BORE POLISHING OF DIESEL ENGINE CYLINDER LINERS

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different methods were employed to simulate the bore polishing and to understand its mechanism. Firstly, by adding diamond paste and secondly, carbon, to the lubricant in order to produce a bore polished surface similar to the surface produced from an engine test.

Examination was carried out to identify the wear mechanism causing running-in and equilibrium wear. An attempt has been made to interpret the information gained from these tests in the light of recent literature in order to further our understanding of the wear mechanism of piston rings and cylinder liners and the bore polishing mechanism.

Optical microscopy and scanning electron microscopy were used to study the surface damage, energy dispersive x-ray analysis (E.p.M.A.) and x-ray photoelectron spectroscopy (xps) or ESCA were used to identify surface films. A Rank Talyor Hobson Talysurf instrument has been used to measure surface finish and obtain a profile of the surface. A rotating particle depositor (R.P.D.) was employed for oil analysis.

ABSTRACT

There are two important omissions in the literature on bore polishing, firstly there is no evidence of the successful development of a reliable tribo test device to simulate bore polishing and secondly, the mechanism of bore polishing has not been fully defined.

The aims of this study were:

1. To establish the principal characteristics of bore polishing in engines.

2. To produce bore polishing in the laboratory.

3. Differentiate between two reference oils in a laboratory tribo test.

4. To understand the mechanism of bore polishing.

The principal characteristics of bore polishing have been identified by the examination of Tornado cylinder bores from an engine test. The graphite structure is visible on the surface which has a surface finish of less then 0.125 μm in C.L.A. value.

The components used in these tests were a grey cast iron piston ring running on a grey cast iron cylinder bore typically used in commercial engines.

A reciprocating tribo test was used to distinguish between the two reference oils. The result showed higher friction, wear and a smoother surface with the oil causing bore polishing compared to the other oil which did not produce bore polishing. Adding carbon, taken from the wall of a piston used in an engine test, to the lubricant in the laboratory tribo test produced a phenomenon resembling bore polishing.

Comparisons have been made between the tribo test results and service engines and a good correlation has been obtained. Several analytical techniques have been used and the knowledge of bore polishing has been advanced. In particular, it is suggested that a combination of two processes, one mechanical and the other chemical, are associated with bore polishing. Four wear mechanisms were identified during this investigation; abrasion, delamination, corrosion and adhesion.

DEDICATION

TO MY WONDERFUL MOTHER AND MY HUMBLE FATHER, FOR THEIR UNLIMITED SUPPORT AND PATIENCE

GOD BLESS YOU

YOUR LOVING SON

GHAZI

AUGUST, 1987.

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CHAPTER ONE

INTRODUCTION

In National Express Coach engines, cylinder liners have a useful life of 250,000 miles before wear causes significant performance deterioration. After approximately 500,000 miles, these coaches are transferred to local use, therefore two complete engine overhauls are usually necessary to replace or refurbish cylinders. Piston rings, which wear at five times the rate of the liner, require replacement after only 50,000 to 100,000 miles.

Improvement in wear performance of piston rings would therefore be more beneficial than similar gains for liner wear and ultimately it would be ideal if the ring life could be improved to match that of the liner. Improvement in ring life will not be achieved in isolation. Consideration must be given to the whole system to obtain increased ring life.

There is evidence of abrasion, corrosion, delamination, and scuffing on rings and liners from diesel engines and in some cases all four may operate together but usually in different positions. Scuffing is particularly catastrophic and is associated with initiation on the piston ring, usually at top dead centre (TDC), which subsequently transfers to and damages the bore.

Abrasion and corrosion are usually considered to be responsible for normal engine wear. Abrasion is caused by surface asperity interactions and loose abrasive particles produced by combustion or wear. Corrosion results from the oxidation of fuel sulphur during combustion which subsequently condenses as a sulphuric acid on liner surfaces. It is not clear whether there are additive or synergistic effects between abrasion on the one hand and corrosive attack on the other. Surface coatings, especially chrome plate is used for diesel bores in marine and other applications in order to increase corrosion and wear resistance. Delamination wear is catastrophic, producing large metallic debris plates, but only occurs during engine development and is normally designed out by the time production of industrial engines commences. Grey cast iron is usually used for piston rings and cylinder liners and surface coatings are becoming more widely used to bring about some improvements in both running-in and longer life. Scuffing is less frequently encountered now that barrelled rings are more widely used and with the improvements in honing of the cylinder bore to improve both oil supply and its retention on the running surface. Both chrome-plated and molybdenum sprayed rings are now more widely used to improve wear life and also to eliminate scuffing.

Since the introduction of turbo-charged diesel engines, there has been frequent reference to bore polishing, which develops after a long period of service and leads to blow-by, increased oil consumption, and difficulties in lubrication which then initiates scuffing. Bore polish is evidenced by clearly defined areas of bright, highly reflective surfaces.

There is no clear evidence in the literature of a tribo bench test which can adequately simulate bore polishing. This has hindered the development of the understanding of bore polishing, since it has meant that the phenomenon can only be studied in expensive and lengthy engine tests. The Ford Tornado engine test is a widely recognised bore polish test, and the Mercedes-Benz OM 352 A engine is also used.

This investigation attempts to establish and characterise the principal features of bore polishing. A simple tribo test has been developed which produces a phenomenon similar to bore polishing observed in engines. From this an improved understanding of the mechanism emerges.

With regard to the serviceability of working diesel engines, it has been suggested that when polishing is less then 14% (measured by polished area divided by swept area) this is acceptable, whilst 45% or more is unacceptable. The onset of bore polishing is characterised by increased oil consumption and blow-by whereby exhaust gases leak past the piston ring pack, which then leads to a reduction in power output and eventually, because of lubrication failure, scuffing and seizure is likely. The onset of heavy bore polishing is clearly a problem worthy of further research in an energy conscious society.

This investigation attempts to differentiate between two reference oils, one preventing bore polishing and the other causing bore polishing, by carrying out a series of tests over a wide range of load and temperature.

The tribo test closely reproduced the type of damage often encountered on the piston ring and cylinder liner of a diesel engine. This was established by a direct comparison of the surface damage produced by the tribo test with the surface from test engines. Two

CHAPTER TWO

LITERATURE REVIEW

2.1 WEAR MECHANISMS

Wear is one of the most commonly encountered industrial problems which may cause a reduction in efficiency and increase power losses, oil consumption, and also the rate of the component replacement. The simplest definition of wear is "the removal of material from surfaces in relative motion by mechanical and/or chemical processes" (1).

It has proved possible to identify several common mechanisms which contribute to wear, although they are rarely encountered in isolation. The main types of wear are abrasion, adhesion, erosion, fretting, corrosion, delamination and rolling contact fatigue.

Abrasion

Abrasion accounts for some 50% of wear found in industrial equipment (1). Although this type of wear is rarely catastrophic, the cumulative effect often results in loss of efficiency and necessitates component replacement. Abrasion may be defined as:

"The removal of material from one or both surfaces in relative motion as a result of the presence of hard asperities on one of the contact surfaces or as a result of hard particles trapped between the surfaces or embedded in one of them" (2).

These particles may be foreign bodies or products from the surfaces. These two basic manifestations of abrasive wear are termed two-body and three-body situations. The former is caused when the abrasive rubs against a second surface and the latter when trapped between two moving surfaces.

The literature on abrasive wear may be divided into that which concentrates on two-body and that which deals with three-body wear.

The early research of Kruschov and Babichev (3) in two-body wear found a direct proportionality between relative wear resistance and bulk hardness for pure metals and some annealed metals and a different linear relationship for heat treated steels. Two main processes were identified when abrasive grains make contact with a wearing surface. These are firstly the primary removal of metallic chips and secondly, the plastic deformation to form grooves on the surface, without any removal of metal.

The influence of hardness on the wear resistance has been examined by Richardson (4,5) who defined hard abrasives as those whose hardness is greater than the contacting surface. For hard abrasives the wear resistance is dependent on the hardness of the abrasive to a critical value above which the wear becomes independent of the abrasive hardness. This occurs at above 1.3 x the hardness of the metal surface. Table 1 shows the hardness of particles which cause abrasive wear.

Sedricks and Mulhearn (6) reviewed the effect of the angle which the cutting face of the abrasive made with the wearing surface. A critical angle was found at which the mechanism changes from a rubbing to a cutting action. The angle was also important when determining the likely embedding of particles in the surface. It was also found that the critical angle at which rubbing changed to cutting depended on the materials under test and is influenced by the coefficient of friction between the abrasive and the surface.

Nathan and Jones (7), Avient, Goddard and Wilman (8) and Kruschov and Babichev (3) have all shown that volumetric wear is directly proportional to the nominal load up to a critical value which is limited by deformation of the speciman and degradation of the abrasive particles. Nathan and Jones (7) also found that the critical load occurs at a lower value for small abrasives. Although the diameter of the wear scars does not depend strongly on the load, the number of contacting points increases linearly with the increase in load. Kruschov and Babichev (3) and others (7) found a small increase in volume wear with increase in speed. In both cases, however, these increases were thought to be due to experimental variables.

Mutton and Watson (9) have attempted to explain the influence of the microstructure on the abrasion resistance of ferrous metals. Mutton and Watson (9) also conclude that increases in the hardness, due to the alloy additions, causes a change in the chip formation and a transition from ploughing to cutting.

The importance of grit size has been investigated by many workers. The dependence of wear on the abrasive particle size is an important characteristic of abrasion. It has been well established that the volume of wear increases rapidly with abrasive size from $1\mu m$ to $70\text{-}80\,\mu m$ and then increases at a much reduced rate with further increases in grit size. This critical size is apparent in both two-body and three-body wear. Rabinowicz and Mutis (10) have shown that the critical size for three-body wear is less than for two-body, at about 50 and that the wear rate was a factor of 10 less than for two-body (11). This was not confirmed by Miller (13) who used diamond abrasive in contrast to silicon carbide used by Rabinowicz (10). Rabinowicz (10) explained the transition effect to be due to the interference of the abrasive mechanism by the formation of large adhesive particles.

Richardson (5) thought the small particles were more prone to fracture than large particles. This theory was first proposed by Mulhearn and Samuels (12). Avient, Goddard and Wilman (8) thought that this characteristic grit size depended on the amount of abrasives embedded in the wearing surfaces.

Miller (13) investigated the abrasive size effect up to

70μ by using diamond abrasive. He concluded that the K value in the wear relationship was a value which related to the efficiency of cutting in abrasive wear.

$$K = \frac{3VH}{LS}$$

The relationship proposed by Kruschov (14) was:

$E = bM$ E = wear resistance = 1/wear volume

M = bulk hardness

b = coefficient of proportionality

This related wear to hardness of the material. Subsequently Rabinowicz (15) derived the following expression:-

$$\frac{dv}{dl} = \frac{L \tan \theta}{p} \qquad \text{Figure 1}$$

The load $L = r^2p$ where p = hardness of materials.

$$r^2 = \frac{L}{p} \qquad (1)$$

The volume of material removed during motion from A to B, $dv = rh\ d(I)$

$$dv = r(r \tan\Phi)dI = r^2 \tan\Phi dI$$

$$dv = \frac{L \tan\Phi\ dI}{p} \quad \text{- substituting from 1}$$

$$\frac{dv}{dI} = \frac{L \tan\Phi}{p} = \text{wear rate}$$

when $L \tan\Phi = K$

the wear resistance, $E = Kp$

This value of K takes into account the impinging angle of particles onto the surface and the possibility that all the loaded particles do not cause the same volume of wear. Although Rabinowicz and Mutis (10) took only $\tan\Phi$ to be variable, assuming that all the loaded grains produce wear, and referred to the $\tan\Phi$ as the coefficient of abrasive wear.

Polishing is a special form of abrasive wear characterized by the use of very small abrasive grains in the order of magnitude 5μ or less (16). Three main proposals have been advanced at various times to describe the mechanism by which a specularly reflecting or "polished" surface is produced when a metal is worked against a fine abrasive, namely:

a) **Flow mechanisms** where material is supposed to be smeared across the surface to fill pre-existing depressions. Proposals of this type were first formalized by Beilby (17) but he did not propose any mechanism for the smearing process. They were further developed by Bowden and Tabor (18) who proposed that asperities in the original surfaces are heated locally (possibly to melting point) then an abrasive particle rubbed over them, thus causing the heated material to be transferred into adjoining depressions. There are three facts which prove that this mechanism does not operate under polishing conditions (19).

Firstly, was the fact that no structure could be discerned in the polished surface. This was due to the limitations of the microscopical techniques available at the time. Modern microscopical techniques indicate that all polished surfaces contain fine scratches. Indeed when a series of progressively finer abraded and polished surfaces are examined, the results show that the scratches in the different types of surface differ only in their absolute dimensions.

Secondly, a flow mechanism proposes that material is merely transported from one point on the surface to another - this would mean no removal of material but experimental work shows that material is removed at significant rates during the polishing. Thirdly it is sometimes observed that a set of coarse scratches which have been obliterated during a finer polishing seem to reappear when the polished surface is subsequently etched. Beilby's explanation was that the coarse scratches are merely covered over by a Beilby layer during polishing. He then supposed that the Beilby layer is chemically more reactive than the crystalline base and hence is preferentially dissolved out of the previous scratches during etching.

b) **Mechanical mechanisms** in which polishing and abrasion are considered essentially to differ only in degree. This proposal implies that the contacting abrasive particles act as single point cutting tools which form shallower and narrower grooves during polishing than during abrasion because a much smaller load is applied to the abrasive particles under the conditions which are characteristically employed to produce a polish. A model of polishing suggested by these observations is sketched in Figure 2. The model indicates that the abrasive particles are held against the specimen surface by being mounted in an elastic cantilever. The force with which they are held against the surface being determined by the stiffness of the fibres of the nap of the polishing cloth. This agrees with the practical observation that the highest polishing rates are obtained with polishing cloths which have short thick nap fibres. Mechanisms of this type were initially proposed by Hooke, Newton and Herschell, but lost favour during the period of ascendancy of the Beilby-Bowden theory. They were revived in more recent times by Samuels (20) who recognised that the effects of polishing and abrasion (the "damage" caused to the surface) differred in degree and not in kind.

c) **Molecular removal mechanisms** (molecular-by-molecular removal) which have been advanced by Rabinowicz (16). This theory is also based on indirect evidence and on rejection of mechanical mechanisms because they require that very small chips be formed. Rabinowicz maintains that this is impossible because the surface-to-volume ratio of the small ratio of the small chips would be so high that an excessive amount of energy would be involved in their formation.

It is the author's view that polishing and abrasion are two similar processes differing only in the degree of the damage to the surface. A plausible suggestion is that the abrasive particles play a definite role in polishing by continuously wiping any corrosive product and protective films, thus enabling more rapid and more uniform chemical attack than would otherwise have been possible. This might be referred to as a wipe-chemical mechanism.

Erosion

Erosion occurs when particles carried in a gas or liquid impinge on a solid and gradually remove material from its surface. Erosion is normally considered to be similar to abrasion as both processes involve hard particles impacting and sliding over a solid surface. Differences between the two are however, apparent as important features of erosion, are the angle at which the particles impinge on the surface (21), the ductility and hardness of that surface (22) and the Kinetic energy of the incident particles (23). The mechanisms of removal by erosion change according to the impact angle. At a low angle (15-30^{o}), cutting is predominant. Under these conditions Eyre (23) indicated the use of high hardness as a precaution against wear. At angles approaching 90^{o} different precautions may be necessary as deformation of the surface may occur with particles shattering on impact.

Adhesion

When two surfaces are in contact it has been shown that the real asperity contact area is $1x10^{-2}$ to 10^{-4} of the nominal area (18). The resultant high pressures which develop in these areas are sufficient to deform the asperities to the extent that they are able to support the applied load, and may cause adhesion. As in any form of pressure welding, adhesion is favoured by clean surfaces, non oxidising conditions and by chemical and structural similarities. Rabinowicz (16) has expressed the dependence of adhesion of pure metals on their ability to form solid solutions. Lead for example has an extremely low solubility with chromium and iron and therefore would be a good choice for a counterface material. There are, however, other constraints particularly in its low strength and it may be necessary to use lead in alloys (i.e. lead, bronze) or as a thin overlay. The ultimate, however may be to select a polymer or ceramic material to slide against metals.

When a junction fractures to transfer material from one surface to another, or fractures at both surfaces to produce debris, wear occurs. Should the junction fracture at the original interface no wear occurs, although local plastic deformation and work hardening have both occurred. Welsh (24) was one of the first to identify changes in wear behaviour due to thermal effects caused by variation in load and sliding speed. These transitions occur between periods of so called "severe" (metallic) and "mild" (oxidative) wear conditions, see Fig.3. Wear in the oxidative regime, although classified under adhesive wear, clearly separates the wearing surfaces. Wear occurs by the removal of oxide debris from an oxidised surface supported on a work hardened substrate (23). At the T2 transition, the surface temperature is high enough for phase hardening to produce a hard "white layer" structure which prevents further deformation and helps to establish an oxidised surface once more.

Fretting wear

Fretting describes wear which often occurs when closely fitted parts are subject to low amplitude vibration. Although considered a basic wear mechanism by Waterhouse (25), fretting is often thought to amalgamate some of the main wear types.

The relative movement was thought by Moore (26) to break down the stable films on the surface. The freshly exposed surface would once again oxidise and be broken away. The generation of large volumes of oxide debris in combined areas was then thought to accelerate abrasion between the components. The pressure of any lubricant was believed to reduce the scale of the damage although Macpherson (27) reported that special lubricants were necessary to control the oxidation of the surface. It is often the case that oxide completely seals the area from lubrication which can lead to excessive noise or eventual seizure.

Macpherson (27) saw two methods of combatting fretting:

(i) Prevention of relative vibration by modification of design to remove fretting sites.

(ii) Prevention of oxidation of the surface by a bulk change in material to a non-metallic (for example), by use of surface coatings or by control of the environment with specially formulated lubricants.

Corrosive Wear

Corrosive wear covers situations where chemical reaction of the components with the environment dictate the wear process. Corrosion, although much of the time detrimental to wear, may work to improve wear behaviour. In some cases reaction with air to form oxides or hydroxides reduces wear as found in severe to mild transitions, but can often have the opposite effect, for example in fretting. Lubrication exerts a marked effect on corrosive wear. Burwell (28) separates the effect into two types:

(i) It may protect surfaces from the corroding environments, thus reducing the corrosive wear that would otherwise result.

(ii) The lubricant itself may react chemically with the surface, thus altering the type of compound and amount of wear that would otherwise result. In addition to the intentional reactions of lubricant and additive it is clear that reactions due to chemical deterioration, decomposition and contamination of the lubricant can be detrimental.

Delamination

The delamination theory of wear by Suh (29) describes the following stages either independently or in sequence.

(i) When two sliding surfaces come into contact, normal and tangential loads are transmitted through the contact points by an adhesive and ploughing action. Asperities in the softer surface are easily deformed and some are fractured by the repeated loading action. Once the surface becomes smooth, the contact is not just asperity-to-asperity contact, but rather an asperity-plane contact, each point experiences cyclic loading as the asperities of the harder surface plough through the softer surface.

(ii) The surface traction exerted by the harder asperities on the softer surface cause plastic shear deformation which accumulates with repeated loading. The material which is very near the surface cold works less than that of the subsurface layer. With continued sliding there will be a pile-up of dislocations at a finite distance from the surface. In time this will lead to the formation of voids. The formation of voids will be enhanced if the materials contain a hard second phase against which dislocations pile. Voids form primarily by plastic flow of the matrix. With time, the voids coalesce, either by growth or by shearing of the metal.

(iii) Once cracks are present, further loading and deformation causes cracks to extend and to propagate, joining neighbouring cracks. The cracks tend to propagate parallel to the surface at a depth governed by the material properties and the coefficient of friction. When cracks propagate because of either limited deformation or an externally small tangential traction at the asperity contact, crack nucleation is the rate controlling mechanism.

(iv) When these cracks finally shear to the surface (at certain weak positions) long and thin shear sheets "delaminate". The thickness of the wear sheet is controlled by the location of subsurface crack growth which is controlled by the normal and tangential loads at the surface.

Surface fatigue

The mechanism of fatigue caused by rolling contact is well known. This type of failure which is characterised by pitting or flaking of the surface, results in a relatively rapid deterioration in performance. Application of the Hertz equation for the elastic deformation of solid bodies to rolling contact has shown that the maximum shear stress occurs just below the contact surface. If fatigue cracks initiate they will do so in this zone and generally run parallel to the surface. This eventually leads to the characteristic spalling of fragments of metals from the surface.

2.2 MATERIAL AND MANUFACTURE OF PISTON RINGS AND CYLINDER LINERS

Piston Rings

Piston rings must satisfy certain requirements (30).

(i) A gas seal must be formed between the piston and cylinder bore.

(ii) Allow correct amount of oil to pass and lubricate the upper cylinder.

(iii) Conduct heat away from the piston.

(iv) Withstand the mechanical and physical stresses without deterioration.

(v) Be resistant to corrosion.

The cast iron piston ring is again a compromise of all these requirements. Experience has shown that a grey iron with flakes of "A type" graphite in the 3-5 size range, performs well in diesel engines. There should be a pearlitic matrix with less than 0.5% free ferrite and some hard phase. This phase may be of carbide, controlled by addition of chromium, molybdenum or vandadium up to about 0.5% and or phosphide eutectic with phosphorus additions of between 0.3 to 0.5%. For better mechanical properties nodular iron may be used but some wear resistance may have to be sacrificed. The metallurgy of both ring and liner is closely linked to the casting process, in particular to the cooling rate for piston rings. There are two methods of manufacture in general use. These are individually cast and rings machined from pots of cast material. It is believed that for rings less than 200mm diameter, single casting produces an optimum microstructure while large bore rings, greater than 400 mm, are superior when pot cast (31). Additions of carbide stabilizing elements are made to suppress the formation of ferrite normally associated with slower cooling rates.

The rings are then finished to the requirements of shape and profile. The shape of the ring is critical to maintain an even pressure on the liner surface when the engine is assembled. Large bore rings are formed to the correct profile after cooling on a mandrel. The smaller ones are normally cast with the required shape. The rings are accurately machined using sophisticated lathes before the gap is cut and the final profile is ground. Although rings are often coated to increase efficiency, liners are not coated at the manufacturing stage. The rings, however, may have either long term or short term coatings applied during manufacture. Coatings which retard corrosion during storage and assist during the running-in period of an engine are commonly applied. These may be oxide coatings which provide a mild abrasive action to ensure good conformity between ring and liner while providing good resistance to further oxidation during storage.

A second type are diffusion treatments to a depth of 0.4mm, e.g. Tufftriding and Sulfinuz. Soft coatings of manganese phosphate are also often used giving a low shear strength layer which has a granular appearance and holds the lubricant.

Running-in coatings of copper have been applied to the surface and have proved useful protection against scuffing. A soft electroplating of cadmium or tin, whilst not having any wear resistant characteristics, will aid running-in by melting and acting as emergency lubrication. Long term coatings designed to last for the life of the component are still applied to cast iron rings. The most common coating for piston rings is electro deposited chrome. It is intrinsically hard (800-900 Hv) and resistant to both abrasive and corrosive wear. The plated surface is usually treated by etching to produce irregularities on the surface which will retain oil. Other techniques are grit-blasting, phonograph finish and channel cracking. The thickness of the deposit varies up to 0.3 mm for large engine rings and is sometimes applied to the flanks as well as the face to control the groove wear in the piston.

Plasma sprayed coating is a modern technique which has not been fully evaluated but it does offer the inherent advantage of being more porous, e.g. chromium molybdenum carbides rather than electro-plated chromium deposited coatings.

Cylinder Liners

Most cylinder liners for diesel engines are manufactured using cast iron. The basic material is grey cast iron with a pearlitic matrix, although the structure may be modified by alloying elements and casting techniques. The basic requirements of the materials used are:

(i) Thermal stability

(ii) High thermal conductivity

(iii) Resistance to abrasive, adhesive and chemical wear.

(iv) Adequate mechanical properties

(v) Low cost

(vi) Good castability and easy machining.

Grey cast iron gives a compromise of these requirements. Experience has shown that optimum mechanical properties are obtained from a fully pearlitic iron with a maximum of 5% free ferrite to at least grade 17 (BS1452-1961). In highly stressed environments grade 23 cast iron is commonly used which may be hardened. The mechanical properties are obtained by additions of tin or chromium which promote the formation of pearlite. Phosphorous is used initially up to 0.7% in large bore cylinders, and forms a phosphide eutetic which is known to improve wear resistance under marginal lubrication conditions. Additions of 0.3% Vanadium have been used to reduce the phosphorus content to around 0.3% which gives improved castability with no loss of wear resistance. The effect of the Vanadium is to produce a dispersed phosphide/carbide complex and to change the iron's stability in sulphuric acid. Additions of copper are sometimes made on the basis of corrosion resistance.

A typical composition range for a large diesel engine cylinder (32) is:

Total carbon	3.1 - 3.4%
Silicon	1.8 - 2.2%
Maganese	0.7 - 0.9%
Phosphorus	0.15 - 0.2%
Chromium	0.2 - 0.4%

The iron is cast into cylinder pots in either resin or CO_2 bonded sands. These pots are then machined to the required dimensions. The surface of the finished cylinder is very important during the early life of the engine. To avoid scuffing which is predominantly a running-in phenomena and therefore, to achieve satisfactory long term performance, the surface should have:-

(i) Accurate dimensions

(ii) Consistent finish over all the bore

(iii) Grooves to help spread the oil from the lubricators over all the bore.

(iv) Freedom from embedded or loose particles and deformed layers.

There are two main finishing techniques and their application depends on the cylinder size. Large bore engines of about 0.5m diameter or greater are turned with a single point round nosed tool which gives a "wave-turned" surface. A smoother surface is obtained by cross honing with diamond or silicon carbide embedded in rubber or cork, or a combination of the two, and is generally used on smaller bore cylinders which produces a plateau groove finish.

2.3 LUBRICATION

Correct lubrication and careful maintenance are essential aspects of successful machine operation. Both have received considerable attention in recent years.

The lubricant in diesel engine cylinders should fulfill several requirements (33) :-

(i) It must reduce sliding friction between rings and liner to a minimum, thereby minimising metal to metal contact and reducing friction and wear.

(ii) It must possess adequate viscosity at high working temperatures and still be sufficiently fluid to spread rapidly over the entire working surfaces to form a good absorbed film.

(iii) It must form an effective seal in conjunction with the piston rings, preventing gas 'blow-by', which causes burning away of the oil film and leads to a loss of compression.

(iv) It must burn cleanly, leaving as little and as soft a deposit as possible. This is especially true of high additive content oils, as unsuitable types can form objectionable ash deposits.

(v) It must effectively prevent the build-up of deposits in the ring zone.

(vi) It must effectively neutralise the corrosive effects of mineral acids formed during combustion of the fuel.

The primary requirement is influenced by the speed, load and temperature conditions over the whole stroke. A useful indication of the influence of these variables on the lubrication of plane surfaces is given by the Stribeck (34) curve, which relates the Sommerfeld Number to the coefficient of friction (Fig.4). The Stribeck curve may be divided into the following three areas. Boundary and hydrodynamic lubrication are the basic mechanisms and the transition area between the two is normally referred to as mixed lubrication.

Hydrodynamic lubrication

If the lubricant film is sufficiently thick to prevent the opposing solids from coming into contact, the condition is described as 'fluid film lubrication' (Fig.5). This condition is often referred to as the ideal form of lubrication since it provides low friction and high resistance to wear. The factors which promote the formation of hydrodynamic lubrication film separating the surfaces are:

(i) high vicosity

(ii) high sliding speed

(iii) low loads

(iv) a wedge between the sliding surfaces.

Mixed lubrication

Mixed lubrication is dependent upon the roughness of the surface for its onset. The main part of the load is carried by the hydrodynamic film in the microgrooves or surface pores, whilst most of the friction and the wear results from asperity contact (Fig.5).

Boundary lubrication

The literature suffers from a lack of an agreed definition of boundary lubrication. Lansdown and Hurricks (35) describe boundary lubrication as 'that type which occurs when the fluid film is insufficient to keep the solid bearing surfaces separated and interaction occurs between the asperities'.

Boundary lubrication was thought to occur only where 'asperities' are in contact (Fig.5). Lubrication under these conditions does not depend on the bulk lubricant or material properties but on the surface and subsurface materials and the formation of molecular dimension films on the surfaces (18).

The conditions which are likely to produce boundary lubrication effects are opposite to those promoting hydrodynamic films. These are low vicosity, low speeds and high loads.

There are several general requirements of a cylinder lubricant:

Primary - control wear, friction and surface damage.

Secondary - sweep heat, dirt and wear debris.

- prevent corrosion.

Effective boundary lubrication is likely to be as a result of chemical reaction between the oil additive and the metals surface. The behaviour of such additives and the resulting films is schematically described in Fig.6. This diagram shows an example of an antiwear additive such as Zincdialkyldithiophosphate or tricresyl phosphate. Another effect which the additive may have is to increase the load carrying capacity, in which case they are referred to as extreme pressure (EP) additives.

2.4 DIESEL ENGINES

Diesel engines are widely used for propulsion in railway, automotive and marine applications. As might be expected a wide range of engines have been developed to meet the needs of each application. Diesel engines may be loosely divided into four categories (33):

(i) Slow speed crosshead engines -

2-stroke, mainly pressure-charged. Output 3,000-4,800 brake horse power (bph). Speed 100-280 r.p.m. Bore 500-1050mm.

(ii) Medium speed engines -

Trunk piston. 4-stroke or 2-stroke. Pressure-charged or normally-aspirated. Output 1000-20000 b.h.p. Speed 350-750 r.p.m. Bore 254-520mm.

(iii) Medium/High speed engines -

Trunk piston. 4 stroke or 2-stroke. Pressure charge or normally-aspirated. Output 400-4000 b.h.p. Speed 850-1800 r.p.m. Bore 200-280mm.

(iv) High speed engines -

Automotive type. Trunk piston. 4-stroke or 2-stroke. Pressure charged or normally-aspirated. Output 250-800 b.h.p. Speed 1500-3000 r.p.m. bore 127-178mm.

Before 1956 2-stroke engines were normally-aspirated. The largest engine developed approximately 8200 b.h.p. In 1956 the introduction of pressure charging increased output by 33.3% Then a new design with bore 760mm. at 115 r.p.m. gave an increased output of 11000 b.h.p. Pressure charged, larger bores, pushed this up to 2500 b.h.p. per cylinder with a maximum of 30000 b.h.p. from a 12-cylinder engine. In 1968 larger size engines were introduced with a bore of 1050mm. and stroke 3200mm. which developed a maximum continuous power output of 4000 b.h.p. per cylinder, so that an eight-cylinder engine developed 3200 b.h.p. and it is possible to build a 22-cylinder engine with an output of 48000 b.h.p.

Therefore over a 15 year period, the cylinder diameter increased by about 95% and by the use of supercharging pressures, cylinder output increased fivefold and what is equally significant is that specific fuel consumption was reduced by improved design. This is one example of similar patterns of development that have been achieved by most of the major engine builders.

The increase of bore and stroke size together with an increasing degree of pressure-charging, has resulted in an inevitable increase in the mechanical and thermal stresses within the engine. This only serves to emphasise the importance of cylinder liner wear and, while the literature does not clearly show improvements if wear rates are considered alone, the fact that wear rates have been held at acceptable limits is an indication of the advances that have been made in the development of cylinder liners.

Operating Conditions

Successful operation of the diesel engine depends largely on the lubrication of the piston rings and cylinder liner. There have been numerous attempts to obtain information about lubrication in a running engine. The various approaches suggest a model which is supported by an elementary analysis of the Stribeck curve. Along the centre of the stroke the lubrication is predominantly hydrodynamic, i.e. the oil film thickness generated by the 'wedge' effect of the piston ring is greater than the asperity height.

Hydrodynamic lubricant films are a result of pressure generated in the oil. This can be produced by a combination of wedge, squeeze and stretch effect (Fig.7). Friction may be attributed to the internal shear of the lubricant film. Towards Top Dead Centre (T.D.C.) and Bottom Dead Centre (B.D.C.) of the stroke, the reduction in piston speed causes a decrease in the film thickness. At or just after T.D.C., where gas pressures and temperatures are at a maximum and the direction of ring travel changes, the reduction in film thickness can result in asperity contact. Friction is no longer caused by shear of the entrained film and is due to boundary contact (Fig.8). The model of lubrication outlined above is complicated in the diesel engine by contamination from air, fuel, wear debris, and hard carbon deposit build up on the top crownland of the piston.

Without the correct maintenance of the filter, air-borne abrasive particles can find their way into the combustion chamber. Particles of 1 μm are nearly always present in the atmosphere and particles of up to 200 μm are not uncommon (36). Abrasives resulting from the deposits remaining after incomplete combustion of poor quality fuels, in addition to the debris produced by wear of the surfaces, can penetrate oil films and interfere with the efficiency of the lubricants in separating the moving surfaces. Particles generated between ring and liner were thought by Poppinga (37) to be in the range of 0.5 - 6μm. In addition particles of less than 2μm may be brought into the cylinder with the fuel as they are normally outside the limit of fuel cleaning. Other products such as asphalt, soot and carbonacesus matter can cause the formation of carbon deposit on the crown land of the piston and this deposit is mainly responsible for the polished mirror finish which leads to an increase in oil consumption and blow-by or even seizure through its effect on liner lubrication.

One further complication is that much of the abrasive, whether from combustion or wear debris, tends to be concentrated in the upper part of the cylinder where lubricant films are at their minimum thickness. Although abrasive problems can be reduced by filtration, the presence of sulphur in the fuel leads to the formation of sulphur dioxide during combustion. Under certain conditions sulphur trioxide can form and condense as sulphuric acid which can corrode the liners and rings. The formation of the acid along with the presence of ash is one of the major factors influencing engine wear. In addition to the control of surface temperatures high enough to restrict the condensation, neutralising additives are incorporated into the cylinder lubricant. The high surface temperatures involved can have a detrimental effect on lubricant viscosity. In addition to higher piston ring loads, modern engines have much higher thermal stresses imposed. The piston crown subsurface temperatures have been measured up to 450^{o}C. Of the total amount of heat generated in the engine from combustion, 41% is used as work, 37% is lost through heat contained in the exhaust gas, and 22% is dissipated by the cylinder components and oil system.

2.5 ENGINE WEAR

The mechanisms of wear which have been identified in diesel cylinders are abrasion, scuffing and corrosive attack, abrasion being the most common. In certain cases all three may operate together but usually in different positions. Recent literature regards scuffing as the most detrimental form of wear. For the most part, scuffing is regarded as a running-in phenomena which occurs before full fluid film lubrication has been achieved.

Since the mid 1970's there have been more frequent references to bore polishing which appears to have arisen since the introduction of turbo charged diesel engines.

Scuffing

Scuffing which has been defined by the Institute of Mechanical Engineers (1957) (38) as:

> "Gross damage caused by the formation of local welds between the surfaces."

It is not commonly encountered in practice (39). When it occurs between liner and rings it often leads to increased gas blow-by and oil consumption, which results in an appreciable loss of power.

The literature on diesel engine wear, in particular, is somewhat confused over what does and does not constitute 'scuffing'. Identification depends largely on surface appearance. Neale (40) described them as torn and smeared in the direction of motion in the cylinder. Rogers (41) on the other hand, describes scuffing as "dull vertical stripes". These two examples illustrate the subjective nature of identification.

It is clear from the literature that very little is known about the mechanism of scuffing.

Wiborg (42) for example describes scuffing as "a form of wear which is defined as the adhesion and successive shearing of junctions in metallic contact."

The much quoted view proposed by Neale (40) in the early 1970's has not changed significantly. The large number of interacting factors, the dynamic nature of the process and the range of scuffing severity, appear to have confounded both experimental and theoretical research into the problem. A recent attempt by Shafia (43), however, concluded that in some cases adhesion plays no part in the process but proposed delamination as the mechanism responsible. His results, however, far from indicating sub surface initiation of cracks, show plastic flow of the surface layers to form lamellae, which are separated by extruded graphite from a flake structure.

