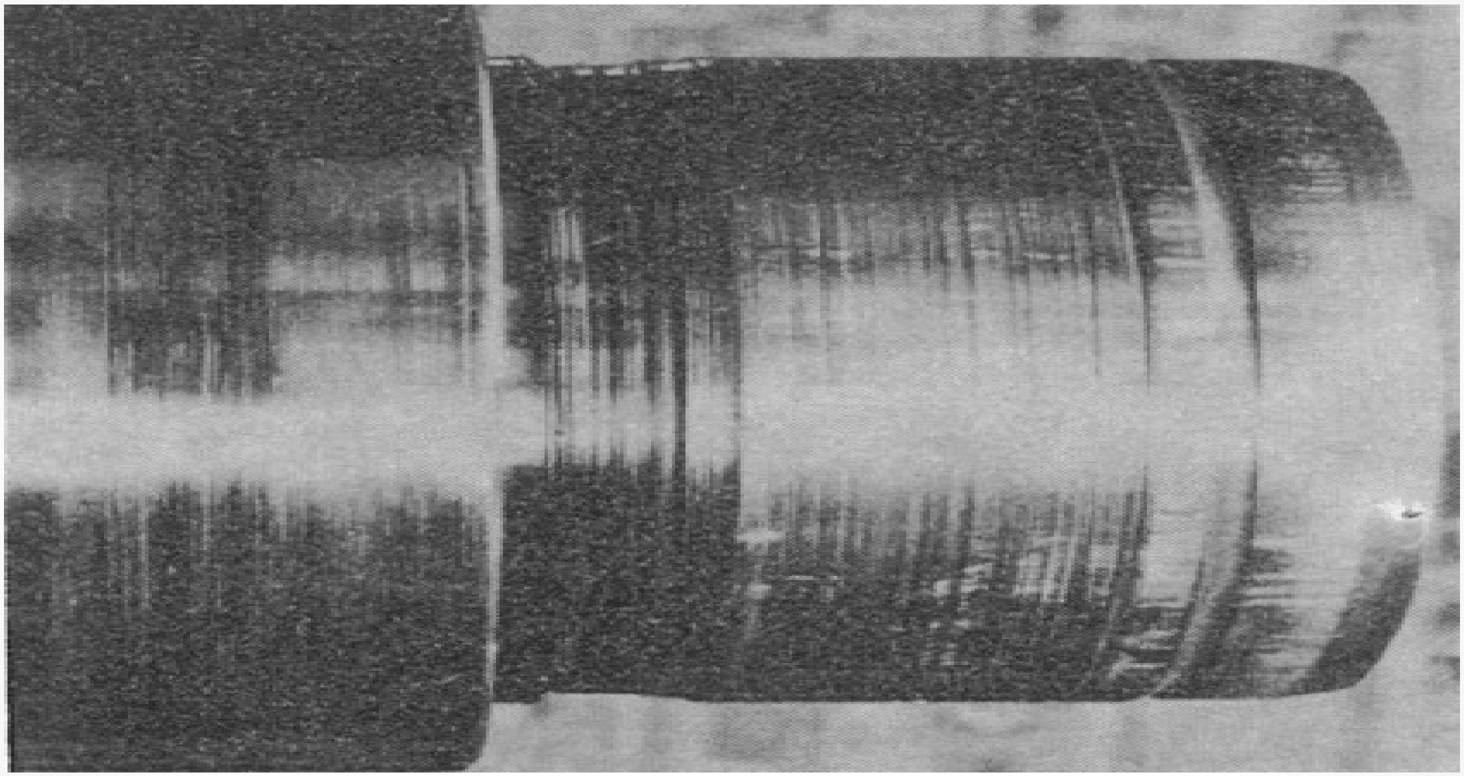


- Causes: Attack by sulphur-compounds from oil additives or fuel combustion products.