This type of plastic surface deformation was also recognised by Nadel and Eyre (44) who were able to identify transverse cracks in the flowed layers due to high surface stresses which could lead to delamination of 'plate-like' debris.

Aue (45) proposed that heat generated between the ring and liner surfaces by friction when the oil film breaks down, can cause local expansion of the surfaces. This in turn causes more contact and more heat to be generated. If the wear rate becomes greater than the surface expansion rate severe wear stops, if not, scuffing will continue.

Neale (40) also believes that the generation of high local temperature plays as important role in the mechanism, whereas Nadel and Eyre (44) saw no indication of excessive surface heating in their examination of scuffed liner surfaces. In addition they could find no evidence of 'white layer' formation on liners, although it is common in piston rings. Dyson (39) and others (42, 44, 46) however, identified the production of hard etch resistant phases on the surface of ferrous materials as characteristic of scuffing. These so called 'white layers' play an influential part in severe wear. Wiborg (46) et al describe the layer itself as scuff-resistant, but prone to fracture and spalling from the surface. Due to high hardness (up to 5 x bulk hardness) it can cause severe abrasion.

There is sufficient contradictory evidence in the literature concerning scuffing of rings and liners to suggest that, at least, there is a progression through a range of scuffing severity and damage, but more important perhaps, there are a number of separate mechanisms producing surfaces of similar appearance which are all identified as scuffed.

Running In

Piston rings and cylinder liners provide a good bearing couple which performs well under extreme conditions. The eventual successful operation depends to a large extent on the lubrication conditions existing between the two surfaces - in order to separate the ring and liner over the whole stroke with a hydrodynamic oil film, the ring has to have an optimum parabolic profile, the liner should be flat and smooth and an adequate supply of oil should be available over all the liner surface. The accommodation of the first two requirements is difficult (which means expensive), if not impossible, to achieve during manufacture. It is fortunate that the engine itself can be induced to do the job successfully.

Sreenath and Raman (47) called the beneficial wear of relatively moving interacting surfaces, resulting in their gradual conformence and improved performance, running-in. Sreenath and Raman (48) also found that there were two stages in the running-in of liner surfaces. While the engine is run at constant load and speed, the asperities are mechanically removed as debris. When the load and speed are increased in steps the remaining valleys are filled, either by debris formed during the second stage, or by production of surface films. Neither of the wear mechanisms were identified. Although running-in and long term wear of liner and ring are synonymous, Sreenath and Raman made no attempt to see how the piston rings changed during the liner running-in process.

Wakuri and Ono (49) examined this change in ring profile over the running-in period. They found that the ring profile is formed by the removal of metal from the ring edges due to interaction with the liner, as the ring tilts during the stroke.

The majority of piston ring and cylinder liner problems occur early in the running-in life of an engine and this corresponds to the periods where there is a substantial amount of material removed from the running face of the piston ring in order to form their operating profiles. At this time the cylinder liners usually require some adjustment to their local surface shape because they are prone to some dimensional distortion due to the pressure and temperature gradients to which they and their restraints are subjected.

Neale and Eyre (50) reported that of all the running-in processes in an engine, that of the piston rings and cylinder liners generate the greatest amount of wear debris, indicating the greatest amount of material removed from any component. Neale and Eyre state (50):

> "In spite of all the precautions which may be taken in terms of material microstructure and surface finishing, the running-in process of piston rings and cylinder liners is still locally severe in triobiological terms and if any engine is stripped and examined during the first few hours of running-in, local scuffed areas will invariably be found on the piston ring and sometimes on the liner as well".

A certain rate of mild wear during the early engine life is beneficial because it will minimise the blow-by and oil consumption and reduce the localised pressure between the liner and rings which can occur due to tolerance in manufacturing and thereby decrease the probability of detrimental scuffing.

Hogmark and Alexander (51) stated that the two most important mechanisms responsible for running-in are abrasion and deformation. Abrasion is caused by microscopic hard particles and the majority of these hard particles originate from the inlet air or from the combustion products, wear debris from the liner/ring, or other friction systems may also produce particles which collect in the crank case oil.

Plastic deformation of colliding asperities is also known to be a common mechanism during running-in, particularly in areas of high localized pressure where the lubrication is insufficient. These two mechanisms operate more or less simultaneously and there are various opinions as to which one predominates.

Sreenath and Raman (47) stated that metal to metal contact takes place between piston ring and cylinder liner at the T.D.C. position of the ring during running-in due to the effect of the initial surface roughness of the liner.

Due to the roughness of the original liner surface which is necessary to shape the ring to the correct profile, metal to metal contact at asperities is inevitable. Wiborg et al (46) believed that once the oil film had been disrupted and rubbing had destroyed any protective surface layers, adhesion could occur between the apserities. Adhesion is regarded by Eyre (23) as the predominant wear mechanism during running-in. It is also thought to occur during start-up and load changes where surface contact occurs.

Measurements of oil film thicknesses in a 4-stroke diesel engine have been made by Moore and Hamilton (52) who found that, up to a period of 22 hours, running at low loads, metal to metal contact occurred between liner and ring, particularly on the compression and firing stroke just after T.D.C. At higher loads, contact still occurred after 62 hours running.

Ostvisk and Christensen (53) observed that running-in was due to squashing of asperities, whilst Montgomery (54) described the production of a glaze formed on the liner surface. Montgomery attributed the smoothing of the surface during running-in to the covering of apserities with glaze, which he stated had iron oxide as an important constituent.

A glaze (55) defined as a skin, or coating, forms chemically on the liner surface from iron oxide, graphite and decompositon products from the fuel and lubricant. It can be 15-20μm in thickness. Pursund (56) also reports finding this glaze in cylinder liners. The wear particles which Atlas (57) describes as a mixed shear layer of 1m m thickness which covers scratches and irregularities during running-in, are also of iron oxide characterisation. There is clearly a wealth of evidence for the formation of glaze formed during running-in, although there is no evidence to suggest that this could be a possible abrasive contaminant helping to promote bore polishing.

Sarkar (58) showed the pattern of the curve which he produced in the laboratory (Fig.9) to be the same as that in actual machine parts and hence there is due emphasis on the importance of running-in before a sliding component takes up its full load during its life of operation.

Wear of the Cylinder Liner at Different Positions

For engines with acceptable wear rates Nadel and Eyre (44), in their survey of a large number of liners, found evidence of abrasion, corrosion and scuffing. Abrasion was thought to be responsible for normal mechanical wear, although corrosion could indirectly lead to the intensification of abrasion, while scuffing appears to be particularly catastrophic in producing metallic plates of debris by a process of delamination.

Sudarshan and Bhaduri (59) reported that the wear at the top portion of the cylinder occurs due to a three-step process involving adhesion, corrosion and abrasion and also in order to achieve minimum wear rates, the parameters which need to be controlled are (i) surface topography, (ii) hardness of the materials in contact and their microstructures and (iii) lubrication.

Nadel and Eyre (44) identified the positions on a liner of a low wear rate engine where the mechanisms of wear occurred. They judged abrasion to predominate at the top and bottom of the stroke, but more severe at the top where hard phases stood in relief above the matrix. This was thought to be due to a differential wear mechanism. Abrasion at the bottom of the stroke was found not to have removed all the original machining marks. At the mid stroke they found no abrasion evidence, although corrosion in the form of widespread etching of the material structure was identified.

Golothan (60) found maximum wear occurred at, or just below, the top of ring travel, the lowest wear around the middle of the cylinder and a small increase in wear at the bottom of ring travel. The maximum wear at T.D.C. he thought was due to a combination of corrosion and wear arising from the breakdown of the oil film due to high gas pressures and high cylinder temperatures which led to oil films with low viscosity.

Dent (61) saw the two mechanisms of abrasion and corrosion combining to give a wear profile with peaks. A peak at, or just below, T.D.C. is due to a predominantly abrasive mechanism and an increase in wear at the middle of the stroke is due predominantly to the action of acid.

The minimum wear was thought to occur at B.D.C. (Fig.10). This view appears to be confirmed by the evidence presented by Nadel and Eyre (44). Dent considered the abrasion to be directly related to the pressure in the cylinder which is maximum just below T.D.C. and gradually declines to B.D.C. This seems to be a rather simplistic view as the load on the abrasive is not from the gas pressure directly but through the piston ring which is separated from the liner by an oil film of variable thickness. It is more likely that it is the oil film thickness which governs the abrasive damage. When the oil film is thinnest the range of abrasive particle sizes, which can cause damage, is greater than when the oil film is at its maximum. Neale (62) has shown that the oil film thickness typically reaches a minimum value just after T.D.C. but increases sharply through the centre of the stroke to decrease again just after B.D.C. The analysis takes into account gas pressure, temperature, piston speed and ring geometry (Fig.8).

From this it might be expected that abrasive wear is maximum just after T.D.C., minimum along the centre of the stroke but increasing slightly just before B.D.C. Cotti and Simonetti (63) also relate maximum pressure with maximum mechanical wear but found that the position of greatest wear did not coincide with the highest gas pressure (Fig.11). It should be emphasised however, that the position of minimum oil film thickness does not necessarily occur at maximum pressure but is determined by the fluid dynamics of ring lubrication and is likely to vary from engine to engine. Cotti and Simonetti were forced to conclude that the peak in cylinder wear just below T.D.C. was due to the dominance of a corrosive mechanism.

One aspect of abrasive wear in engines which has received no attention in the literature is the importance of the distribution of abrasive particles over the liner surface. It is likely that there would be a higher concentration of abrasive debris around the position of maximum wear which may coincide with minimum oil film thickness. The generation of debris at this point would tend to accelerate abrasive damage. The distribution of abrasives from induction and combustion may also be important.

The evaluation of worn components is made difficult by the complex interaction of wear mechanisms which can produce surfaces which appear to have been affected by only one mechanism.

2.6 BORE POLISHING

Definition and Characterisation

Bore polishing in cylinder liners of diesel engines is evidenced by clearly defined areas of a bright mirror finish (reflective surface) on the liner surface (64). This is caused by local mechanical wear and usually occurs after a long period of running-in at high speeds and loads (55). One example where this phenomena is encountered is turbo-charged diesel engines. Wilson and Calow (64) stated that bore polishing is characterised by a bright smooth (mirror) finish of less than 0.125μ m in C.L.A. value, with a total absence of cross hatch pattern. Different intensities of polishing can occur and a broad scale of reference have been agreed as follows (64):

Light Polish - mirror finish overlaid on the original honing pattern.

Medium Polish - mirror finish showing faint overlaid honing pattern.

Heavy Polish - mirror finish showing no traces of the original honing pattern.

Many difficulties have been, and still are being encountered in bore polishing work, in the assessment of polished areas and in identifying differences between polishing intensities. One method of recording polished areas which has been found to be particularly useful is tracing as follows (65):

(i) The cylinder liners are split along the wrist-pin axis.

(ii) The polished areas are outlined with a marking pencil.

(iii) A clear plastic sheet is placed on one side of the liner and the polished areas are traced onto this sheet.

Tracing method has been criticized (55) by illustrating how a numerical assessment of areas polished does not always give a true indication of the liner appearance. Wilson and Calow (64) have used a visual ratings of intensity and area of polishing in favour of Talysurf profiles and surface roughness measurements in terms of C.L.A. values, but stated that even by this method, the rating of lightly polished areas can present problems. It has been difficult in many instances to decide whether a lightly polished area was a normally run-in zone or was in fact showing the early stages of heavy polishing (64).

Attempts have been made to define the severity of polishing by means of Talysurf profiles and surface roughness measurements in terms of C.L.A. values (64,65). This method (55) has not been successful and it was found that the C.L.A. values for the polished areas were the same or less than those obtained for normally worn surfaces after a 480 hour caterpillar 1A test (66). It has been reported (55) that visual ratings of polished areas in the liner did not correlate with Talysurf profiles or C.L.A. values for the polished areas.

McGeehan (67) used optical microscopy with the other methods (visual and Talysurf) for identifying the bore polishing and different intensities of polishing but he did not use scanning electron microscopy.

Hogmark and Alexander (51) used scanning electron microscopy with a visual method to identify the polishing but they could not differeniate between the intensities of bore polishing. It is apparant at this stage that considerably more work remains to be done on the rating of polishing and the assessment of bore polishing.

Mechanism of Bore Polishing

A review of the limited amount of literature on the subject of bore polishing gives very little conclusive information about the mechanism of bore polishing. Various authors referred to the hard carbon on the piston crown land acting as a fine abrasive producing a polished liner (55,64,67-72).

Haggh and Holmer (68) first identified the carbon polishing problem when the piston collected carbon from the top of the liner and polished the liner and Soule (69) called this bore polishing 'bore glazing'. This occurred if engines were run continuously, 24 hours a day, at maximum output for long periods where the thermal condition appeared to create a very hard carbon around the piston. Hollinghurst (70) blamed the carbon on the top land for causing bore polishing. Wilson and Calow (64) reported that polishing can occur below the bottom of ring travel with relatively clean piston skirts. Knight and Weiser (71) reported that although top land carbon seemed to be the main cause of bore polishing, bore polishing can occur in all liner sections, from the areas below the ring belt zone at bottom dead centre, up to the areas of the liner above the first ring at top dead centre. In the literature (55) it states that "polishing has been found to extend above the area of ring travel'.

McGeehan (67) stated that excessive top deposits cause higher cylinder bore polishing and that these deposits cause cylinder bore polishing (wear) through fine scale abrasion. Parsons (73) suggests that (i) carbon deposits on the piston act as a lap (grinding paste) which polishes the liner. This suggestion was proved by work done by Ishizaki, Sato and Takase (74), (ii) the deposit on one side of the piston pushes the piston over, thus causing the piston to bear heavily on the opposite side of the bore.

Ayel, Roux and Tahon (75) suggested that bore polishing is caused by abrasive 'grinding paste' composed of hard carbon in the oil. This acts between the ring and liner and they observed abrasive wear in a bore polished liner. Hogmark and Alexander (51) stated that polishing of the cylinder liner occurs by abrasion with hard metallic and non-metallic particles embedded in the soot layer on the piston top land. These particles, probably silicon rich sand particles originating from the air inlet, are also contained in the lubricating oil or in the combustion gas and can be squeezed between the ring and liner and slide against the liner surface. This indicated that three-body abrasive wear mechanism was involved.

Three-body abrasive wear in worn liners has not been given enough attention in the literature of bore polishing as being important. Having said that, it is possible to identify a lot of abrasive debris between the piston ring and cylinder liner especially at the Top Dead Centre where the oil film thickness is minimal.

Knight and Weiser (71) reported that in three engines out of nine, bore polishing occurred during the running-in period. They referred to it as individual manufacturer's problems. Hogmark and Alander (51) stated that it was more likely that the major polishing occurs during the early stages of the test (running-in).

Much still remains to be answered concerning the association of running-in with bore polishing in the cylinder liner.

Parsons (73) stated that there is no conclusive proof for the proposed mechanisms. He reports that with two different oils producing 14% bore polish and 40% bore polish (measured by polished area divided by swept area), the piston deposits were found to be identical. In conclusion it may be said that whilst carbon deposits on the piston are a factor in the mechanism of bore polishing, there is insufficient evidence to show that this mechanism is of fundamental importance. Parsons (73) reported on an investigation carried out where two oils were tested, one had been centrifuged to remove insolubles, while the other had insolubles present in it. The device which he used produced a polished wear surface for the uncentrifuged oil and a non polished surface for the centrifuged oil. The conclusion drawn being that oil insolubles, whether carbonacous or wear promoting contaminents, could enhance bore polishing.

Wilson and Calow (64) suggested that it is reasonable to anticipate a lubricant's influence through its ability to affect the characterisation of the deposit later on. Ishizaki (74) and McGeehan identified the effect of lubricant on the carbon build up. Stevens (72) states certain additive packages have been developed which will control the bore polishing but they did not give any information about the package. Lubricant, therefore, does play an important part in causing bore polishing.

Wilson and Calow (64) comment that bore polishing can occur early in engine life with almost deposit-free pistons. This suggests a polishing mechanism occurred by metal to metal contact between piston ring and cylinder liner. The literature (55) reports that some polishing can occur as a result of piston ring face/liner interaction through normal ring face pressures and carbon packing behind rings which increases face pressures and liner polishing can occur. Hogmark and Alexander (51) found plastic deformation and metallic adhesion between the liner and the ring and they reported that this occurred especially nearest the point of zero velocity when the piston and liner are almost completely in contact, at which time the liner material transferred to the ring surface may seriously abrade it during the polishing process.

Oil film thickness between the ring and the liner has been measured at 0.5μm to about 2μm on the power stroke (76) while in another investigation, Moore and Hamilton (52) found a minimal oil thickness of 0.2μm. These values of oil film thickness indicate that the mechanism of piston ring and cylinder interaction cannot be ignored.

McGeehan (67) stated that cylinder bore polishing can also be caused by corrosion when high sulphur oils lead to the formation of sulphuric acid from sulphur trioxide and water on the liner surface as a result of combustion products. However, some diesels, e.g. marine diesels have high sulphur fuels and corrosive wear is often found as a consequence but bore polish is not encountered. Hogmark and Alexander (51) also identified the corrosive wear on the polished liner but they did not associate this as a mechanism involved with bore polishing. McGeehan and Knight and Weiser (65,71) reported the piston ring polishing as a possible mechanism for causing bore polishing in the cylinder liner, but they did not offer any explanation.

Effect of Bore Polishing

The onset of bore polishing is characterized by increased oil consumption and blow-by occurs (64,65), i.e. exhaust gases leak past the piston rings and eventually the lubricant breaks down which leads to scuffing of the liner surface and eventually to seizure. The effect of blow-by in diesel engines is to reduce the compression ratio, i.e. the ratio of pressure at top dead centre to that at bottom dead centre. Since the efficiency of the engine is closely linked to the compression ratio, blow-by causes the engine to be less efficient. For small bore polishing percentage, blow-by is not a serious problem, however for large bore polishing, percentage blow-by is associated with loss of power and efficiency as well as high oil consumption. Further more, large areas of bore polishing increase the likelihood of scuffing failure. Clearly bore polishing on a large scale is unacceptable while on a small scale it may be tolerated.

At the present time there is no clear understanding of how polished areas of cylinder bores lead to increased oil consumption and scuffing/seizure problems. Wilson and Calow (64) stated that there appeared to be a change in the mode of oil distribution at the surface; perhaps with a lessening of the ability of the surface to retain oil accompanied by loss of control of the oil flow up the bore and past the rings. These local areas of oil sealing failure will also allow excessive blow-by to occur and eventually lubrication breakdown occurs leading to scuffing and eventual seizure. McGeehan (67), Haggh and Holmer (68) stated that cylinder bore polishing exacerbates oil consumption because a polished surface is susceptable to scuffing due to loss of lubricant retention ability. Knight and Fuhrmann (77) stated that bore polishing will be a component of three modes of services failure, namely:

Engine seizure - where bore polishing prevents good oil film retention and piston ring scuffing follows leading to seizure of rings, piston and the liners.

Excessive oil consumption - where the extent of bore polishing increases rapidly taking oil consumption rates from an acceptable designed level to the point where additions are so frequent and great that operators complain.

Excessive oil consumptions from carbon build up around the crown land where the gas pressure on the top oil control ring is lost and bore polishing follows.

Reduction in Bore Polishing

Various methods have been employed to reduce cylinder wear and bore polishing. Furuhama and Hiruma (78) report on a series of tests in which the crown land shape was altered. The crown land shape A (Fig.12) gave very high oil consumption whilst profile B (Fig.12) gave very low oil consumption. The main oberservation from these tests was that oil on the cylinder wall in case A was scraped up by the piston land edge and burnt by the combustion flame. In Case B the oil was not scraped up and was not burnt by the combustion flame. Consequently carbon deposits were found with A but not with B. Dinsmore (79) also mentioned cutting back the crown land and stated that it minimizes hard carbon build-up on the crown land. It is clear that the crown land profile is important.

Engine manufacturers have used 'cut back' toplands (Fig.13) that improved oil consumption due to the quick seating of the top ring and more favourable pressure distribution among the rings. The combination of these factors increased the radial pressure at the start of the down stroke and thus decreased oil consumption. Low oil consumption with the cut back top land appears to be due to the fact that it minimizes contact between the carbonaceous deposit and the cylinder wall (74). A cut back piston, defined as a piston which has much larger than usual clearance at the top land (74), was made for the same engine. A cut back piston with a nominal clearance of more than twice that of the non cut back piston was tested (Fig.14) and this showed a wear pattern. After 2000 hours operation this was compared with the results of the non cut back piston. It is clear from Figure 14 that there is almost no wear with the cut back piston except normal wear caused by the top ring. Also there was no oil consumption and this confirmed that carbon polishing wear can be prevented by the cut back piston.

The effect of piston misalignment has been investigated by several experiments, in particular Furuhama and Hiruma (78). The piston was slightly offset and an increased oil consumption noted. The effect was also noted by Wilson and Calow (64) and considered to effect bore polishing. The cylinder liner surface finish has an effect on the performance of the liner. Willn and Brett (80) mentioned that a diamond honed finish usually gives poor performance when compared with silicon carbide honed finishes. They attribute the difference to loosely adhering flakes of liner material formed during the diamond honing process which act as an abrasive contaminant. Wilson and Calow (64) observed that the depth of cross hatching was particularly relevant in engine tests.

The effect of lubricant and its additives has received little attention with reference to bore polishing. Parsons (73) states that by careful choice of oils and additives, bore polishing may be alleviated, McGeehan (67) stated that low bore polishing can be achieved with lubricant formulations which minimize the top land deposit and provide sufficient alkalinity to minimize the corrosion aspect of bore polishing. McGeehan (67) found that a commercial SAE15W.40 oil provided lower oil consumption in engines than SAE30.

Lohuis and Thompson (81) stated that both monograde and multigrade formulations show excellent bore polishing protection equivalent or superior to the good reference oil RL47, while Schmidt and Michael (82) report upon tests with 6-cylinder turbo charged diesel in which multigrade oils show slightly lower liner wear than SAE30 oils at load levels typically found in field use.

The formation of soot in a diesel engine can seriously affect the properties of the oil as well as forming an abrasive deposit on the crown land of the piston. Rounds (83) states that above a soot concentration of 3.5%, the antiwear properties of zinc dithiophosphate (ZDP) are completely destroyed. McGeehan (84) showed that soot can significantly increase carbon deposit formation at high exhaust smoke densities. The soot is directly caught on the top land and increases carbon deposit, or comes back after once having been caught by oil, also increasing carbon deposit.

By reducing the soot level, bore polishing could be partially alleviated, although legislation in many parts of the world would complicate this aim since greater engine efficiency has a number of environmental drawbacks such as pollution. Cloud and Blackwood (85) state that the major diesel fuel constituent governing wear is sulphur content. Their tests indicate that an increase of sulphur content from 0.2 to 1% can result in a two to sixfold increase in liner wear.

Ishizaki and others (74) stated that fuels could increase the cylinder liner carbon polishing wear due to the effect of formation of soot on the carbon deposit. Knight and Fuhrmann (77) referred to the design solutions which have contributed to the current reduction of problems in service. Among these changes which engine builders have revealed are:

(i) filtering charge air coolers.

(ii) reducing peak firing pressures.

(iii) introducing piston offset.

(iv) reducing supercharged air pressure.

Research into Bore Polishing

CEC (Coordinating Engineering Council) IGL-5 Investigation Group was set up in mid 1973 to examine the lubrication requirements of high speed diesel engines in Europe.

Its first task was to carry out a survey of European high speed diesel engine manufacturers with the object of reviewing current and future lubricant requirements and predicting those factors and problems which were likely to have most influence on future lubricant quality requirements. (Table 2).

This survey showed that wear problems were the largest source of difficulties, ranging from valve train wear to ring sticking and scuffing, and bore polishing was listed as being a significant potential problem in high output turbo-charged diesels. As a result of this survey CEC IGL-5 decided to focus its attention on bore polishing and formed the project group EL-20 to try to develop an engine test using reference oils of known performance. The two reference oils were RL47 and RL48 - RL47 is a high quality oil which is known to produce low levels of bore polishing and RL48 is known to produce high levels of bore polishing. Knight and Weiser (71) reviewed the suitability of existing lubricant test methods for European automotive needs and they recognised the necessity for the development of specific test methods which could evaluate bore polishing in respect of the lubricant (the reference oils RL47 and RL48).

It was reported (55) that single cylinder engines such as the Caterpillar 1-G and Petter AVB and a multi-cylinder engine such as the Volvo TD120 and Ford Tornado should be considered for a possible bore polishing test. The conclusion was that single cylinder engines had poor reproducibility and were consequently abandoned whilst the turbo-charged Ford Tornado differentiated well between the two reference oils (RL47 and RL48). The Ford Tornado was tentatively adopted by the Coordinating Engineering Council (CEC) (86) as a standard industrial test.

Wilson and Calow (64) conducted bore polishing tests on a range of engines and stated that the single cylinder Caterpiller 1Y73 showed promise as a standard test since it differentiated well between reference oils, although its repeatability was not fully acceptable. Schmidt and Michael (82) describe tests with a 6-cylinder turbo-charged diesel in which the range of performance of several commercial oils was investigated in terms of piston deposits, but liner bore polishing was not considered in the tests.

Davies, Stevens and Walters (72) used Ford Tornado engines as a laboratory engine test procedure and differentiated very well between the reference oils (RL47 and RL48) in respect of the bore polishing problem in highly turbo-charged direct-injection diesel engines.

CHAPTER THREE

EXPERIMENTAL

3.1 EXAMINATION OF FORD TORNADO CYLINDER LINERS

Two Ford Tornado liners were obtained for examination, one of which had been used in an engine test to reproduce bore polishing and the other was in an unused condition. Before the liner surfaces were examined they were both cut into two halves, cleaned in white spirit, dried, then cleaned in acetone, then dried again. After which the polished halves were traced onto a plastic sheet to identify the different intensities of bore polishing. Information was then obtained by using the replica technique. This technique is applied by fixing the lower end of a 2cm. strip of acetate film to the liner wall with plasticine and then softening one side of the film with acetone and allowing it to spring into contact with the wall and dry insitu. Attempts have been made to define the severity of polishing by means of Talysurf profiles and surface roughness measurements in terms of C.L.A. values (centre line average).

Further examination was carried out using optical, scanning electron microscopy and energy dispersive x-ray analysis (E.P.M.A.) applied to both the new and bore-polished liners. Taper sections were prepared at an 11.5^{o} angle through both the unused and bore-polished surfaces.

3.2 RECIPROCATING TRIBO TEST MACHINE

A laboratory Tribo test machine has been designed to simulate the reciprocating motion in the engine bore (Fig.15). Essentially the machine reciprocates a flat under a loaded pin. The flat represents the cylinder liner while the pin represents the piston ring (Fig.16). The flat was fastened to the base of an aluminium bath by two screws through holes at either end of the specimen (Fig.17). This bath was fastened to the plate of aluminium by two brass screws through holes at either end of the bath edges. The capacity of the aluminium bath was 50 ml. (Fig.17). Plates of asbestos were fitted underneath the plate of aluminium and a linear bearing frictionless table fixed underneath the plate of aluminium to allow free movement (Fig.18). A friction transducer was fitted in one side of the aluminium plate with the holder of the transducer fixed to one end of the connecting rod and the other end of the connecting rod to a rotating spindle. Reciprocation was achieved by linking one end of the aluminium plate through the holder of the friction transducer to the rotating spindle with a connecting rod. The rod was fixed by screws which clamped through the centre of the ballraces housed at either end of the rod which were free to rotate. The amplitude of reciprocation was selected by positioning the connecting rod in one of several positions at different radii from the centre of the spindle. This spindle was driven by a variable speed 1.5h.p. motor. The Kopp setting was geared to a given speed range of 8.3 to 70 r.p.m. From the rotation speed, the sliding speed was calculated to be 0.008 $msec^{-1}$ (Fig.19). Further anlaysis of the speed data is shown in Figure 19 where at the terminal points the sliding speed was zero and at the middle the sliding speed is equivalent to 0.008 $msec^{-1}$.

A pin specimen is fixed in the holder of the pin (Fig.20) which consists of two parts. Firstly the body holder, and secondly the tracks which are fitted in the body holder so that the holder of the pin fits on the wheel of the slide bearing as shown (Fig.21). This part of the DUA-L-VEE System (Guide Wheel System), which obtains precision linear motion, was fitted in the body holder so that the body holder of the pin fitted on the wheel of the DUA-L-VEE System. The pin was fixed half way along the arm from the pivoted end. A 2:1 advantage of the suspended load was gained at the pin/flat interface. Frictional force was measured in both directions by linear voltage displacement tranducers. The vertical displacement of the arm, representing the combined wear of the pin and the flat, was also measured by a linear voltage displacement transducer. The oil bath was heated by three cartridge heaters located underneath the bath, controlled by a thermocouple underneath the bath and maintaining oil at the required temperature inside the bath (Fig.22). This temperature was monitored by a thermocouple in the oil and displayed as a digital output, after amplification it was continuously recorded on a chart recorder. Two other channels on this chart recorder collected the amplified force and wear signals from the two (L.V.D.T.). The fourth channel recorded the amplified contact resistance between the pin and flat from a contact resistance box. The contact resistance between the pin and the flat was measured by the contact resistance box with the pin and flat separated electrically. A short circuit occurred when the pin contacted the flat, while a higher contact resistance was provided when the lubricant separated the pin from the flat (Fig.23).

Specimen preparation

Pin and flat were machined from the same standard cast iron as piston ring and cylinder liner respectively (Figs.24,25 and Tables 3,4 show the microstructures and compositions respectively). A 7mm. curved length was cut from the ring radius and milled to make a straight square section. Both ends of the pin were turned into a rectangular shape with coordinates of 2mm. and 4mm. in length. After the specimens had been cut to length they were ground to give a surface finish transverse to the sliding direction in a jig using new 300 wet abrasive papers so that abrasive grooves ran parallel to one side of the pin. The specimens were then thoroughly cleaned in an ultrasonic bath, degreased and stored in a desiccator until required. Flat material was cut from the wall of the cylinder liner and milled roughly to shape. Holes were accurately drilled at each end of the flat with the aid of a jig. Finally the surfaces were ground using a freshly prepared wheel on a horizontal grinding machine.

The lay of the final surface to be tested was orientated so that the grooves were 60^{o} to the intended direction of rubbing.

Calibration

Before testing commenced the two transducers, measuring the dimensional change and the friction force, were calibrated over the linear portion of their output. Amplification was set for each signal to give a known displacement on the chart recorder from which measurements of wear and frictional force could be taken after the test. These amp-settings were checked periodically throughout the test programme to maintain reproducibility. The box measuring the contact resistance was made to receive a certain amount of voltage. Amplification was set for the signal from the box to give a known displacement on the chart recorder from which measurements of contact resistance could be taken after each test. This amplification setting was checked periodically throughout the test programme to maintain reproducibility.

Kopp motor variable speed settings were calibrated against the final spindle speed using a stroboscope. All the thermocouples were checked against each other over the range of intended temperature use. The lever arm was balanced with counterbalance weights to give zero load on the pin when only the pin was suspended in the arm.

Test Procedure

The pin and flat were thoroughly cleaned ultrasonically, degreased using white spirit, then dried, cleaned in acetone and weighed before placing them in the bath. The flat specimen was clamped into the aluminium bath using two brass screws. The pin was clamped into a pin holder and fitted into the wheel of the slide linear bearing. 40ml. of the lubricant was then added to the bath, the lever arm was brought down on the pin holder so that the pin and the flat were in contact. Using the threaded adjustable stop, the arm was raised so that there was a 4mm gap between the pin and the flat before the load was suspended from the end of the lever arm. Once a constant temperature of 80°C has been maintained for 45 minutes, the Kopp motor was started and the pin lowered onto the flat specimen. During the time taken for the bath temperature to reach equilibrium the chart recorder was allowed to warm up and was set to run at 60 mm per hour. Wear and friction transducer output, bath temperature and contact resistance, were continuously maintained on the chart recorder. Tests were allowed to run under set conditions for several hours to allow steady state wear to stabilize between the specimens. The time was determined as the maximum time for the surface to run-in which was monitored by the change in friction and wear over the length of the test. The duration of the test was 20 hours.

In order to remove the specimens from the tribo test machine as quickly as possible, the following procedure was followed. Stop the motor, the test being terminated with the arm being lifted off the holder of the pin, and the pin lifted from the flat from its holder. The oil was then removed from the bath and stored for further examination when the motor was switched off, pin and flat specimens were removed and were thoroughly cleaned in an ultrasonic bath, degreased using white spirt, then dried and cleaned in acetone and finally weighed, then the worn specimens were stored in a desiccator for future examination. After each test the bath was flushed with white spirit and then acetone before re-use to remove all traces of oil and debris.

3.3 TEST CONDITIONS

Two series of tests have been carried out on the machine to differentiate between two different lubricants which have become established in the lubricant industry to discriminate bore polishing. These were referred to as RL47 and RL48 as defined by a standard Tornado test (86). Two different oil temperatures were investigated namely, 80^{o} and 150^{o}. Five different loads were also evaluated, these being:

300 N 600 N 900 N 1200 N 1500 N

All these conditions applied for both lubricants in two series, one for RL47 and the other for RL48.

Further tests were also carried out for short terms such as 1, 3, 5 hours and for long terms such as 50 and 100 hours. A series of tests were performed on the machine in the selected range of speed.

3.4 DIAMOND TEST

Oil RL48 which is known to promote bore polishing in Ford Tornado engine tests (86) does not produce bore polishing in the tests previously described. It was decided therefore to promote a smoother surface by adding into RL48 a controlled amount of 0.25 μm diamond paste to produce such a surface. Some short tests were also carried out for 1, 3, 5, 6 hours to give more information.

Test Procedure

The pin and flat were thoroughly cleaned ultrasonically, degreased using white spirit, then dried, cleaned in acetone and weighed before placing them in the bath. The flat specimen was clamped into the bath, then the pin clamped into the pin holder and fitted into the wheel of the slide linear bearing. 40ml of RL48 was added to the bath and once a constant temperature of 80^{O} had been maintained for 45 minutes, then 1gm of diamond paste, size 0.25 μm added to the oil in the bath and left for 10 minutes to mix with the oil. Then the 900 N load was applied on the pin, and the Kopp motor started and left running for 8 hours during which time the friction and wear tranducers output, bath temperature and contact resistance were continuously monitored on the chart recorder. In order to remove the specimens from the bath, the previous procedure used in the Test Procedure section was applied.

3.5 CARBON TEST

It is somewhat artificial to produce bore polishing with diamond paste which is not present in a running engine and therefore samples of hard carbon were obtained from used pistons, because the literature referred to hard carbon on the piston acting as a fine abrasive producing a polished liner (67,68). Therefore it was decided to promote bore polished surfaces in the laboratory by adding into the RL48 a controlled amount of hard carbon to produce such a surface - some short tests of 1, 3 5, 8, 10, 12 hours were carried out.

Test Procedure

The pin and flat were thoroughly cleaned ultrasonically, degreased using white spirit, then dried and cleaned in acetone and weighed before placing them in the bath. The flat specimen was clamped into the bath, the pin clamped into the pin holder and fitted into the wheel of the slide linear bearing. 40 ml. of RL48 was added to the bath, and once a constant temperature of 80^{o} had been maintained for 45 minutes, then 0.8 gm of hard carbon was added to the oil in the bath and left for ten minutes to mix with the oil. Then a load of 1200 N was applied on the pin and the Kopp motor started and left running for 14 hours during which time the friction and wear transducers, bath temperature and contact resistance were continuously monitored on the chart recorder. The carbon conditions were different from those used for diamond paste because the former did not produce bore polish as readily. In order to remove the specimens from the bath the previous procedure in the Test Procedure section was applied.

3.6 SURFACE FINISH TESTS

This test was designed to gain information about the relation between the smoothness of the surface and the breakdown of the oil film under boundary lubrication conditions for the pin and flat. Grey cast iron specimens were metallography prepared.