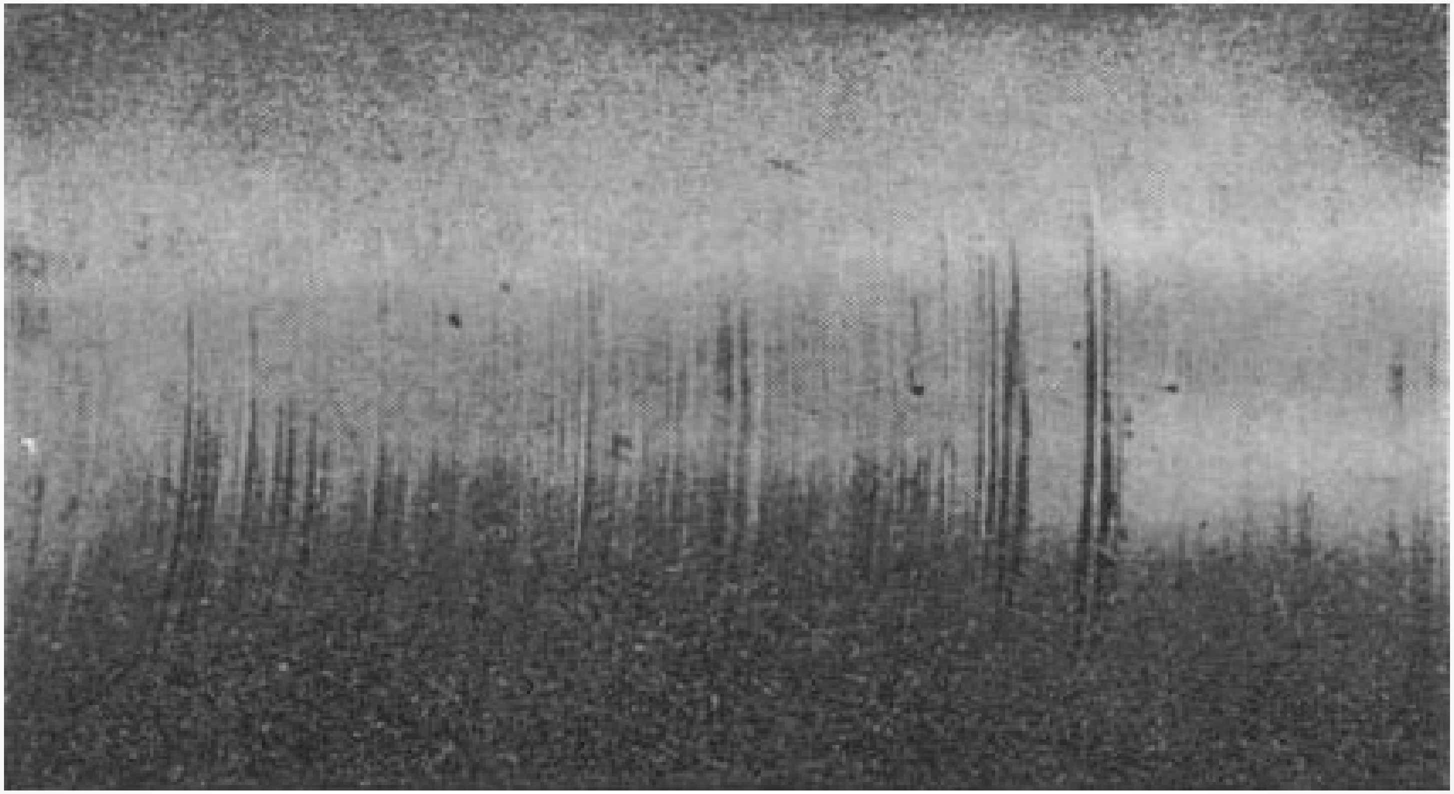
Wire wool damage



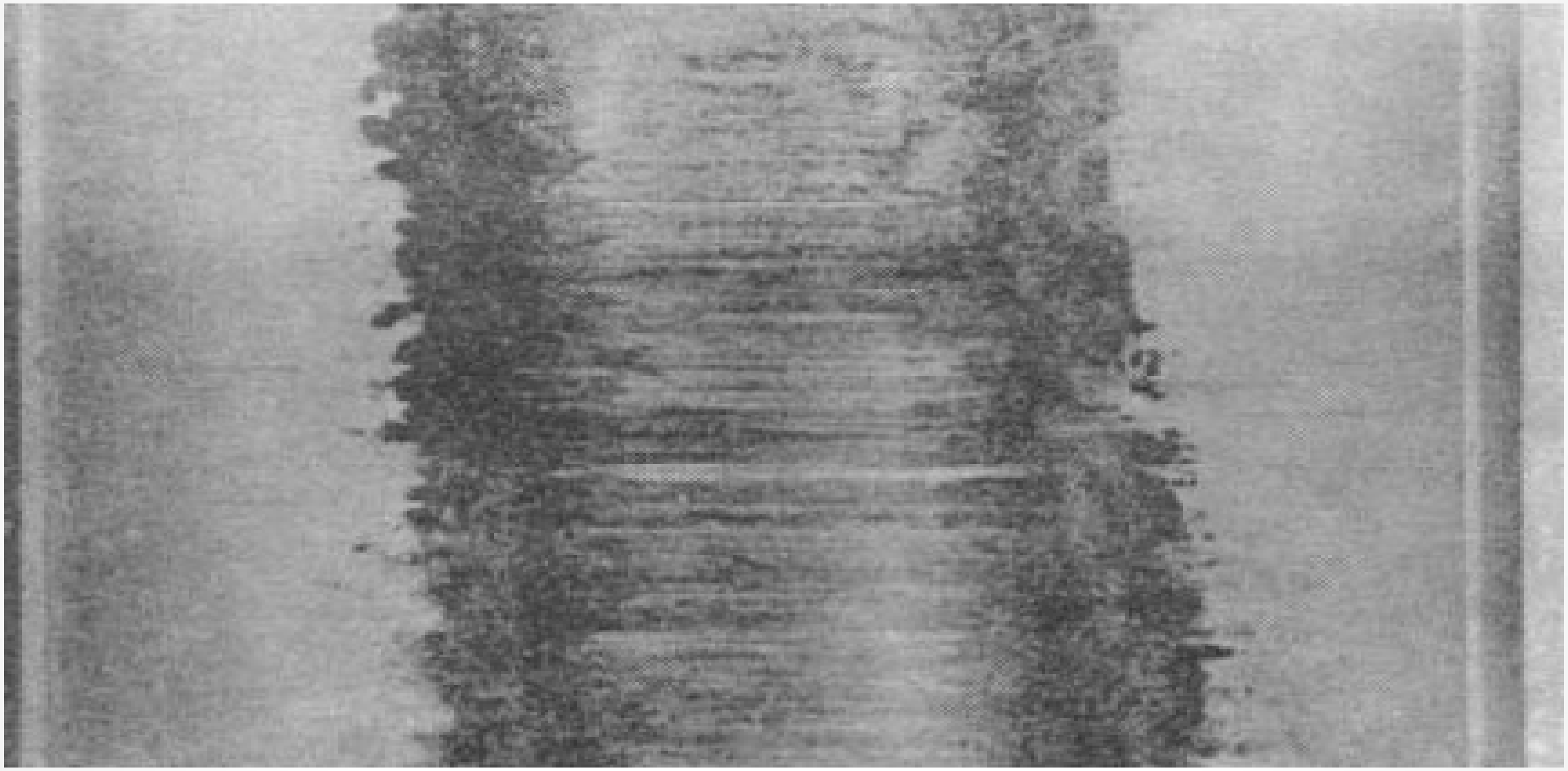
Wire wool damage



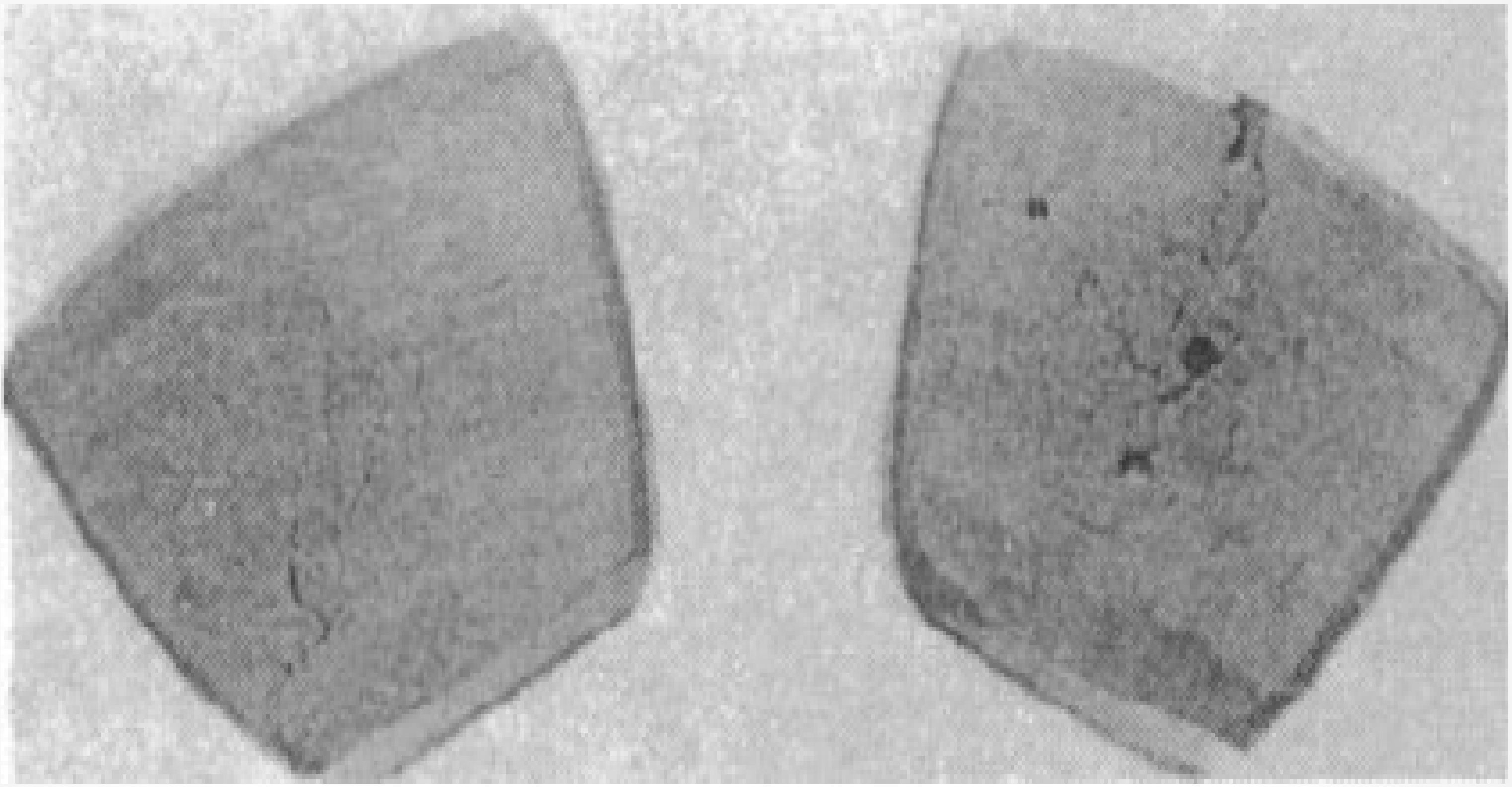
Electrical discharge



Fretting due to external vibration



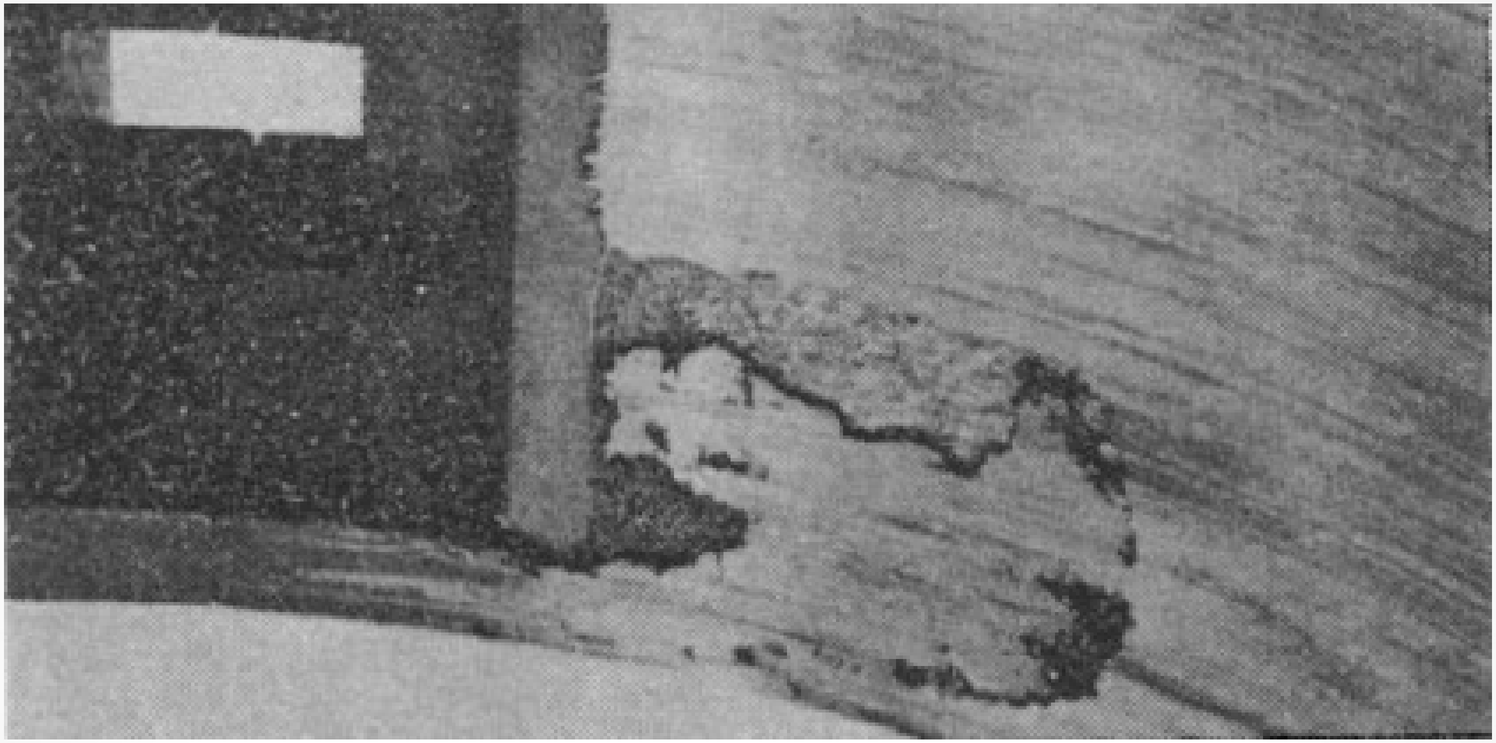
Overheating



Thermal cycling

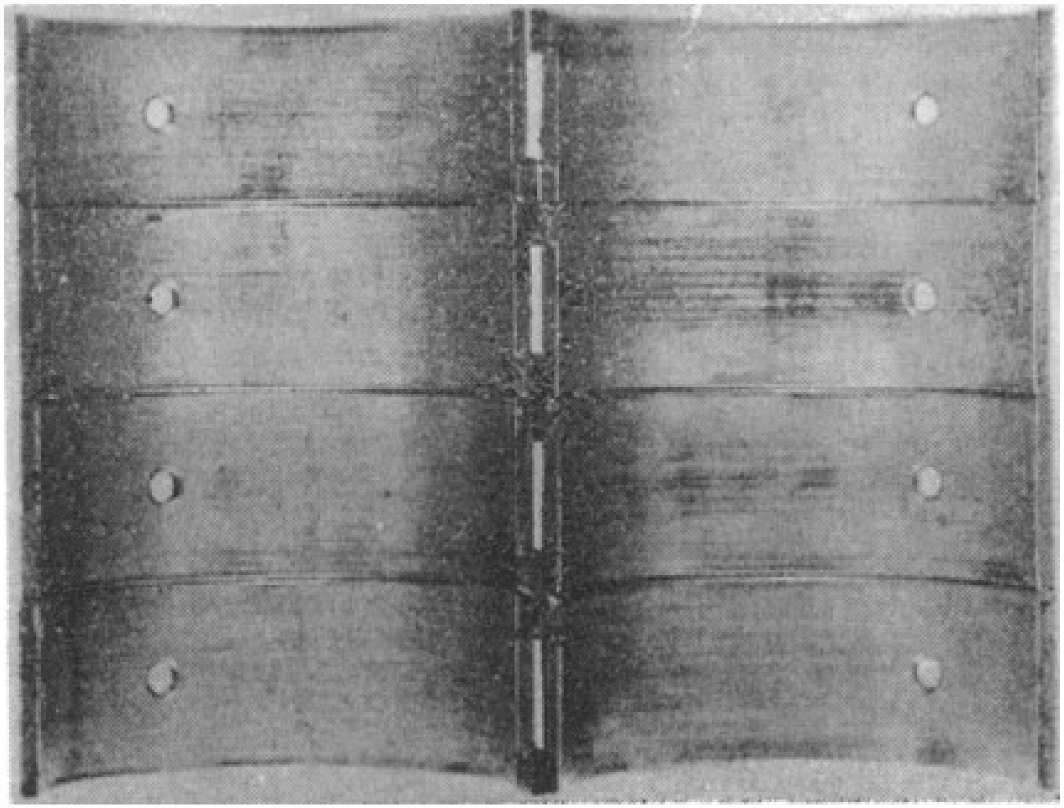
- Causes: Thermal cycling in service, causing plastic deformation, associated with the non uniform thermal expansion of tin crystals.

Faulty assembly



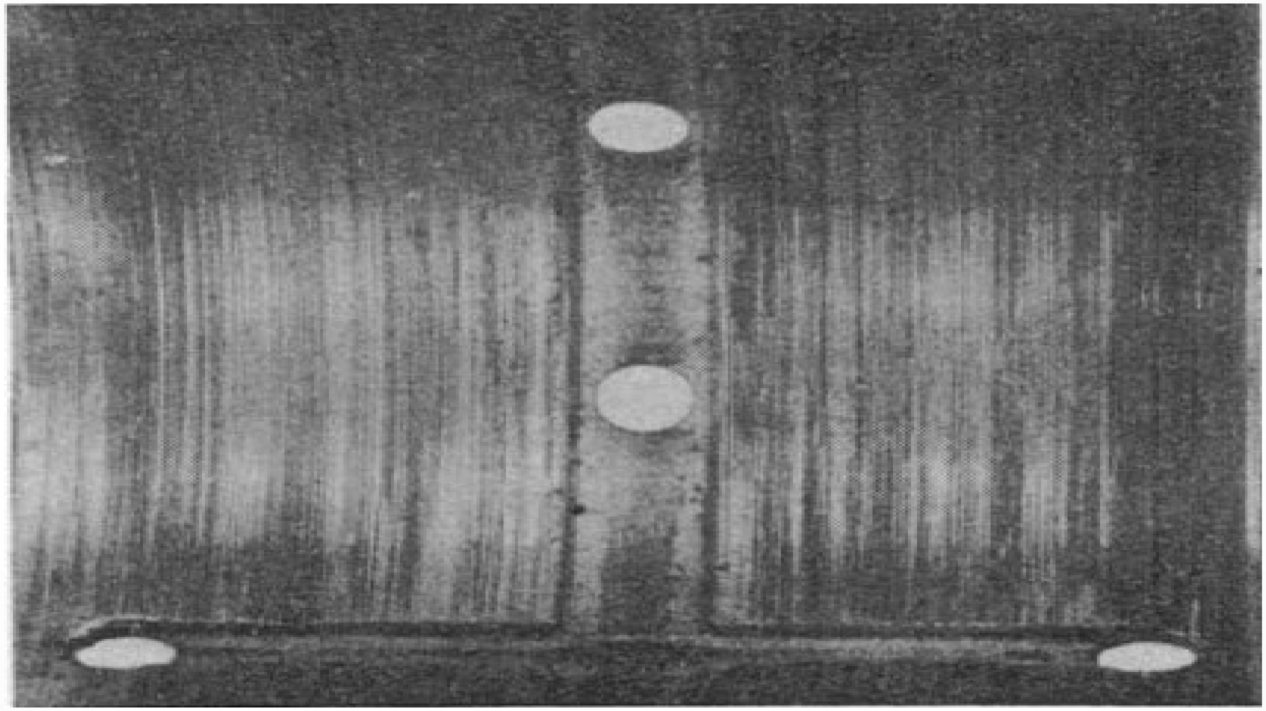
- Causes: Stagger at joint faces during assembly, due to excessive bolt clearances, or incorrect bolt disposition (bolts too far out).

Faulty assembly



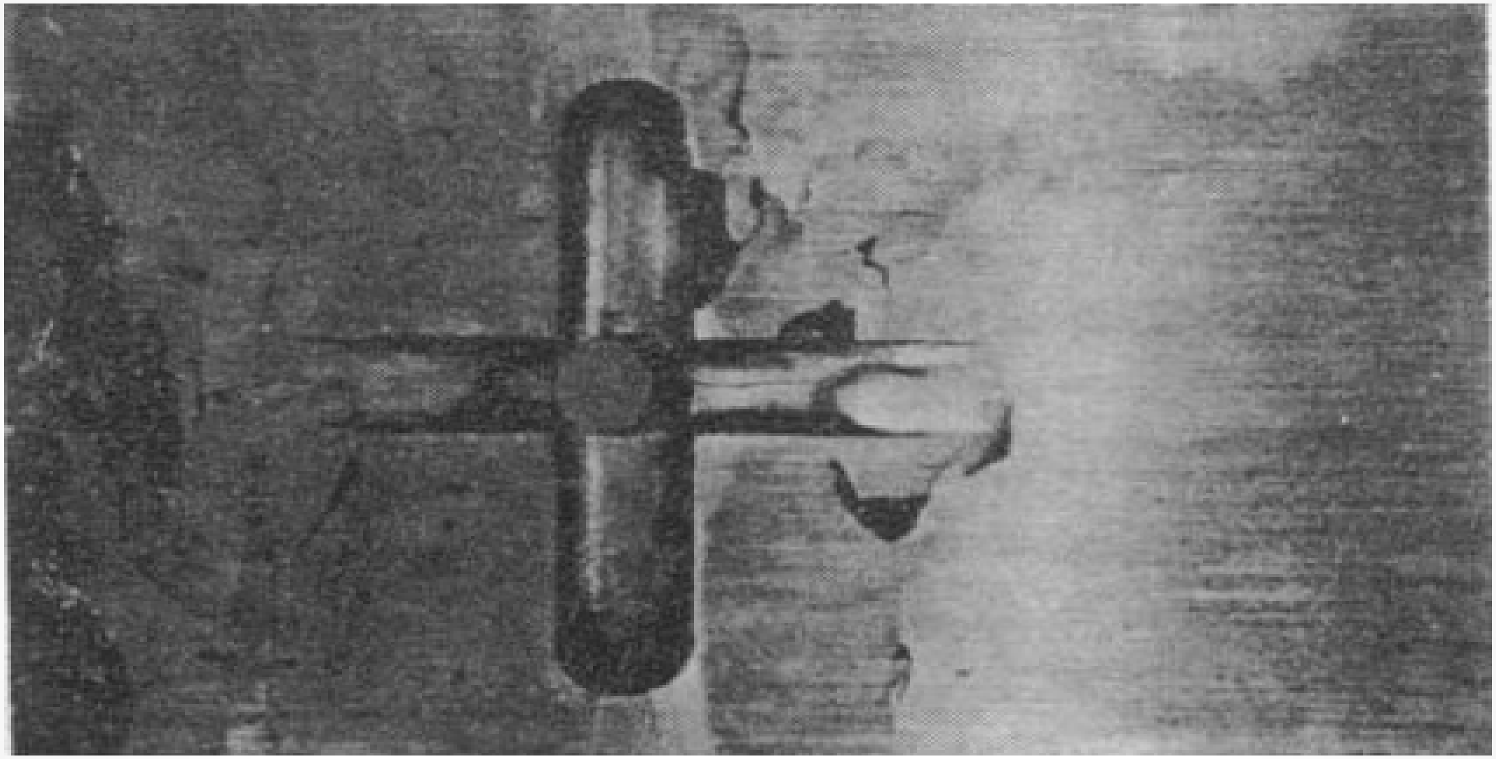
- Causes: Incorrect grinding of journal radii, causing fouling at fillets,
-

Incorrect journal grinding



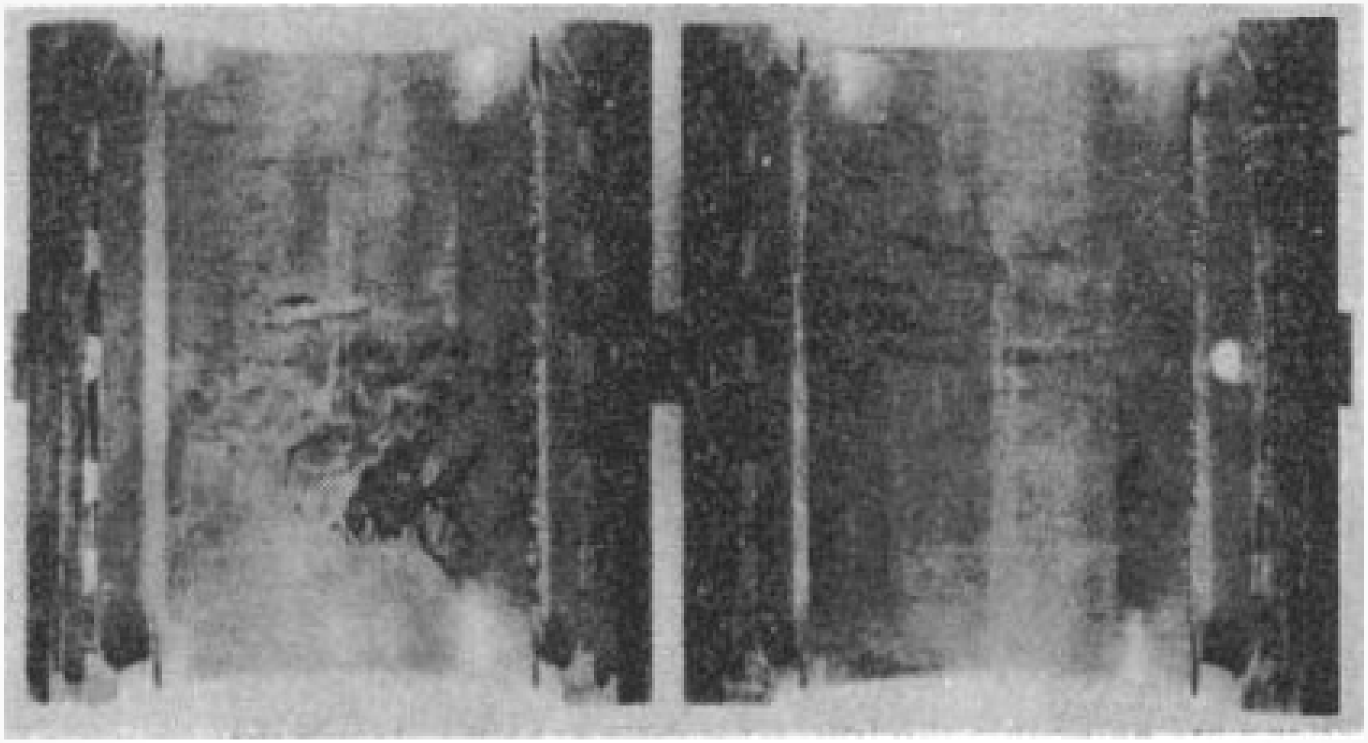
- Causes: Too coarse a surface finish, or in the case of SG iron shafts, the final grinding of journal in wrong direction relative to rotation in bearing.