Test Procedure

A standard procedure of four grades (180, 320, 400 and 600) of silicon carbide papers followed by 6 μm and μm diamond paste respectively were used for metallography preparation for both the pin and the flat, then thoroughly cleaned ultrasonically, degreased using white spirit, then dried, cleaned in acetone and weighed before being placed in the bath. The flat specimen was clamped into the bath, the pin clamped into the pin holder and fitted into the wheel of the slide linear bearing. 40 ml. of RL48 was added to the bath and once a constant temperature of 80^{o} had been maintained for 45 minutes, the load was then applied on the pin and the Kopp motor started and left running for 20 hours during which time the friction, wear transducer, bath temperature and contact resistance were continuously monitored on the chart recorder. In order to remove the specimens from the bath, the previous procedure in the Test Procedure section was applied.

3.7 EXAMINATION OF WORN SURFACES

Before the worn surfaces were examined, they were ultrasonically cleaned in white spirit, then dried and cleaned in acetone to remove all trace of oil. The main tool for the surface examination was a scanning electron microscope (SEM) which allows a depth of focus superior to optical examination. Both the pin and flat specimens were examined in the optical microscope before sectioning and after. In addition to the routine examination, areas of interest were analysed using energy dispersive x-ray analysis (E.P.M.A.) Films formed on the specimens' surface have been examined by both wave length dispersive x-ray and x-ray photoelectron spectroscopy (XPS) or ESCA. The surface roughness measurements and the profile of the pin and flat specimens were measured by using a Rank Taylor Hobson Talysurf instrument. The surface roughness measurements were taken in terms of C.L.A. value (centre line average) for both the pin and flat before the test and after the test. In the case of the flat three positions in the wear track, i.e. at both ends of the wear track and in the middle of the wear track. Several taper sections were made of the pin and the flat in order to examine surface and subsurface effects. The angle of the section was 11.5° to the surface which introduced a factor of 5 times increase in the apparent depth of the surface layer.

Some oil samples were comprehensively analysed by a rotating particle depositor (R.P.D.). Optical and scanning electron microscope (SEM) were employed for examination of wear debris separated out from the oil by means of a rotary particle depositor.

CHAPTER FOUR

RESULTS

4.1 EXAMINATION OF FORD TORNADO CYLINDER LINERS

Two Ford Tornado liners were examined, one had been used in an engine test to reproduce bore polishing and the other was in an unused condition. One method of recording polished areas is to trace outlines onto a clear plastic sheet. The results of tracing the polished area are shown in Figure 26 confirming its concentration around TDC (Top Dead Centre).

Attempts have been made to define the severity of polishing by means of Talysurf profiles and surface roughness measurements in terms of C.L.A. value (Centre Line Average); on the basis of optical examination, three areas were selected for obtaining Talysurf traces and these are shown in Figure 27 which confirms the very smooth nature of the polished condition.

Photographs taken of both the unused and the polished liner are shown in Figure 28. From the photographs it appears that this technique was not entirely satisfactory.

Sections were cut through the thickness of the liners and examined in the polished and etched condition (Fig.29a). In the unetched condition it shows an undercooled graphitic structure, A.S.T.M. 267 Type B/D, in the size range 6 to 8 (Fig.29b). After etching it shows a normal pearlitic structure with some phosphide but no free ferrite. Table 5 shows the composition of the liner.

An examination was carried out using a plastic replica; optical and scanning electron microscopy techniques used on both the unused and the polished liner enable more information to be obtained about the bore polishing.

A photograph of the cross-honed structure of the unused bore is shown in Figure 30 and this represents a finish which has an ideal plateau groove surface to retain an oil film. A section from a heavily polished area of the bore is shown in Figure 31, and it will be observed that all the cross honing has been completely removed and abrasion marks are evidence, running in the ring travel direction. The graphitic structure of the cast iron can be observed quite clearly and there is no evidence of corrosive attack or surface deposits. Figure 32 shows a section of medium polish, taken from the polished bore, with faint original honing pattern, while Figure 33 shows the lightly polished section with a mirror finish on the cross honing pattern. Taper sections were prepared at an 11.5^{o} angle through both the unused and the heavily polished surfaces. Figure (34a) shows evidence of the cross honing at the surface while Figure 34b shows that on the polished surface the cross honing has been completely removed and the surface is now much smoother than before. There is no evidence of deformation of the surface or of a glazed surface deposit. Etching shows the pearlitic structure to be free from plastic deformation and there is no evidence of any white transformation layer which would have indicated scuffing (Fig.34c).

A Talysurf trace was taken through from an unworn area to a polished area. Figure 35 shows that the trace passes from a cross honed area into a run-in area before the polished area, with considerable smoothing occuring, but this trace also shows that wear of only approximately 1 μm in depth has occurred.

4.2 WEAR RESULTS FOR BOTH OILS

The vertical movement of the lever arm on the tribo test machine (Fig.15) as measured by a displacement transducer was recorded along with temperature, frictional force and contact resistance, using a four channel chart recorder for the duration of the test. A section of this chart is shown in Figure 36.

The total weight loss of both the worn pins and flats for the first series of tests, using the fully formulated oil RL47, are given in Table 6. These results show that, in general, wear of the specimens resulted in weight loss, increasing with the severity of the test (i.e. for increases in load and temperature). However, on further analysis of these results and comparison with the visual inspection of the worn samples, it was concluded that weight loss was not an adequate criterion to evaluate wear. The main problem being attributed to:

1. Surface pick-up of the wear debris and other extraneous matter, despite ultrasonic cleaning of the specimens.
2. Deformation of the pin resulting from the wear process which causes a skirt to form around the pin.
3. Oil absorption causing an increase in specimen weight.

Thus throughout these series of tests the wear of the specimen, evaluated as a volume reduction from the chart recorder, 'the volume loss of both pin and flat' was continuously monitored and finally converted into the wear volume. The variation of volume loss against sliding distance as a function of load for two oil bath temperatures of 80^{o}C and 150^{o}C, are shown in Figures 37,38. The analysis of the wear curves show that irrespective of either load or temperature, two basic periods of wear existed:

1. An initial higher wear rate running-in period (higher wear rate calculated as a total volume of material removed from the pin and flat expressed in mm^3 for relative sliding of the pin over the flat which depends on the test conditions, and may extend to approximately 200m sliding distance).
2. The equilibrium wear period. This region of wear had a lower magnitude than that of the running-in period. This period is referred to as equilibrium wear.

The running-in and equilibrium wear portions were then identified so that the equilibrium wear could be calculated and the rate of wear of the equilibrium period was evaluated as shown in Table 7 and expressed graphically in Figure 39 which shows the higher wear rate with higher temperatures and wear rate increased with increasing loads (Fig.39).

Further analysis of wear curves suggested the increase in the oil bath temperature from 80^{o}C to 150^{o} provided an increased wear rate which is mainly due to the viscosity of lubricant reduced with increasing the oil bath temperature.

Further analysis suggested that increasing the load from 300N to 1500N resulted in increased wear which is clearly due to the condition of the boundary lubrication which applied during the test. The change in wear rate over the range of loads tested was similar in all cases. The curves exhibit an exponential increase in wear rate as the load increases. The second series of tests was identical to the first except that the lubricant was a fully formulated RL48. The total weight loss of pin and flat are given in Table 8. Examination of these results show a similar trend to that for the RL47 lubricant, i.e. wear of specimens resulted in weight loss that increased with the severity of the test (increasing load and temperature). Further analysis of the results revealed that in general the weight loss for RL48 was greater compared to the RL47 lubricant, although some specimens actually increased in weight. The evaluation of wear by weight loss was considered inadequate.

The variation of volume loss against sliding distance for the two oil bath temperatures 80^{o}C and 150^{o}C, is shown in Figures 40,41 respectively. These curves also show the same two basic periods of wear as discussed previously for RL47. Comparison of these wear curves with Figures 37,38 revealed an apparent increase in wear rate when using RL48 compared with RL47.

Further analysis of the volumetric curves resulted in the calculation of the equilibrium wear period. These values are given in Table 9 and expressed graphically in Figure 42. The general form of equilibrium wear rate curves are similar to those observed for the RL47 lubricant, i.e. a higher wear rate with increasing load and temperature (Fig.42) and a higher wear rate with RL48 (Fig.39) compared to the RL47 (Fig.42).

The two fully formulated lubricants, the RL48 which produced bore polishing and the RL47 which prevented bore polishing when used in a Ford Tornado engine test as a standard method, were compared in terms of wear under laboratory test conditions and the difference between the two oils, although very small at lower loads, was quite marked at higher loads. The material removed from both surfaces (pin and flat) was greater with the RL48 as a lubricant compared to the RL47 as a lubricant (Tables 6,8).

Tribo test machine results showed that the design and test procedure were sufficiently sensitive to measure the modification in wear rate corresponding to the changes in operating conditions. A higher wear rate was found when the speed of the wear machine was increased (Fig.43) obviously due to an increase in the sliding distance of the pin over the flat.

4.3 FRICTION RESULTS FOR BOTH OILS

The frictional force between the pin and flat is transmitted through the rod which contacts the bath with the spindle and the movement of this rod was monitored by L.V.D.T. (Linear Voltage Displacement Transducer). In this way the frictional force could be monitored along the stroke length in both directions and the frictional force recorded by the transducer in both directions with accurate measurement due to the position of the transducer. A continuous recording of this force was retained on a chart recorder (Fig.36). A storage oscilloscope was also used to retain some of the force traces over the length of the pass. The stored traces could then be transferred to a chart paper at a slow speed to ensure that the response of the recorder was not responsible for the loss of any relevant information. A trace showing the variation of friction along the pin transverse is shown in Figure 44. The friction force measured between the pin and flat for unlubricated tests, run at room temperature on the same machine, did not show a similar increase at the end of the stroke (Fig.45).

The average frictional force throughout the test duration for the first series of tests using the fully formulated lubricant RL47 was calculated and is illustrated graphically in Figures 46,47 for the two oil bath temperatures 80^{o}C and 150^{o}C respectively. Examination of these graphs showed that in the majority of cases the friction peaked at the initiation of the test and then decreased to a state of constant value. Further analysis revealed that while the load increased, the frictional force increased, which may be due to the reduction in the oil film thickness, resulting in the surface to surface contact being increased. Further observation suggested that the increase in temperature from 80^{o}C to 150^{o}C resulted in an increase in frictional force which may be associated with the viscosity of the lubricant. The coefficient of friction was calculated by the simple formula:

$$\mu = \frac{F}{L}$$

μ	=	Coefficient of Friction
L	=	Normal Load
F	=	Frictional Force

From the formula the coefficient of friction was calculated as shown in Figure 48.

$$\mu = 0.08 \text{ to } 0.13$$

The average friction force for the second series of tests for the fully formulated RL48 are illustrated in Figures 49,50 for the two oil bath temperatures, $80^{o}C$ and $150^{o}C$. These curves show the same trend of curves for RL47 except that the frictional force for RL48 is higher than than for RL47, with the same increase in frictional force produced by increasing the severity of the test (increasing the loads, and temperatures). The coefficient of friction for RL48 is in the range of:

$$\mu = 0.10 \text{ to } 0.16.$$

as can be seen in Figure 48. This range is higher than that recorded for the RL47 lubricant (Fig.48).

A lower friction force was obtained when the speed of the motor was increased. Figure 51 shows the frictional force against the sliding distance for different speeds. The tribo test machine differentiates between the two oils in terms of friction. Figures 46,47,49 and 50 show a higher friction force with RL48 compared to RL47 and a higher coefficient of friction with RL48 compared to the coefficient of friction of RL47 (Fig.48).

4.4 CONTACT RESISTANCE FOR BOTH OILS

In order to obtain an indication of the oil film thickness between the pin and the flat, it was decided to measure the contact resistance between the pin and the flat. A potential divider with 100 logarithmic potential was designed to obtain an estimate of the contact resistance on the chart recorder.

A 16mv potential difference is applied across the contact and is also monitored on the chart recorder. A reading of 0 Ω indicates metallic contact whilst 100 Ω corresponds to the separation of the pin from the flat by an oil film thickness which is called thick boundary oil film. Throughout the first series of tests where RL47 was used as a lubricant, the contact resistance evaluated from the chart recorder and plotted against sliding distance of the pin over the flat for two oil bath temperatures, 80°C and 150°C, is shown in Figures 52, 53. With a low load (300 to 600N), at both temperatures an oil film builds up between the pin and the flat at the higher loads (900, 1200 and 1500N) for both temperatures, resulted in a very high metallic contact indicated by a lower contact resistance between the pin and the flat. But there is no difference between the contact resistance in terms of the two different oil bath temperatures (Figs.52,53). In the second series of tests where RL48 was used as a lubricant, the contact resistance was evaluated from the chart recorder and plotted against the sliding distance of the pin over the flat for the two oil bath temperatures, 80°C and 150°C (Figures 54,55).

There is a similar trend for the contact resistance for the RL47 which at low load shows an oil film build up separating the pin from the flat, and at high loads where the contact resistance was very low (higher metallic contact). There is no difference in contact resistance with the increase of temperature from 80^{o}C to 150^{o}C. There is no difference between the two lubricants, RL47 and RL48, in terms of contact resistance as seen in Figures 52-55.

4.5 WEAR EXAMINATION

In the two oils tested there was no difference between the tests run with RL47 or RL48 in terms of wear mechanisms. Examination of the flat surfaces shows that wear is predominantly by abrasion and delamination wear in both oils.

Abrasion is shown for both oils in Figures 56,57 and it will be observed that both consist of regular abrasion grooves running in the rubbing direction, all of the machining marks have been removed. Figures 58,59 show flat surfaces with delamination wear, which occurred in both oils, with a plate-like debris, especially with very high loads, while there is no sign of a graphite structure on the surface of the flats for both oils.

The pin surfaces in both oils also contained both wear mechanisms, abrasion and delamination wear; abrasion grooves on a fine scale which are smooth (Figs.60,61) and delamination wear which produced plate-like debris (Figs.62,63) for both oils. There is a difference between the test run with a high load (Fig.58) where the delamination wear and abrasion wear were predominant; while abrasion and plastic deformation were predominant with low loads (Fig.64). From such examinations the capability of the tribo test machine to differeniate between the low wear condition and severe wear condition proved possible. The length of the wear track on the flat surface was 30mm (Fig.65) and the cross section of the pin was 2mm in length on each pass. Therefore the pin received a sliding distance of 15mm where each point on the wear track had a sliding distance of only 2mm. The difference between the two oils in terms of C.L.A. value (Centre Line Average) for the flats at 80°C (Fig.66) showed at a lower load the surface in both oils is slightly smoother than the unworn surface (0.55 μm being the unworn surface value) and rougher at higher loads particularly for the RL47. For the RL48 oil the surface was slightly smoother than the RL47 surfaces for 80°C, while for 150°C (Fig.67) at higher loads the surface becomes very rough especially for RL47 compared to RL48. Overall the surfaces were smoother with the RL48 compared to RL47. In general the final surface finish of the flats become rougher with increasing oil bath temperatures and loads, however, the effect of load was often masked due to the very high wear rate, thus it may be concluded that the final surface is also dependent on the test conditions (temperatures and loads).

Figures 66,67 show there is no indication of C.L.A values less than 0.125 μm, which is the C.L.A. value for a polished bore (65) and also shown in Figure 27. For the pins the difference between the two oils, in terms of C.L.A. values, showed smooth surfaces with the RL48 compared to the RL47 (Figs.68,69) and smooth surfaces with low load and temperature compared to the higher load and temperature (increasing the severity of the test "load and temperature") which showed an increase in the roughness of the surface.

In lubricated tests an oil deposit film developed on the surfaces which was not present on the unused surfaces. This film had a dark grey appearance when examined optically (Fig.70) and was restricted to the wear track. Although no quantative analysis was available from electron microprobe analysis (E.P.M.A.), it was possible to compare qualitative information from three samples of the same size. One sample was unworn, the second sample was lubricated with RL47 and the third sample was lubricated with RL48. The first sample (Fig.71) shows the composition of the material (grey cast iron) unused flat; the second sample (Fig.72) shows the film building up on the surface when using RL47 as a lubricant. There is one peak of sulphur (S) and very small peaks of calcium (Ca) and phosphorus (P). While the third sample (Fig.73) shows the presence of three large peaks, one for sulphur (S), one for calcium (Ca) and one for phosphorus (P) and these are larger than the peaks in the second sample which is lubricated with RL47.

The films which were formed on the two samples have been examined by x-ray photo electron spectroscopy. Figures 74,75,76 show that the sulphur (S) which forms in both oils, is in the form of oxide which is SO_3, and the same for calcium (Ca) and phosphorus (P) which are also are in the forms of oxide which are CaO and PO.

Examination of the flats for both lubricants (RL47 and RL48) showed no sign of a bore polished surface, i.e. no appearance of graphite structure on the surface, with C.L.A. values of not less than 0.125 μm as seen in Figures 27,31 for Ford Tornado liners.

Further examination revealed wear behaviour changes on the flat surface where it could be seen that the deposit area was concentrated in the middle of the wear track (Fig.77). Both ends of the wear track had predominent wear by abrasion and plastic deformation (Fig.78), whilst almost all the original machining marks were removed at both ends of the wear track compared to the middle of the wear track where the original machining marks were still visible (Fig.78). Evidence of this is the measurement of the C.L.A. value for the three positions as seen in Figure 79. The running-in and equilibrium wear stages are both identified in the wear result but the behaviour of the surfaces of the pin and flat during these stages was examined by running two different tests, first by a short term test and then a long term test.

Figures 80,81 show the unused pin and flat with the original machining mark appearing on both surfaces (pin and flat). After one hour the pin surface becomes smooth (Fig.82) with no sign of the original machining mark but some marks of abrasion and some cracks (which led to initiation of delamination wear) appeared on the surface of the pin (Fig.83). After one hour the surface of the flat became smoother (Fig.84) but still some of the original machining marks can be seen on the surface of the flat with two wear mechanisms appearing. These are abrasive wear and plastic deformation (Fig.85). As well as the removal of material from both the pin and the flat it showed that this removal increased with time for both the pin and the flat (Table 10) and it also showed that the removal of material from the pin was much higher than the removal of material from the flat. After running the test for three hours, the same conditions applied for the pin and the flat and both became smoother than after the one hour test. The results of the one hour test are shown in Figures 83, 85 and for the three hour test in Figures 86,87. After five hours both the pin and flat surfaces became smoother (Figs.82,84) with two wear mechanisms operating on the surface of the pin, abrasive wear and delamination wear (Fig.88). The flat showed some evidence that the plastic deformation is concentrated at the end of the wear track (Fig.89) which may be due to the complete contact between the pin and flat which leads to delamination wear. Another wear mechanism, abrasive wear, also appeared on the flat (Fig.89). After eight hours the pin surface operated under two wear mechanisms, abrasive and delamination (Fig.90), with all the original machining marks removed giving a smooth surface.

For the flat the surface was smooth with both abrasive wear and plastic deformation appearing on the surface (Fig.91) and also some sign of delamination wear. After a ten hour test, both surfaces became rougher (Figs.82,84) with both abrasive wear and delamination wear appearing on both the pin and the flat surface (Figs.92,93). These two mechanisms continued up to 100 hours of testing (Figs.96,97). At 20 hours the surfaces become smoother with still both abrasive and delamination appearing on the surfaces (Figs.94,95) while Figure 98 shows the wear curve for 100 hours with two wear stages, the running-in stage and terminating with the equilibrium wear stage. The surfaces became rougher with the increase in time, which may be due to debris trapped between the two surfaces.

Examination of the debris extracted from the lubricant using a R.P.D was carried out and the large size debris being produced during the course of the experiment was plate-like particles, in both oils, RL47 (Fig.99) and RL48 (Fig.100), which indicated that delamination wear was operating during the test. Figure 101 shows the debris produced after a one hour test as plate-like particles less than 10μm in size, which suggests that adhesive wear was operating during the test and that significant debris was produced after just one hour. Figure 102 shows the debris produced after an 8 hour test as plate-like particles which suggests that adhesive wear is the mechanism responsible during the test. Figure 103 shows the debris produced in a 100 hour test as large plate-like particles, larger than 50μm, which suggested that delamination wear was operating during the test.

4.6 BORE POLISHING PRODUCED BY DIAMOND PASTE

Oil RL48 is known to promote bore polishing in the Ford Tornado test but it does not produce bore polishing in the test described in section 4.5 although the surface produced with RL48 was smoother than that produced by RL47 (Figs. 67,68). It was therefore decided to promote smoother surfaces by adding a controlled amount of diamond paste (Fig.104) size 0.25μm to RL48. The smoothest surface produced in the series of tests with RL48 lubricant at 900N (load) and 80^{o}C temperature (oil bath temperature) was chosen as a test condition for attempting to produce a bore polished surface in the laboratory. Two tests were carried out, one was run with RL48 + diamond paste and the other was run with RL48 only. The results showed that much more material was removed from both the pin and flat in the test run with RL48 + diamond paste compared to the test run with RL48 only (Table 11). The results showed higher wear (Fig.105) and higher friction (Fig.106) with RL48 + diamond paste compared to RL48 only, and a lower contact resistance with RL48 + diamond paste compared to RL48 only (Fig.107).

The Talysurf profile and C.L.A. value revealed that a smoother surface resulted with RL48 + diamond paste (Fig.108) compared to RL48 only (Fig.109); with C.L.A. values less than 0.125 μm using diamond paste compared to 0.20 μm on the test run using only RL48. Smoother surfaces were produced at both ends of the wear track compared to the middle of the wear track.

Further examination using both an optical microscope and scanning electron microscope (Fig.110) revealed a smooth surface with a graphite structure appearing on the surface with the diamond paste test, especially at both ends of the wear track (Fig.111) but there was no sign of a graphite structure appearing on the surface in the test run with only RL48 (Fig.91). Abrasive wear appeared on the surface of the diamond paste test (Fig.110) and the beginning of delamination wear also appeared on the surface (Fig.112). It was observed that abrasive wear has clearly revealed a dispersed hard phase by differential wear (Fig.113). Figure 114 shows some mottled areas with a thumb-print structure and hard phase appearing on the surface. This indicates a corrosive action selectively attacking the different constituents in the cast iron. In some areas the attack is rather patchy with a bright smooth surface on one part where the graphite structure appeared while the other part is etched (Figs.115,116).

Further examination was carried out using microprobe analysis. Two samples were taken, the first for a flat lubricated with RL48 + diamond paste and the second sample lubricated with RL48 only. The examination revealed that the first flat (Fig.117) showed three large peaks, one for S, one for Ca and one for P, while the second sample (Fig.118) showed three peaks also for S, Ca and P. When the peaks of the second sample were compared with those of the first sample, the peaks were higher in the first sample. Further examination using E.S.C.A. analysis (Fig.119) showed that these peaks in the form of oxide were SO_3, Ca0 and P0. A taper section was taken through the diamond paste flat and an unused flat. It revealed (Fig.121) that there was no glaze or deposit but there were some marks of the original machining mark on the surface, and no deformation or white layer appeared on the surface but some grooves of abrasive wear appeared on the surface of the diamond paste flat. The unused flat (Fig.120) showed only the original machining marks were visible on the surface.

Different sizes of diamond paste were also used. Using 1μm diamond paste, the results showed there was higher wear (Fig.105), higher friction (Fig.106) and lower contact resistance (Fig.107) when compared to a test using 1/4μm diamond paste. Removal of material from both the pin and the flat was higher with 1 μm diamond paste compared to using 1/4μm diamond paste(Table 11). The optical and SEM examination showed a smooth surface with a graphitic structure appearing on the surface of the 1μm diamond paste test (Fig.122), but the Talysurf profile and the C.L.A. value showed (Fig.123) higher values than 0.125 μm was produced. Therefore using 1μm diamond paste did not produce a surface with the characteristics of bore polishing as seen in Figures 27,30, with graphite structure appearing on the surface and C.L.A. value less than 0.125 μm.

Short and long term tests were also carried out. After one hour the flat became smoother (Fig.124) with some of the original machining marks starting to be removed, some graphite structure appearing on the surface with both abrasive grooves and plastic deformation appearing on the surface (Fig.125). After a three hour test a comparison was made with the one hour test. This showed that after three hours the surface had become smoother (Fig.124), the removal of material increased (Table 12) and abrasive wear and plastic deformation (Fig.126) appeared on the surface with the graphite structure also appearing on the surface. After five hours the surface was smoother and much more of the original machining marks had been removed (Fig.127), with abrasion; plastic deformation and a graphite structure appearing on the surface (Fig.128). After eight hours the surface was as described before (Figs.108,110,112) but after a ten hour test the surface became rougher with a graphite structure appearing on the surface (Fig.124,129) and a C.L.A. value higher than 0.125 μm which did not agree with the characteristics of bore polishing (which is less then 0.125 μm). This rougher surface may be due to the debris which is trapped between the surfaces. Examination of the debris produced by R.P.D revealed that after one hour, plate-like particles were produced (Fig.130) and the same kind of debris was produced after the eight hour test (Fig.131) which suggests that the wear mechanism involved was delamination.

4.7 BORE POLISHING PRODUCED BY CARBON

It is somewhat artificial to produce bore polishing with diamond paste which is not present in a running engine, so samples of hard carbon (Fig.133) were obtained from a used piston; the hard carbon referred to in the literature (64,67) appears to be one of the reasons for causing bore polishing.

This section is divided into two. The first section is concerned with trying to produce a bore polished surface using RL48 + hard carbon and the second section with trying to differentiate between the two lubricants RL47 and RL48 when carbon was used in both lubricants to reproduce the bore polishing surface. By doing so the conditions which exist in the engine are simulated.

Two tests were carried out, one using hard carbon + RL48 and the other using RL48 only with a 1200N load on the pin and an oil bath temperature of 80^{o}C. The test was run for 14 hours. The test conditions used for the diamond paste test did not produce a bore polished surface when carbon was used instead of diamond paste, and when the load was increased, different conditions appeared. After running the two tests, the results showed that higher wear (Fig.134), higher friction (Fig.135) and lower contact resistance (Fig.136) were obtaining using RL48 + carbon compared to the test using only RL48.

In the test using RL48 + carbon, the Talysurf profile measurements showed a smooth surface with a C.L.A. value less than 0.125 μm (Fig.137) compared to the test run with RL48 only which showed a relatively smoother surface with a C.L.A. value higher than 0.125 μm (Fig.138), and a smoother surface at both ends of the wear track compared to the middle of the wear track (Fig.137).

Examination with the optical microscope and scanning electron microscopy showed a smoother surface with almost a total absence of the original machining marks, with the graphite structure appearing on the surface (Fig.139) of the carbon test, while for the test run with RL48 only, there was no sign of graphite structure (Fig.140) appearing on the surface. Further examination of the carbon test showed evidence of two wear mechanisms, abrasive wear (Fig.139) and delamination wear (Fig.141). There was also differential wear involved in the carbon test as seen in Figure 142 where the hard phases stand out clearly on the surface. There is also some mottled areas with a thumb-print structure appearing on the flat surface (Fig.143), with hard phase. Therefore using carbon with RL48 produced a surface (Figs.137,138,139) similar to the surface produced on the Tornado liner (Figs.27,31), which did not occur when RL48 only was used (Figs.138,140). This would indicate that the carbon is responsible for producing the bore polishing surface in the laboratory conditions.

Further examination was carried out using microprobe analysis. Two samples of the same size were examined, one lubricated with RL48 + carbon and the other using RL48 only. With the first sample using RL48 + carbon the results showed three large peaks of S, P and Ca (Fig.144) while with the second sample using RL48 only, the results showed the same peaks (Fig.145) but smaller. The E.S.C.A. examination also emphasized these peaks in the form of oxide (Fig.146). A taper section taken through the flat of the carbon test (Fig.147) showed no sign of the original machining marks, no sign of glaze or any deposit, and no deformation or white layers appearing on the surface (Fig.147).

Short term and long term tests were also carried out using carbon. After one hour the flat surface became smoother (Fig.147) with some of the original machining marks removed and a graphite structure starting to appear, with abrasive wear and plastic deformation (Fig.148). The same process continued up to 5 hours testing, after which the surface became relatively rougher (Fig.148) with some graphite structure appearing on the surface with abrasive wear, plastic deformation and the start of delamination wear (Fig.150). After the test had run for eight hours the surface became smoother (Fig.148) with the same mechanism as for 5 hours, with a smooth surface, graphite structure, abrasion and delamination wear (Fig.151) appeared on the flat surface.

Table 14 shows the removal of material from the pin and flat during the short and long term tests showing that the removal of material increased with time, and removal of material from the pin was much higher than from the flat. After running the test for 10 hours up to 14 hours (Fig.152) results showed the same mechanism as previously shown for the 14 hour carbon test, while after running the test for 16 hours results showed that the delamination wear was the predominant wear (Fig.153) on the surface.

The same conditions for producing bore polishing surfaces using hard carbon + RL48 was repeated using RL47 instead of RL48. Results shows that no graphite structure appeared on the surface (Fig.154) with C.L.A. values greater than 0.125 μm (Fig.155), with less wear (Fig.134), friction (Fig.135) and higher contact resistance (Fig.136), and also the material removed from the pin and flat was relatively less when compared to the test using RL48 + carbon (Table 13). The microprobe analysis revealed three peaks of S, P and Ca (Fig.156) appearing on the surface using RL47 + carbon but these were smaller peaks compared to the peaks using RL48 + carbon (Fig.144). Therefore using RL47 with carbon (Figs.154,155) did not produce a surface similar to the Tornado liner surface (Figs.27,31) "which is the exposure of the graphite structure and C.L.A. values less than 0.125 μm ". This indicated that carbon is not the only cause of bore polishing, but the lubricant and its additives also play an important part. Examination of the debris for RL48 + carbon test showed large plate particles with abrasive wear grooves appearing on the surface of the debris which indicated that the wear mechanisms involved during the test were delamination and abrasion (Fig.157).

4.8 SURFACE FINISH RESULTS

Samples of the pin and flat were prepared by a standard procedure using four grades of silicon carbide papers followed by 6 μm and 1 μm diamond for metallography preparation as shown in Figures 24,25 and then the pin was reciprocated against the flat under test conditions of 1200N load applied on the pin at 80^{o}C (oil bath temperature); the fully formulated lubricant RL48 was used and the test was run for 20 hours under boundary lubrication conditions with both surfaces very smooth (Fig.158).

The wear results (Fig.159) indicate a running-in period with a high wear rate continuing up to 150m sliding distance and then a period of very little wear due to the oil film which built up separating both surfaces. This kind of film is called the boundary oil film as indicated by the contact resistance measurement for the pin and flat (Fig.160). The oil film then breaks down which leads to an increase in the wear rate, then the start of the equilibrium period, continuing up to 350m sliding distance, and then the two surfaces are separated by the boundary oil film again, then this breaks down and wear increases and so on, as shown in Figure 159.

Examination with the optical microscope and scanning electron microscope showed on the surface of the flat evidence of abrasive wear and delamination wear (Fig.161) and there are etched hard phases as shown in Figure 162.

Two samples of the same size, one unused and the other already worn, were taken for examination by microprobe analysis and this showed an oil film build up on the worn surface where there are three large peaks of S, Ca, and P on the surface of the flat (Fig.163). The running-in period appears to be associated with abrasive wear and corrosive wear from the examination of the flat (Fig.164). Equilibrium wear is associated with abrasive wear and delamination of the flat (Fig.165).

Examination of the pin showed abrasive wear with corrosive wear (Fig.166) for the running in period. Examination of the debris produced after 20 hours testing showed plate-like particles as shown in Figure 167 which suggests adhesive wear as the wear mechanisms associated with this test.

From the results shown in Figures 158,159,160, there appeared to be a relationship between the surface finish and the oil film separating the surfaces; where there is an optimum surface finish. The oil is retained on the surface thus reducing wear but as the surface becomes smoother and smoother is gets more difficult to retain the oil film and lubricate the surface. These tests were carried out on smooth surfaces as seen in Figure 158 with the wear increasing (Fig.159) due to the difficulty of lubricating such a smooth surface.

HAPTER FIVE

DISCUSSION

1 INTRODUCTION

The literature of diesel engine wear has been associated with abrasion, scuffing and corrosive attack and of these abrasion is the most common. Abrasion and corrosion are believed to account for the majority of wear during normal engine operation. Scuffing is usually associated with the piston ring, particularly at top dead centre (TDC), which subsequently transfers to and damages the bore. From time to time problems occur with new designs but these are usually eliminated before going into production and other developments have taken place in materials, design and surface finishing in order to eliminate other problems which have arisen in the past.

Since the mid 1970's there have been more frequent references to bore polishing which appear to have arisen since the introduction of turbo charged diesel engines. In this investigation the main characteristics of bore polish have been identified, establishing the method for identifying characteristics and understanding the bore polishing problem.

A tribo test machine was used to simulate bore polishing. This work was aimed at producing bore polishing and improving the basic understanding of the mechanism of bore polishing using hard carbon. This avenue of study culminated with a tribo test machine, designed and built to incorporate the wear mechanisms in a lubricated environment.

The first use of the tribo test machine was to investigate the difference between the two fully formulated oils, RL47 and RL48, in terms of wear, friction, contact resistance and surface roughness. In a second series of tests on the tribo test machine particular attention was paid to produce bore polishing using an abrasive particle (diamond paste) to produce wear surfaces similar to those encountered on cylinder components during the normal Ford Tornado engine's operation. A third series of tests on the tribo test machine paid particular attention to producing a bore polishing surface using hard carbon thus producing the wear surfaces similar to those encountered on cylinder components of the Ford Tornado engine.

In addition to this approach there was involvement in the specification of the parameters which included a provision for monitoring facilities to obtain the maximum amount of information relating to the wear of the piston ring, cylinder liner and bore polishing. Although some of the analysis work was conducted elsewhere, examination and interpretation of the ring and liner surfaces and replicas were performed by the author to give a complete picture of the wearing process. Results from the various sources have been collected to form the basis for the engine test discussion.

.2 EXAMINATION OF THE FORD TORNADO CYLINDER LINER

Bore polishing in cylinder liners of diesel engines is self evident by the clearly defined areas of bright mirror-finish (highly reflective) on the liner surface. This is caused by local mechanical wear and usually occurs after a long period of service at high speeds and loads.

Different intensities of polishing can occur and a broad classification has been agreed as follows:

Heavy polish: A mirror finish showing no traces of the original honing pattern (Fig.31)

Medium polish: A mirror finish showing faint signs of original honing pattern (Fig.32).

Light polish: A mirror finish overlaid on the original honing pattern (Fig.33).

Two Ford Tornado liners were obtained for examination, one had been used in an engine test to reproduce bore polishing and the other was in an unused condition (Fig.28). One method of recording polished areas which has been found to be particularly useful is to trace their outline on a clear plastic sheet laid out in the bore. This has the advantage of producing a permanent record and also accurately locates the position of the different types of polish in the liner. The result of tracing the polished areas are shown in Figure 26, confirming its concentration around TDC, and this may suggest that the interaction between the piston ring and cylinder liner play an important role in the cause of bore polishing. The direct tracing technique appears to be satisfactory although it has been difficult in many instances to decide whether it is a lightly polished area or a normal partially run-in zone or the early heavy polish stage.

Another technique for assessing the extent and severity of bore polishing is rated as a percentage of the liner swept area (above ring bottom reversal). Area rating has presented many problems in the field of bore polishing because of the difficulty in determining the boundaries of the polished area. The most effective technique for rating the bore polishing is the SKF laser scanning method. By using this equipment, a whole liner has its bore surface scanned by laser and the polished areas mapped. This is the fastest and most accurate method so far considered but unfortunately the high capital cost of the equipment makes it unlikely to be widely adopted.

Photographs are not a very satisfactory method of recording polished surfaces, mainly due to lighting and light reflection problems, coupled with the curvature of the liner surface as seen in Figure 28, for both the bore polished liner and the unused liner.

Attempts have been made to define the severity of polishing by means of Talysurf profiles and surface roughness measurements in terms of C.L.A values (Centre Line Average). However, characteristics of the original bore surface finish and the condition of normally run-in surfaces are found to affect interpretation of the results on this basis.

On the basis of the optical examination three areas were selected for obtaining Talysurf traces and these showed results fairly typical of bore polished Tornado engines (Fig.27). These results also confirmed the very smooth nature of the polished condition. From this examination it was identified that there is a correlation between the optical examination (Figs.31,32,33) and the Talysurf profile, C.L.A. values, for the polished area (Fig.27). It has been stated (55) that visual rating of polished areas in the liner did not correlate with Talysurf profiles or C.L.A. values for the polished areas. This examination showed that a heavy bore polish surface can be identified by visual rating and relate to the

C.L.A. Therefore there is a correlation between heavy polish visual rating and the Talysurf profile C.L.A. value.

The structure and composition of the two liners is the same and represents a good quality commercial product for this type of application as seen in Figure 29 and Table 5.

The replica technique shows the cross honed structure of an unused bore (Fig.30a) and this represents a finish which produces an ideal plateau grooved surface to retain an oil film. When a comparison is made between the replica technique (Fig.30a) and optical technique (Fig.30b), for the unused liner, this showed a good correlation has been established and it is recommended that replication should be more widely used in the study of friction and wear.

A section from a heavy polished area is shown in Figure 31, all of the cross honing has been completely removed, abrasion marks can be seen running in the ring travel direction (Fig.31), the graphite structure of the cast iron can be observed quite clearly and there is no evidence of corrosive attack or surface deposit. This examination to obtain the Talysurf profile and C.L.A. values produced a very good technique for identifying bore polishing in the liner. McGeehen (67) did not use the scanning electron microscope for identifying bore polishing and different intensities of polishing, Hogmark and Alexander (51) did not use C.L.A. Values or optical techniques for identifying the bore polishing.

Using the following techniques therefore provides a good approach to rating, identifying, understanding bore polishing and its mechanisms.

1. visual and tracing techniques,
2. Talysurf profile and C.L.A. value,
3. optical, replica and S.E.M.,
4. taper section,

Taper section (Fig.34.6) indicated that the polished liner had these characteristics:

1. The smooth surface with cross honing had been completely removed.
2. No sign of glaze or deposit.
3. No evidence of a white layer which would have indicated scuffing.

The onset of bore polishing is characterised by increasing oil consumption and blow-by, in which exhaust gases leak past the piston ring pack leading to a reduction in power output and because of lubrication failure, eventual seizure.

There has been some suggestion that blow-by and seizure are a result of excessive local wear in the bore polish area which leads to a loss of seal. This hypothesis has been tested by taking a Talysurf trace through from an unused to a polished area (Figure 35). This trace shows that wear of only approximately 1 μm depth has occurred and it is not clear whether the problem is caused by the amount of local wear leading to excess clearance and loss of the seal, or whether it is due to the difficulties associated with the retention of a lubricant on a smooth surface. It is believed that the latter is the dominating factor. The bore polish process also involved the removal of material as seen in Figure 35 and this agrees with Parsons (73) who stated that bore polishing involved the removal of very small amounts of material during the process.

It has been stated (55) that there appears to be a change in the mode of oil distribution at the polished surface, perhaps with a lessening of the ability of the surface to retain oil, accompanied by a loss of control of oil flow up the bore and past the ring. Wilson and Calow (64) stated that there appeared to be a change in the mode of the oil distribution at the surface, perhaps caused by a lessening of the ability of the surface to retain oil or perhaps with the establishment of an oil flow up the bore between the worn polished areas and the piston ring. Knight (77) stated that it has been generally agreed that bore polishing will be a component of three modes of service failure or complaint. Engine seizure occurs where bore polishing prevents good oil retention and piston ring scuffing follows. Scuffing in turn leads either to a shorter life between overhauls or seizure of the rings, piston and liner. Finally Haggh and Holmer (68) stated that it was actually a carbon polishing problem, i.e. the piston collected carbon from the top of the liner and polished the liner and the resulting mirror surface was unable to retain the lubricating oil.

A smooth bore could be produced by a number of different processes and three of these are shown schematically with the original view of the surface in Figure 168A. Figure 168B shows the effect of closing over the graphite structure which would therefore not be observed in the optical microscope. Figure 168C shows the effect of building up a surface deposit layer which might be described as a glaze or 'polymeric film' and although both would produce a smooth surface, the graphite structure would not be observed. Figure 168D shows the effect of polishing which would remove material but not obscure direct observation of the graphite structure and it is this mechanism which is proposed to produced the bore polished correlation shown in Figure 31 (heavy polishing). This condition would be expected to be produced by an abrasive/polish process in which the surface is gradually exposed to smaller and smaller abrasives similar to laboratory metallographic polishing. Comparing Figure 29 (microstructure) prepared by metallographic polishing with Figure 31 (heavy polishing) the bore polished surface shows that Figure 28 has been polished more efficiently and the graphite structure is more clearly observed, where the structure is only partly observed in Figure 31. It was decided therefore to metallographically polish an identical cast iron sample less efficiently to see if a polished condition even more closely comparable to the above polish could be produced and this is confirmed by Figure 169.

׀.3 TRIBO TEST MACHINE DESIGN

During 1974 problems associated with high speed diesel engines were investigated by the Co-ordinating European Council (CEC) investigation group IGL-S. This investigation took the form of a survey covering production, statistics, design types, and the results of the survey are summarised in Table 2. The CEC group used the Ford Tornado test as a tool both for evaluating oil and additive performance and for investigating the fundamental nature of the problem. Two reference oils had been used, RL47 with a good bore polishing performance and RL48 with a bad bore polishing performance. The tests showed a clear discrimination between the reference oils used and were presented to ELTC for tentative method approval which was subsequently granted. The tribo test machine has been designed to be used as a tool in laboratory conditions to discriminate between RL47 and RL48; to produce bore polishing and a better understanding of its mechanisms.

The project group CL.20 was therefore formed to develop an engine test for the evaluation of bore polishing using reference oils of known bench and field performance for the Ford Tornado.

Examination of worn cylinder liners and piston rings has shown that maximum wear occurs at, or just below, the top dead centre of the stroke (TDC) for the cylinder (Fig.26). This position coincides with the minimum oil film thickness occurring on the expansion stroke. It is impossible to define exactly the condition of lubrication in this area because the proportions of boundary and hydrodynamic lubrication are not fixed. For controlled simulation tests however, the same variation in conditions cannot be tolerated and a single lubrication mechanism should be employed. As hydrodynamic lubrication introduces a gap between the surfaces which would reduce the wear, the test was defined to achieve a boundary lubrication condition.

Examination of the Sommerfeld Number indicated the conditions likely to promote boundary lubrication were as follows:

$$Z = \frac{\mu}{P} N$$

Z = generalised Sommerfield Number

μ = dynamic viscosity

N = load

P = pressure

Small values of Z lead to these conditions which can be achieved with a miniumum speed and maximum load. In boundary lubrication the bulk viscosity does not play any part but for a mixed system the hydrodynamic component would be minimised by a low oil viscosity. This may be achieved by careful temperature control. Too high a temperature may lead to a breakdown of the boundary lubrication film. In view of the speed requirements, a reciprocating test offers the best conditions as at the end of the stroke the speed reaches a zero value which is where boundary lubrication is most likely to occur (Fig.19). The test involves two specimens representing the piston ring and the cylinder liner, materials which are reciprocated relative to each other under a normal load, with lubricated conditions and at elevated temperatures. The design requirements were:

a) Reciprocating motion
b) Accessable specimens
c) Provision of an oil bath
d) Control of (i) speed, (ii) load, (iii) temperature
e) Measurement of (i) wear, (ii) friction (iii) temperature, (iv) contact resistance.
f) Continuous operation
g) Ability to mix particles in the oil bath to produce a surface similar to the surface produced in service.

In an attempt to separate the wear of the pin and flat materials, a conical pin with a taper of 30^o was evaluated. Measurement of the pin cross section was made at intervals throughout the test but these were of little value. Increases in the wear of the pin cross section meant that pressures at the surface reduced throughout testing. New material was constantly being run-in at the edges of the track and the test had to be interrupted in order for the measurement of the pin cross section to be made. A 2 x 4mm cross section pin did not allow separation of wear but it was simple to manufacture and produce the required pressures with only moderate suspended loads. As the metallurgy of the pin and flat material were fundamentally similar, natural characteristics could be isolated to indicate the individual wear rates from the debris analysis. Expensive techniques such as thin layer activation to separate wear could not be justified. The overall objective of the research to simulate the bore polishing was not jeopardised by the decision to adopt the measurement of combined wear. Use of the same pin material throughout the tests meant that this wear could be used as a valid basis for comparison of flat material wear.

The oil bath system was designed so that the aluminium bath (Fig.17) could be filled with 50 ml. of lubricant of the type recommended for the investigation of bore polishing when used in diesel engines and this was also used in the trib test machine. The lubricant was heated by three cartridge heaters in the base of the bath controlled from a thermocouple underneath the oil bath. The thermocouple reads the temperature of the oil bath and records this on the chart recorder (Fig.36). Lubrication conditions in the tribo test machine were flooded over the whole track length which did not result in the partial or total oil starvation sometimes found at T.D.C. in diesel engines, which may cause collapse of lubrication (39) and lead to intensification of wear. Temperatures in the tribo test machine were modest compared to average surface temperatures near T.D.C. which may be as high as 240^{o}C and have significant effects on oil properties. The pin and flat (Fig.16) were fixed in the machine which continuously reciprocated the same areas of the specimens against each other, producing smooth conforming surfaces which were able to support effective lubrication. Surfaces in an engine cylinder liner become smooth (Fig.31) which may be attributed to a combination of the nature of the ring and cylinder liner contact, three body abrasive effects, and the lubricant effect. Changes in areas of contact between rings and liner can be caused by several factors:

1. The ring tilting at the end of each stroke.
2. Position of rings in the piston grooves.
3. Thermal expansion and contraction of cylinder components.
4. Tilting of the piston.

Using an arm fixed at one end and loaded at the other, with the pin specimen fixed half way along the lever (Fig.15). The liner bearing frictionless table (Fig.18) was fixed underneath the oil bath for free movement and gave an extremely low coefficient of friction which allowed a more accurate reading for the friction between the pin against the flat.

In order to obtain an indication of the oil film thickness between the pin and the flat, it was decided that a 16mv potential should be applied between the two components and the contact resistance between the surfaces were monitored and 0 was taken to indicate metal contact, whereas 100Ω indicated a thick oil film. Measurement of electrical contact resistance between the piston ring and cylinder liner cannot be used for assessing the oil film thickness because sporadic metallic contact between the ring and liner prevents accurate measurement of instantaneous resistance. However, this breakdown indicates the degree of contact between the ring and liner, hence resistance measurements are useful for identifying different lubrication regimes.

The DUA-L-VEE System (three components) as in Figure 21 is used as the holder of the pin and it provides a very rigid support for the pin when it slides over the flat. This device provided excellent linear motion throughout the experiments. Friction force was measured in both directions by a linear voltage displacement transducer. The bath plate was fixed on a frictionless table and the friction transducer was positioned to prevent interference by the wear transducer also of the same type.

The repeatability of the test was found to be about 10%. The material used for this work, for both the pin and the flat, was grey cast iron and the A.S.T.M. classification for both the pin and the flat is as follows:

Pin type A uniform distribution size 2-7
Flat type A uniform distribution, size 2-7, and the microstructure is shown in Figures 24,25 and composition in Tables 3,4.

Grey cast iron was used because of its specific properties and microstructure and it is the material most commonly used for these industrial components.

Friction and wear outputs were found to be extremely sensitive to bath temperature variation of $\mp 3^{o}C$. This was thought to be due to changes in oil viscosity over this range.

To improve this it was necessary to modify the temperature control system. When these modifications had been made this allowed a more accurate measurement of friction, wear, temperature and contact resistance throughout the tests. The advantage of this method, where stopping and starting occurs at the beginning and end of each stroke, is that it reproduces the conditions of motion in service. The main advantage of this type of testing used to study piston ring and cylinder liner tribological behaviour are summarised below.

1. The ability to measure friction, wear, contact resistance, and the temperature which closely relate to those encountered in service.
2. Size of the sample is easy to examine.
3. Simple test component, allowing manufacture in a wide range of materials and control of surface roughness.
4. Inexpensive test in comparison to other test bed field engines.
5. Producing a surface which is similar to the surface produced in the engine.
6. To be able to collect the wear debris and examine it to identify the cause of failure.

5.4 THE RESULTS FOR THE TWO OILS

Weight loss was an inadequate criterion to evaluate wear, therefore wear of the specimens were evaluated as a volume loss computed from the chart recorder. This is the total wear of the pin and the flat, the variation of volume loss against sliding distance as a function of loads for the two oil bath temperatures of $80^{o}C$ and $150^{o}C$ for RL47 lubricant as shown in Figures 37,38; there are two distinct regions:

1. **Running-in**

   A high initial wear rate during the running-in period is due to the wear of the pin and the flat, especially the pin (Fig.82 and Table 6). This gradually reduces as the interaction between the asperities becomes less frequent and less severe. During the running-in process surfaces are conditioned from the original ground finish to a smoother surface which is more amenable to lubrication (Figs.82,84)

2. Equilibrium Wear

Equilibrium wear is established when the energy input due to the sliding conditions is balanced by the processes of the surface topography changes and the film formation to yield an essentially constant rate of material removal from the surface and is dependent on the applied load; the equilibrium period having a lower wear rate than the running-in period. The analysis for this period indicated that the wear rate increased with both load and temperature (Fig.39).

Hirst and Lancaster (87) have thoroughly examined wear curves and the difference between the running-in and equilibrium wear, and they classified three types of wear curves. Type I is characterised by a high wear rate during the running-in period and they suggested this high wear rate is due to the predominant wear of the pin transforming to a lower wear rate with time (equilibrium wear). Type II shows no variation throughout the running-in and equilibrium wear. Type III is the converse of Type I. The examination of the wear curves produced by using the tribo test machine have been shown to be of a similar Type I of Hirst and Lancaster (87).

Sarkar (58) showed the pattern of curves which he produced in the laboratory (Fig.9) to be the same as those in actual machine parts and hence there is due emphasis on the importance of running-in for sliding components. The Sarkar curves look similar to the ones produced by the tribo test machine as seen in Figures 37 and 38. In the case of RL48, the wear curves (Figs.40,41) have the same form as those for RL47 - two periods of wear (Figs.40,41), running-in wear period and equilibrium wear period.

The wear plotted throughout the test followed a similar pattern for both lubricants, RL47 and RL48 (Figs.37,38,40,41). The initial high wear gradually reduced over a number of hours to a lower equilibrium wear. This running-in phenomena has been studied by Sreenath and Rama (48) who identified two stages in the process. The first was thought to be due to rapid removal of the surface peaks by deformation and severe mechanical wear. During the second stage the remaining valleys were filled in by debris or films generated on the surface.

Tests were run for up to 20 hours in order to identify any further changes in wear behaviour. Deviation from a running-in and equilibrium wear pattern only occurred during the early stages of equilibrium wear for high loads. After a period of normal running-in and equilibrium wear the rate was found to increase which culminated in the fracture of the pin. The fractures appeared to have initiated at the edges of the pin. Although the resulting fracture of the material from the pin should not have changed the real area of contact between the specimens, deformation of the surfaces and dissipitation of the lubrication between the surfaces rapidly increased the wear rate. Apart from this, the equilibrium wear was representative of the long term wear behaviour for the test conditions (Fig.98) of a 100 hour test. The time taken for the specimens to complete the running-in process varied from test to test despite close control of the specimen preparation. These variations were thought to be caused by minute differences in topography and metallurgy. As the reproducability of the equilibrium wear rate was acceptable, variation in the running-in behaviour did not appear to significantly influence the equilibrium wear conditions.

Initially the wear rate was high because large asperities were removed from both surfaces but this gradually reduced as the interaction between the asperities became less frequent and less severe. When running-in changed to equilibrium wear there was no further improvement in surface roughness. Wear continued due to slight asperity interaction and the debris carried in the lubricant which was trapped between the counterfaces. These mechanisms are thought to have combined to prevent any further smoothing of the surface.

As with the ring wear in the engine, the pin wear rate was greater than the flat. This was clear from the original grinding evidence remaining on the surface of the specimen (Fig.83) and Table 10 shows the removal of material from both the pin and the flat. This wear may be attributed mainly to the difference in the sliding distance of the components, although the effect of traversing the conditions at the end of the stroke on each pass should not be ignored. The wear of the pin was fifteen times higher than the wear of the flat (Fig.65) due to the cross section of the pin being fifteen times the length of the flat. In addition to changes in wear which occurred during the running-in and equilibrium wear, friction between the pin and the flat also varied (Figs.46,47,49,50). The high initial frictional force was a result of deformation and mechanical interlocking of the surface grooves. As the surfaces became smoother, the proportion of metallic friction reduced and lubrication became more stable. The friction recorded by an oscilloscope for one pass during equilibrium wear shows a decrease along the middle of the stroke (Fig.44). In identical speed conditions for an unlubricated test, where metal to metal contact occurs over the whole stroke, showed no such variation in friction (Fig.45). This was borne out by Bowden and Tabor (18) who showed that the coefficient of friction was independent of speed for true boundary lubrication. Values for the coefficient of friction in boundary lubrication are usually quoted in the range of 0.05 to 0.2 whilst hydrodynamic friction of 0.001 to 0.03 , is much lower. The range of coefficient of friction for the lubricated tests is 0.08 to 0.16 (Fig.48). Figure 170 shows the friction force against the load and a linear relationship between the friction force and the load for both lubricants (RL47 and RL48) is produced and these graphs satisfy the formula

$$\mu = \frac{F}{L}$$

where μ is the coefficient friction, F is the friction force and L is the load.

Both the wear and friction performance of the formulated oils would be expected to be superior to base oils because of its high viscosity and the influence of its boundary additives. The effect of these factors is evident in the microprobe analysis results (Figs.71,72,73).

Wear rates for both lubricated tests on the reciprocating tribo test machine (Figs.37,38,40,41) showed that conditions which led to an increase in wear rate were lower oil viscosity and low sliding speed or higher loads. Small increases in the oil temperature also caused a higher wear rate because as the temperature of the oil increases, its kinematic viscosity decreases.

In mixed lubrication it is thought to be the summation of all hydrodynamic effects over the surface which controls the proportion of solid contact and indirectly the amount of wear. An increase in hydrodynamic lubrication results in less solid contact as more of the load is supported by hydrodynamic pressure within the oil film.

It is possible to show theoretically that the hydrodynamic pressure between two relatively moving surfaces is a function of viscosity, speed and variation of the film thickness in the direction of the sliding (26).

$$\text{Hydrodynamic pressure} = 6uU\frac{dh}{dx} \qquad (5:1)$$

u = absolute oil viscosity

U = relative velocity

h = distance between surfaces

x = direction of sliding

However, this does not indicate a relationship between pressure and film thickness. To measure the effectiveness of the lubrication it is necessary to refer to the generalised form of the Sommerfeld Number which is defined as:

$$So = \frac{uU}{PL}$$

U = relative velocity

u = absolute oil viscosity

P = pressure

L = Length of the bearing

After assuming Newtonian behaviour, the Sommerfield Number becomes:-

$$So = \frac{\text{Viscosity x speed}}{\text{load}} \qquad (5:2)$$

As the tribo test machine results and generalised Sommerfeld Number indicate, increases in viscosity or speed improve lubrication. Equation (5:1) shows that these effects increase the hydrodynamic pressure which is likely to decrease the proportion of solid contact and lead to a lower wear rate. Increasing the load reduces the lubrication effectiveness. Providing the surfaces remain rigid, load does not alter the generation of hydrodynamic pressure (5:1). Increased solid contact to accommodate the load change can then lead to a higher wear rate. Wear curves for all the tests using both lubricants are similar in shape and display an exponential increase in wear rate with load (Figs.39,4 2). Reduction in the Sommerfeld number brought about by an increase in load, indicates more support through solid contact and therefore more severe boundary lubrication. The significance of these friction and wear results was that they proved the reciprocating tribo test machine and test procedure was sensitive enough to respond to small changes in conditions. As a whole, therefore, the test was shown as a good, but not perfect, basis for the second stage of the work which was to simulate the bore polishing surface in laboratory conditions.

The pin surface (Fig.83) after a one hour test featured predominant abrasive wear with no sign of the original machining marks, with fine parallel abrasive grooves running the length of the pin. Figure 85 shows the minimal wear of the flat with most of the original machining marks still visible (after one hour) with abrasive grooves running along the wear track with plastic deformation appearing on the surface. Debris examination (Fig.101) shows plate-like particles after a one hour test and suggested that the wear mechanism involved was adhesive wear due to the colliding of both surfaces' asperities when the test started and during the one hour test. The surface of the specimens just after the running-in process featured two wear mechanisms for the pin with a total absence of the original machining marks (Fig.90); the two wear mechanisms were abrasive grooves and delamination wear with a smooth surface (Fig.82). Figure 91 shows the flat specimen just after the running-in process with a smooth surface (Fig.84) and two wear mechanism involved; abrasive wear and plastic deformation (Fig.91). Examination of the debris for the running-in process (Fig.102) revealed plate-like particles suggesting that adhesive wear was involved during the running-in process. The wear mechanisms involved during the equilibrium wear period are abrasion and delamination wear (Figs.94,95), with debris analysis identifying two kinds of debris which were large plate-like particles (Fig.103) which suggests that delamination wear was involved. Small particles are associated with running-in.

Hogmark and Alexander (51) stated that the two most important mechanisms responsible for running-in are abrasion and deformation. Eyre (23) regarded adhesive wear as the predominant wear mechanism during the running-in process. Nadel and Eyre (44) stated that abrasion appears to be responsible for the normal mechanical wear that occurs in the majority of liners and referred to the fact that corrosion may also contribute indirectly to increased abrasion. They also identified the delamination wear when they examined actual cylinders in service. Eyre (1) in his review about the wear mechanism referred to 50% of wear in industrial equipment being due to abrasion.

Figure 78 shows the change in wear on the wear track with a smooth surface at both ends of the wear track (Fig.79) and some of the original machining marks visible in the middle of the track and with deposits of the oil concentrated in the middle (Fig.77).

Sudarsham (59) and others reported that wear at the top dead centre was due to the wear mechanisms of abrasion, adhesion and corrosion, while Nadel and Eyre (44) identified positions on the liner where the abrasive wear mechanism was predominant at TDC and BDC and more severe at the top due to the high pressure at the TDC.

Fully formulated oils contain additives, the molecules of which are intended to adhere to the surface. The action of these molecular layers is to lubricate and protect surfaces which come into contact under marginal lubrication conditions. It is thought therefore that the difference between the two lubricants (RL47 and RL48) was due to these additives under extreme boundary conditions and evidence of this was the microprobe analysis, wear and friction results for both lubricants (RL47 and RL48).

The difference between the two oils (RL47 and RL48) in terms of wear are featured in Tables 6 and 8. The removal of material was higher with RL48 compared to RL47 (Figs.37,38,40,41). This may be due to the following:

1. Higher S with RL48 compared to RL47 as shown from the microprobe analysis (Figs.71,72,73) and S in the form of oxide may initiate the formation of sulphuric acid when water vapour is present resulting in corrosive wear. Cloud and Blackwood (85) during their examination also observed that the wear of cylinder liners increased when S was increased in the fuel.
2. Higher Ca with RL48 compared to RL47 as shown from the microprobe analysis (Figs.71,72,73) and Ca transforms to the relatively soft sulphate resulting in a smooth surface and increased wear. This was also predicted by Ayel (75) who also found the Ca higher in RL48 compared to RL47.
3. The stability of the film for the oils due to the extreme boundary conditions, when there is a difference between RL48 and RL47.

A review of the literature on this subject confirms that sulphur seems to be associated with cavities on heavily polished liner surfaces and not on non-polished surfaces (67). Ayel (75), Coy and Quinn (88) also found sulphur to be associated with smooth load bearing areas of specimens taken from a four ball machine and concluded that the E.p. activity is attributable to small amounts of iron sulphide. The result of the analysis in this section may indicate that polishing is associated more with an E.p. action than with an antiwear action. In particular it is possible that the formation of an iron sulphide film and its continual removal as described by Mills and Cameron (89) is partly responsible for the smoothing of the surface. In this respect a chemical corrosive wear process could be partly responsible for the smoothing of the surfaces with abrasive particles also being formed. This idea is further supported by Forbes (90) who also identified sulphur with the load carrying area.

Whilst RL48 shows a distinct sulphur peak this was considered to indicate that the formation of an iron sulphide film and its removal by the piston ring could constitute a corrosive wear process. This could lead to a gradual smoothing of the surface by chemical corrosion as mentioned by Barcroft (91) and also initiate abrasive particles which could in turn cause a smoothing of the surface.

Thus the correlation found between sulphur and heavy polished liners is confirmed in practice (67) and also in the engine test liners. There is no difference between the two oils in terms of wear mechanism. Abrasive wear and delamination wear appears on both oil surfaces with very rough abrasion grooves running along the wear track for both oils (Figs.56,57) while the delamination wear with plate-like debris appeared with both oils (Figs.58,59) and this pointed to the fact that both fully formulated oils were similar in behaviour under the same conditions of extreme boundary lubrication and featured the same wear mechanisms. Examination of the debris showed the same kind of plate-like debris (Figs.99,100) which proved that delamination wear is the mechanism involved in both oils.

The Talysurf profile and C.L.A. value measurements revealed smoother surfaces with RL48 (Figs.65,66) compared to RL47, due to the higher wear with RL48 compared to RL47 (Tables 6,8 and Figs.37,38,40,41). C.L.A. value figures (Figs.66-69) show a dependence on the load and temperature conditions during the test.

While the contact resistance results revealed that there is little difference between the two oils, but there is a difference in the contact resistance in terms of high and low loads (Figs.52,53). This clearly proves that under the higher load there is higher contact resistance due to the increase in the solid contact between the two surfaces and with the low load there is very little contact resistance due to the effect of the oil film separating the two surfaces and decreasing the solid contact between the two surfaces.

Both oils (RL47 and RL48) did not produce bore polished surfaces with smooth surfaces of less than 0.125 μm in C.L.A. values, and the appearance of the graphite structure on the surface of the flat, as seen in the Ford Tornado liner (Figs.29,30).

5.5 BORE POLISHING PRODUCED BY DIAMOND PASTE

As the diamond paste tests were designed to simulate only the appearance, characteristics and mechanism of the bore polishing and not the engine environment, the temperature and test conditions are not entirely representative of those encountered inside diesel engines. Cylinder temperatures are usually in the order of 250^{o}C for highly rated engines although surface temperatures at the top of the stroke may be appreciably higher. Good correlation was found between the tribo test machine and engine surfaces which indicated that the differences in temperature and diamond paste conditions had been overcome for the test result.

A phenomenon resembling bore polishing in diesel engines was obtained using the reciprocrating tribo test apparatus. This resemblance had the following significant features:

1. **Visual appearance of liner surface**

In practice heavy polishing is associated with large amount of exposed graphite structure of the cast iron (Fig.31). This was also noticed with the polished diamond paste specimen (Fig.110). In addition the tribo test demonstrated its ability to potentially screen the surface, similar to the surface in the engine (Figs.31,110)

2. **Surface Profile**

The Talysurf profile for polished flat and heavy polished liners had two important similarities:

a) Corresponding similar C.L.A. values (Figs.27,108) with both less than 0.125 μm (64).

b) Similar appearance of surface profile and produced the nature of the polished surface (Figs.27,108) which is very smooth.

3. **Subsurface**

The taper section for polished flat and heavy polished liner produced similar observations (Figs.34,121); with heavy polishing, total absence of the original honing pattern and a smoother surface, while the polished flat showed some of the original machining marks and some abrasion marks but in both there is no evidence of deformation of the surface or a glaze surface deposit, and no evidence of a white layer which would have indicated scuffing. There is also the debris analysis which showed plate-like particles (Fig.131) which suggests adhesive wear is involved, so a similar subsurface was seen in both the flat and the liner surface.

4. **The removal of surface material**

Both processes which occurred in the tribo test and in the engine liner showed that removal of material occurred (Fig.35, Table 11) which proved, in both processes, removal of material rather than filling in of grooves on the surface and also proved that the wear occurred due to the removal of material. This agrees with McGeehan's statement (67) that polishing is a special form of wear.

5. **Wear Mechanisms**

The wear mechanisms involved for polished flats and heavy polished liners had these similarities:

a) Both had abrasive grooves running in the direction of the moving surfaces (Figs.31,110) and the literature referred to abrasive wear (67) as one wear mechanism involved in the mechanisms of bore polishing.

b) On the flat (Fig.111) bore polishing was concentrated at both ends of the wear track and with the heavy polished liner (Fig.26) it was concentrated at the TDC. In both cases complete boundary lubrication, achieved due to the velocity, was approaching zero (Fig.19). Examination of the Sommerfield numbers likely to promote boundary lubrication were as follows:

$$Z = \frac{\mu N}{P}$$

Small values of Z lead to these conditions which can be achieved with a minimum speed. Examination of the debris showed (Fig.131) plate-like particles and the appearance of plastic deformation on the flat surface (Figs.125,126) which suggests that adhesive wear was also involved in the mechanism of bore polishing. So important emphasis must be put on the interaction between the piston ring and cylinder liner in connection with bore polishing mechanisms.

c) Corrosive wear was observed (Fig.114) indicating that it was selectively attacking the different parts of the cast iron (this agrees with Eyre's (92) examination of liners from service). Therefore there is a connection between the bore polishing mechanism and corrosive wear. This agrees with McGeehan's findings (67), namely that bore polishing can be caused by corrosion due to the high fuel used in the oil while Ayel (75) in his work stated there is no corrosion involved with bore polishing mechanism. Plastic deformation and delamination wear has been observed (Figs.112,125) and this also associated these with the mechanism of bore polishing which was also noted by Hogmark and others (51) for plastic deformation.

6. Microprobe analysis and ESCA

Chemical analysis of the polished flat indicated significant traces of sulphur as well as Ca, and P peaks (Fig.118) which are E.p. additives. Further analysis revealed that these peaks of S, Ca and p are in the form of oxide (Fig.119). This is further proof that corrosive wear is involved caused by the high sulphur in the oil forming sulphuric acid from sulphur trioxide and water on the liner surface.

Ayel and others (75) identified sulphur with the polished liner and also found Ca present on the polished liner and suggested it may be converted to either relatively soft sulphate or very hard carbonate. McGeehan (67) stated that bore polishing increased when the sulphur in the fuel was increased.

Sato (74) and others in their work referred to the bore polishing surface as shiny, as if polished by a lapping compound, which has been proved in this work.

Polishing is a special form of abrasive wear characterised by the use of very small abrasive particles (of magnitude 5μm or less). Samuels (20) stated that polishing differs from abrasion only in degree and can apply to the work of diamond paste; the difference between the two tests, one run with RL48 only and the other with RL48 + diamond paste were firstly, a smoother surface was produced with RL48 + diamond paste compared to RL48 only (Fig.109). Secondly, the removal of material was higher with RL48 + diamond paste compared to the test with RL48 only (Table 13). Thirdly, the exposure of a graphite structure with RL48 + diamond paste (Fig.110) compared to

RL48 test (Fig.91), and that is similar to the surface produced by mechanical polishing for the flat in the laboratory (Fig.25),emphasising that the mechanism involved to produce such a surface is similar to the polished surface from the engine (Fig.31), was the fine polish/abrasive mechanism. When considering the mechanisms that are actually involved in a standard polishing process, one must first consider three points (93). First the polished surface is always composed of fine grooves or scratches (Fig.110), secondly the material is removed at a significant rate continuously during polishing (Table 13). Thirdly a plastically deformed layer is produced (Figs.125,131). These observations emphasise that a fine polish/abrasive mechanism took place during the test and in fact, the polishing differs from abrasion only in degree which agrees with Samuel's theory (20) about the polishing mechanism. While it is observed that abrasive wear in Figure 113 has clearly revealed the dispersed hard phase by differential wear, there is also a chemical attack which indicates corrosive wear (Fig.114) as the wear mechanism involved in the bore polishing mechanisms also observed by McGeehan (67). Therefore the operation involved for the polished flat by diamond past is a fine polish/abrasion and chemical process. This was also proved by microprobe analysis and ESCA examination (Figs.118,119). It would appear that a fine polish/abrasion and chemical process has been at work to produce what appears to be a good metallographic finish. Archard (93) in his review of mechanical polishing of metals referred to polishing like-wear as being clearly not limited to a single physical mechanism. While examination of the short term test and debris analysis (Figs.126,130,131) showed that plastic deformation was also involved in the mechanism of bore polishing which appeared on the surface with the diamond paste (Fig.125) and the debris produced plate-like particles which suggested (Fig.131) the mechanism of adhesion and delamination were also involved due to asperity contact and the collision of the asperities during the diamond paste test.

There is evidence to associate the running-in phenomena with the bore polishing mechanism when the comparison was made between wear and friction (Figs.105,106) during the test run with RL48 only and RL48 + diamond paste. So it was agreed that bore polishing was also associated with the running-in problem.

Using 1/4 μm and μm diamond pastethere is a difference in the kind of surface produced. A rougher surface was produced with 1 μm diamond paste compared to 1/4μm diamond paste (Figs.122,110) thus proving that small particles of 1/4μm polished the surface effectively. It must be pointed out that particles of 1/4 μm are nearly always present in the atmosphere and these particles are not uncommon in the engine (36), with abrasive wear also associated with more than 50% (1). Therefore, abrasive wear is also associated with the mechanism of bore polishing. It must also be pointed out that 1/2μ particles appeared and were generated between the ring and liner during the operating condition of the engine (37) and may be brought into the cylinder with the fuel. There is also a concentration of these particles at TDC which may be due to debris or combustion where the oil film thickness is minimal at this position.

5.6 BORE POLISHING PRODUCED BY CARBON

It is somewhat artificial to produce bore polishing with diamond paste which is not present in a running engine. This kind of test is simulative because of the likely cause of bore polishing being due to hard carbon on the piston crown land acting as a fine abrasive and producing polished areas in the bore (75). Therefore carbon has been used to simulate the condition of bore polishing in the engine instead of carbon black. Rounds (83) stated that diesel soot from used engine oil was found to be far more detrimental than carbon black, having the same size, distribution and surface area per gram. Also CEC (86) in their report mentioned that a grinding paste of hard carbon in the lubricating oil may be responsible for the polishing process. A good correlation has been obtained between the tribo test and the engine and despite the different conditions the results showed that the tribo test produced a surface similar to the engine surface.

The phenomenon produced by the tribo test using the carbon was similar to the one produced in the test engine, with a large amount of exposed graphite structure (Fig.139) which appeared on the flat surface similar to the polished liner (Fig.31). Both surfaces (Figs.27,137) are smooth in terms of C.L.A. values and both are less then 0.125 μm (65). The profile of both surfaces (Figs.27,137) are similar and represent the nature of the polished surface "which is a very smooth surface with less then 0.125 μm in C.L.A. values" (65).

The taper section for the polished flat (Fig.147) revealed a smooth surface with no sign of glaze deposit or any kind of deformation or a white layer which would suggest that scuffing had occurred. This is similar to the section taken from the heavy polished liner (Fig.34b).

Removal of material occurs in both processes (laboratory and service) (Fig.35 for the polished liner and Table 13 for the polished flat), therefore, the process in both is the removal of material rather than filling in of grooves in the surface. This agrees with Parsons (73) when he stated that the amount of liner wear in the polished area is remarkably small. Abrasive wear was observed on the polished flat (Fig.139). This agrees with McGeehan's statement (67) that cylinder bore polishing results from the top land deposit causing fine scale abrasion and degraphitisation of the cast iron. Therefore these results showed a good correlation between the polished liner and the polished flat, produced by adding carbon to RL48 with the following similarities:

1. The exposure of graphite structure (Figs. 31, 142).
2. The Talysurf profile and C.L.A. value less than 0.125 μm (Figs.27,137).
3. The process involved removal of material (Fig.35, Table 14).
4. Taper section (Figs.34,147).
5. The heavy polish was concentrated at both ends of the wear track (Figs.26,137).

Therefore, these emphasise that the polished surface produced by the tribo test machine is similar to that of the polished liner from the Tornado engine.