Inadequate oil film thickness



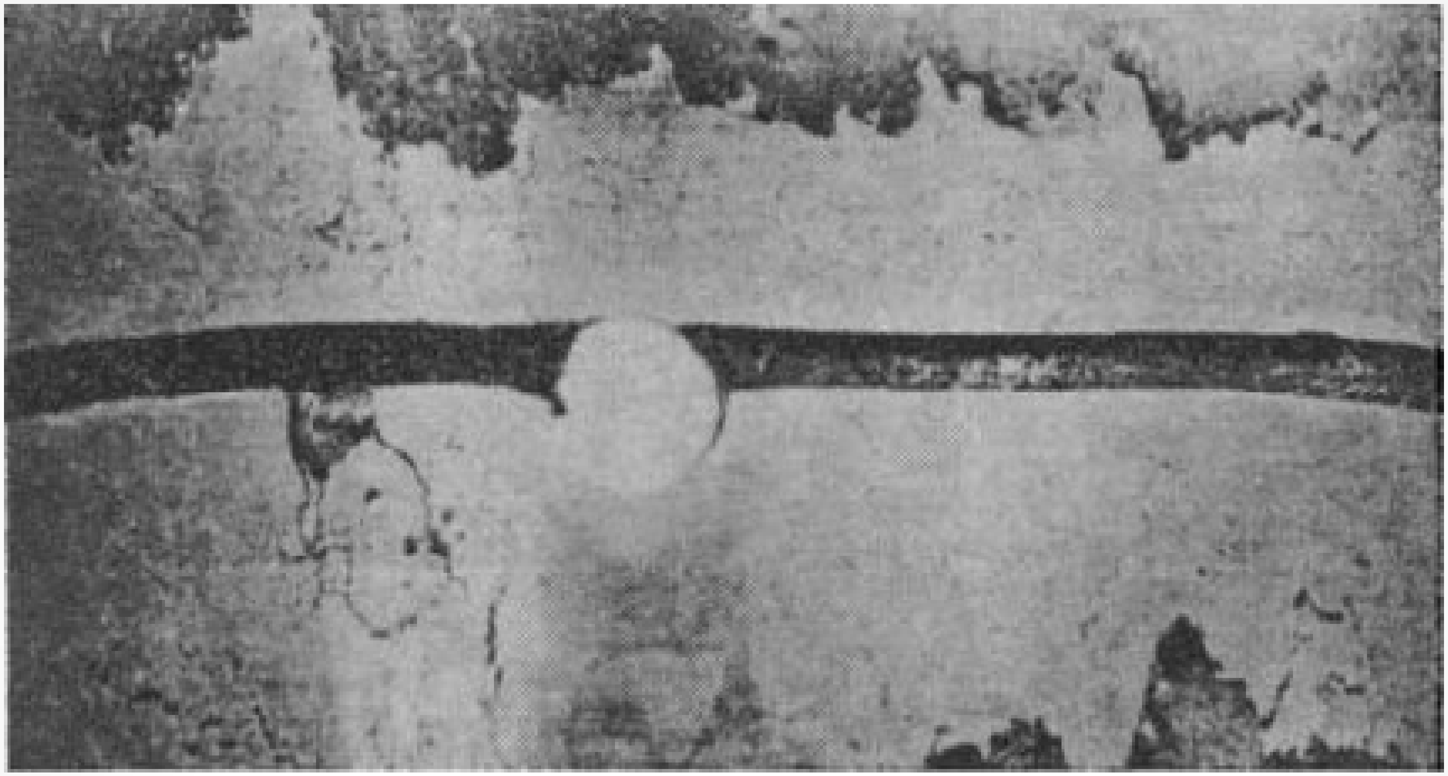
- Causes: Incorrect groove design, e.g. positioning a groove in the loaded area of the bearing.
-

Inadequqte lubrication



- Causes: Inadequate pump capacity or oil gallery or oilway dimensions Blockage or cessation of oil supply.

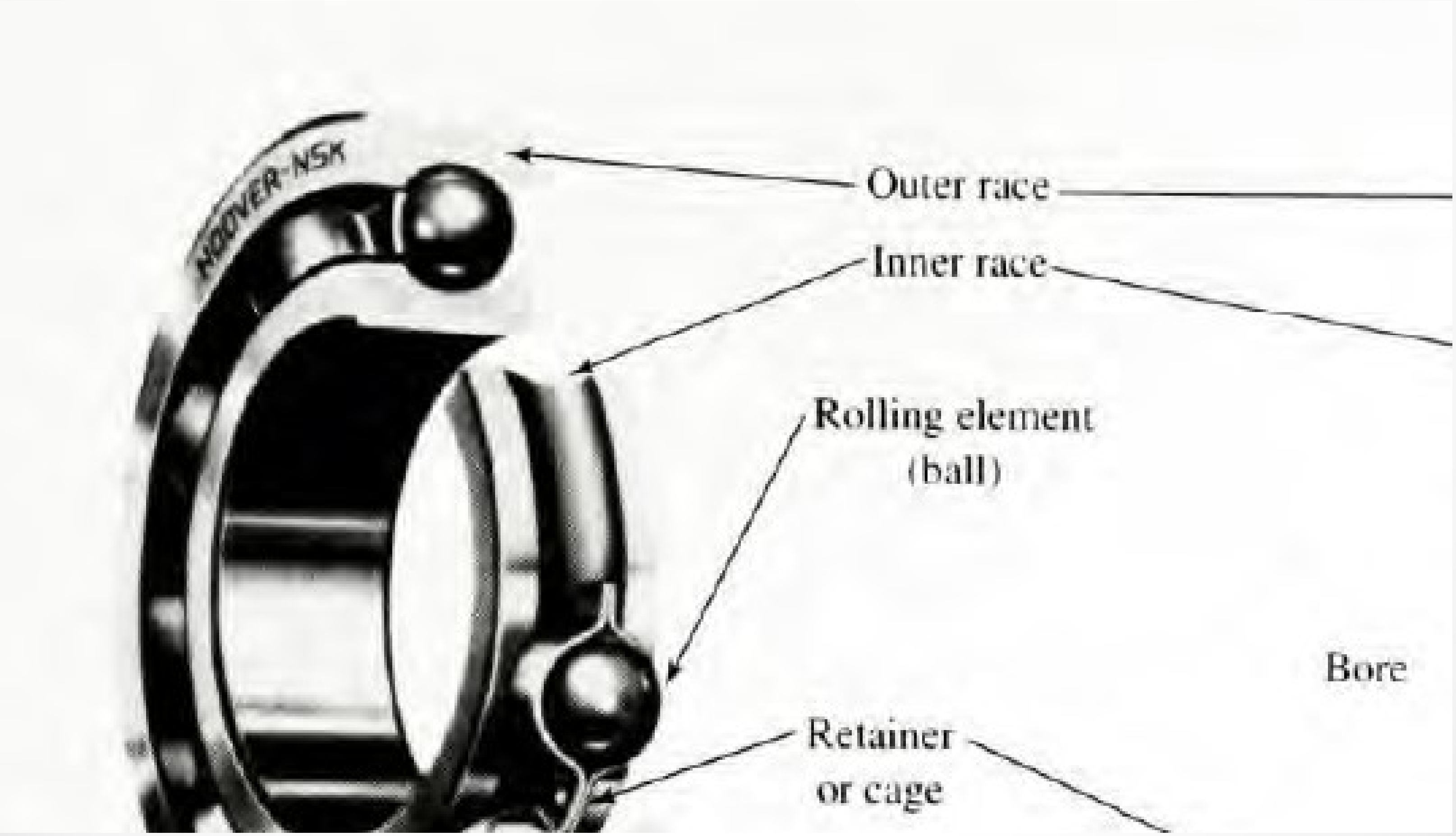
Bad bonding



- Causes: Poor tinning of shells; incorrect metallurgical control of lining technique.