Jenkinson (94) also mentioned that bore polishing occurs as a result of abrasive wear. It is necessary for rubbing surfaces to be in contact with an abrasive of suitable hardness and particle size such as abrasion contained in the deposit. Ayel et al (75) also observed grooves caused by abrasive wear on the polished liner. There was also differential abrasive wear (Fig.142) when it leaves the hard phase standing proud on the surface, as well as plastic deformation observed on the flat during the test (Figs.148,149). This was also observed on the polished liner by Hogmark and others (51) who did not agree with Ayel and others (75) because they mentioned that there was no involvement between the bore polishing mechanism and plastic deformation.

The difference between the carbon test and the test run with RL48 only were firstly, a smoother surface was produced with the carbon test (Fig.137) compared to RL48 only (Fig.138) and secondly the removal of material was higher with the carbon test (Table 13) compared to RL48 only. Thirdly, the exposure of graphite structure with the carbon test (Fig.139) compared to RL48 only (Fig.140) and the surface of the carbon test was similar to the surface produced by mechanical polishing of the flat in the laboratory (Fig.25) thus emphasising that fine polish/abrasion mechanism had taken place. The mechanism of mechanical polishing (20) showed three factors also observed in the carbon test (Fig.137,Table 13) which also emphasised that the mechanism involved was fine polish/abrasive mechanism.

Corrosive wear was also observed (Fig.143) and evidence of high S (Fig.144) also suggested that corrosive wear was an important factor associated with the bore polishing mechanism. This agrees with McGeehan (67) who revealed that bore polishing increases with increased corrosive wear due to an increase of sulphur in the oil. McGeehan (67) stated that it may be the product of the corrosive which acts as a polishing agent.

This work agrees with McGeehan's statement that "in each case the primary wear mechanism is hard carbon rubbing between the surfaces of the piston ring and the cylinder liner".

It is also important to mention that bore polishing was concentrated at both ends of the wear track (Fig.139) due to boundary lubrication conditions applied, which may suggest that adhesive wear was also involved and evidence of this was examination of the debris (Fig.157). This did not agree with the findings of Ayel (75) and others. Wilson et al (64) said that even with the top crown land free of deposit, bore polishing appeared and they referred to metal-to-metal contact between the liner and piston.

The test conditions were that the pin rubbed the flat with abrasive particles (carbon) between them. Three body abrasive wear was also observed. This happens in the actual engine where abrasive fragments, obviously also contained in the lubricating oil or in the combustion gas, are squeezed between the liner and the ring, during sliding. It is believed that hard particles embedded in the piston soot layer are the most devastating. There were two kinds of debris produced from the test; large plate-like particles (Fig.157b) larger than 50μm which suggested that delamination wear was involved in the process and there was also debris associated with adhesive wear due to their size being less than 10μm (Fig.157a). The mechanism of bore polishing is very complicated and involves mechanical and chemical processes. This agrees with Jenkinson's (94) explanation of the bore polishing mechanism that a complex relationship is involved between the mechanical and chemical conditions existing in the engine.

The use of microprobe analysis revealed higher S and P, and Ca (Figs.144,145) in the form of oxides (Fig.146). The sulphur may combine with water produced in the combustion process to form sulphuric acid resulting in corrosive wear (Fig.143). This is detrimental to the sealing properties of the piston ring and may also produce abrasive particles. Increases in sulphur (Fig.144) and subsequent increases in bore polishing carbon may suggest that bore polishing is related to sulphur and this would agree with McGeehan (67).

When running the test with RL47 + carbon instead of RL48 + carbon, bore polishing did not exist on the flat surface and no sign of a graphite structure appeared on the flat (Fig.154) and C.L.A. values were higher than 0.125 μm (Fig.155). Therefore using RL47 + carbon did not produce a bore polished surface suggesting that the carbon is not the only problem. There is also the lubricant effect which causes bore polishing. From these tests emphasis must be put on the effect of lubricant causing bore polishing. This agrees with Parsons (73) who stated that carbon was not the only reason for causing bore polishing but there is also the effect of the lubricant, additives and fuel. This also agrees with Wilson et al (64) who stated that bore polishing is lubrication related, since engine manufacturers consider the lubricant as a component which could be used to assist in overcoming bore polishing. The literature (55) shows that the reduction of carbon deposit (particularly on the crown land) by changes in oil formulation has been accompanied by an almost complete elimination of bore polishing. Further examination was carried out on the carbon used in the test. The examination revealed (Fig.171) that the carbon also combined Fe, Si, S, p, Ca, Zn, Cu and Cl. Table 15 shows the composition of this carbon and from the results show that carbon is composed of organic and inorganic elements - a mixture of fuel soot and oxidation oil - this carbon was formed due to (1) high sulphur in the fuel and (2) the formation of the soot during the combustion process.

Finally the problem between bore polishing and lubricant must be solved by reducing the sulphur and thus indirectly reducing bore polishing.

CHAPTER SIX

CONCLUSIONS

1. The principal characteristics of bore polishing were established by the examination of Tornado cylinder liners from engine tests. These characteristics were:

   a) The exposure of the graphite structure on the surface of the liner.

   b) Smooth surface with C.L.A. values of less than 0.125 μm.

   c) The process involves removal of material.

2. A laboratory tribo test was developed and used to differentiate between the two reference oils, RL47 and RL48, with higher friction, wear and a smoother surface produced with RL48 compared to RL47. The tribo test has demonstrated its ability as a lubricant screening device. The tribo test also shows its capability for producing a full range of friction and wear behaviour for grey cast iron, normally used as a material for engines, with a full range of variables including temperature, load, and speed.

3. A phenomenon similar to bore polishing was produced by adding carbon from a Tornado test engine with RL48 lubricant and this phenomenon had the following characteristics:
   a) The exposure of the graphite structure on the surface of the flat.
   b) Smooth surface with C.L.A. values of less than 0.125 μm.
   c) The process involved removal of material.

4. There is a very good correlation between the engine test and tribo test results in terms of interaction between the piston ring and cylinder liner for causing bore polishing.

5. The mechanism of bore polishing is very complex and consists of a combination of two processes, one mechanical and the other chemical. Four wear mechanisms are involved; abrasion, delamination, corrosion and adhesion.

6. Running-in and equilibrium wear has been identified as well as their associated wear mechanisms with abrasive wear and plastic deformation responsible for the running-in, and abrasive wear and delamination wear responsible for equilibrium wear. The wear of the pin is fifteen times higher then the wear of the flat.

7. Both lubricants, RL47 and RL48, formed films on ring and liner surfaces with higher S, Ca, and P with RL48 compared to RL47 and all of them in the form of oxides.

CHAPTER SEVEN

SUGGESTIONS FOR FURTHER WORK

The work described in this project has highlighted a number of important and new facts associated with bore polishing. However, there is clearly a need for increased investigation into bore polishing, both in engine tests and laboratory tests. The scope for such investigation is vast since bore polishing is clearly a complex problem being influenced by lubricant, additives, fuel, surface finish, and other design factors.

The following suggestions for further work refer to the laboratory tribo test machine.

1. The effect of carbon on bore polishing could be investigated by adding different types of carbon taken from engines with RL48.
2. With the present tribo test, it would be useful to investigate different lubricants.
3. In order to understand the effect of corrosion on bore polishing it would be important to add sulphuric acid to the lubricants.

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The Particles	The Hardness
Steel	Up to 700 Hv
SiO_2	800 - 1,000 Hv
Fe_3O_4	700 - 900 Hv
SiC	1,500 Hv
Al_2O_3	2,000 Hv

Table 1 Hardness of abrasive particles.

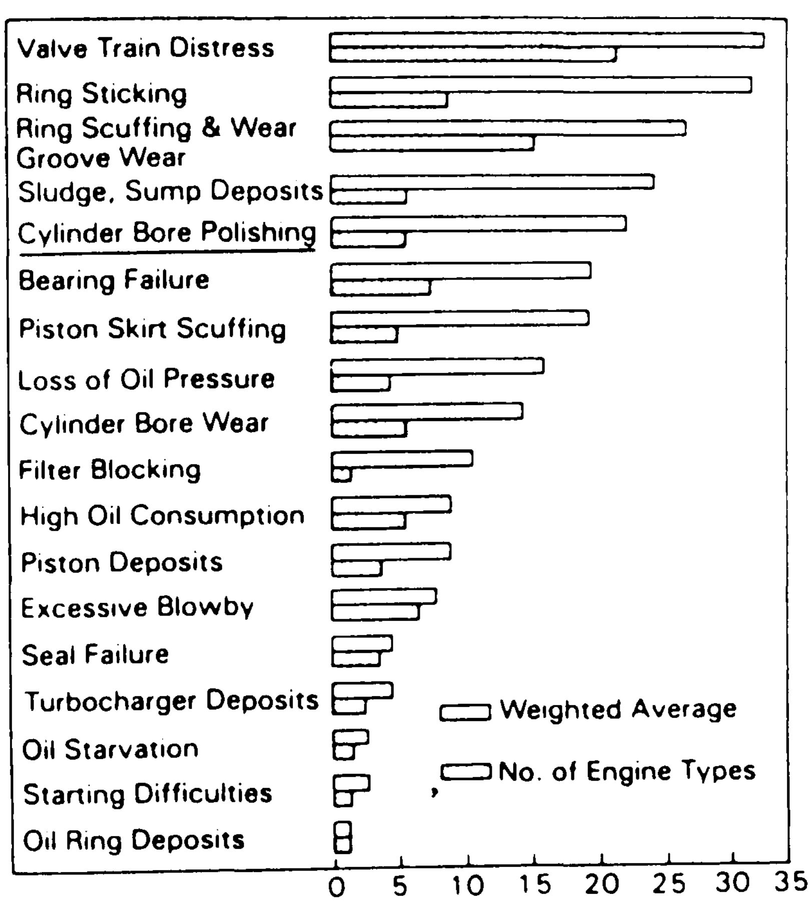


Table 2 Reported lubricant-related problems from CEC Survey.

(Reference 64)

Element	Composition %
C	3.32
Si	1.82
P	0.11
S	<0.1
Mn	0.65

Table 3 Composition of the pin.

Element	Composition %
C	3.20
Si	1.9
Mn	0.5
P	0.15
Cr	0.15
S	0.11

Table 4 Composition of the flat.

Element	Composition %
C	3.35
Si	2.40
Mn	0.65
P	0.18

Table 5 Composition of the Ford Tornado liner.

Test Load (N)	Oil Bath Temperature (°C)			
	80		150	
	Pin (gm)	Flat (gm)	Pin (gm)	Flat (gm)
300	0.0018	0.0001	0.0009	-0.0008
600	0.0012	0.0000	0.0017	0.0010
900	0.0016	0.0008	0.0025	0.0010
1200	0.0018	0.0010	0.0030	0.0022
1500	0.0022	0.0014	0.0035	0.0045

Table 6 Total weight loss of wear pin and flat specimens for RL47.

Test Load (N)	Oil Bath Temperature (°C)	
	80 (mm^3m^{-1})	150 (mm^3m^{-1})
300	7×10^{-7}	1.2×10^{-6}
600	1×10^{-6}	2×10^{-6}
900	7×10^{-6}	1×10^{-5}
1200	1.1×10^{-5}	1.5×10^{-5}
1500	2.9×10^{-5}	2.1×10^{-5}

Table 7 Equilibrium wear rates for RL47.

Test Load (N)	Oil Bath Temperature (°C)			
	80		150	
	Pin (gm)	Flat (gm)	Pin (gm)	Flat (gm)
300	0.0020	0.0008	0.0022	-0.0001
600	0.0021	0.00010	0.0024	0.00012
900	0.0017	0.0020	0.0029	0.00024
1200	0.0038	0.0030	0.0049	0.0015
1500	0.0045	0.0034	0.0058	0.0039

Table 8 Total weight loss of wear pin and flat specimens for RL48.

Test Load (N)	Oil Bath Temperature (°C)	
	80 (mm^3m^{-1})	150 (mm^3m^{-1})
300	7×10^{-7}	1.2×10^{-6}
600	1×10^{-6}	2×10^{-6}
900	7×10^{-6}	1×10^{-5}
1200	1.1×10^{-5}	1.5×10^{-5}
1500	2.9×10^{-5}	2.1×10^{-5}

Table 9 Equilibrium wear rates for RL48.

TIME (Hr)	Pin (gm)	Flat (gm)
1	0.0001	-0.0005
3	0.0010	0.0009
5	0.0011	0.0008
8	0.0014	0.0012
10	0.0019	0.0010
12	0.0011	0.0012
15	0.0024	0.0016
20	0.0041	0.0018
50	0.0061	0.0034
100	0.0197	0.0153

Table 10 Total weight loss of wear pin and flat specimens for different test periods using RL48 at 80^{o}C (load = 900N)

Lubricant	Oil Bath Temperature 80°C, 900N			
	Pin (gm)	Flat (gm)	Pin (gm)	Flat (gm)
RL48 only	0.00016	0.0010	0.00012	0.0009
RL48 + 1/4 μm diamond paste	0.00020	0.00016	0.0024	0.0014
RL48 + 1 μm diamond paste	0.00025	0.0020	0.0030	0.0019

Table 11 Total weight loss of wear pin and flat specimens.

TIME (Hr)	Pin (gm)	Flat (gm)
1	-0.0002	0.0003
3	0.0004	-0.0005
5	0.0010	0.0008
8	0.0024	0.0014

Table 12 Total weight loss of wear pin and flat specimens for different test periods using RL48 + 1/4μm at 80°C (load = 900N)

Lubricant	Oil Bath Temperature 80°C, 1200N			
	Pin (gm)	Flat (gm)	Pin (gm)	Flat (gm)
RL48 only	0.0040	0.0025	0.0035	0.0021
RL48 + Carbon	0.0052	0.0039	0.0048	0.0042
RL47 + Carbon	0.0035	0.00025	0.0030	0.0022

Table 13 Total weight loss of wear pin and flat specimens.

TIME (Hr)	Pin (gm)	Flat (gm)
1	0.0012	-0.0007
3	0.0016	-0.0002
5	0.0020	0.0007
8	0.0022	0.0016
10	0.0028	0.0020
12	0.0035	0.0028
14	0.0049	0.0040

Table 14 Total weight loss of wear pin and flat specimens for different test periods using RL48 + carbon at 80°C, (load = 1200N).

Element	Composition %
Fe	0.498
Si	3.179
S	13.185
P	0.630
Ca	8.642
Zn	0.195
Cu	0.303
Cl	0.455
	Total 27.087

Table 15 Composition of carbon removed from piston top land.

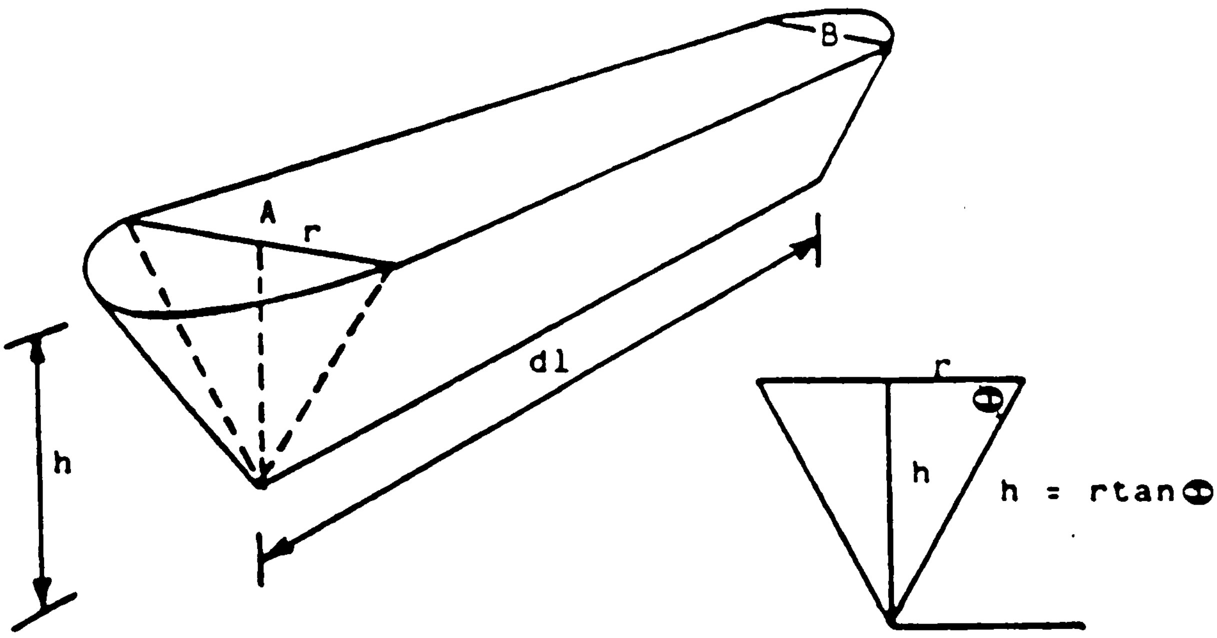


Fig.1 **Abrasive Wear Model.** (Reference 15)

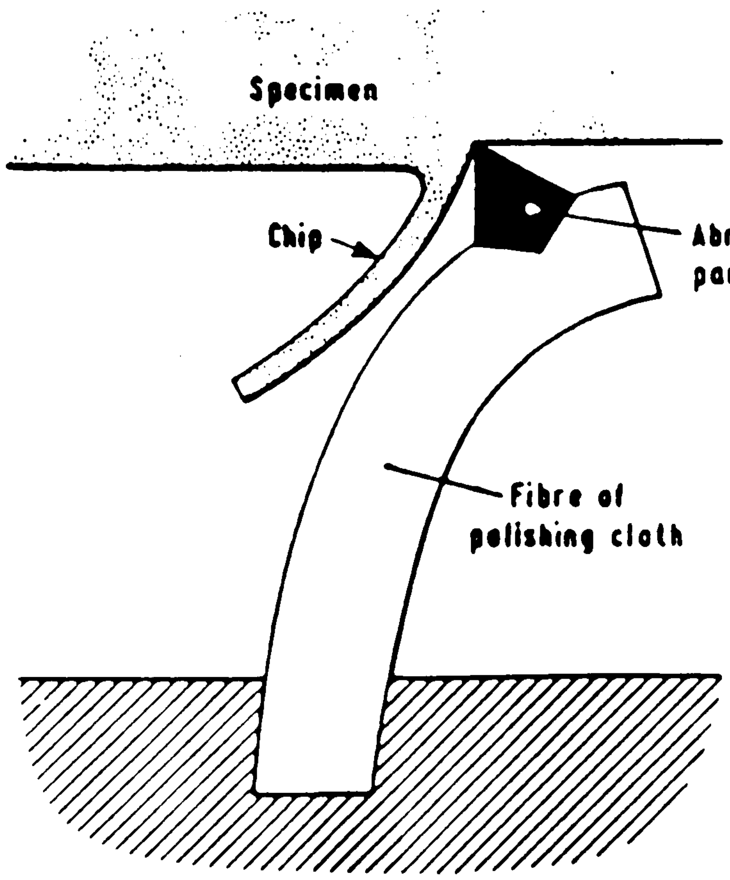


Fig.2 **Diagrammatic illustration of the mechanical mechanism of polishing showing how an abrasive particle is held by the nap of the polishing cloth. The sketch is an abrasive particle which cuts a chip; others would plough a groove.**

(Reference 19)

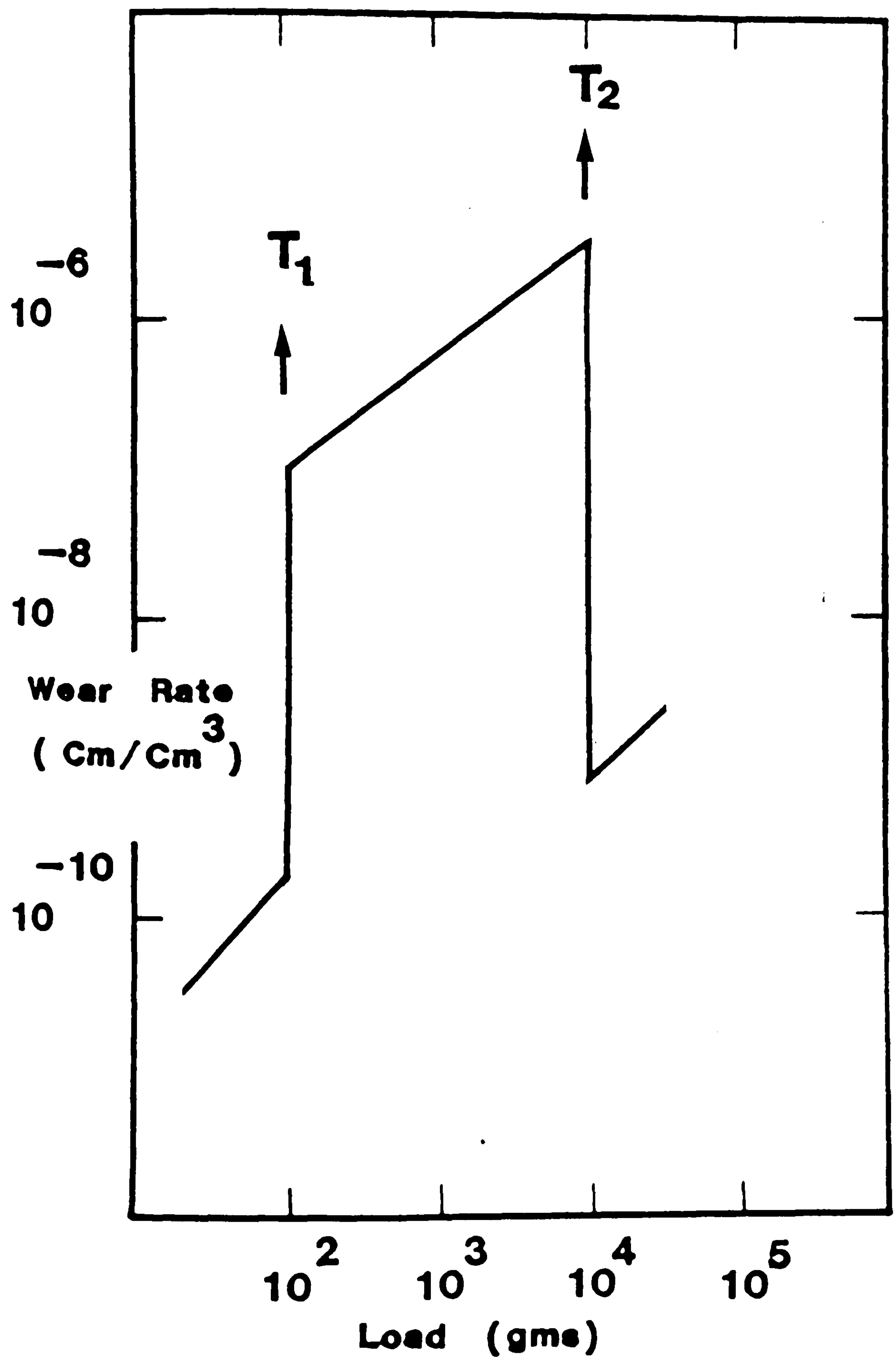


Fig.3 Transitional wear behaviour. (Reference 24)

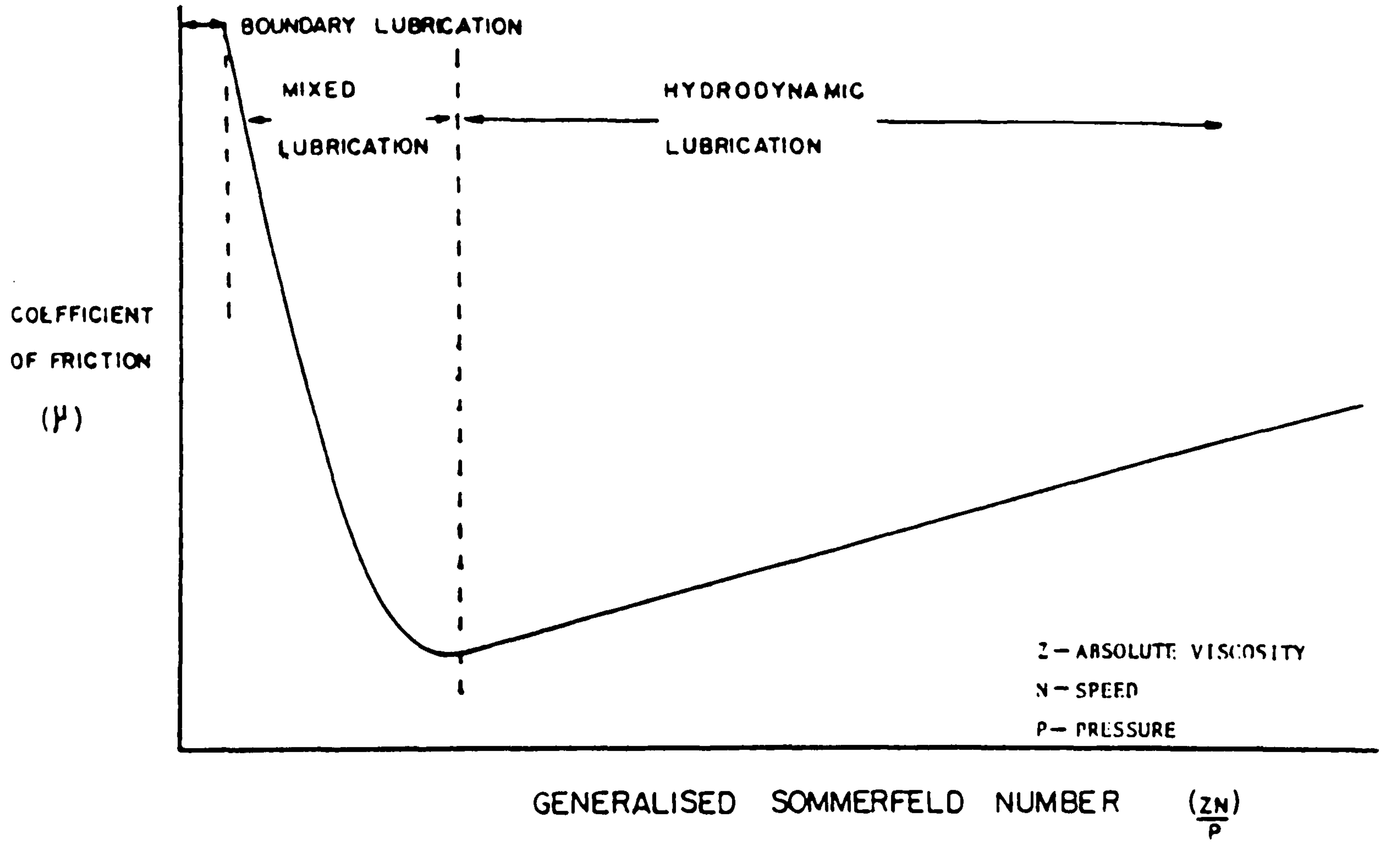


Fig.4 **Stribeck curve.** (Reference 34)

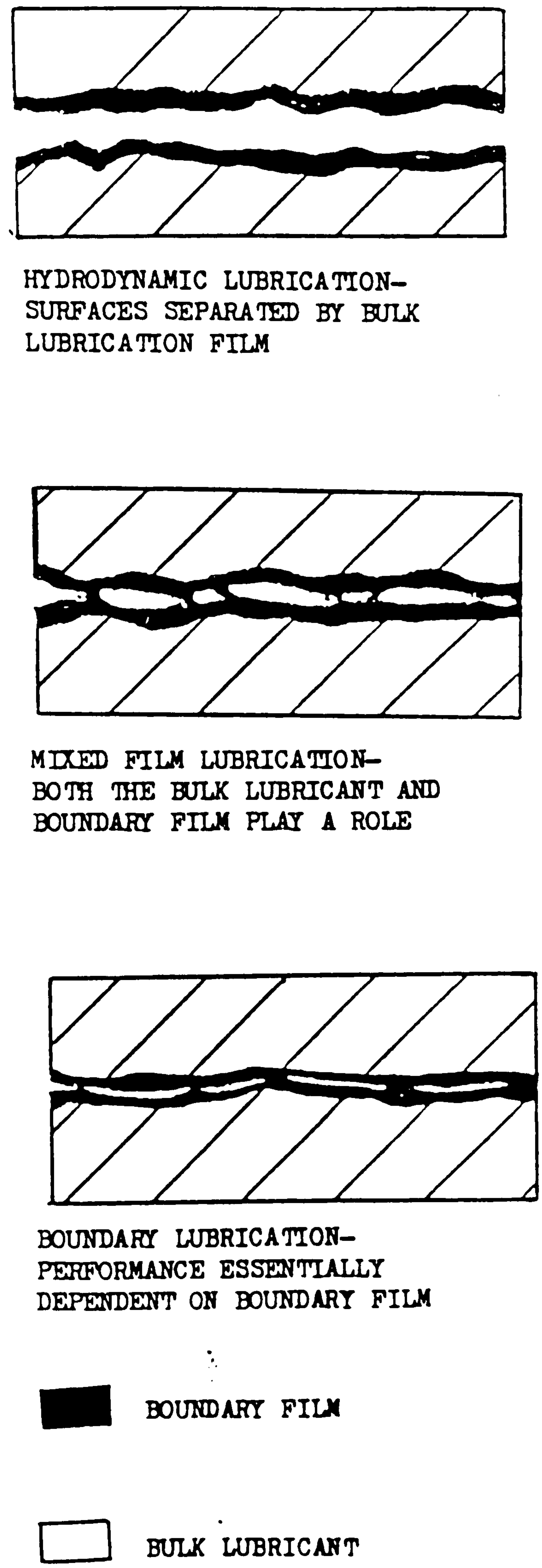


Fig.5 Various regimes of lubrication.

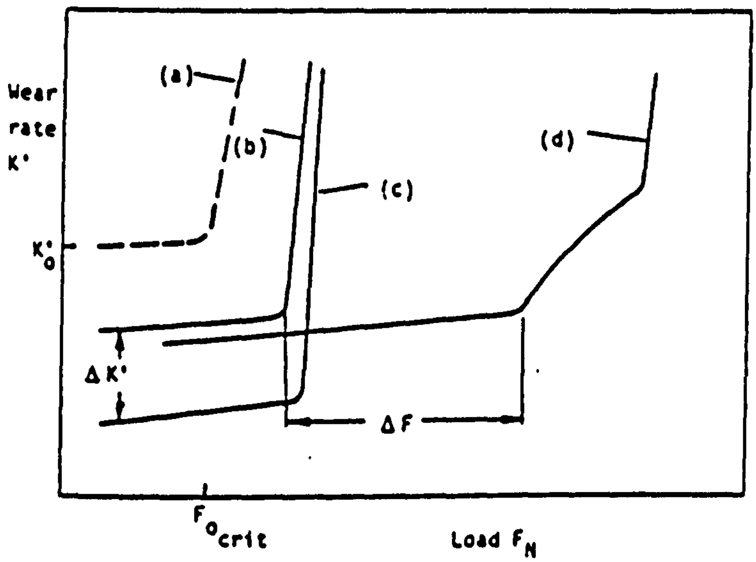


Fig.6 Wear behaviour of boundary lubrication systems (schematic).
a) Dry metal/metal sliding system showing mild/severe transition.
b) System lubricated with base oil.
c) Addition of antiwear additive to the base oil (AW).
d) Addition of additive which increases load carrying capacity (EP).

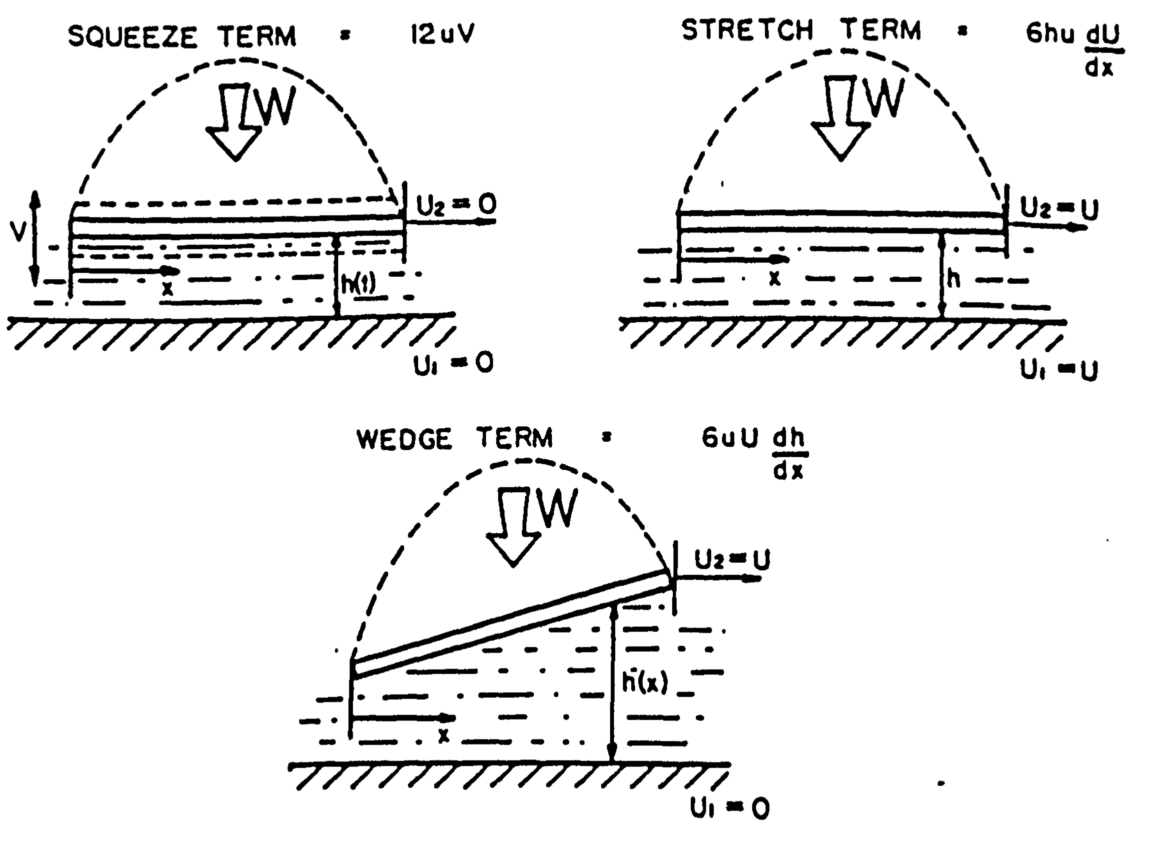


Fig.7 Factors leading to the generation of hydrodynamic pressures in an oil film.
(Reference 26)

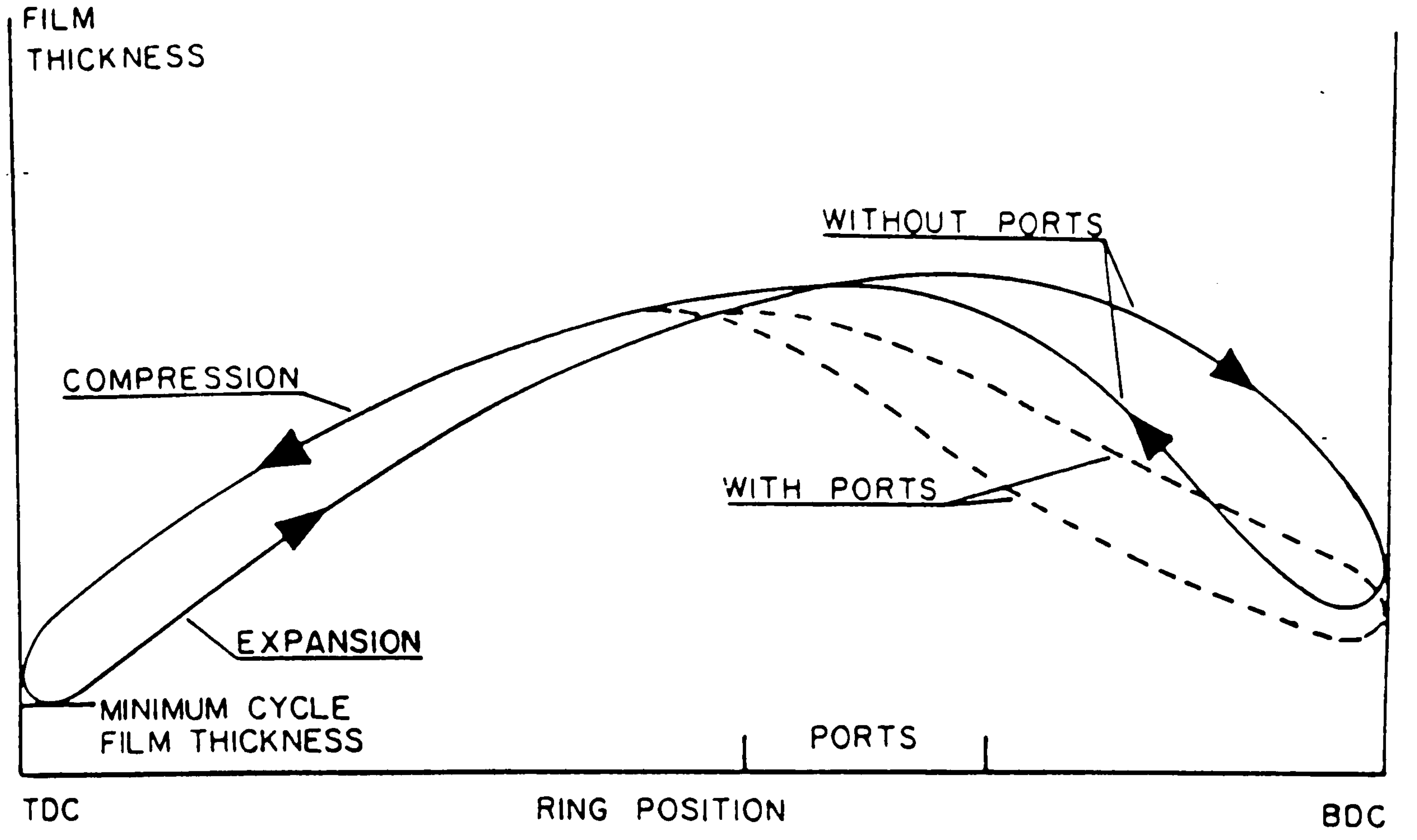


Fig.8 Variation in oil film thickness over the stroke.

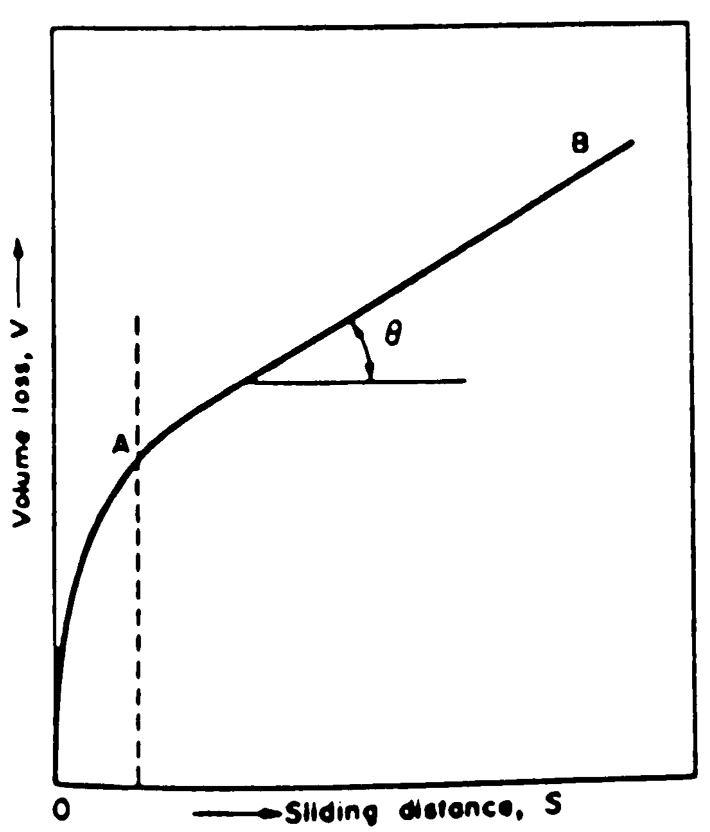


Fig.9 The pattern of a typical wear curve OA is curvilinear and is the running-in wear. AB is linear and is the steady state wear. tan θ measures the rate of wear in loss of volume per unit sliding distance. (Reference 58)

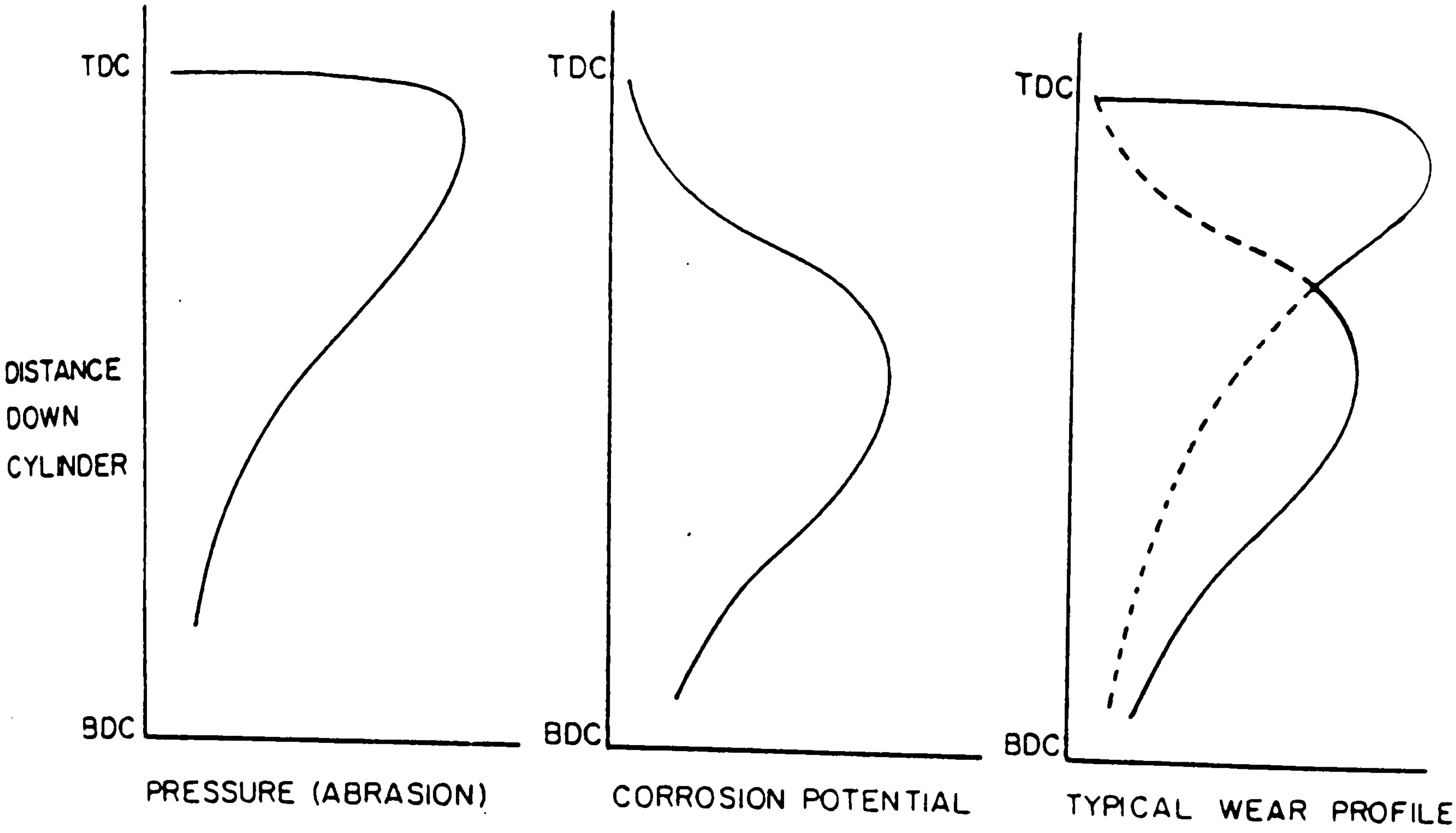


Fig.10 **Liner wear profile down the engine stroke resulting from abrasion and corrosion.** (Reference 61)

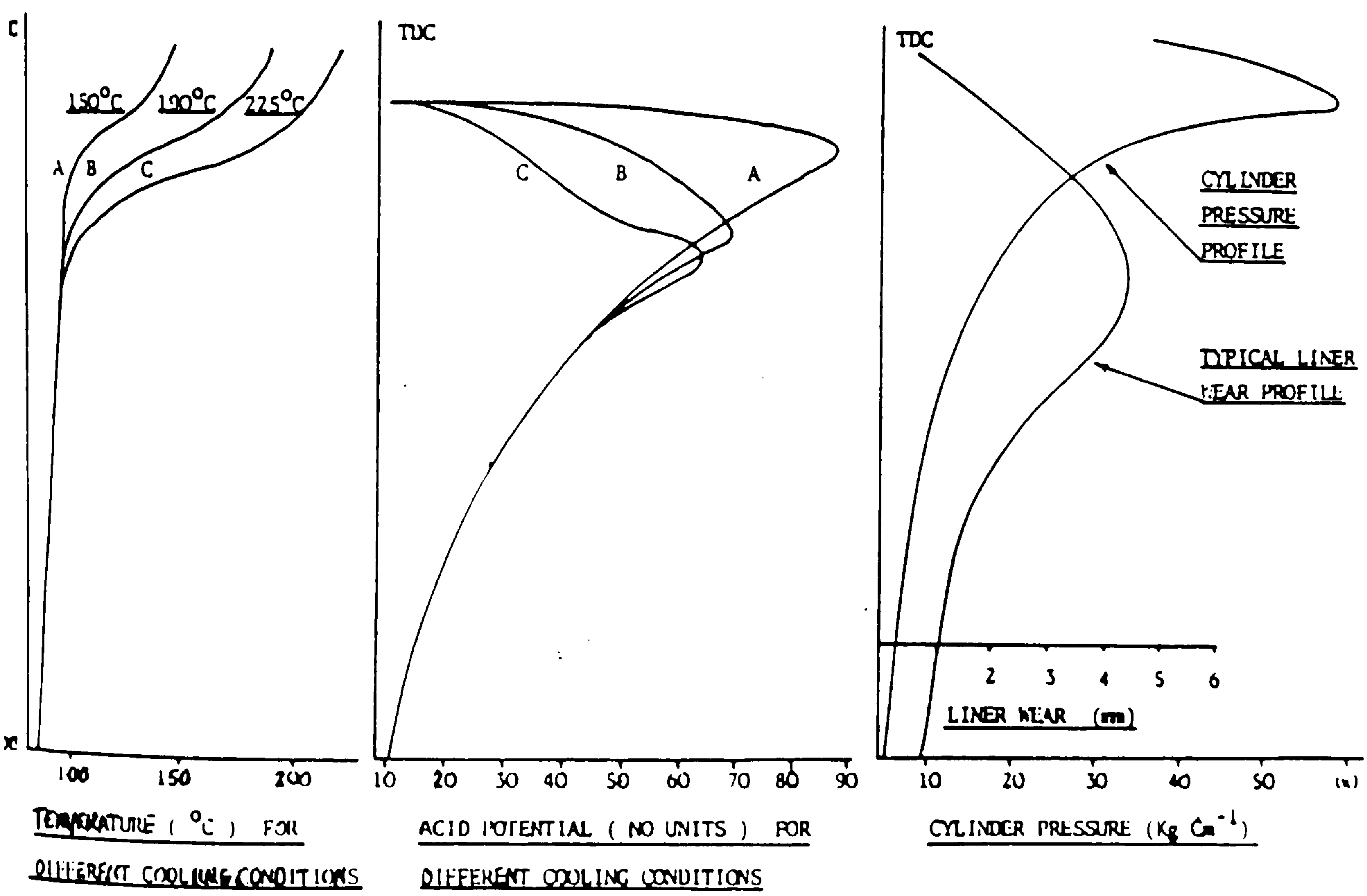


Fig.11 **Temperature, corrosion potential, gas pressure and liner wear profile for an engine running on a 3.0% sulphur residual fuel.**
(Reference 63)

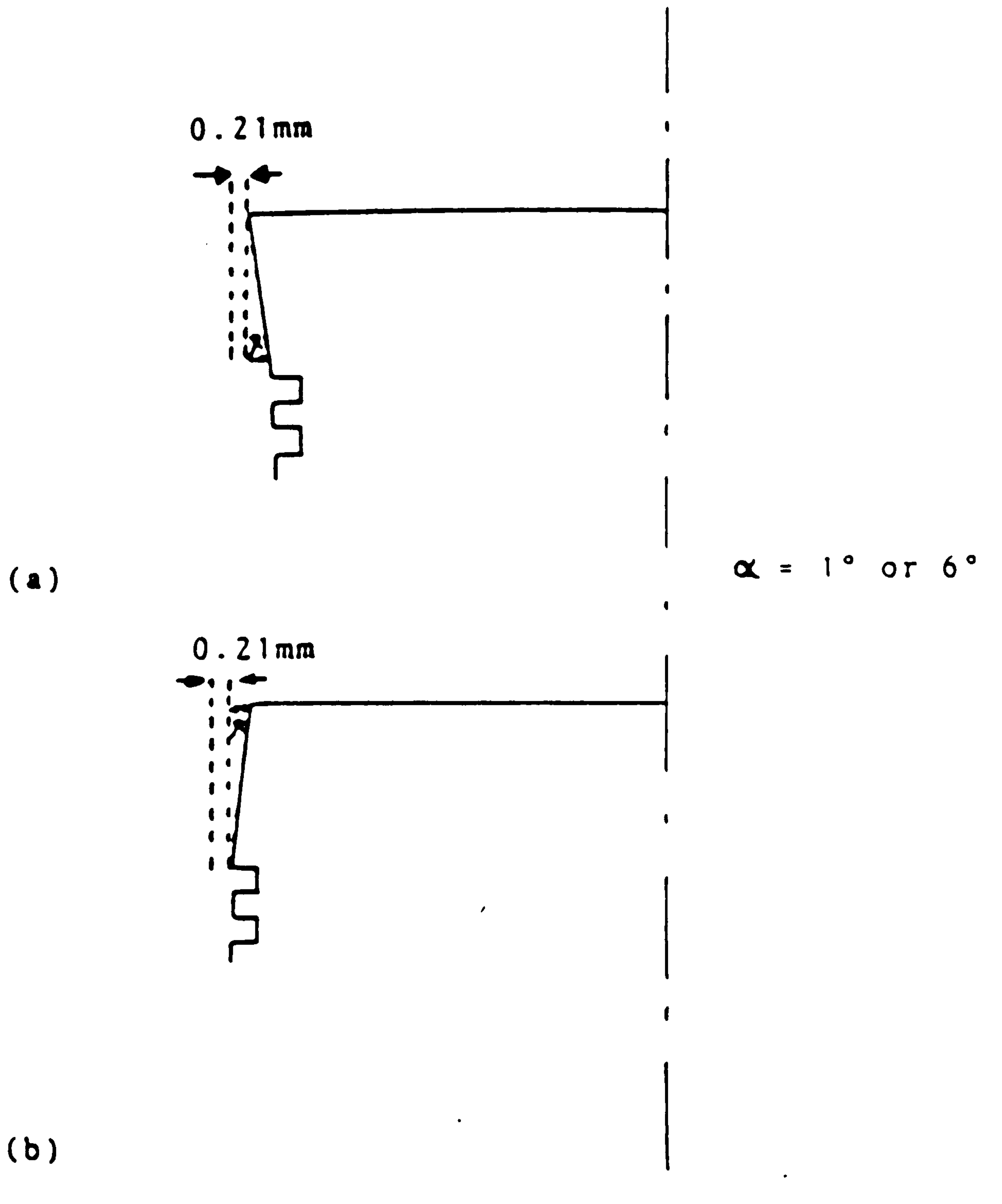


Fig.12 Two different crown land profiles. (Reference 78)

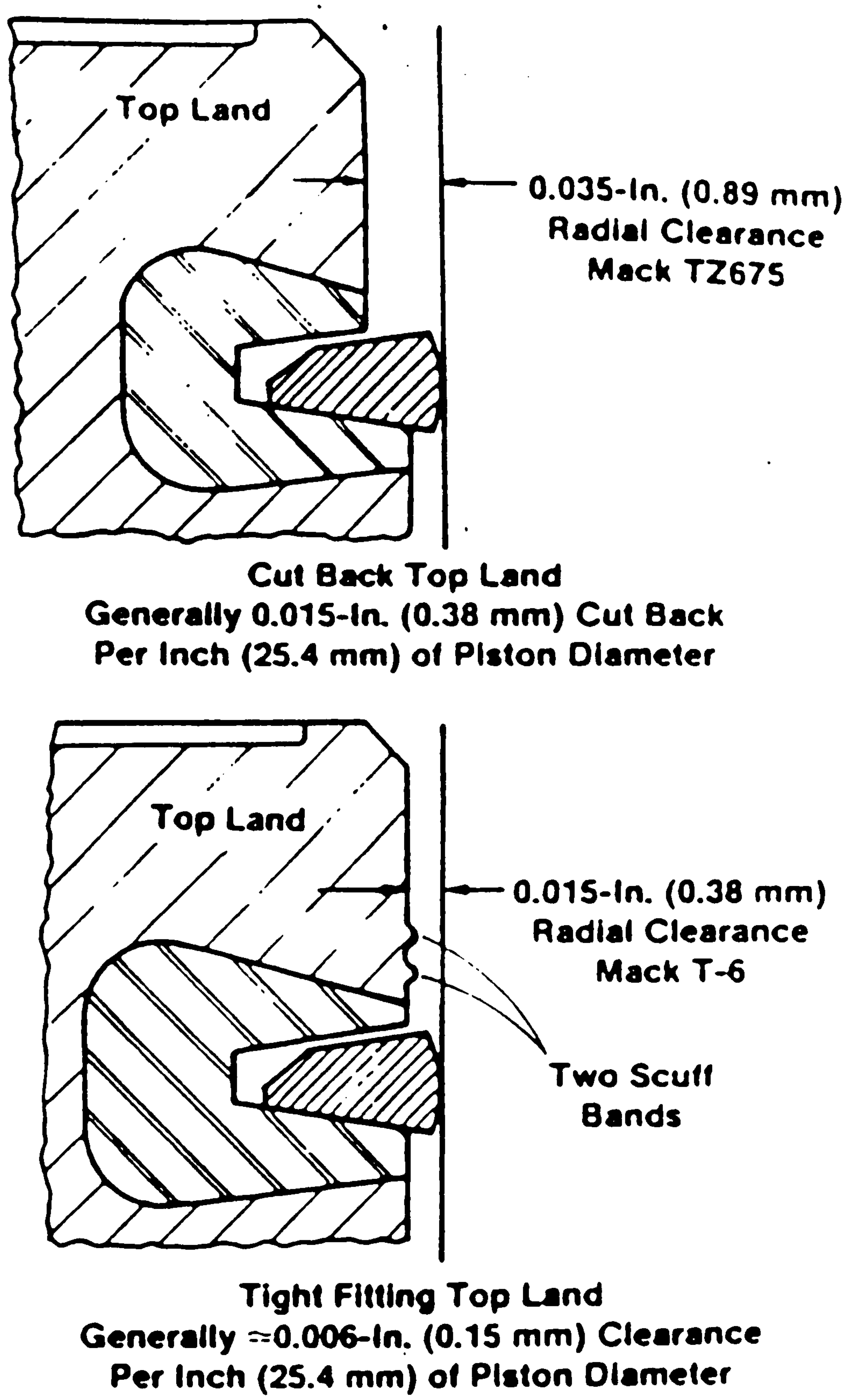


Fig.13 Cut back and tight fitting top land pistons. (Reference 79)

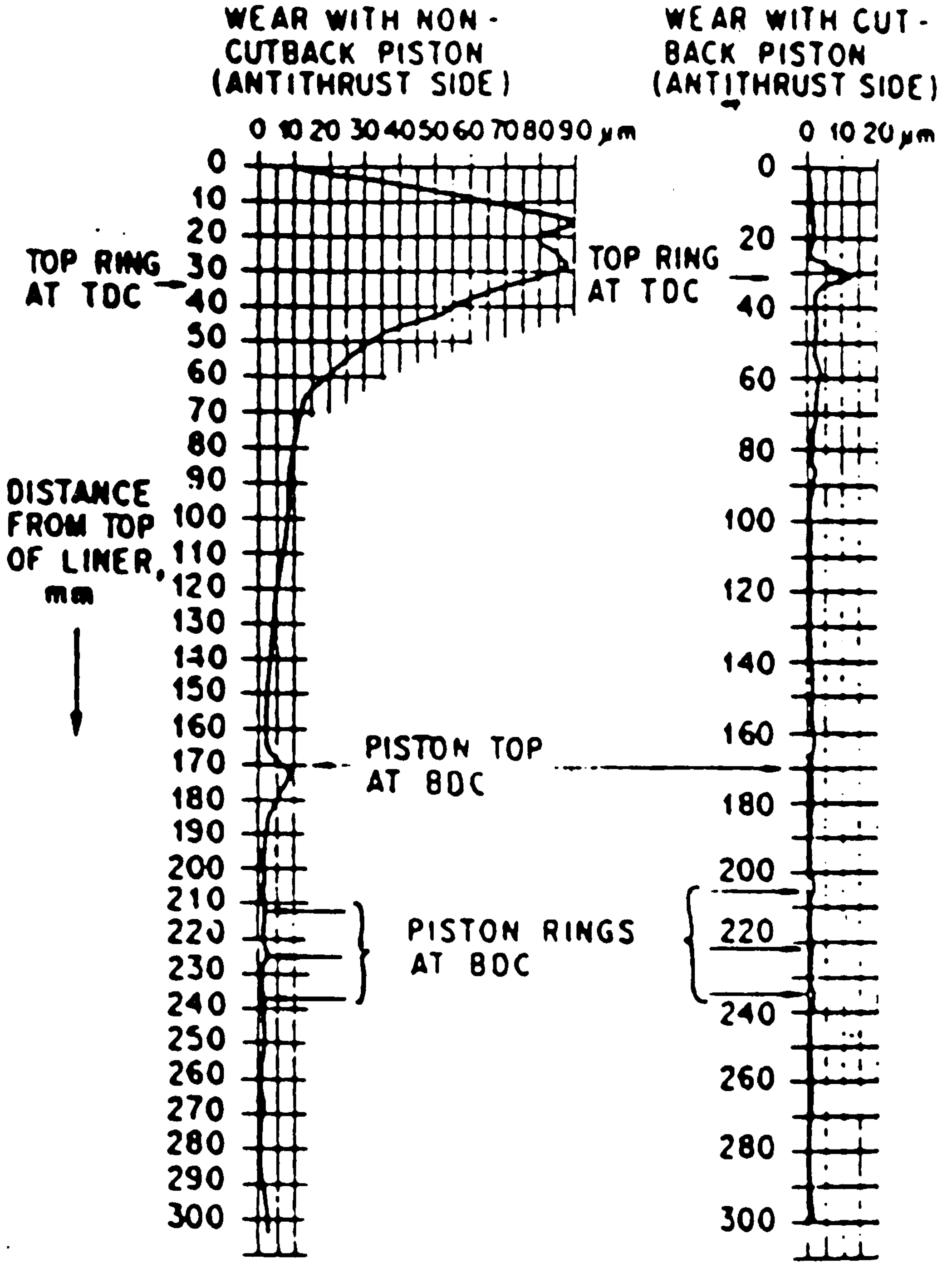


Fig.14 Comparison of liner wear pattern for non-cutback and cutback pistons

(Reference 74)

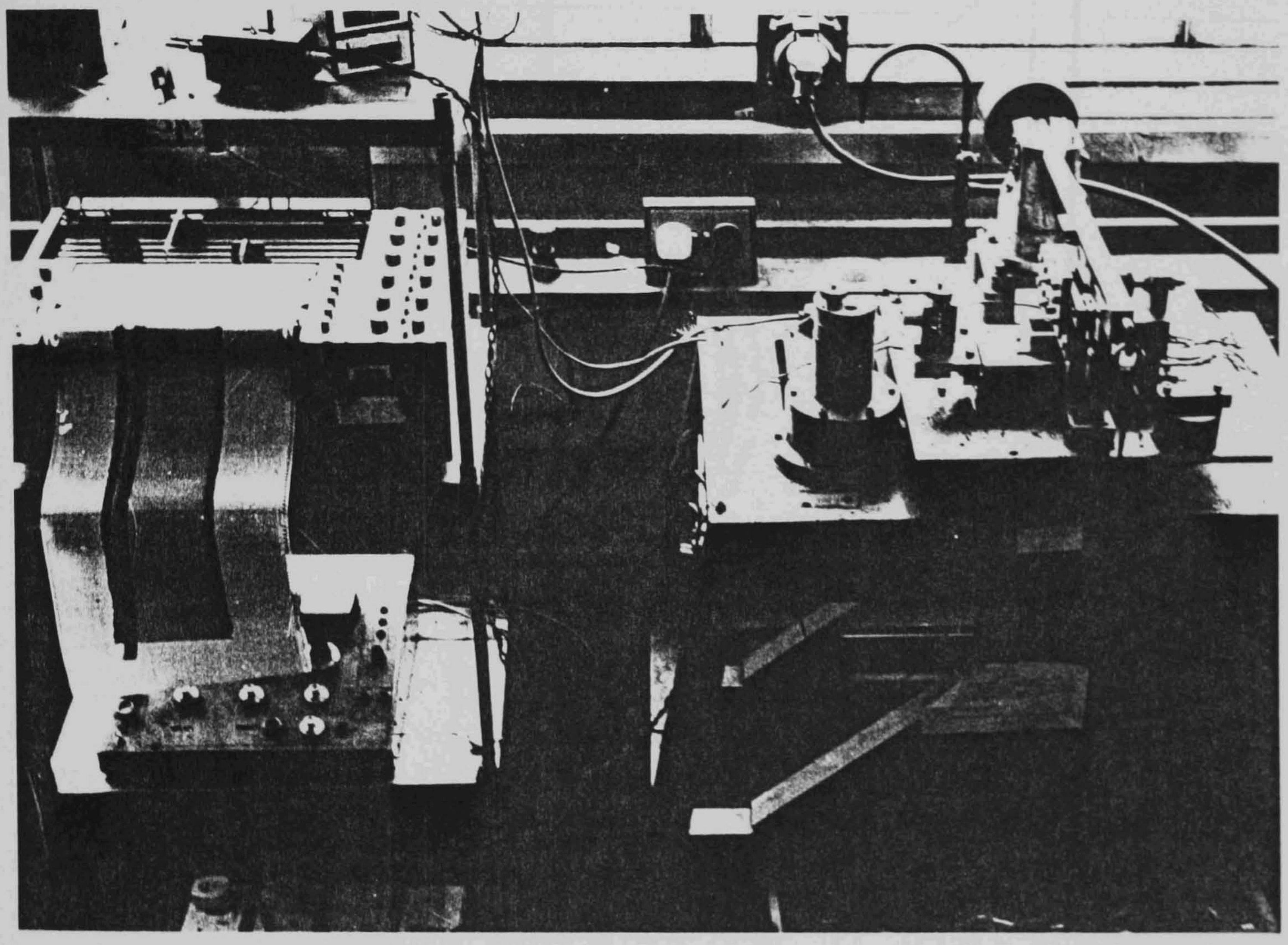


Fig.15 Tribo test machine.

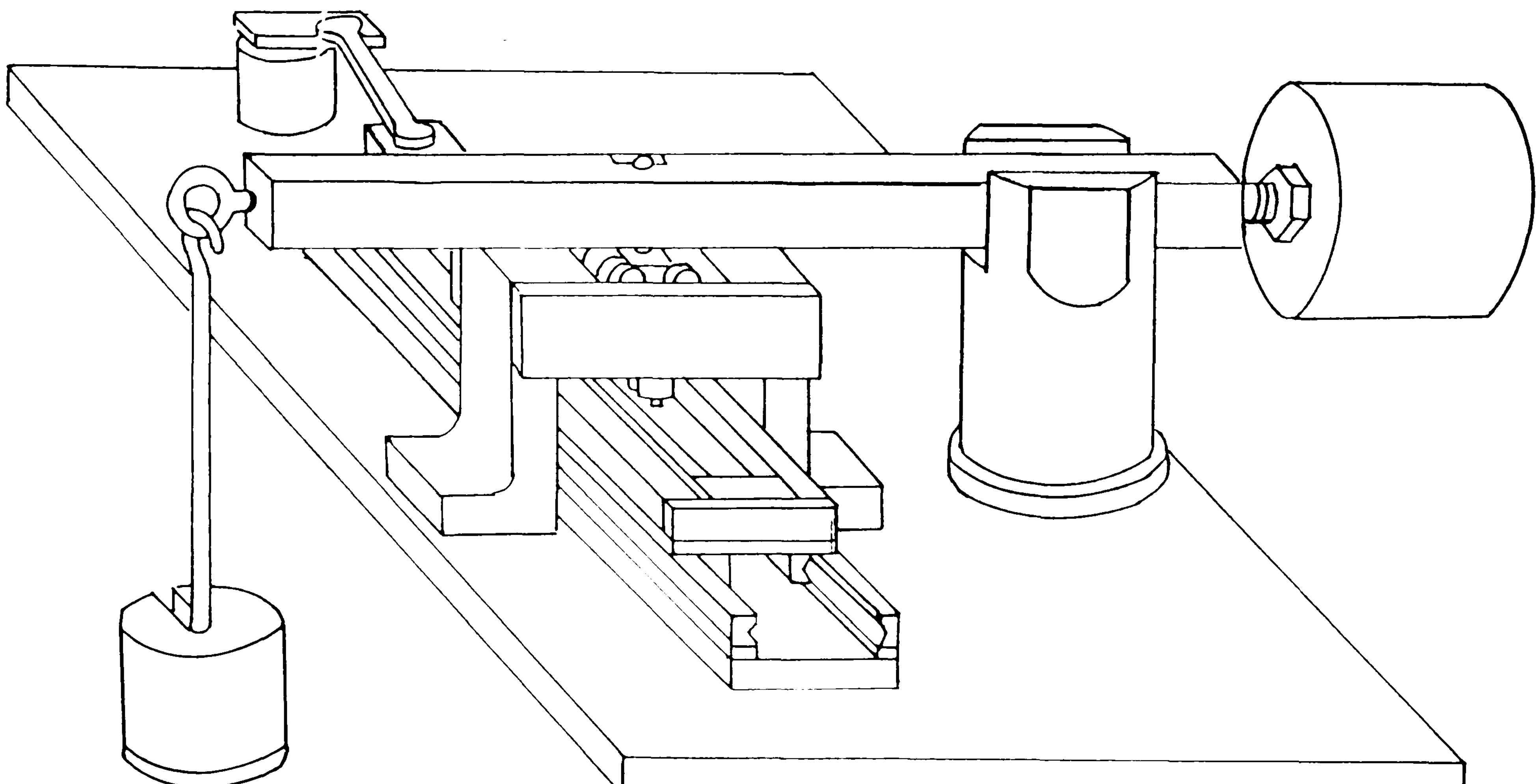


Fig.15 Tribo test machine.

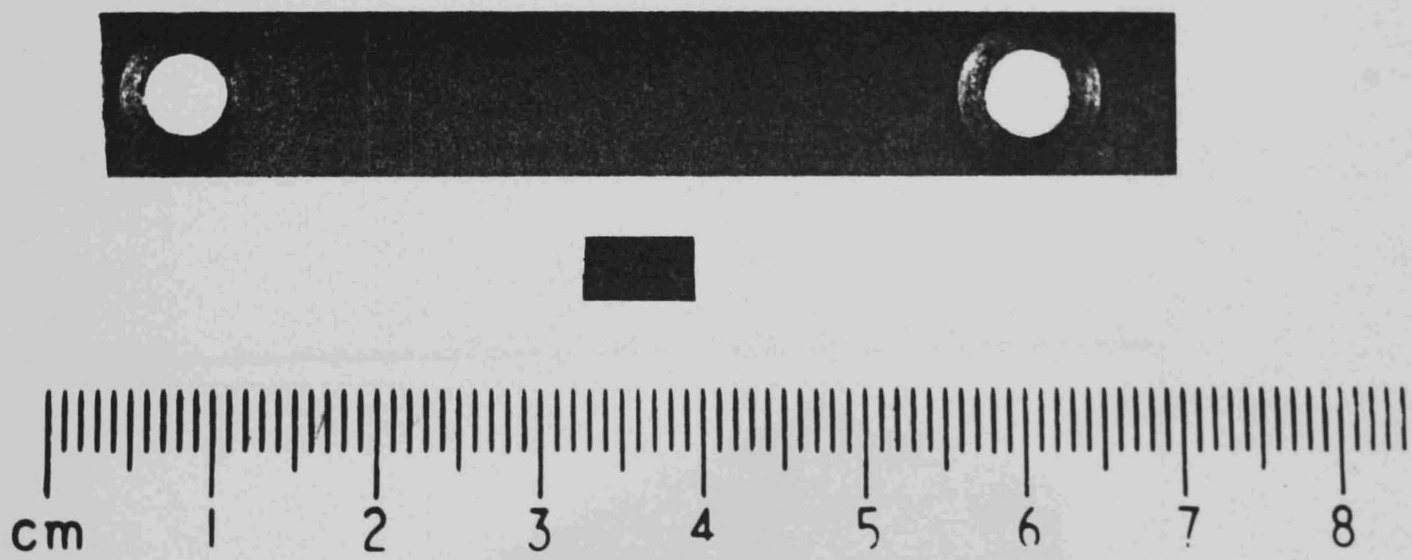


Fig.16 Unused pin and flat.

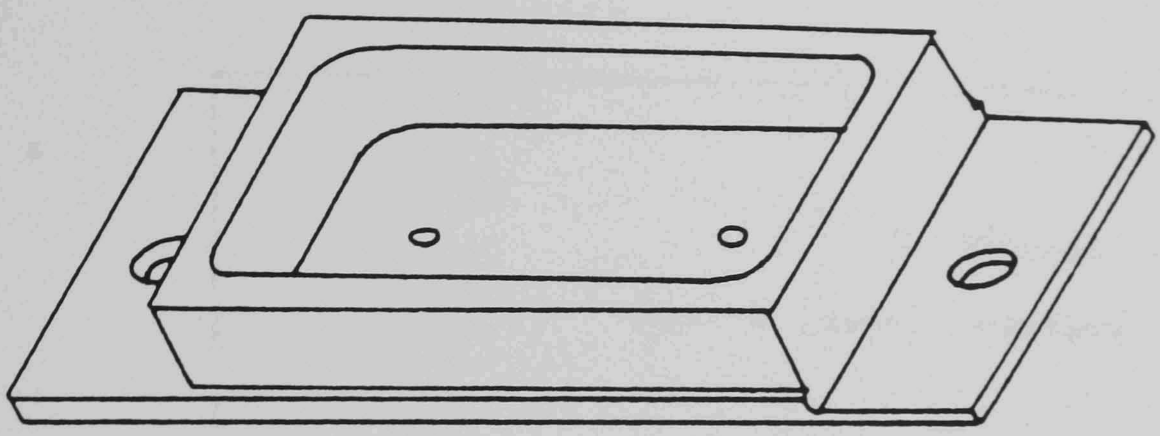


Fig.17 Oil bath.