ROLLING ELEMENT BEARINGS

Single-row, deep-groove ball bearing

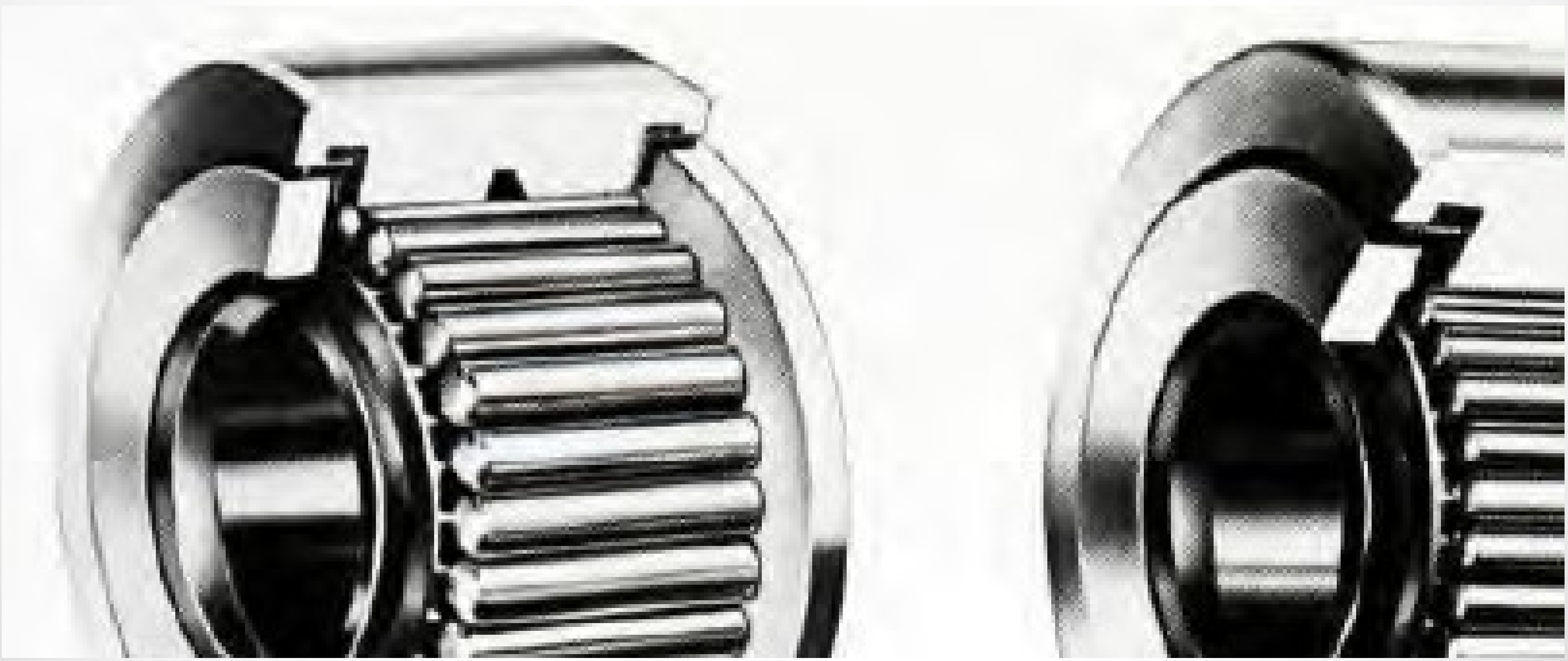


Double-row, deep-groove ball bearing

Angular contact ball bearing

Cylindrical roller bearing

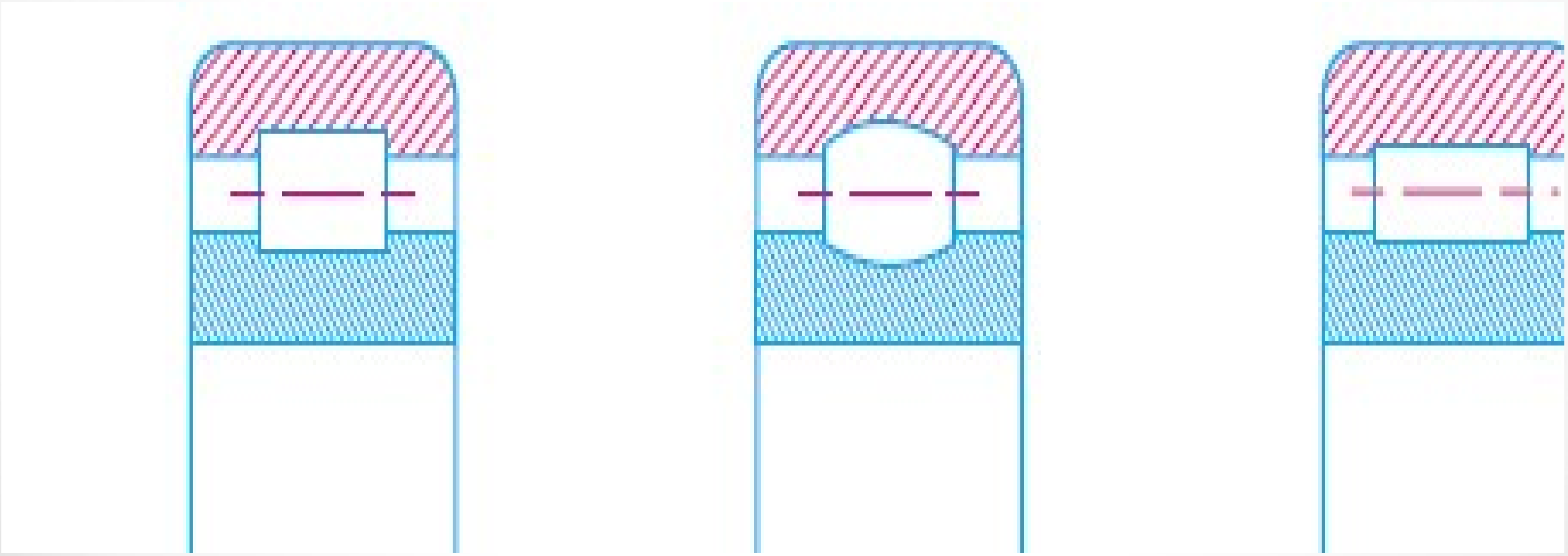
Single- and double-row needle bearings



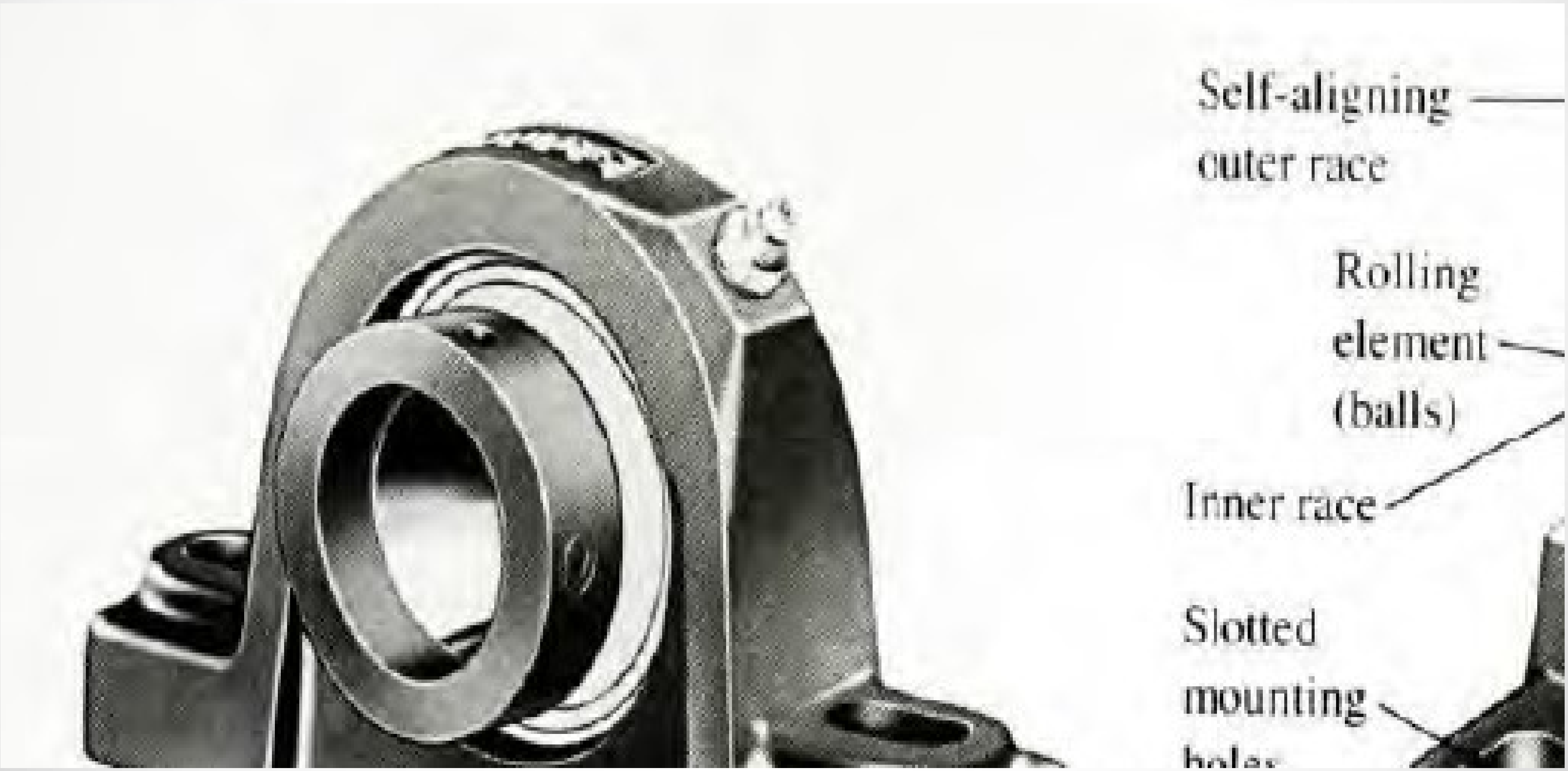
Spherical roller bearing

Tapered roller bearing

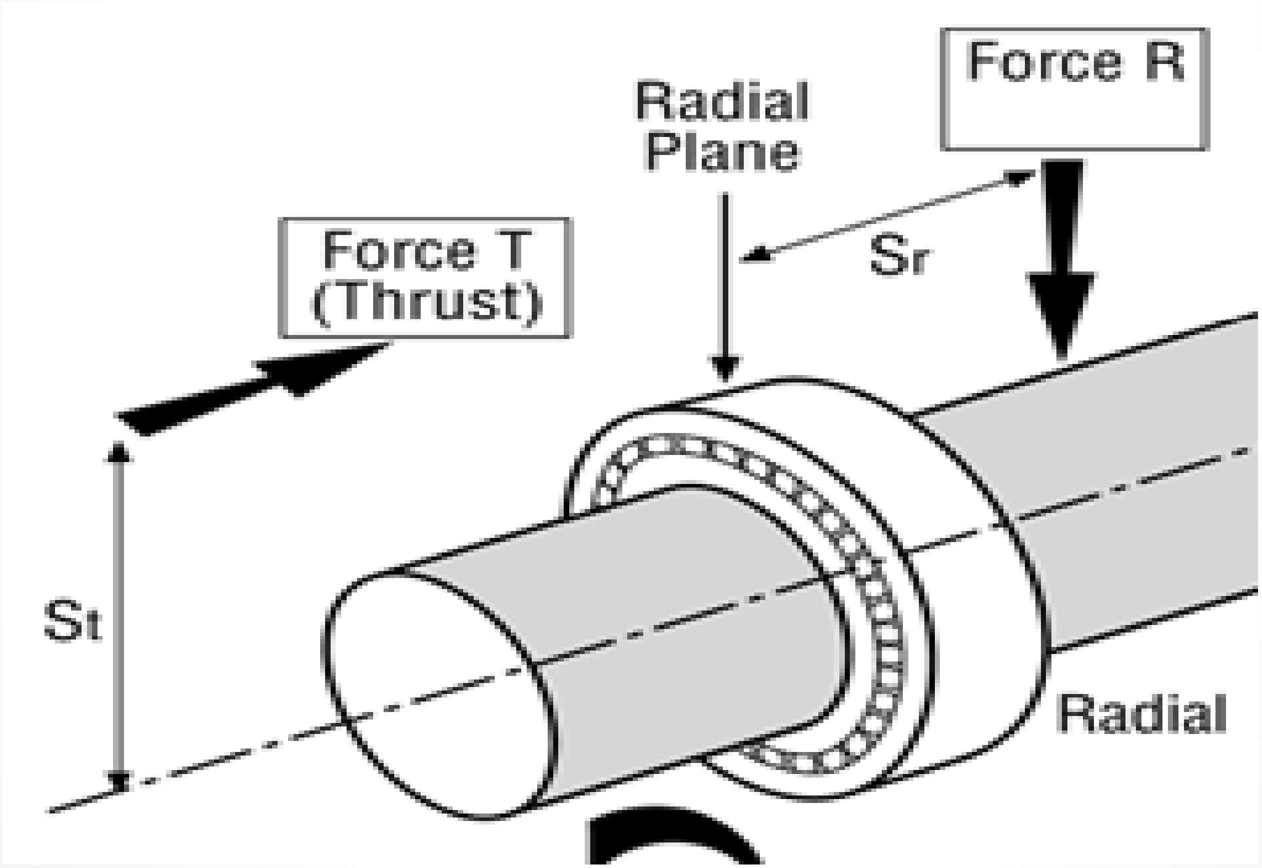
Sectional views



Ball bearing pillow block



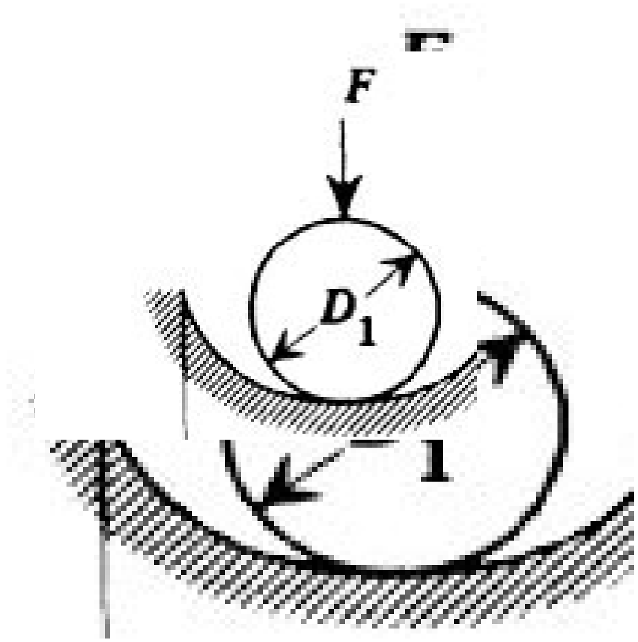
Loads on Bearing



Comparison of Rolling bearing types

Bearing type	Radial load capacity	Thr ca
Single-row, deep-groove ball	Good	Fai
Double-row, deep-groove ball	Excellent	Go
Angular contact	Good	Ex
Cylindrical roller	Excellent	Po
Needle	Excellent	P

Hertzian contact stress



Consider a solid sphere held in a Cup by a force F such that their point of contact expands into a circular area of radius, a

$$a = K_a \sqrt[3]{F}$$

$$\text{where } K_a = \left[3\,(1-\nu_1^2)/E_1 + (\right.$$

Where,

F= Applied force

V1 & V2= Poisons ratios for the sphere and cup

E1 & E2 = Elastic Modulii for sphere and cup

D1 and D2= diameters of sphere and cup

The maximum contact pressure occurs at the center point of the contact area

$$p_{max} = \frac{3}{}$$

TYPES OF LOADING AND STRESS RATIO

The primary factors to consider when specifying the type of loading to which a machine part is subjected are the manner of variation of the load and the resulting variation of stress with time. Stress variations are characterized by four key values:

1. Maximum stress, σ_{max}
2. Minimum stress, σ_{min}
3. Mean (average) stress, σ

The maximum and minimum stresses are usually computed from known information by stress analysis or finite-element methods, or they are measured using experimental stress analysis techniques. Then the mean and alternating stresses can be computed from

$$\sigma_m = (\sigma_{max} +$$

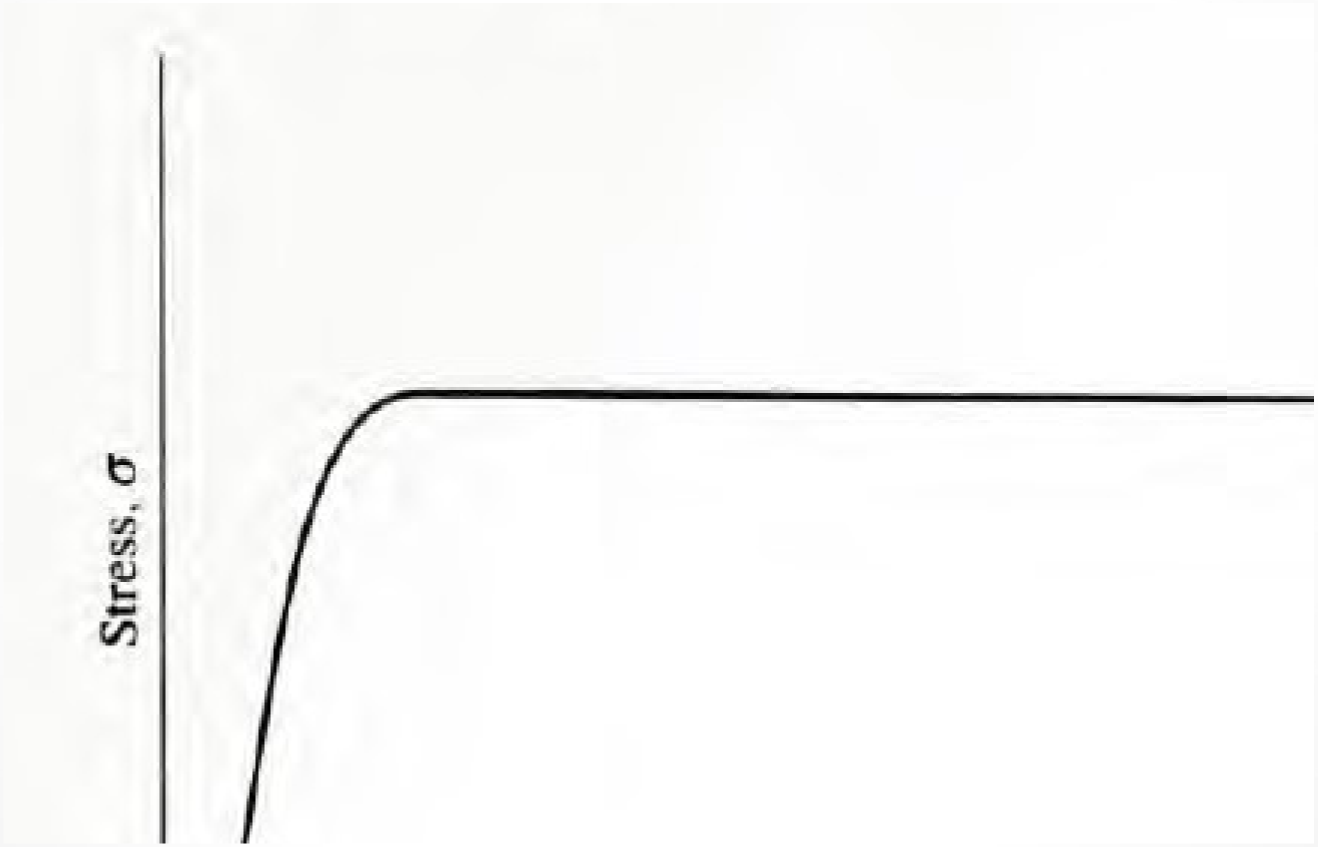
Stress Ratio

The behavior of a material under varying stresses is dependent on the manner of the variation. One method used to characterize the variation is called *stress ratio*. Two types of stress ratios are commonly used, defined as

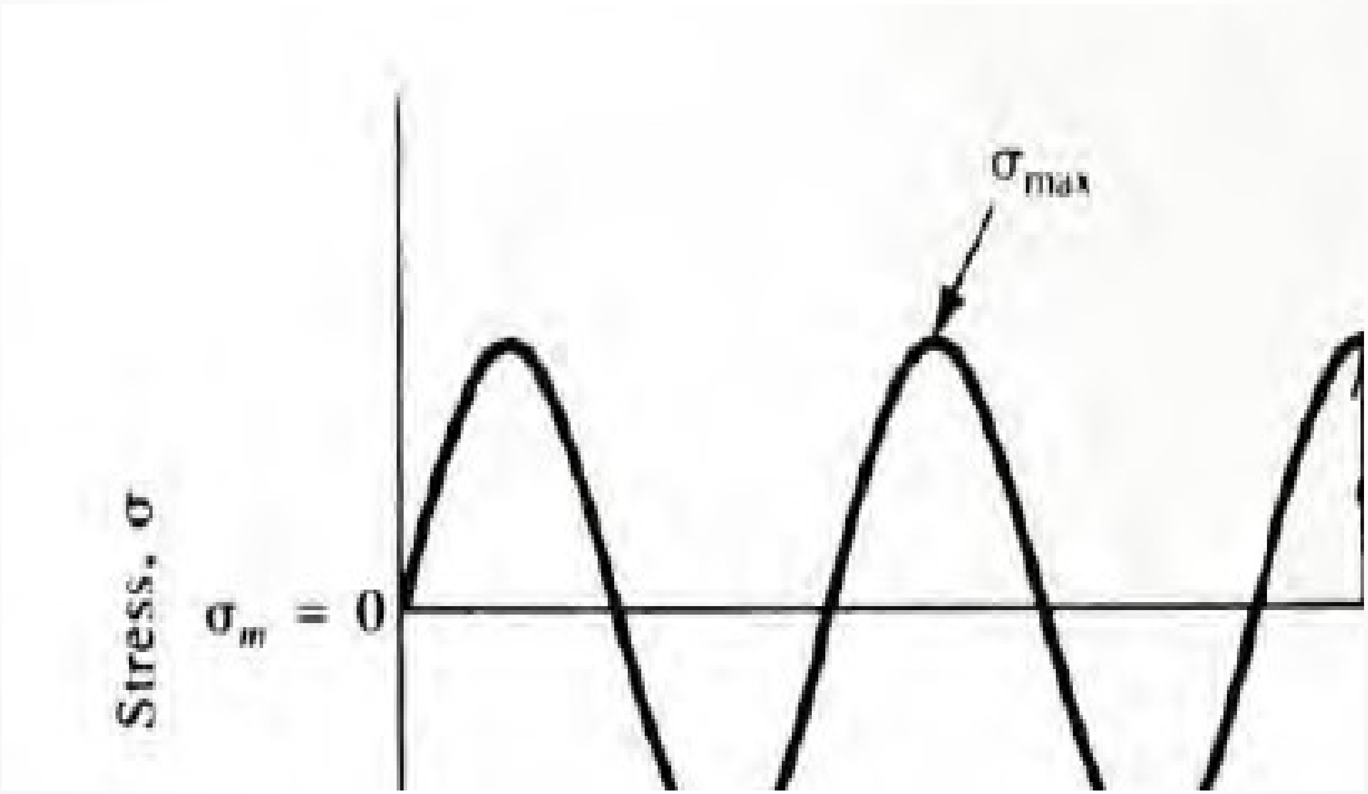
$$\text{Stress ratio } R = \frac{\text{minimum s}}{\text{maximum s}}$$