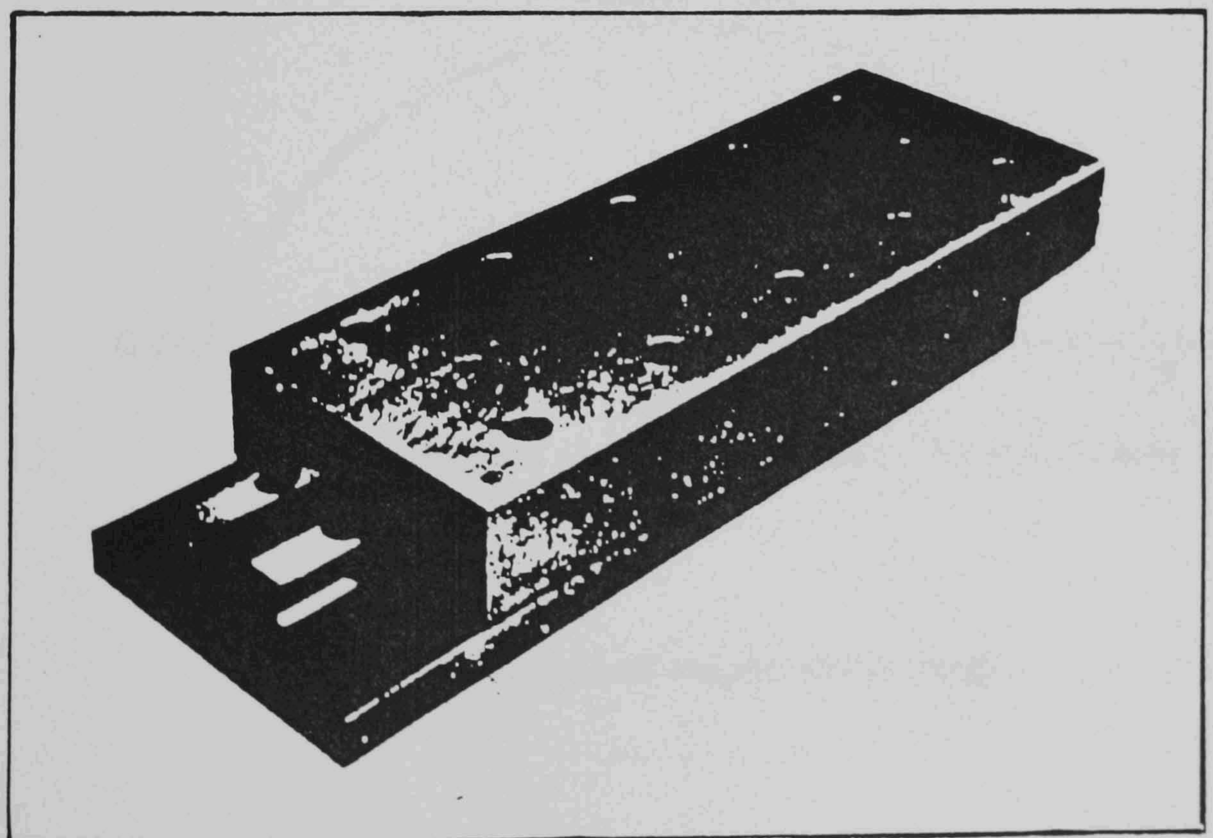


Fig.18 The table underneath the oil bath.

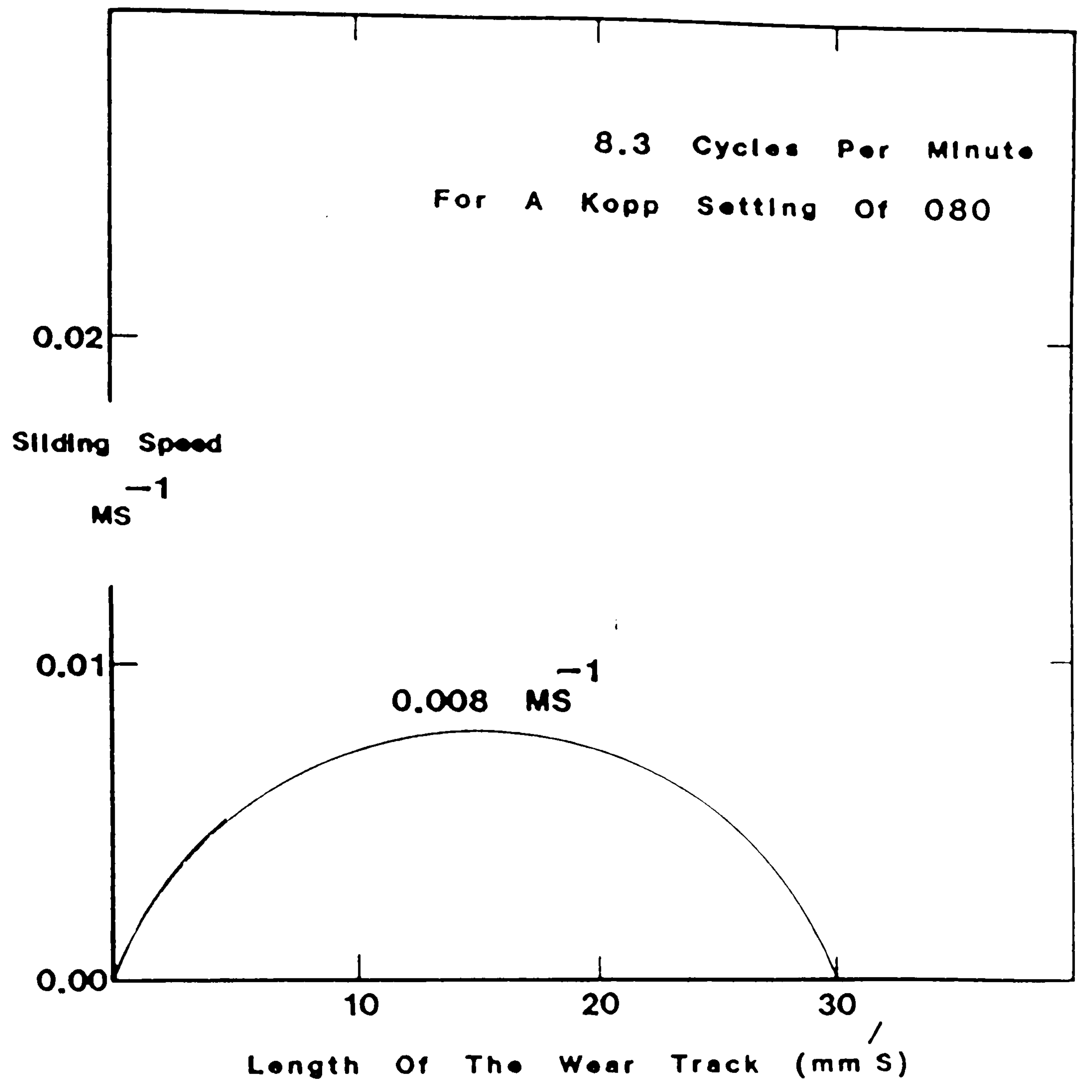


Fig.19 Sliding speed versus track length.

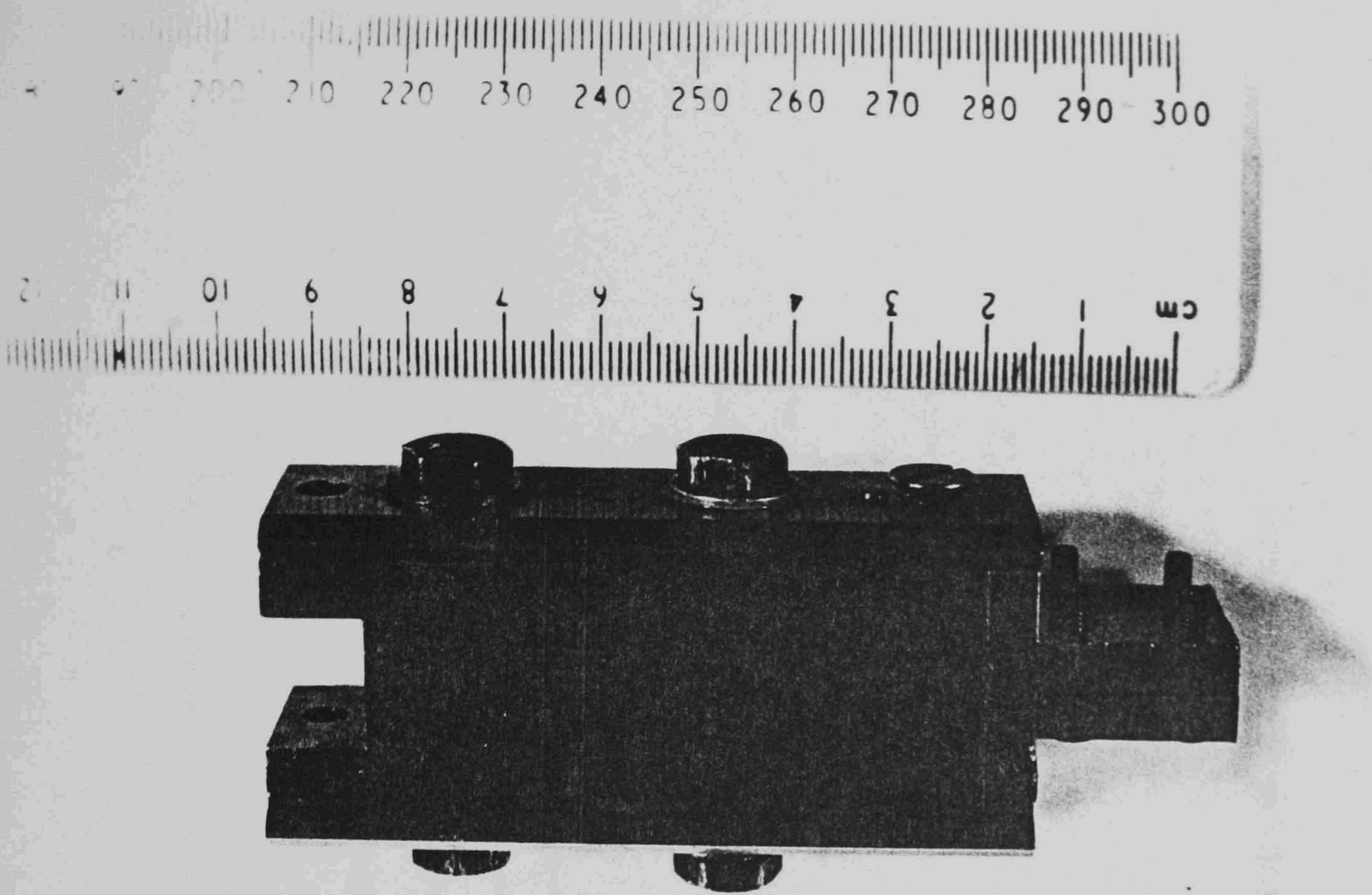


Fig. 20 General view for the holder of the pin.

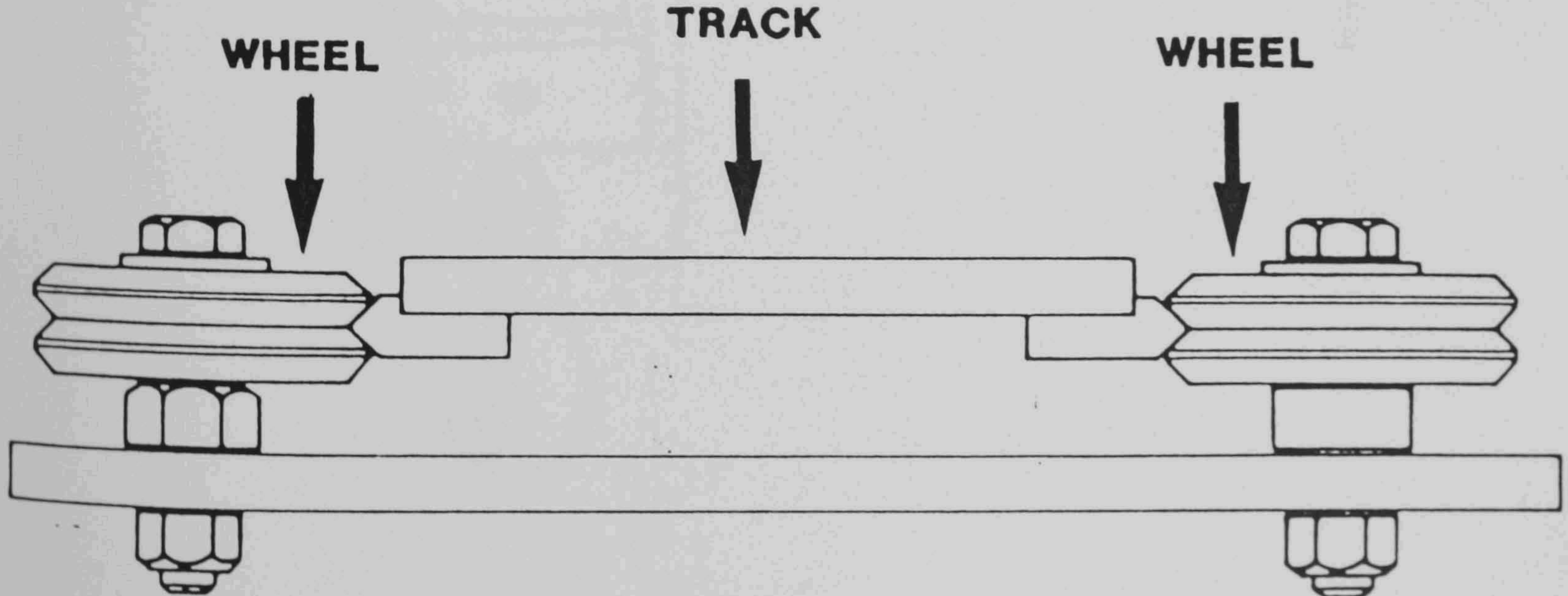


Fig. 21 Section through the pin holder.

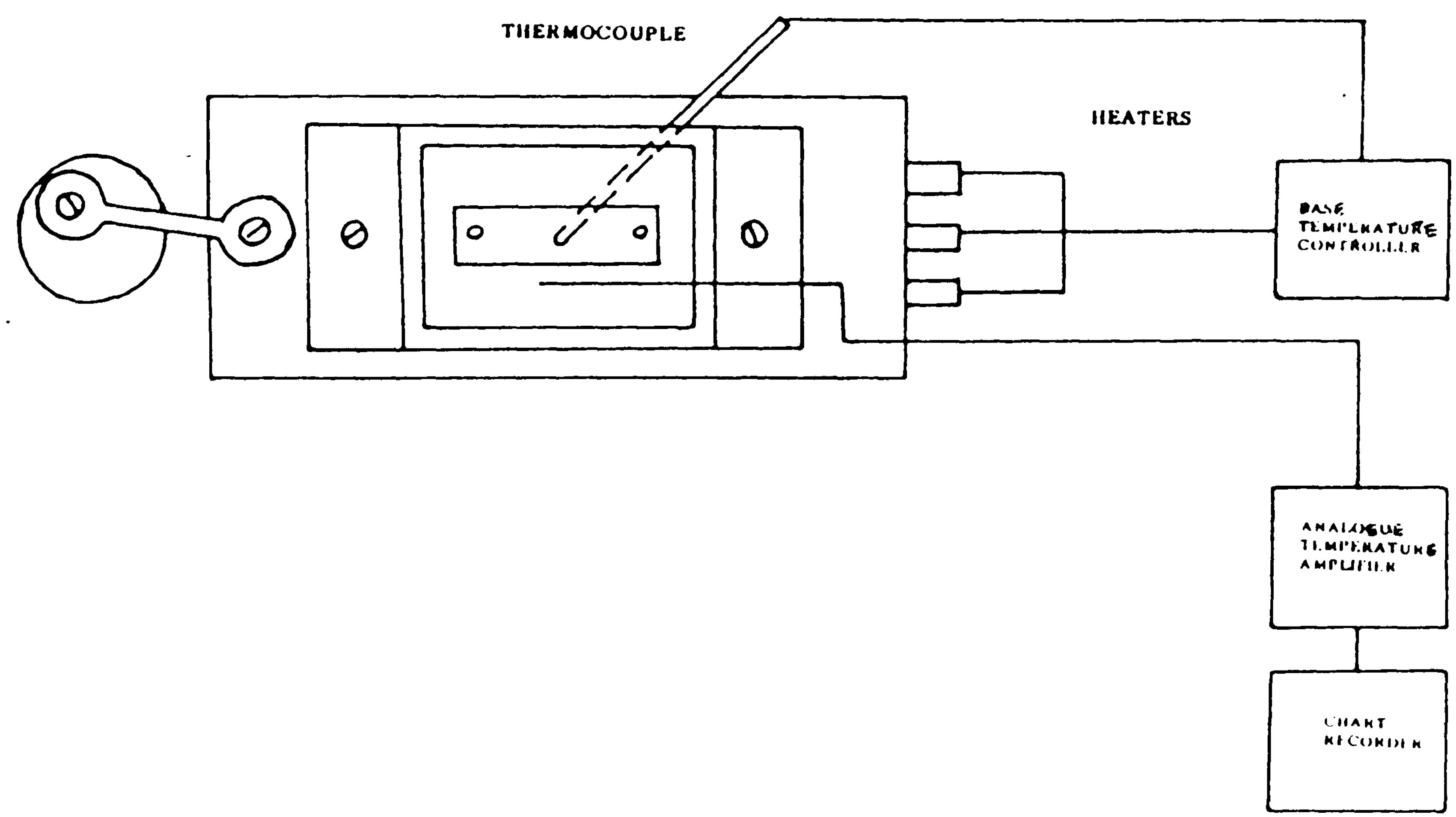


Fig.22 Diagram of oil bath and temperature control system.

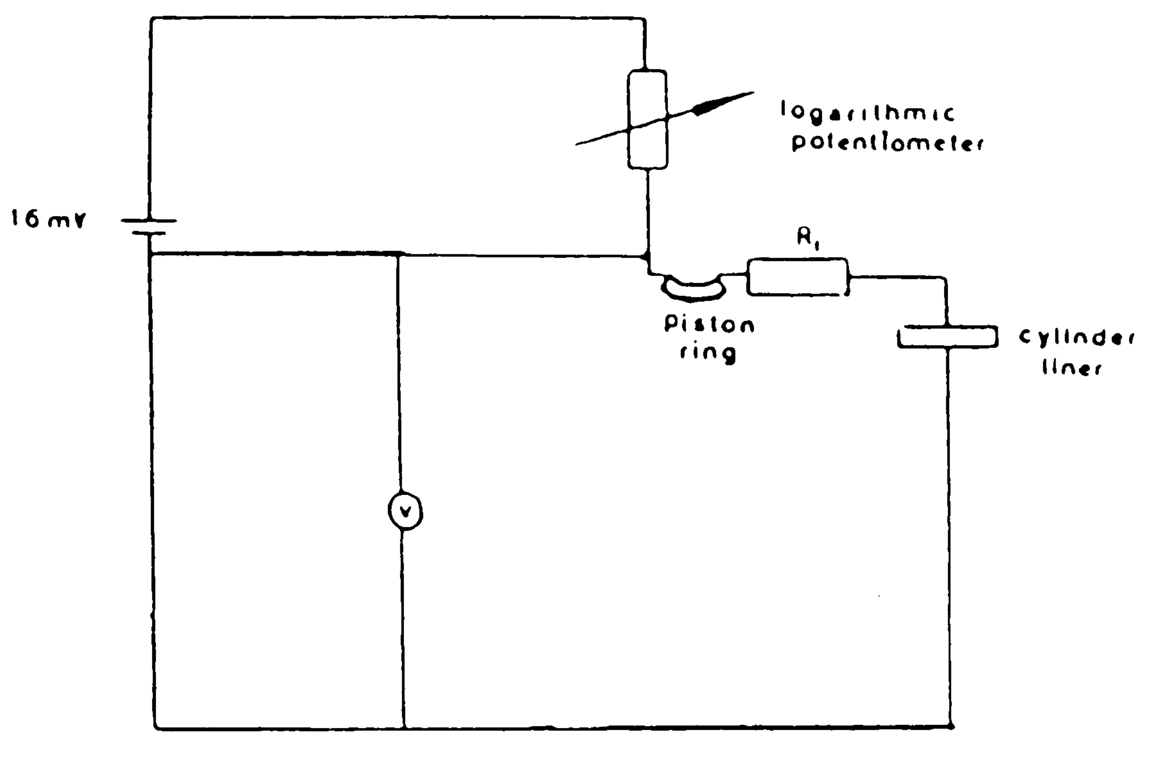
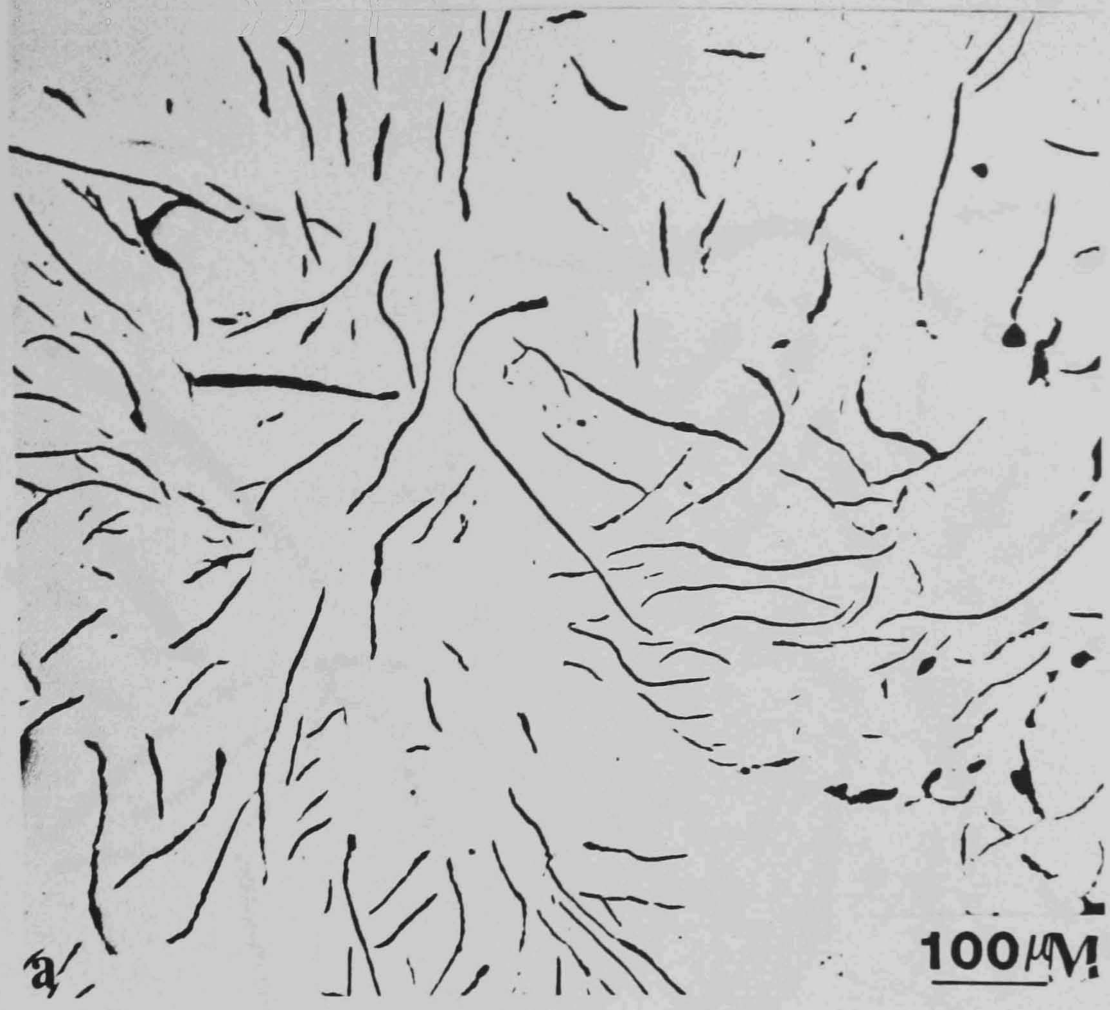


Fig.23   Calibration of contact resistance circuit.



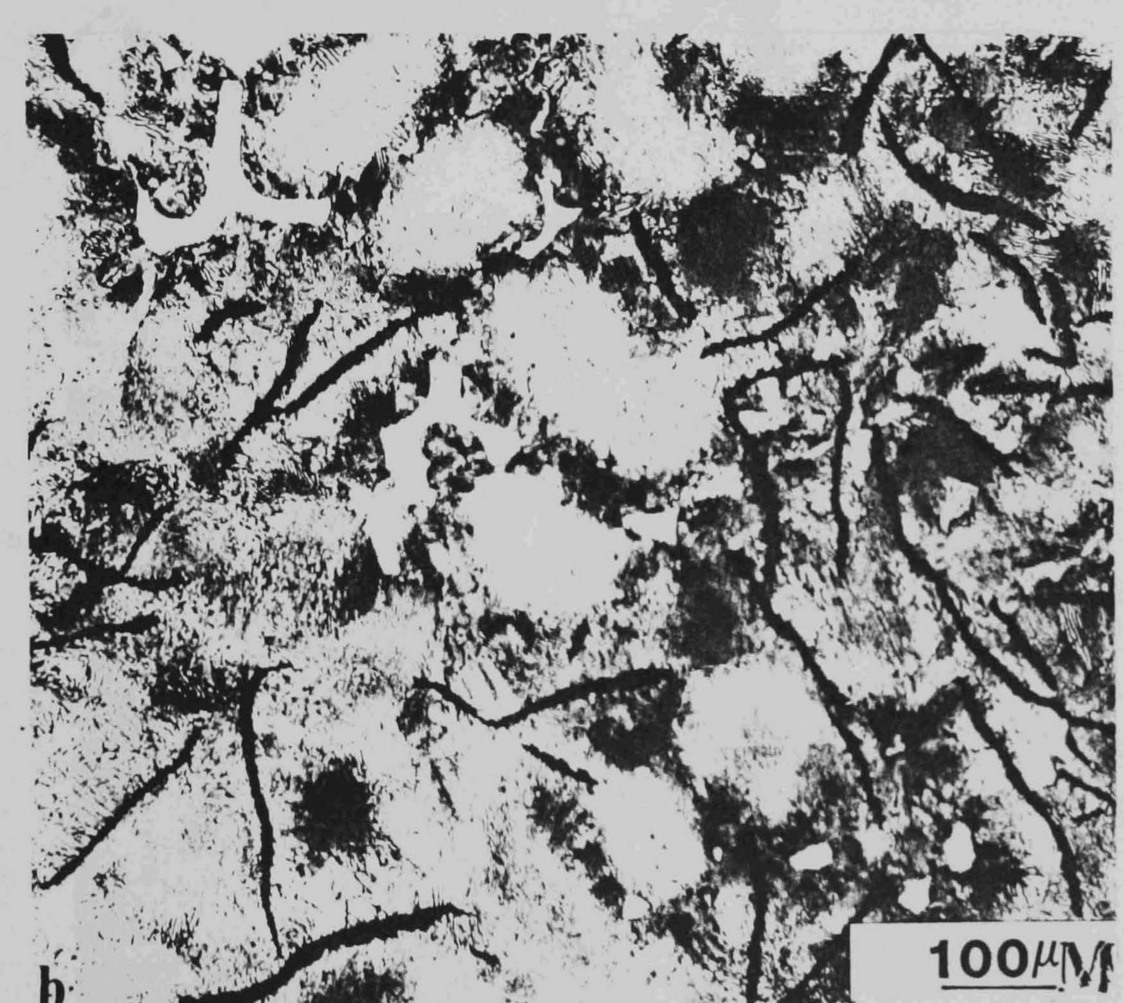
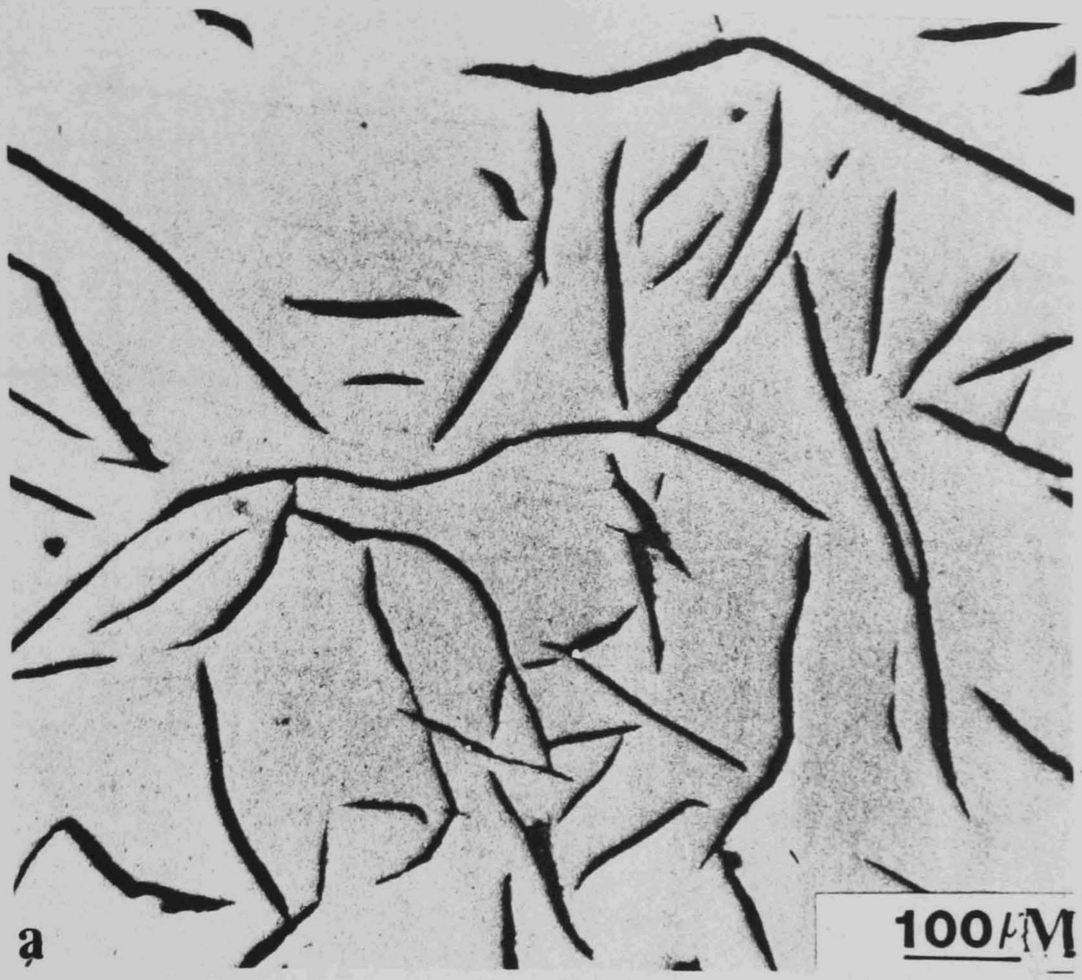


Fig. 24 Microstructure of the pin:
a) Unetched
b) Etched



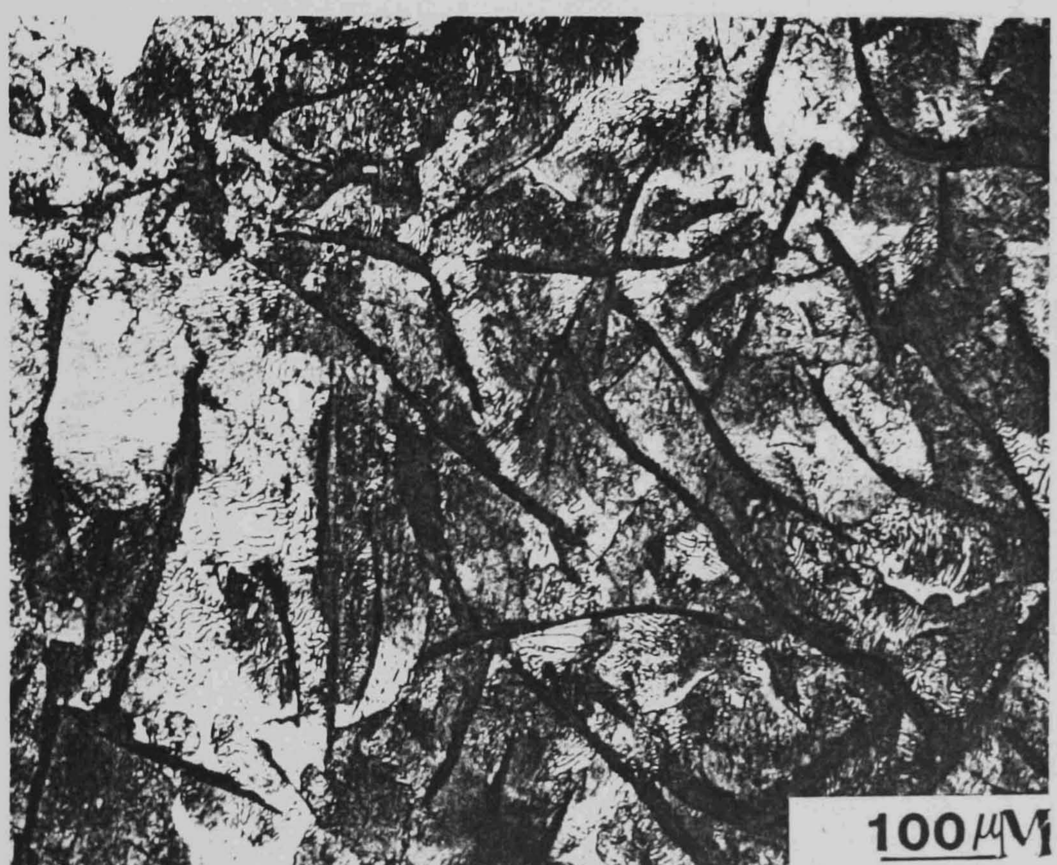


Fig. 25 Microstructure of the flat:
a) Unetched
b) Etched

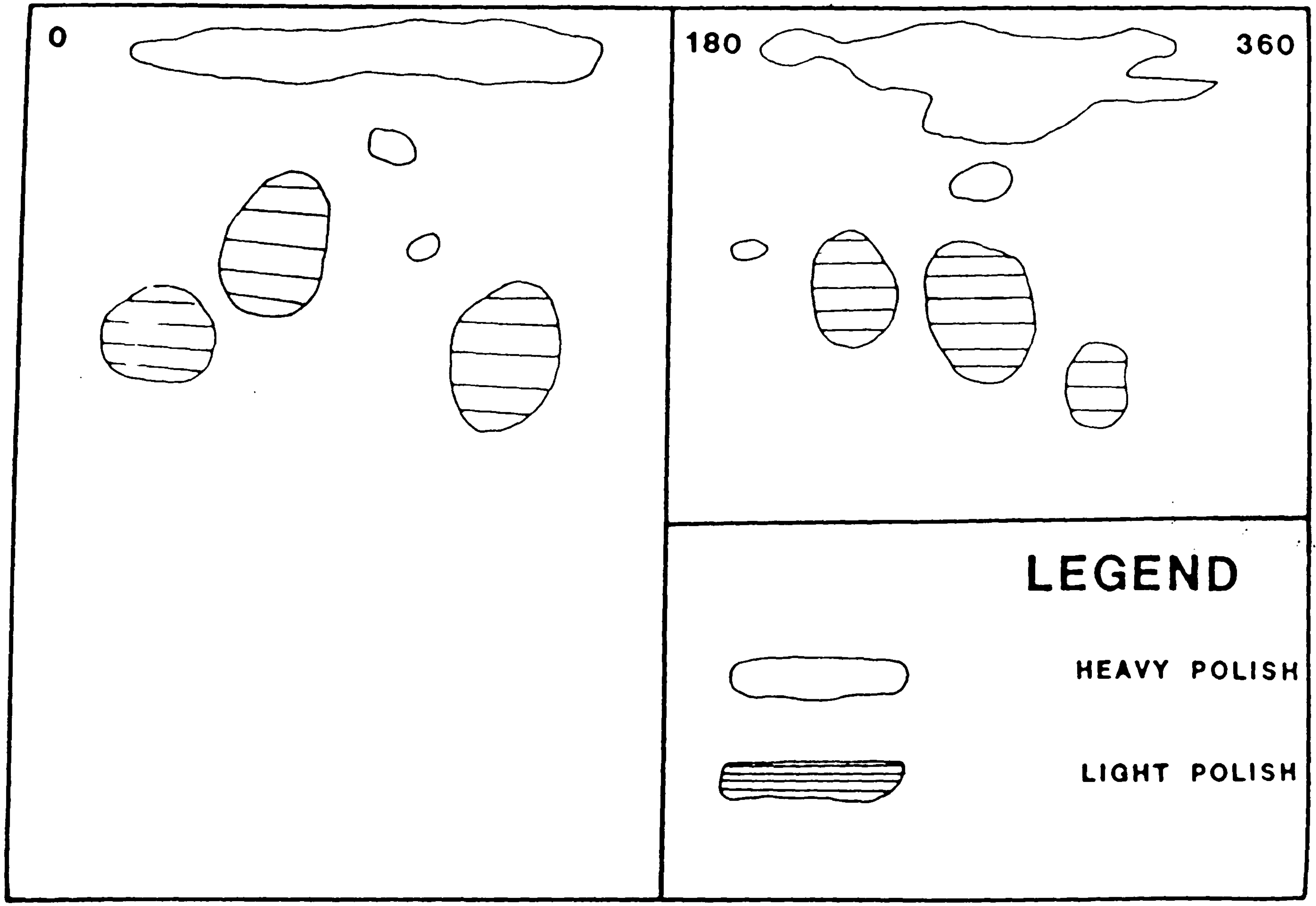


Fig. 26 Tracing of the polished liner.