$$\text{Stress ratio } A = \frac{\text{alternating}}{}$$

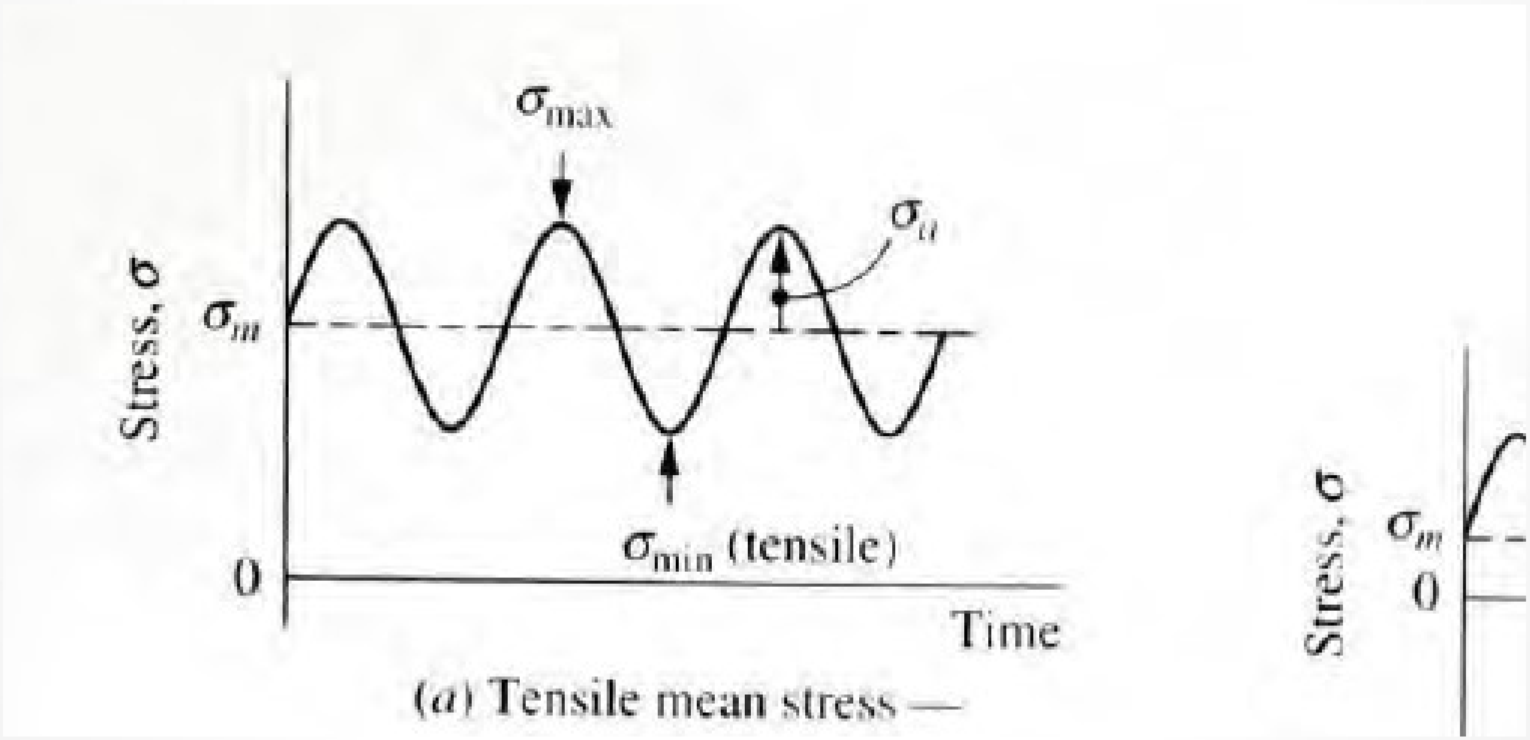
Static Stress



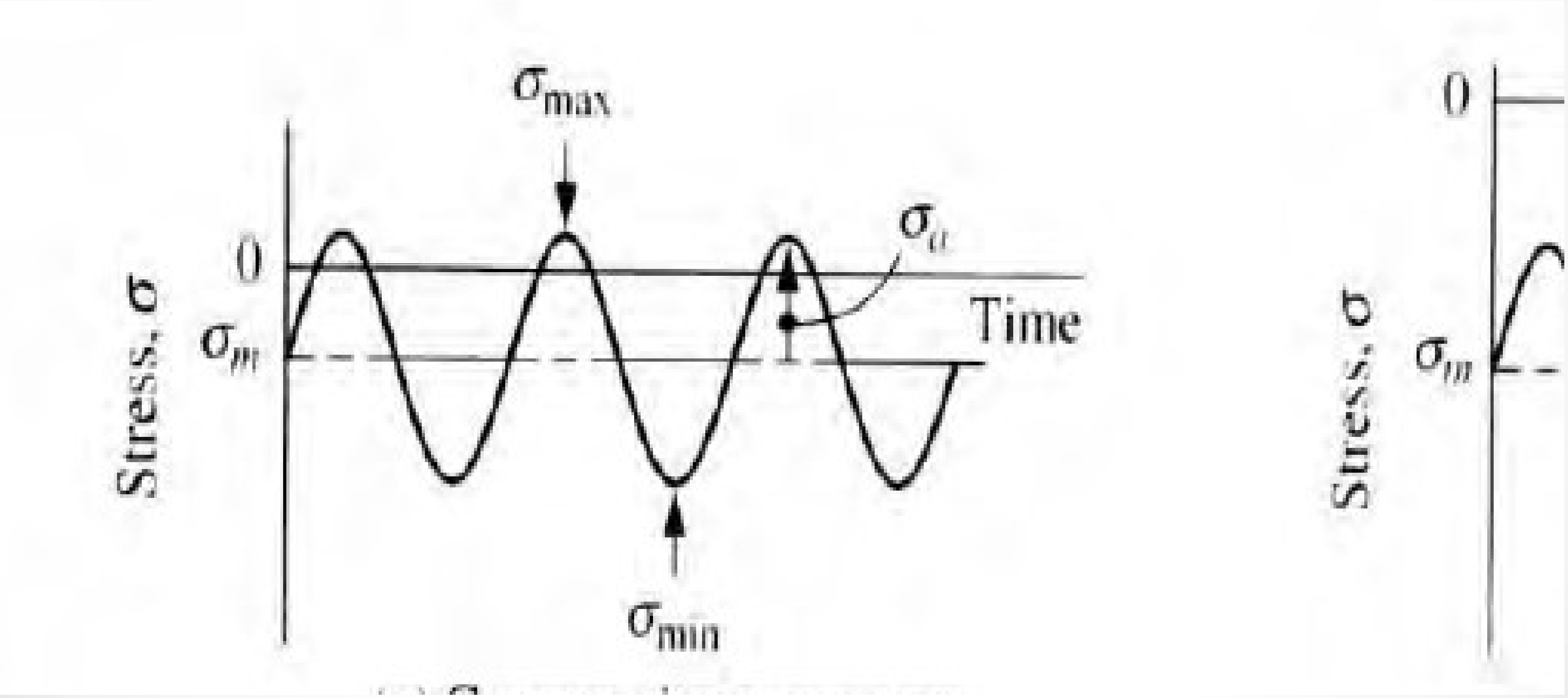
Repeated and Reversed Stress



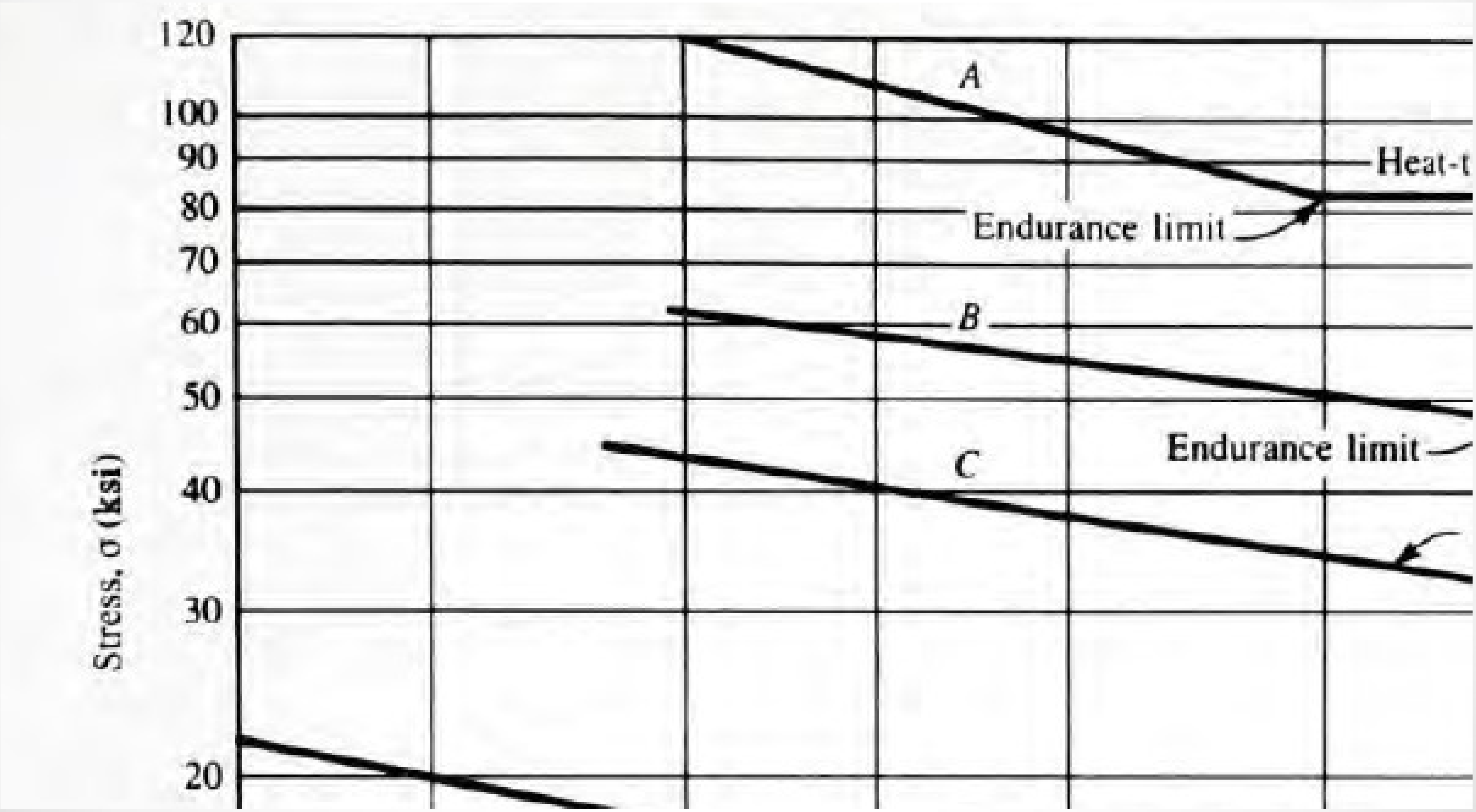
Fluctuating Stress



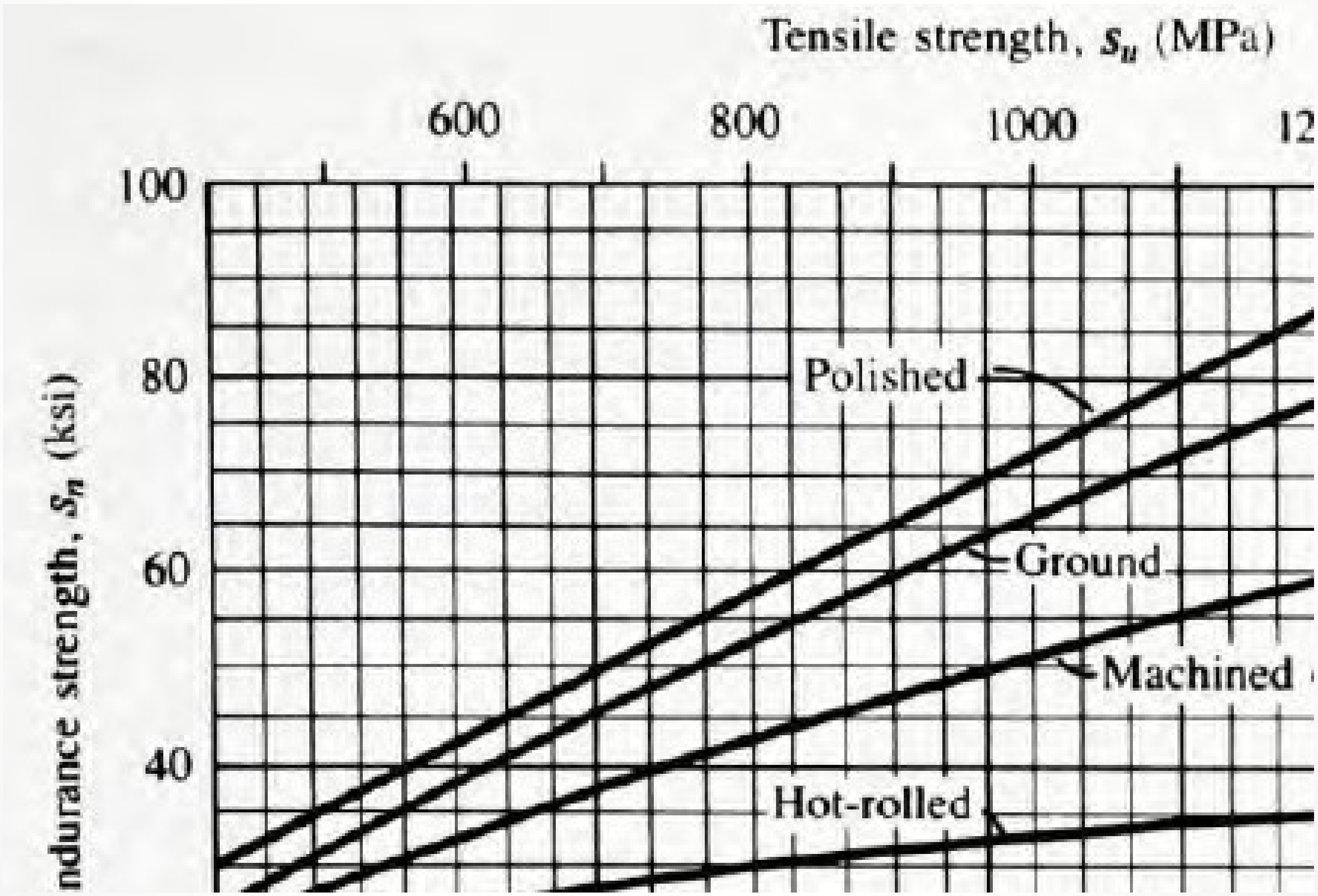
Fluctuating Stress continued…



Endurance Strength

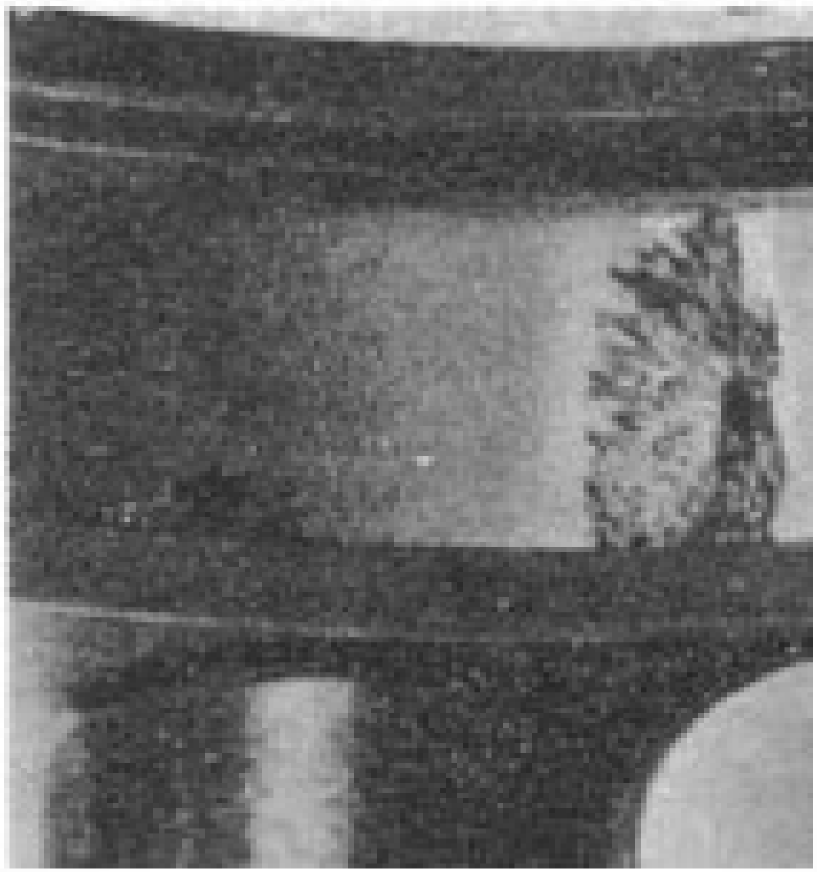


Endurance strength vs. tensile strength for wrought steel for various surface conditions

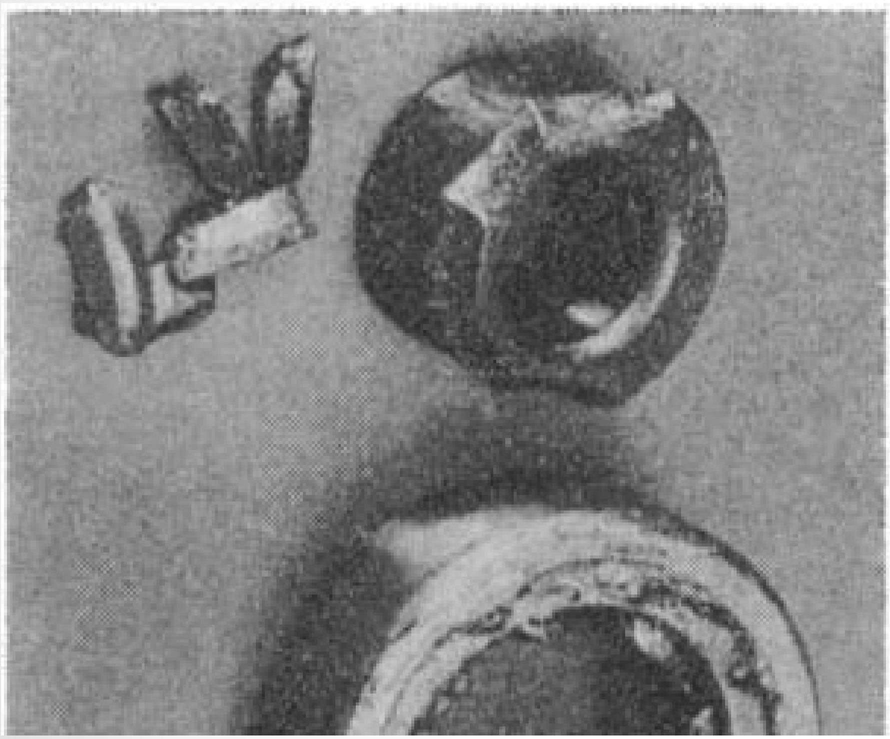


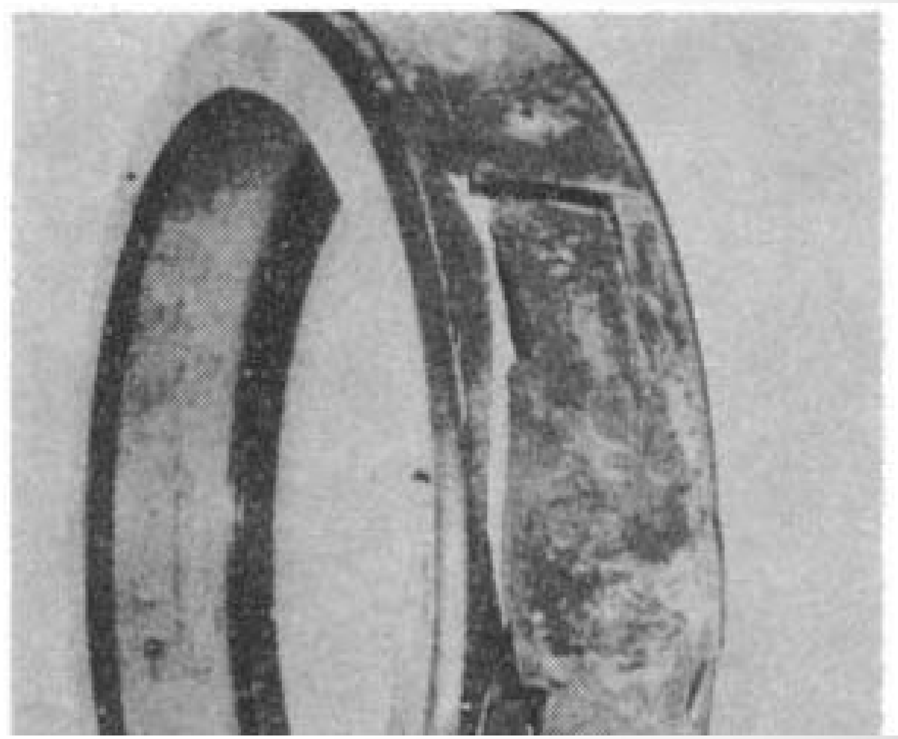
Rolling Bearing Failures

Fatigue failure



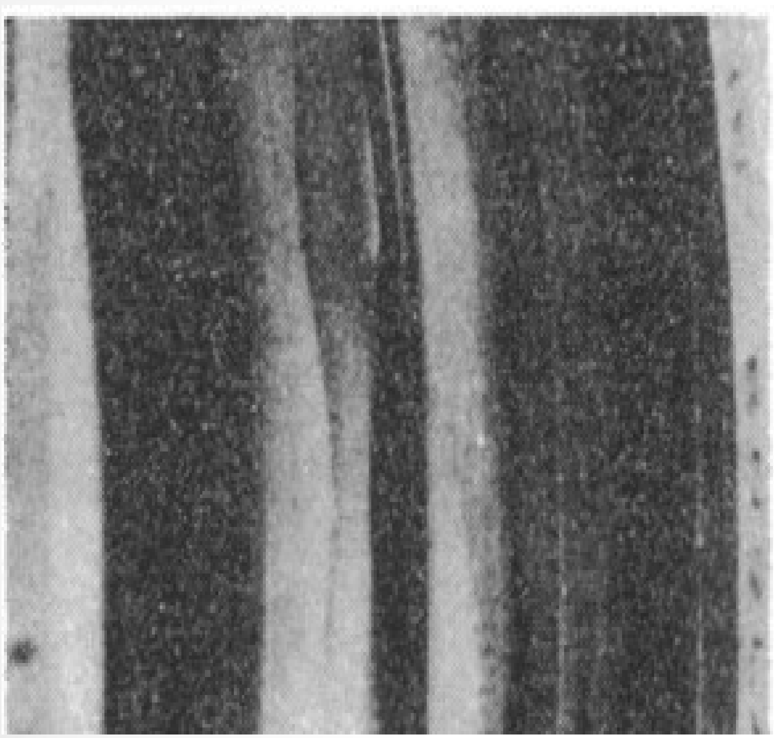
Failures continued….

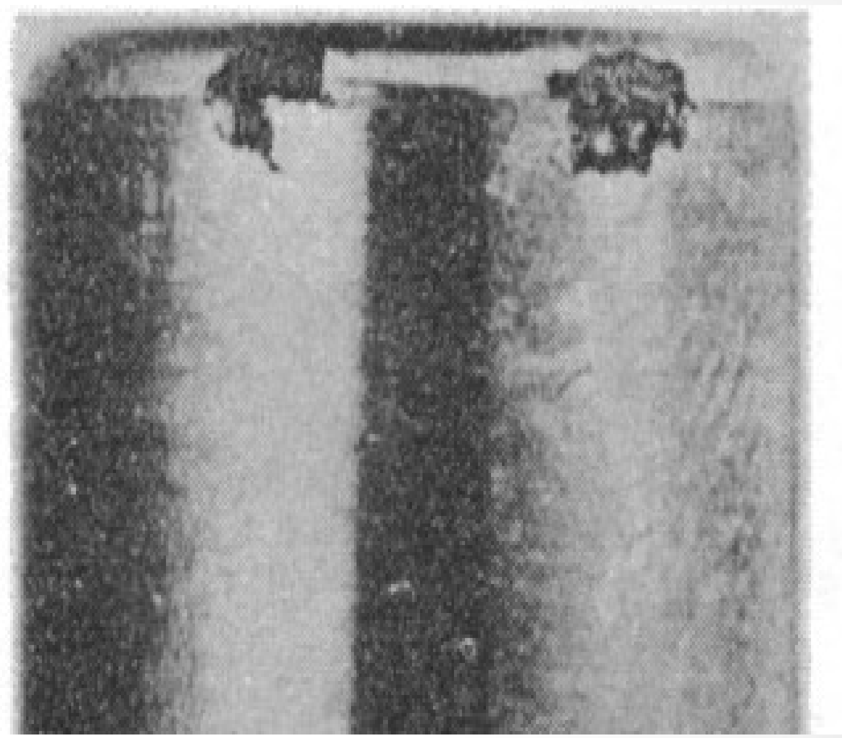




Fractured Flange Outer Race Fretting Inner Race Fretting

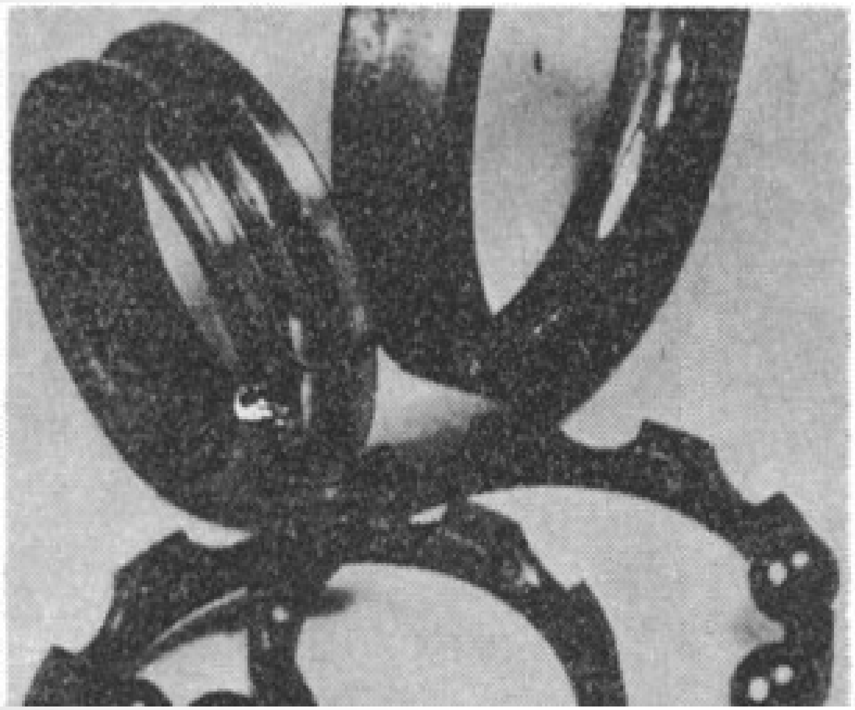
Failures continued….

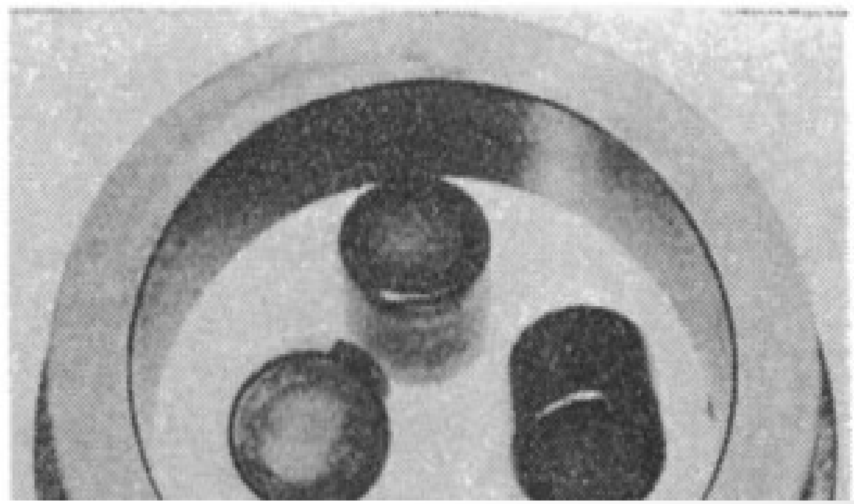




Uneven Wear Marks Roller End Collapse Roller End Chipping

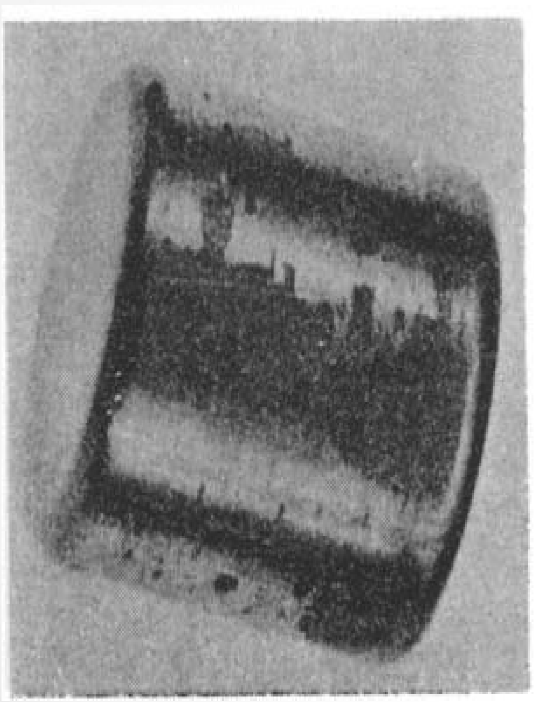
Failures continued….



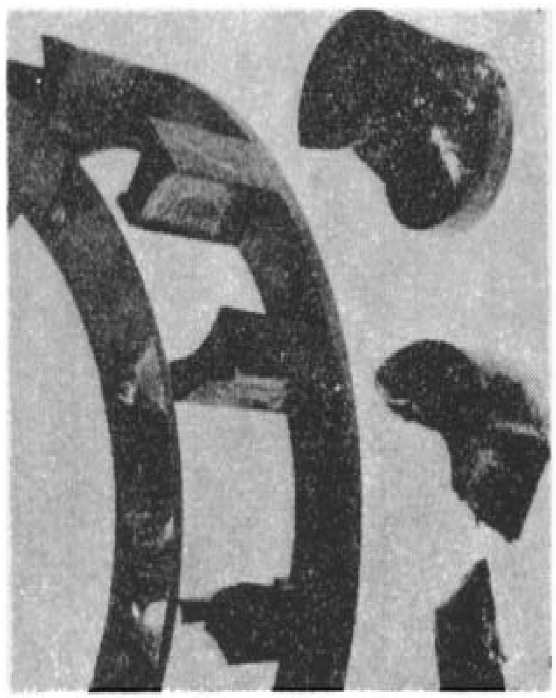


Overheating Smearing Abrasive Wear

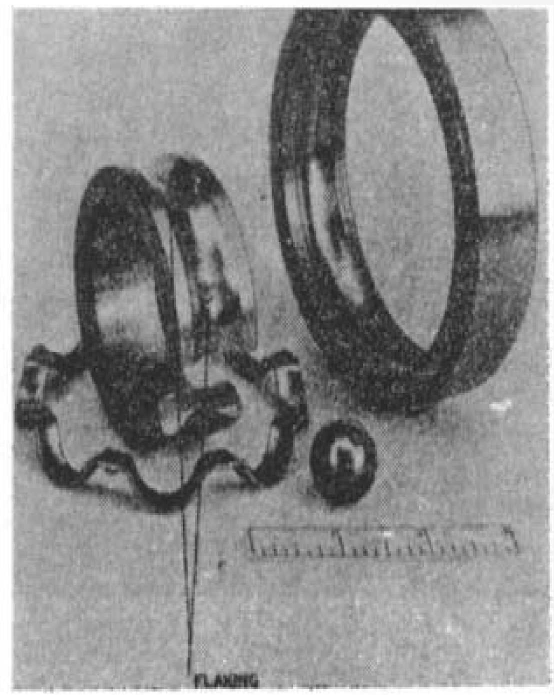
Failures continued….



Roller Peeling



Roller Breakage



Magnetic Damage

Comparison of bearing materials

			Mat
	Silicon nitride	52100 steel	4
Room-temperature hardness, HRC	78	62	
Room-temperature elastic modulus	45×10^6 psi 310 GPa	30×10^6 psi 207 GPa	2

LOAD/LIFE RELATIONSHIP

where $k = 3.00$ for ball bearings

Where

P1 & L1 = rated load and life

P2 & L2= Design load and life

Bearing Selection data for single row, deep groove ball bearings

A. *Series* 6200

Bearing number	Nominal bearing dimensions: d (mm)	d (in)	D (mm)	D (in)	B (mm)	B (in)	r^a (in)	Preferred shoulder diameter: Shaft (in)	Housing (in)	Bearing weight (lb)	Basic static load rating, C_o (lb)	Basic dynamic load rating, C (lb)
6200	10	0.3937	30	1.1811	9	0.3543	0.024	0.500	0.984	0.07	520	885
6201	12	0.4724	32	1.2598	10	0.3937	0.024	0.578	1.063	0.08	675	1180
6202	15	0.5906	35	1.3780	11	0.4331	0.024	0.703	1.181	0.10	790	1320
6203	17	0.6693	40	1.5748	12	0.4724	0.024	0.787	1.380	0.14	1010	1660
6204	20	0.7874	47	1.8504	14	0.5512	0.039	0.969	1.614	0.23	1400	2210
6205	25	0.9843	52	2.0472	15	0.5906	0.039	1.172	1.811	0.29	1610	2430
6206	30	1.1811	62	2.4409	16	0.6299	0.039	1.406	2.205	0.44	2320	3350
6207	35	1.3780	72	2.8346	17	0.6693	0.039	1.614	2.559	0.64	3150	4450
6208	40	1.5748	80	3.1496	18	0.7087	0.039	1.811	2.874	0.82	3650	5050
6209	45	1.7717	85	3.3465	19	0.7480	0.039	2.008	3.071	0.89	4150	5650
6210	50	1.9685	90	3.5433	20	0.7874	0.039	2.205	3.268	1.02	4650	6050
6211	55	2.1654	100	3.9370	21	0.8268	0.059	2.441	3.602	1.36	5850	7500
6212	60	2.3622	110	4.3307	22	0.8661	0.059	2.717	3.996	1.73	7250	9050
6213	65	2.5591	120	4.7244	23	0.9055	0.059	2.913	4.390	2.18	8000	9900
6214	70	2.7559	125	4.9213	24	0.9449	0.059	3.110	4.587	2.31	8800	10 800
6215	75	2.9528	130	5.1181	25	0.9843	0.059	3.307	4.783	2.64	9700	11 400
6216	80	3.1496	140	5.5118	26	1.0236	0.079	3.504	5.118	3.09	10 500	12 600
6217	85	3.3465	150	5.9055	28	1.1024	0.079	3.740	5.512	3.97	12 300	14 600
6218	90	3.5433	160	6.2992	30	1.1811	0.079	3.937	5.906	4.74	14 200	16 600
6219	95	3.7402	170	6.6929	32	1.2598	0.079	4.213	6.220	5.73	16 300	18 800
6220	100	3.9370	180	7.0866	34	1.3386	0.079	4.409	6.614	6.94	18 600	21 100
6221	105	4.1339	190	7.4803	36	1.4173	0.079	4.606	7.008	8.15	20 900	23 000
6222	110	4.3307	200	7.8740	38	1.4961	0.079	4.803	7.402	9.59	23 400	24 900
6224	120	4.7244	215	8.4646	40	1.5748	0.079	5.197	7.992	11.4	26 200	26 900

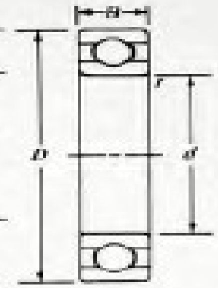


Table continued….

B. *Series 6300, continued*

Bearing number	Nominal bearing dimensions							Preferred shoulder diameter		Bearing weight	Basic static load rating, C_o	Basic dynamic load rating, C
	d		*D*		*B*		r^*	Shaft	Housing			
	mm	in	mm	in	mm	in	in	in	in	lb	lb	lb
6316	80	3.1496	170	6.6929	39	1.5354	0.079	3.622	6.220	7.93	18 300	21 300
6317	85	3.3465	180	7.0866	41	1.6142	0.098	3.898	6.535	9.37	20 400	22 900
6318	90	3.5433	190	7.4803	43	1.6929	0.098	4.094	6.929	10.8	22 300	24 700
6319	95	3.7402	200	7.8740	45	1.7717	0.098	4.291	7.323	12.5	24 900	26 400
6320	100	3.9370	215	8.4646	47	1.8504	0.098	4.488	7.913	15.3	29 800	30 000
6321	105	4.1339	225	8.8583	49	1.9291	0.098	4.685	8.307	17.9	32 500	31 700
6322	110	4.3307	240	9.4488	50	1.9685	0.098	4.882	8.898	21.0	38 000	35 500
6324	120	4.7244	260	10.2362	55	2.1654	0.098	5.276	9.685	27.6	38 500	36 000
6326	130	5.1181	280	11.0236	58	2.2835	0.118	5.827	10.315	40.8	44 500	39 500
6328	140	5.5118	300	11.8110	62	2.4409	0.118	6.220	11.102	48.5	51 000	43 500
6330	150	5.9055	320	12.5984	65	2.5591	0.118	6.614	11.890	57.3	58 000	47 500
6332	160	6.2992	340	13.3858	68	2.6772	0.118	7.008	12.677	58	58 500	48 000
6334	170	6.6929	360	14.1732	72	2.8346	0.118	7.402	13.465	84	73 500	56 500
6336	180	7.0866	380	14.9606	75	2.9528	0.118	7.795	14.252	98	84 000	61 500
6338	190	7.4803	400	15.7480	78	3.0709	0.157	8.346	14.882	112	84 000	61 500
6340	200	7.8740	420	16.5354	80	3.1496	0.157	8.740	15.669	127	91 500	65 500

Rated Life and Basic dynamic load rating

- The rated life is the standard means of reporting the results of many tests of bearings of a given design. It represents the life that 90% of the bearings would achieve successfully at a rated load.
- It also represents the life that 10% of the bearings would not achieve. The rated life is thus typically referred to as the L_{10} *life* at the rated load.
- Now the *basic dynamic load* rating can be defined as that load to which the bearings can be subjected while achieving a rated life (L_{10}) of 1 million revolutions (rev).

Problem: A catalog lists the basic dynamic load rating for a ball bearing to be 7050 lb for a rated life of 1 million rev. What would be the expected L_{10} life of the bearing if it were subjected to a load of 3500 lb?

Solution:

$P_1 = C = 7050 \text{ lb}$ (basi

$P_2 = P_d = 3500 \text{ lb}$ (desi

$L_1 = 10^6 \text{ rev}$ (L_{10} l

$k = 3$ (ball

Then letting the life, L_2, be called the *design life*, L_d,

Procedure for computing the required basic dynamic load rating *C* for a given design load P_d and a given design life L_d

We have already discussed

$$\frac{L_2}{} = \left(\frac{P}{} \right.$$

$$L_2 = L_d = L$$

If the reported load data in the manufacturer's literature is for 10^6 revolutions the above equation can be written as

$$L_d = (C/P_d)^k$$

The required C for a given design load and life would be

$$C = P_d (L_d/1$$

Now, for a specified design life in hours, and a known speed of rotation in rpm, the number of design revolutions for the bearing would be

$$L_d = (h)(rpm)(60$$

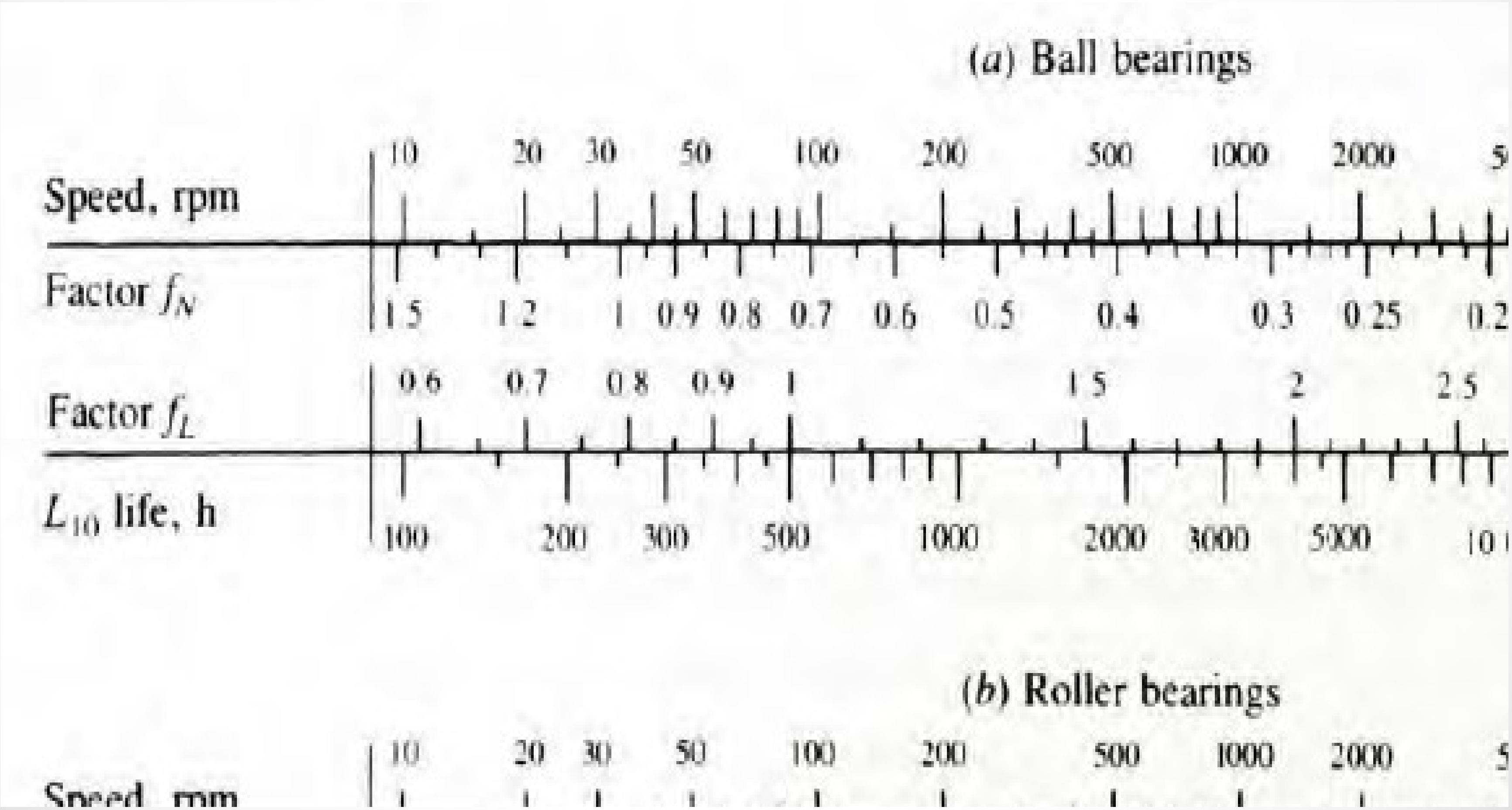
Recommended design life for bearings

Application
Domestic appliances
Aircraft engines
Automotive
Agricultural equipment
Elevators, industrial fans, multipurpose gearing

- The rated life of 1 million rev would be achieved by a shaft rotating at $33\frac{1}{3}$ rpm for 500 h.
- If the actual speed or desired life is different from these two values, a speed factor f_N and a life factor f_L can be determined from charts shown in the next slide.
- The factors account for the load/life relationship.
- The required basic dynamic load rating, C, for a bearing to carry a design load, P_d, would then be

$$C = P_d f_L$$

Life and speed factors for ball and roller bearings



Problem: Compute the required basic dynamic load rating, C for a ball bearing to carry a radial load of 650 lb from a shaft rotating at 600 rpm that is part of an assembly conveyor in a manufacturing plant.

Solution:

let's select a design life of 30 000 h from table .

Then *Ld* is

$$L_d = (\mathrm{h})(\mathrm{rpm})(6$$

$$L_d = (30\,000\ \mathrm{h})(600\ \mathrm{rpm})(60\ \mathrm{min/h})$$

Also, Dynamic load rating

$$C = P_d\,(L_d/1$$

$$C = 650(1.08 \times 10^9/10^6)^1$$

- If we solve this problem by using the charts of speed and life factors, we have

$$f_N = 0.381 \quad (\text{for } 600 \text{ rp}$$

$$f_L = 3.90 \quad (\text{for } 30\,000\text{-}$$

- This compares closely with the value of 6670 lb found previously.

Procedure for Selecting a Bearing—Radial Load Only

Step 1: Specify the design load on the bearing, usually called *equivalent load*. The method of determining the equivalent load when only a radial load, *R*, is applied takes into account whether the inner or the outer race rotates.

$$P = 1$$

- where V = *rotation factor*

  =1.0 if the inner race of the bearing rotates,

  = 1.2 if the outer race rotates

Step 2: Determine the minimum acceptable diameter of the shaft that will limit the bore size of the bearing.

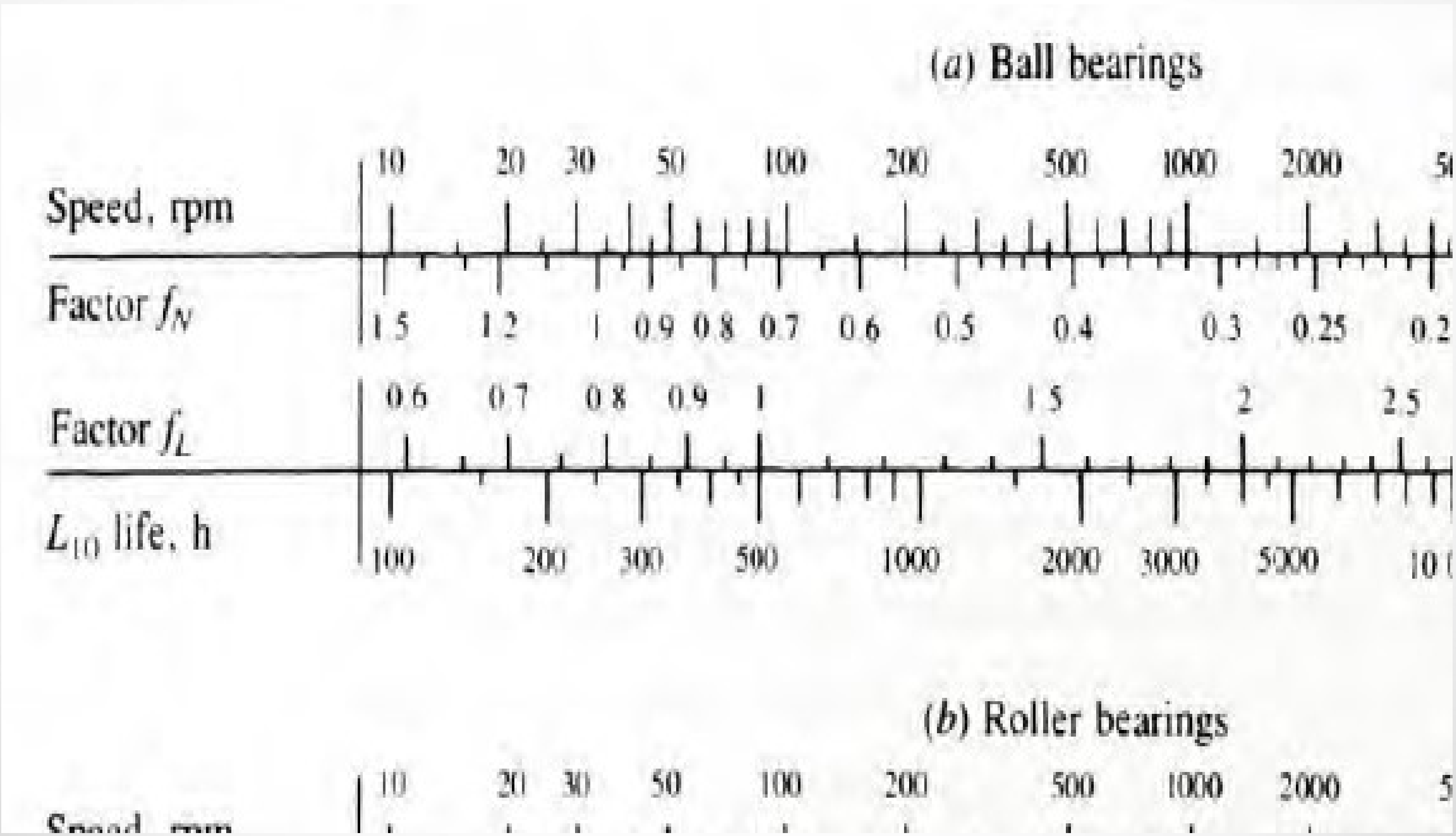
Step 3: Select the type of bearing, using following table

Bearing type	Radial load capacity	Th c
Single-row, deep-groove ball	Good	Fa
Double-row, deep-groove ball	Excellent	G
Angular contact	Good	E
Cylindrical roller	Excellent	P

Step 4: Specify the design life of the bearing, using the following table

Application
Domestic appliances
Aircraft engines
Automotive
Agricultural equipment
Elevators, industrial fans, multipurpose gearing

Step 5: Determine the speed factor and the life factor if such tables are available for the selected type of bearing.



Step 6: Compute the required basic dynamic load rating C from following equations

$$\frac{L_2}{L_1} = \left(\frac{P}{P} \right.$$

$$C = P_d (L_d /$$

or

$$C = P_d$$

Step 7: Identify a set of candidate bearings that have the required basic dynamic load rating.

Step 8: Select the bearing having the most convenient geometry, also considering its cost and availability.

Step 9: Determine mounting conditions, such as shaft seat diameter and tolerance, housing bore diameter and tolerance, means of locating the bearing axially, and special needs such as seals or shields.

Problem: Select a single-row, deep-groove ball bearing to carry 650 lb of pure radial load from a shaft that rotates at 600 rpm. The design life is to be 30 000 h. The bearing is to be mounted on a shaft with a minimum acceptable diameter of 1.48 in.

Solution:

- Note that this is a pure radial load and the inner race is to be pressed onto the shaft and rotates with it. Therefore, the rotation factor V= 1.0 in
- Therefore From equation

$$P =$$

P=R

the design load is equal to the radial load.

We know,

$$L_d = (\mathrm{h})(\mathrm{rpm})(6($$

$$L_d = (30\,000\ \mathrm{h})(600\ \mathrm{rpm})(60\ \mathrm{min/h})$$

Also, Dynamic load rating

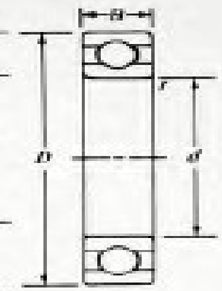
$$C = P_d\,(L_d/1$$

$$C = 650(1.08 \times 10^9/10^6)^1$$

- **Giving design data for two classes of bearings, we find from table that we could use a bearing 6211 or a bearing 6308.**

A. *Series* 6200

Bearing number	Nominal bearing dimensions: *d*		*D*		*B*		r^*	Preferred shoulder diameter: Shaft	Housing	Bearing weight	Basic static load rating, C_s	Basic dynamic load rating, *C*
	mm	in	mm	in	mm	in	in	in	in	lb	lb	lb
6200	10	0.3937	30	1.1811	9	0.3543	0.024	0.500	0.984	0.07	520	885
6201	12	0.4724	32	1.2598	10	0.3937	0.024	0.578	1.063	0.08	675	1180
6202	15	0.5906	35	1.3780	11	0.4331	0.024	0.703	1.181	0.10	790	1320
6203	17	0.6693	40	1.5748	12	0.4724	0.024	0.787	1.380	0.14	1010	1660
6204	20	0.7874	47	1.8504	14	0.5512	0.039	0.969	1.614	0.23	1400	2210
6205	25	0.9843	52	2.0472	15	0.5906	0.039	1.172	1.811	0.29	1610	2430
6206	30	1.1811	62	2.4409	16	0.6299	0.039	1.406	2.205	0.44	2320	3350
6207	35	1.3780	72	2.8346	17	0.6693	0.039	1.614	2.559	0.64	3150	4450
6208	40	1.5748	80	3.1496	18	0.7087	0.039	1.811	2.874	0.82	3650	5050
6209	45	1.7717	85	3.3465	19	0.7480	0.039	2.008	3.071	0.89	4150	5650
6210	50	1.9685	90	3.5433	20	0.7874	0.039	2.205	3.268	1.02	4650	6050
6211	55	2.1654	100	3.9370	21	0.8268	0.059	2.441	3.602	1.36	5850	7500
6212	60	2.3622	110	4.3307	22	0.8661	0.059	2.717	3.996	1.73	7250	9050
6213	65	2.5591	120	4.7244	23	0.9055	0.059	2.913	4.390	2.18	8000	9900
6214	70	2.7559	125	4.9213	24	0.9449	0.059	3.110	4.587	2.31	8800	10 800
6215	75	2.9528	130	5.1181	25	0.9843	0.059	3.307	4.783	2.64	9700	11 400
6216	80	3.1496	140	5.5118	26	1.0236	0.079	3.504	5.118	3.09	10 500	12 600
6217	85	3.3465	150	5.9055	28	1.1024	0.079	3.740	5.512	3.97	12 300	14 600
6218	90	3.5433	160	6.2992	30	1.1811	0.079	3.937	5.906	4.74	14 200	16 600
6219	95	3.7402	170	6.6929	32	1.2598	0.079	4.213	6.220	5.73	16 300	18 800
6220	100	3.9370	180	7.0866	34	1.3386	0.079	4.409	6.614	6.94	18 600	21 100
6221	105	4.1339	190	7.4803	36	1.4173	0.079	4.606	7.008	8.15	20 900	23 000
6222	110	4.3307	200	7.8740	38	1.4961	0.079	4.803	7.402	9.59	23 400	24 900
6224	120	4.7244	215	8.4646	40	1.5748	0.079	5.197	7.992	11.4	26 200	26 900



A. *Series* 6200, continued

Bearing number	Nominal bearing dimensions							Preferred shoulder diameter		Bearing weight	Basic static load rating, C_o	Basic dynamic load rating, C
	d		D		B		r*	Shaft	Housing			
	mm	in	mm	in	mm	in	in	in	in	lb	lb	lb
6226	130	5.1181	230	9.0551	40	1.5748	0.098	5.669	8.504	12.7	29 100	28 700
6228	140	5.5118	250	9.8425	42	1.6535	0.098	6.063	9.291	19.5	29 300	28 700
6230	150	5.9055	270	10.6299	45	1.7717	0.098	6.457	10.079	25.3	32 500	30 000
6232	160	6.2992	290	11.4173	48	1.8898	0.098	6.850	10.886	32.0	35 500	32 000
6234	170	6.6929	310	12.2047	52	2.0472	0.118	7.362	11.535	38.5	43 000	36 500
6236	180	7.0866	320	12.5984	52	2.0472	0.118	7.758	11.929	41.0	46 500	39 000
6238	190	7.4803	340	13.3858	55	2.1654	0.118	8.150	12.717	50.5	54 500	44 000
6240	200	7.8740	360	14.1732	58	2.2835	0.118	8.543	13.504	61.5	60 000	46 500

B. *Series* 6300

Bearing number	d (mm)	d (in)	D (mm)	D (in)	B (mm)	B (in)	r* (in)	Shaft (in)	Housing (in)	Bearing weight (lb)	C_o (lb)	C (lb)
6300	10	0.3937	35	1.3780	11	0.4331	0.024	0.563	1.181	0.12	805	1400
6301	12	0.4724	37	1.4567	12	0.4724	0.039	0.656	1.220	0.13	990	1680
6302	15	0.5906	42	1.6535	13	0.5118	0.039	0.781	1.417	0.18	1200	1980
6303	17	0.6693	47	1.8504	14	0.5512	0.039	0.875	1.614	0.25	1460	2360
6304	20	0.7874	52	2.0472	15	0.5906	0.039	1.016	1.772	0.32	1730	2760
6305	25	0.9843	62	2.4409	17	0.6693	0.039	1.220	2.165	0.52	2370	3550
6306	30	1.1811	72	2.8346	19	0.7480	0.039	1.469	2.559	0.76	3150	4600
6307	35	1.3780	80	3.1496	21	0.8268	0.059	1.688	2.795	1.01	4050	5800
6308	40	1.5748	90	3.5433	23	0.9055	0.059	1.929	3.189	1.40	5050	7050
6309	45	1.7717	100	3.9370	25	0.9843	0.059	2.126	3.583	1.84	6800	9150
6310	50	1.9685	110	4.3307	27	1.0630	0.079	2.362	3.937	2.42	8100	10 700
6311	55	2.1654	120	4.7244	29	1.1417	0.079	2.559	4.331	2.98	9450	12 300
6312	60	2.3622	130	5.1181	31	1.2205	0.079	2.835	4.646	3.75	11 000	14 100
6313	65	2.5591	140	5.5118	33	1.2992	0.079	3.031	5.039	4.63	12 600	16 000
6314	70	2.7559	150	5.9055	35	1.3780	0.079	3.228	5.433	5.51	14 400	18 000
6315	75	2.9528	160	6.2992	37	1.4567	0.079	3.425	5.827	6.61	16 300	19 600

- Either has a rated C of just over 6670 lb.
- But note that the 6211 has a bore of 55 mm (2.1654 in), and the 6308 has a bore of 40 mm (1.5748 in). The 6308 is more nearly in line with the desired shaft size.

Summary of data for the selected bearing:

Bearing number: 6308, single-row, deep-g

Bore: d = 40 mm (1.5748 in)

Outside diameter: D = 90 mm (3.5433 in)

Width: B = 23 mm (0.9055 in)

BEARING SELECTION: RADIAL AND THRUST LOADS COMBINED

For this case equivalent load is given by

$$P = VXR \; -$$

where P = equivalent loa

V = rotation factor

R = applied radial

T = applied thrust

- The values of *X* and *Y* vary with the specific design of the bearing and with the magnitude of the thrust load relative to the radial load.

- For relatively small thrust loads, X = 1 and Y = 0, so the equivalent load equation reverts to the form for pure radial loads.

i.e.,

$$P = $$

- To indicate the limiting thrust load for which this is the case, manufacturers list a factor called *e*.
- If the ratio *T*/*R* > *e* Equation

$$P = VXR -$$

must be used to compute *P*.

- If *T*/*R* < e. Equation

$$P = V$$

must be used to compute *P*.

Radial and thrust factors for single-row, deep-groove ball bearings

e	T/C_o	Y	e
0.19	0.014	2.30	0.34
0.22	0.028	1.99	0.38
0.26	0.056	1.71	0.42
0.28	0.084	1.55	0.44

where C_o is the static load rating of a particular bearing.

Procedure for Selecting a Bearing—Radial and Thrust Load

Step 1: Assume a value of *Y* from Table . The value *Y* =1.50 is reasonable, being at about the middle of the range of possible values.

e	T/C_o	Y	e
0.19	0.014	2.30	0.34
0.22	0.028	1.99	0.38
0.26	0.056	1.71	0.42
0.28	0.084	1.55	0.44

Step 2: Compute

$$P = VXR -$$

Step 3: Compute the required basic dynamic load rating C.

Step 4: Select a candidate bearing having a value of C at least equal to the required value.

Step 5: For the selected bearing, determine C_o.

Step 6: Compute T/C_o.

Step 7: From Table determine e,

Step 8: If $T/R > e$, then determine Y from Table.

Step 9: If the new value of Y is different from that assumed in Step 1, repeat the process.

Step 10: If $T/R < e$, use equation $P = VR$ to compute P, and proceed as for a pure radial load.

Problem: Select a single-row, deep-groove ball bearing from Table 14-3 to carry a radial load of 1850 lb and a thrust load of 675 lb. The shaft is to rotate at 1150 rpm, and a design life of 20000 h is desired. The minimum acceptable diameter for the shaft is 3.10 in.

Solution:

Step 1. Assume $Y = 1.50$.

Step 2. $P = VXR + YT = (1.0)(0.56)(1850) + (1.50)(675) = 2049$ lb.

Step 3. From Figure the speed factor $f_N = 0.30$, and the life factor $f_L = 3.41$. Then the required basic dynamic load rating C is

$$C = Pf_L/f_N = 2049(3.41)/(0.30) = 23\,300 \text{ lb}$$

Step 4. From Table we could use either bearing number 6222 or 6318. The 6318 has a bore of 3.5433 in and is well suited to this application.

Step 5. For bearing number 6318, $C_o = 22\,500$ lb.

Step 6. $T/C_o = 675/22\,500 = 0.03$.

Step 7. From Table , $e = 0.22$ (approximately).

Step 8. T/R = 675/1850 = 0.36. Because $T/R > e$, we can find Y = 1.97 from Table by interpolation based on T/C_0 = 0.03.

Step 9. Recompute P = (1.0)(0.56)(1850) + (1.97)(675) = 2366 lb:

$$C = 2366(3.41)/(0.30) = 26\,900 \text{ lb}$$

The bearing number 6318 is not satisfactory at this load. Let's choose bearing number 6320 and repeat the process from Step 5.

Step 5. C_0 = 29 800 lb.

Step 6. T/C_0 = 675/29 800 = 0.023.

Step 7. e = 0.20.

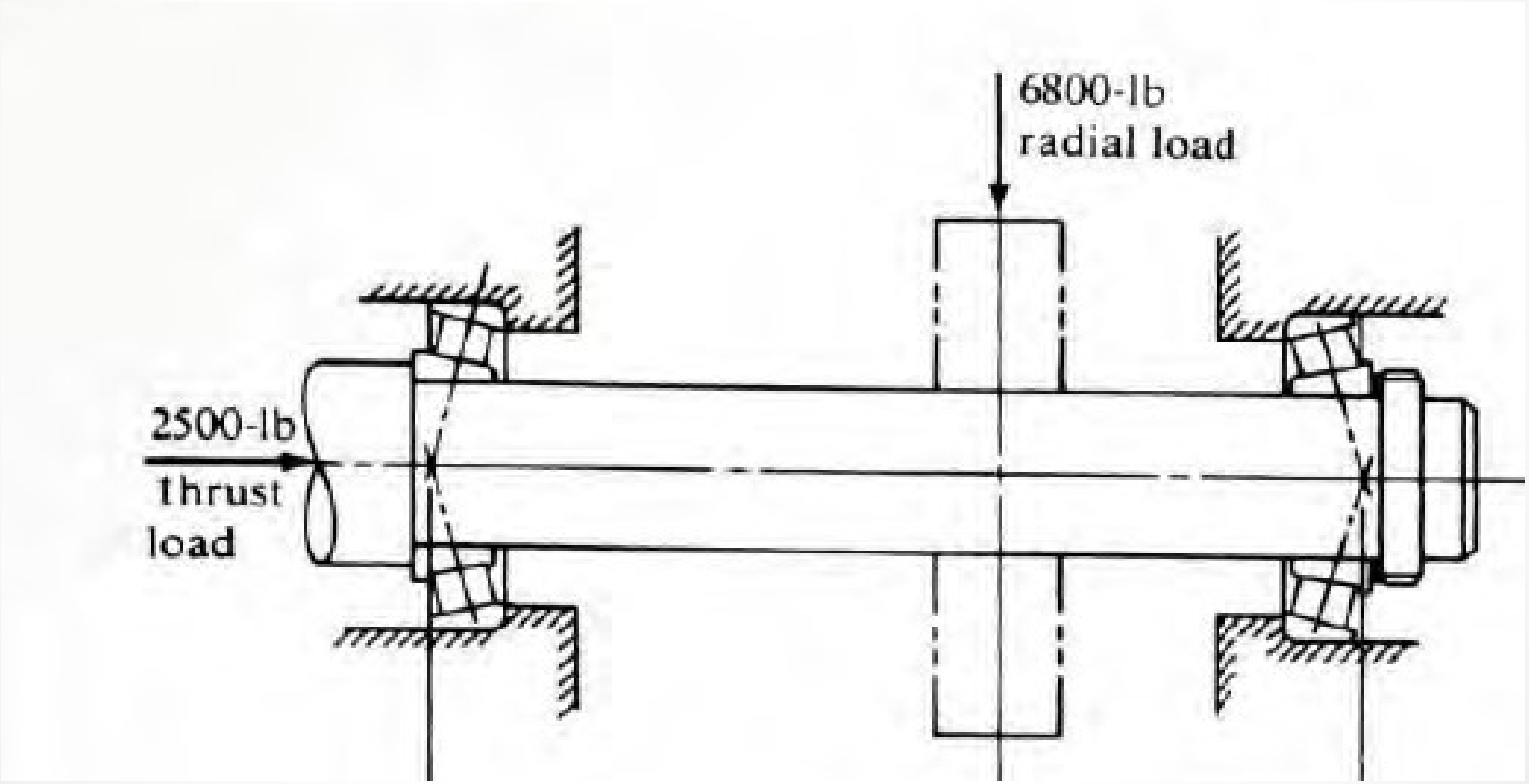
Step 8. $T/R > e$. Then $Y = 2.10$ using $T/C_o = 0$

Step 9. $P = (1.0)(0.56)(1850) + (2.10)(675) =$

$C = 2454(3.41)/(0.30) = 27\,900$ lb

Tapered roller bearing

Tapered roller bearing installation



The American Bearings Manufacturers" Association (ABMA) recommends the following approach in computing the equivalent loads on a tapered roller bearing:

$$P_A = 0.4F_{rA} + 0.5$$

$$P_B = F_{rB}$$

where P_A = equivalent radial load on bearing

P_B = equivalent radial load on bearing

F_{rA} = applied radial load on bearing A

F_{rB} = applied radial load on bearing B

Tapered Roller Bearing data

Bore	Outside diameter	Width	a
1.0000	2.5000	0.8125	0.583
1.5000	3.0000	0.9375	0.690
1.7500	4.0000	1.2500	0.970
2.0000	4.3750	1.5000	0.975
2.5000	5.0000	1.4375	1.100

Problem: The shaft shown in previous figure carries a transverse load of 6800 lb and a thrust load of 2500 lb. The thrust is resisted by bearing *A*. The shaft rotates at 350 rpm and is to be used in a piece of agricultural equipment. Specify suitable tapered roller bearings for the shaft.

Solution:

The radial loads on the bearings are

$$F_{rA} = 6800(4 \text{ in},$$

$$F_{rB} = 6800(6 \text{ in}.$$

we must assume values of Y_A and Y_B.

Let's use $Y_A = Y_B = 1.75$. Then,

$$P_A = 0.40(2720) + 0.5\,\frac{1.75}{1.75}\,4080 + 1.7$$

- Using Table as a guide, let's select 4000 h as a design life.

$$L_d = (4000\ \text{h})(350\ \text{rpm})(60\ \text{min/h})$$

The required basic dynamic load rating can now be calculated , using k = 3.33

$$C_A = P_A(L_d/10^6)^{1/k}$$

$$C_A = 7503(8.4 \times 10^7/1$$

Similarly,

From following Table , we can choose the bearings.

Bore	Outside diameter	Width	a
1.0000	2.5000	0.8125	0.583
1.5000	3.0000	0.9375	0.690
1.7500	4.0000	1.2500	0.970
2.0000	4.3750	1.5000	0.975
2.5000	5.0000	1.4375	1.100

Bearing A

$$d = 2.5000 \text{ in} \qquad D = 5.0000$$

$$C = 29\,300 \text{ lb} \qquad Y_A = 1.65$$

Bearing B

$$d = 1.7500 \text{ in} \qquad D = 4.0000$$

$$C = 21\,400 \text{ lb} \qquad Y_B = 1.50$$

We can now recompute the equivalent loads:

Life of bearing

mean effective load, F_m:

$$F_m = \left(\frac{\sum_i (F_i)^p N_i}{N} \right)^{1/p}$$

where F_i = individual load among a series of i loads

N_i = number of revolutions at which F_i operates

N = total number of revolutions in a complete cycle

p = exponent on the load/life relationship; $p = 3$ for ball bearings, and $p = 10/3$ for rollers

Alternatively, if the bearing is rotating at a constant speed, and because the number of revolutions is proportional to the time of operation, N_i can be the number of minutes of operation at F_i, and N is the sum of the number of minutes in the total cycle. That is,

$$N = N_1 + N_2 + \cdots + N_i$$

Then the total expected life, in millions of revolutions of the bearing, would be

$$L = \left(\frac{C}{F_m}\right)^p$$

Problem

A single-row, deep-groove ball bearing number 6308 is sub
loads for the given times:

Condition	F_i	
1	650 lb	3
2	750 lb	1
3	250 lb	2

Solution:

$$F_m = \left(\frac{\sum_i (F_i)^p N_i}{N} \right)^{1}$$

$$F_m = \left(\frac{30(650)^3 + 10(750)^3 + 20(}{} \right.$$

$$L = \left(\frac{C}{F_m}\right)^p$$

From Table for the 6308 bearing, we find that $C = 7050$ lb. Then

$$L = \left(\frac{7050}{597}\right)^3 = 1647 \text{ million rev}$$

At a rotational speed of 600 rpm, the number of hours of life would be

$$L = \frac{1647 \times 10^6 \text{ rev}}{1} \cdot \frac{\text{min}}{600 \text{ rev}} \cdot \frac{\text{h}}{60 \text{ min}} = 45\,745 \text{ h}$$