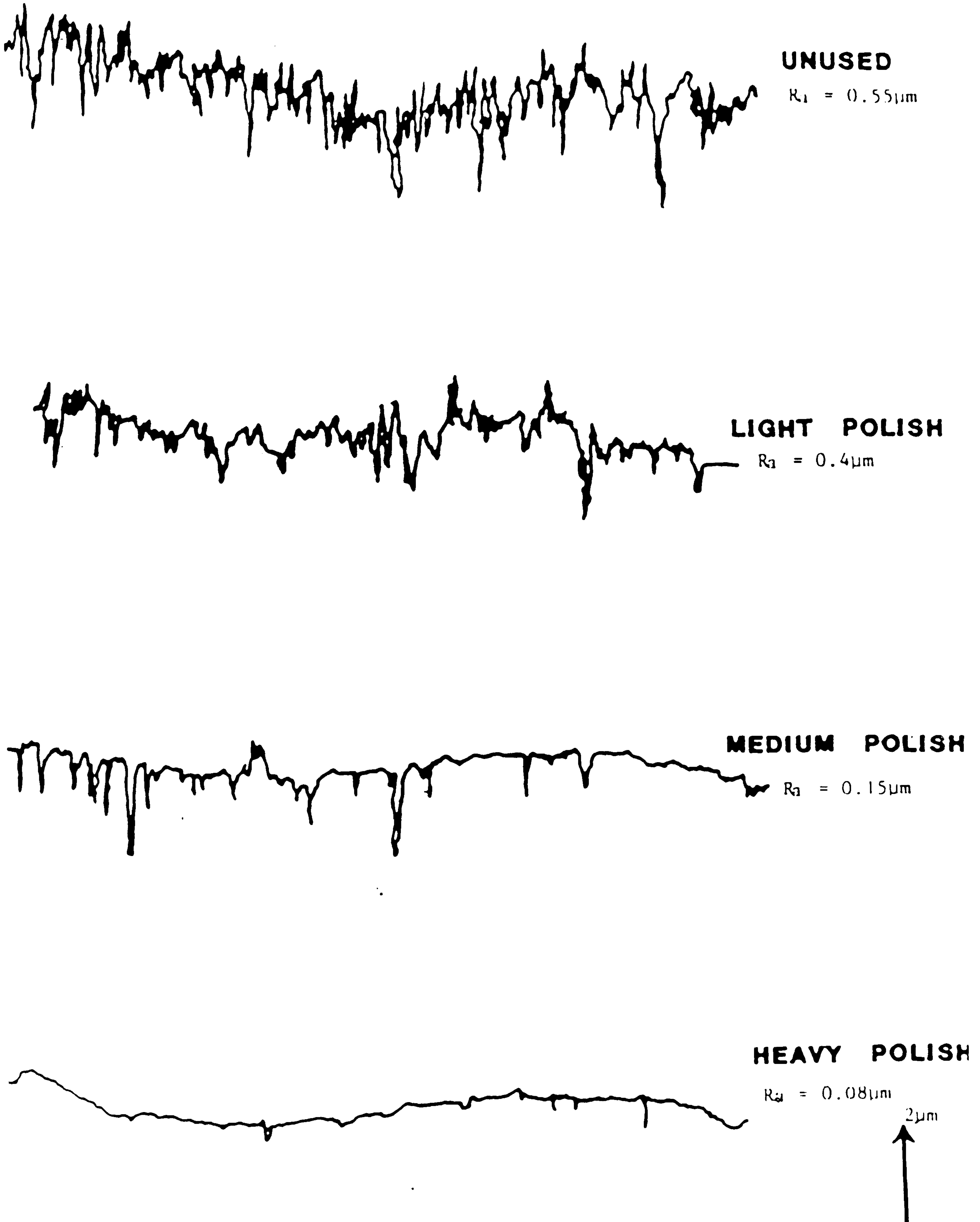
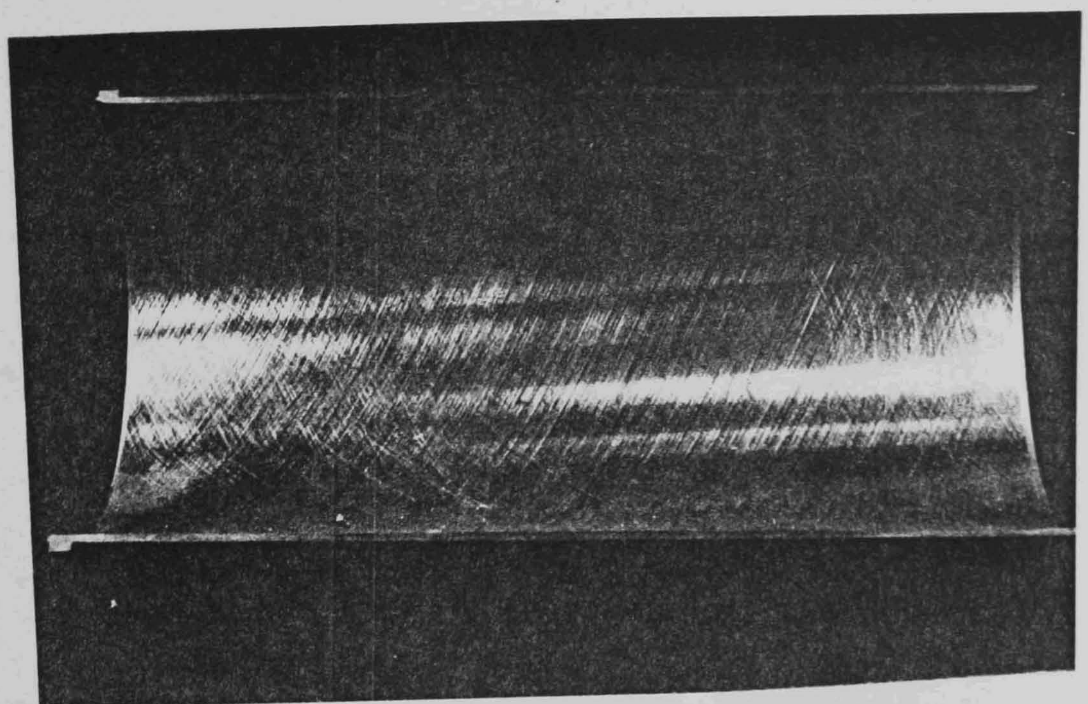


Fig.27 The C.L.A. value for polished Tornado liner.



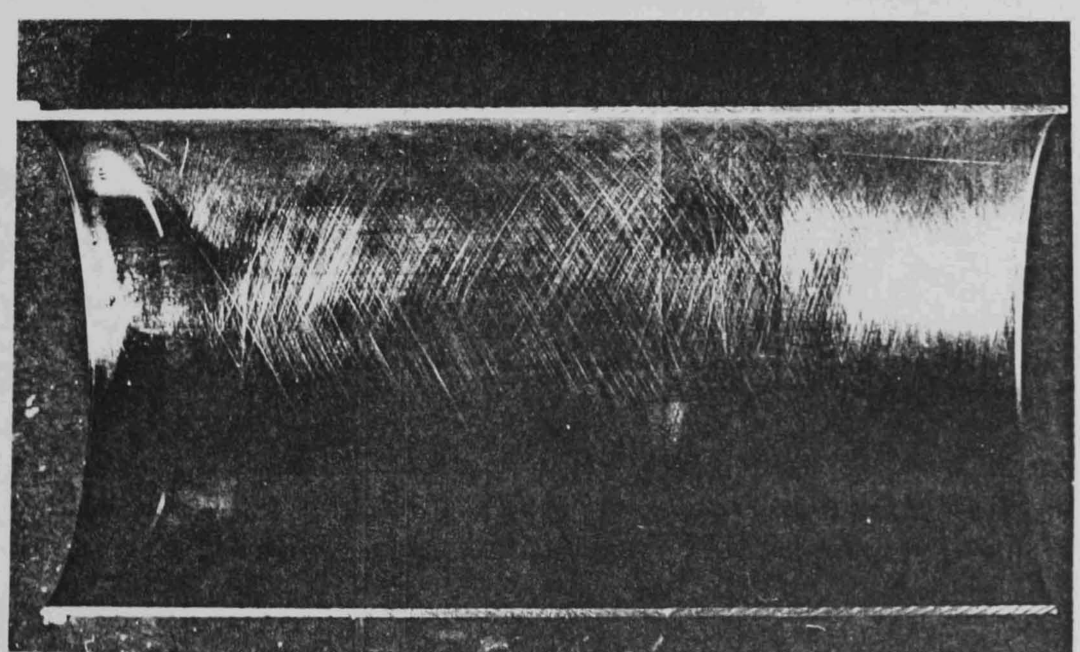


Fig. 28   General view for unused and polished Tornado liner.

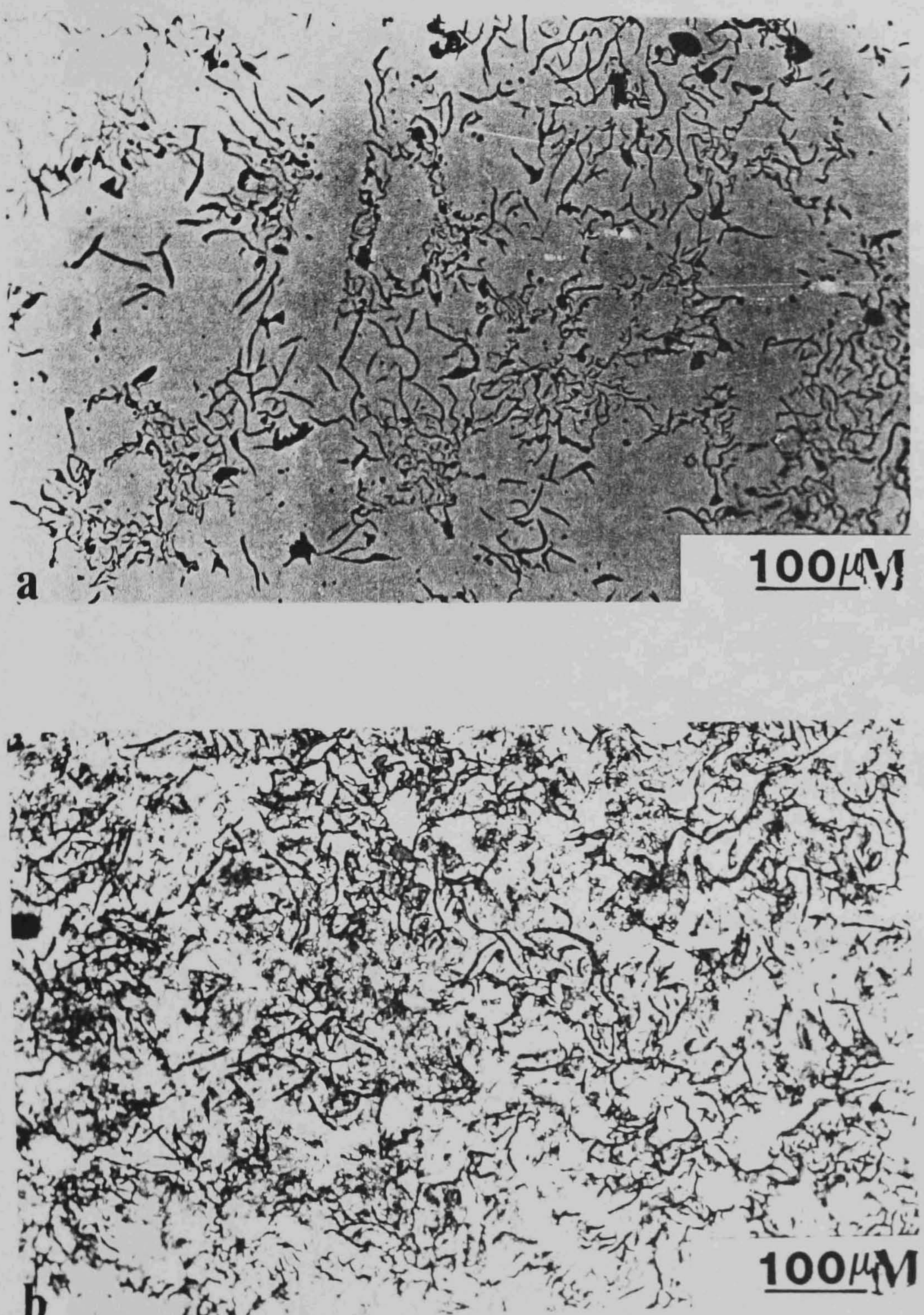


Fig. 29 Microstructure of the liner:
a) Unetched
b) Etched

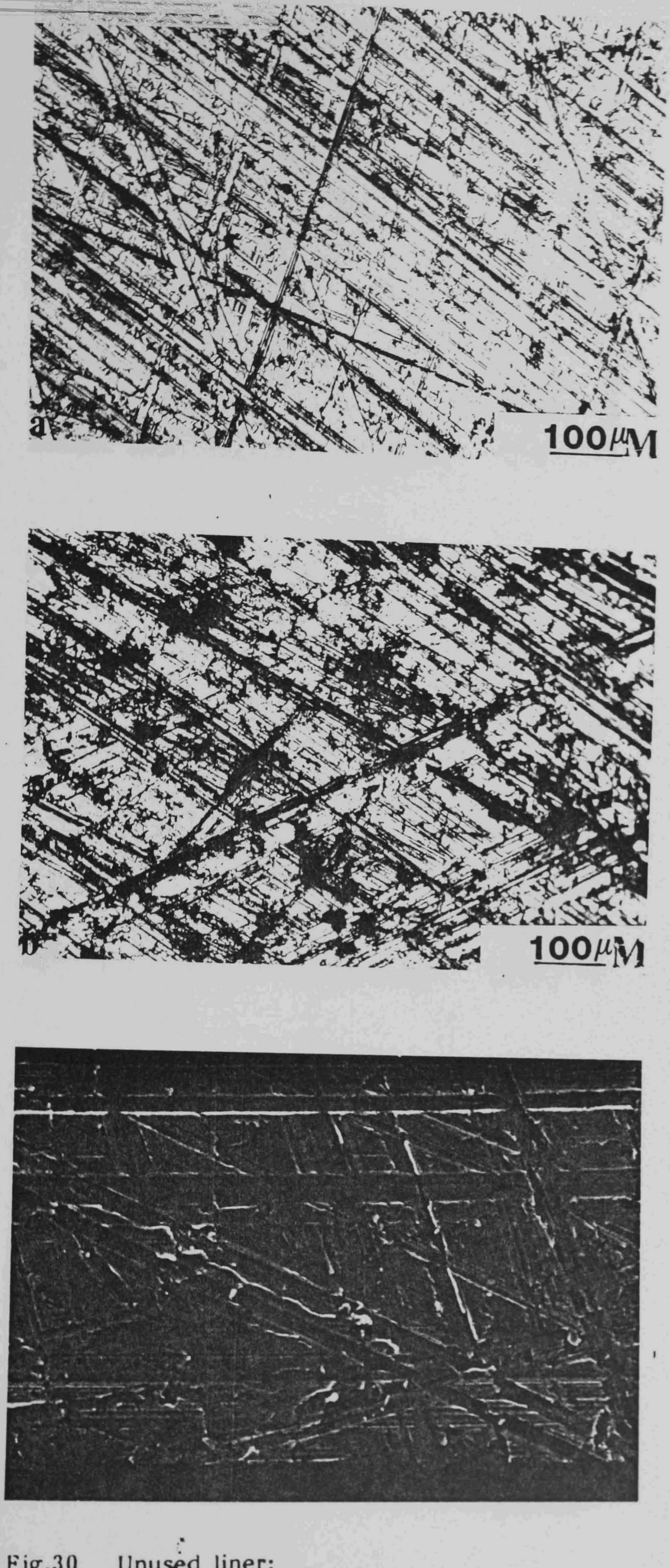


Fig.30 Unused liner:
a) Replica
b) Optical
c) Scanning electron microscopy

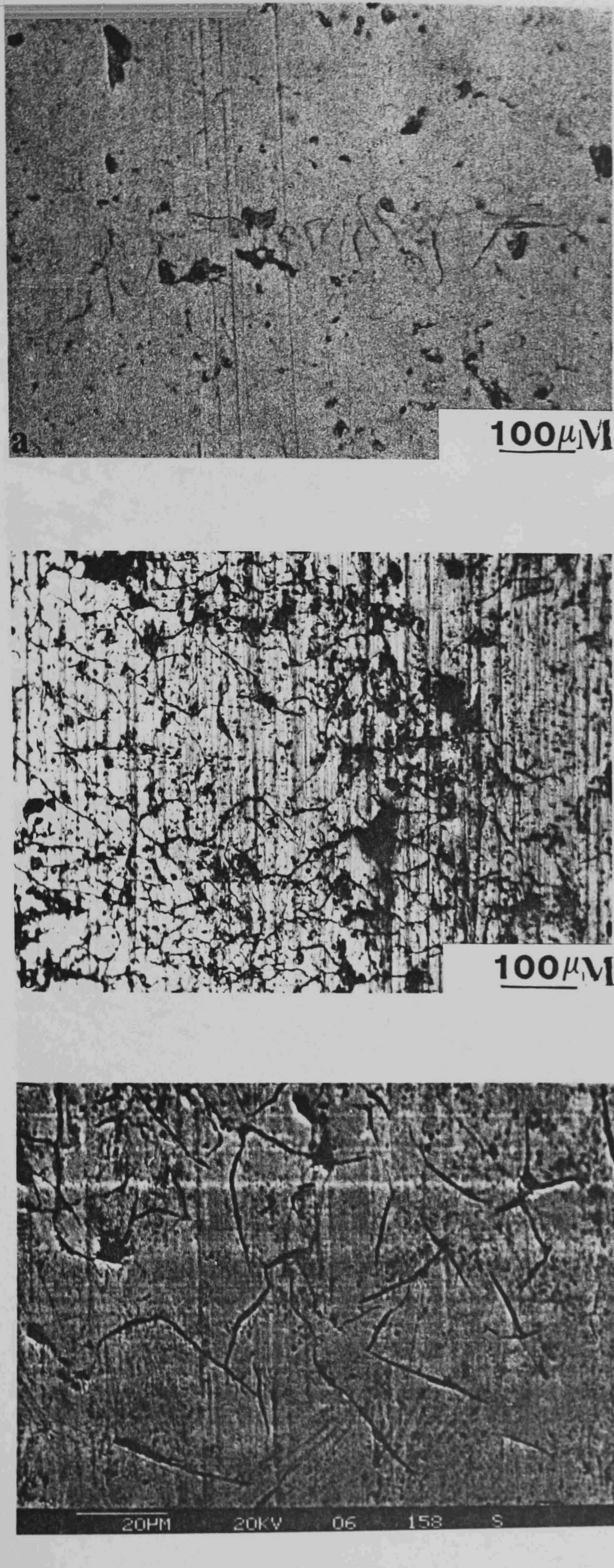
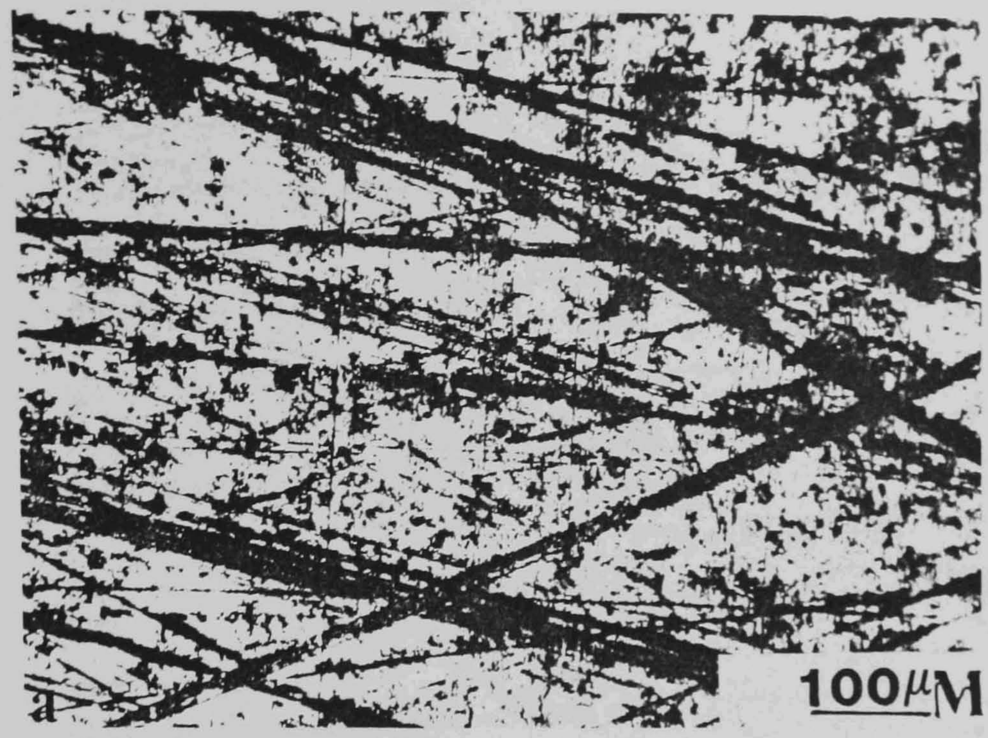


Fig.31 Heavy polish liner:

a) Replica
b) Optical
c) Scanning electron microscopy



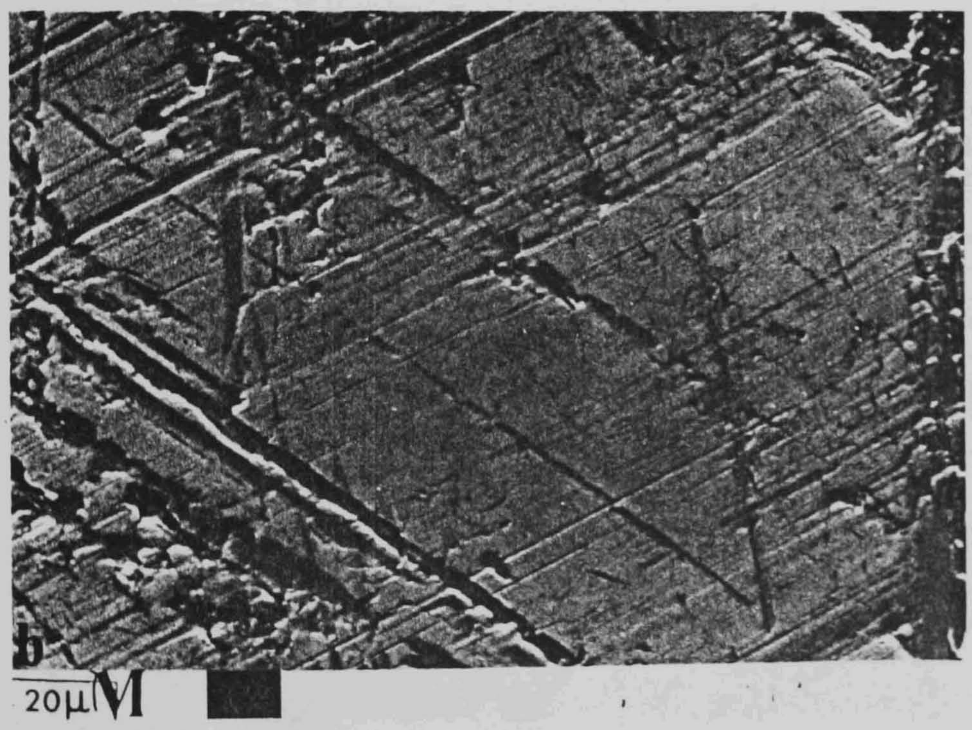


Fig.32 Medium polish liner:
a) Optical
b) Scanning electron microscopy

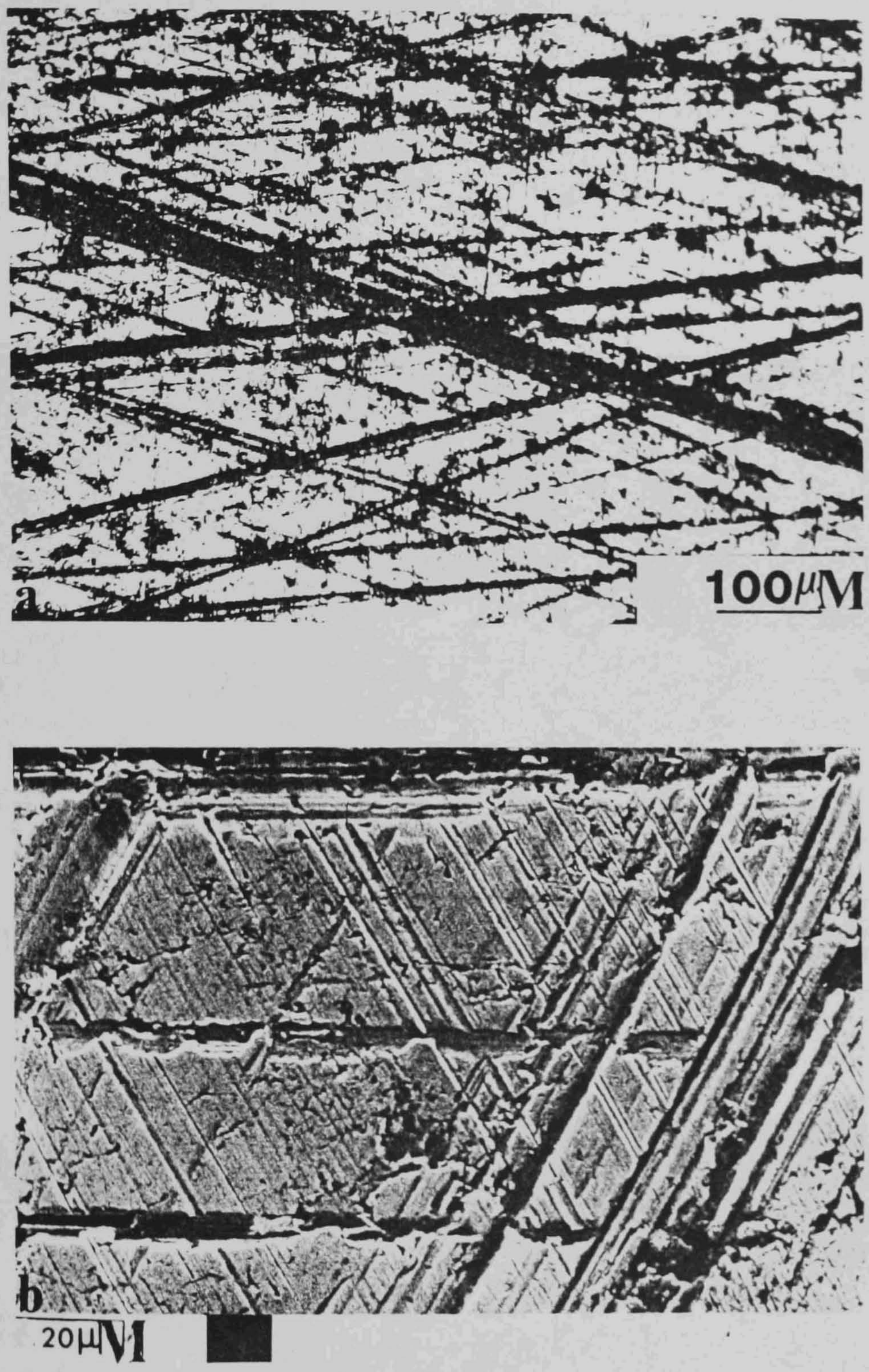
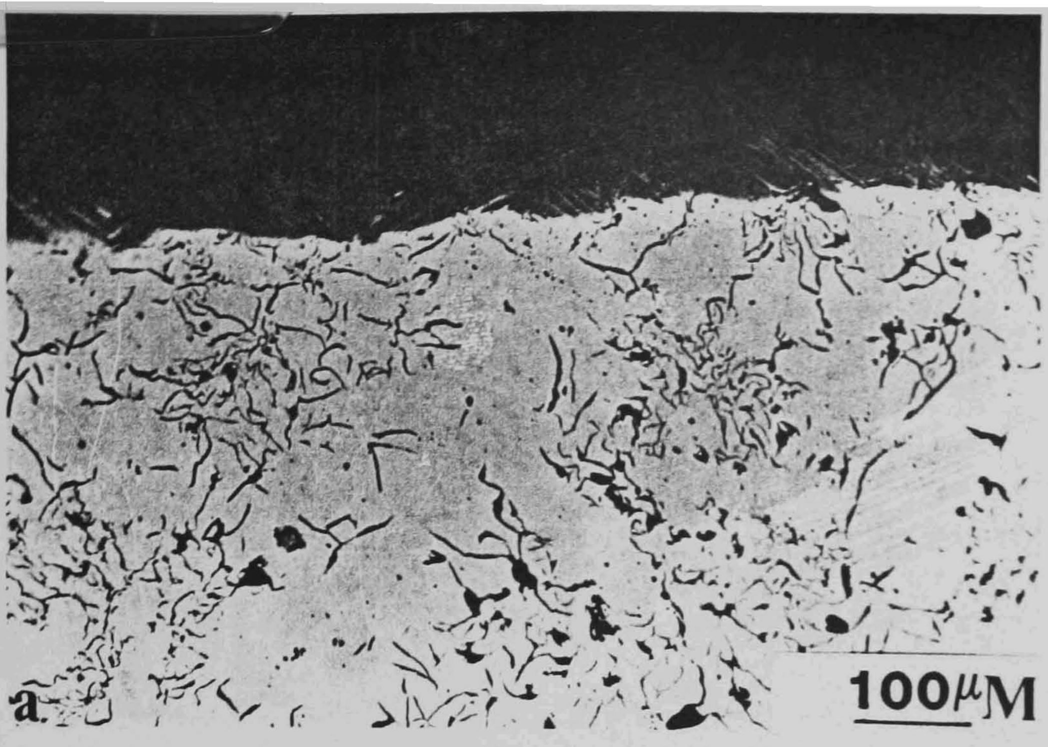
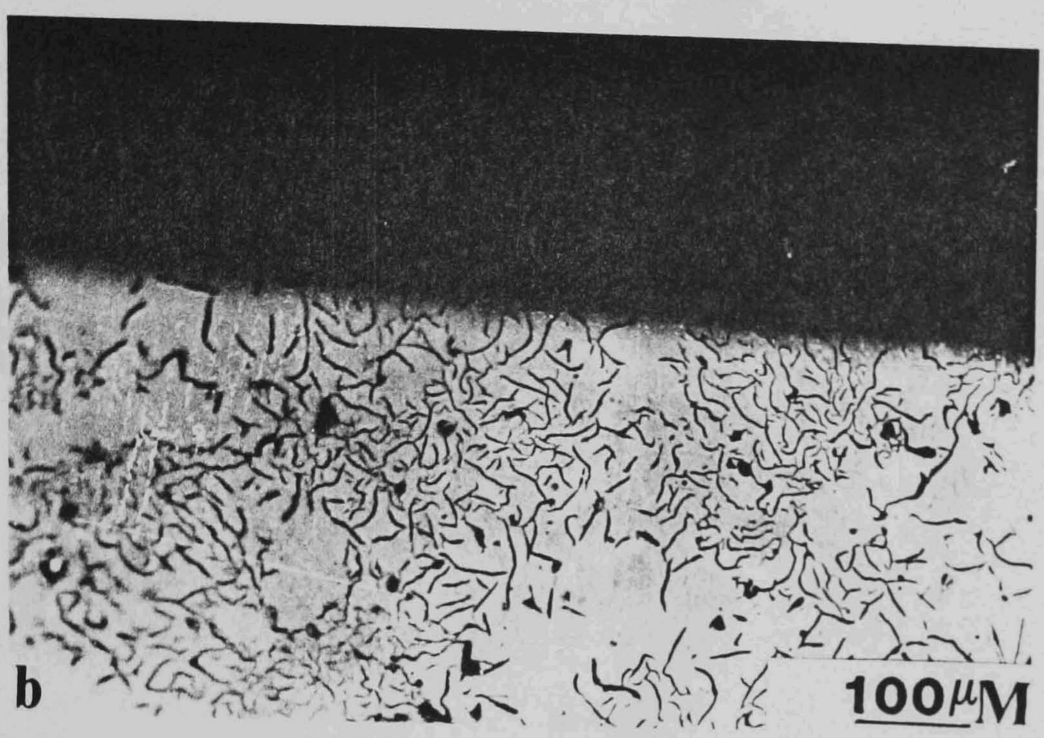


Fig.33 Light polish liner:
a) Optical
b) Scanning electron microscopy





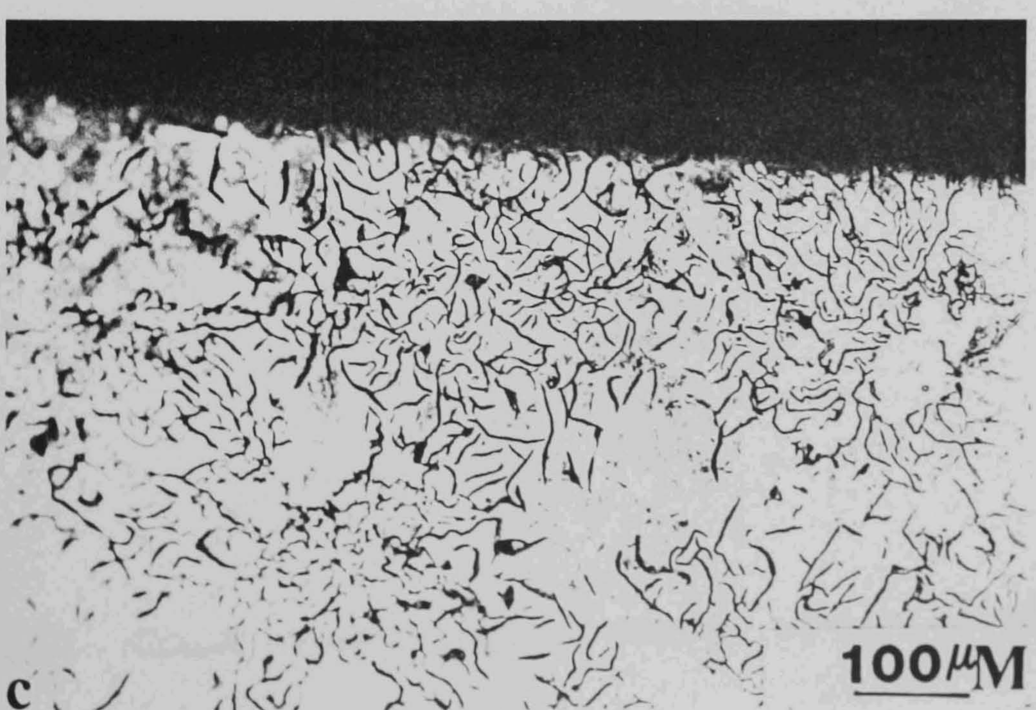


Fig.34 Taper section through:
a) Unused liner
b) Heavy polish liner
c) Etched heavy polish liner

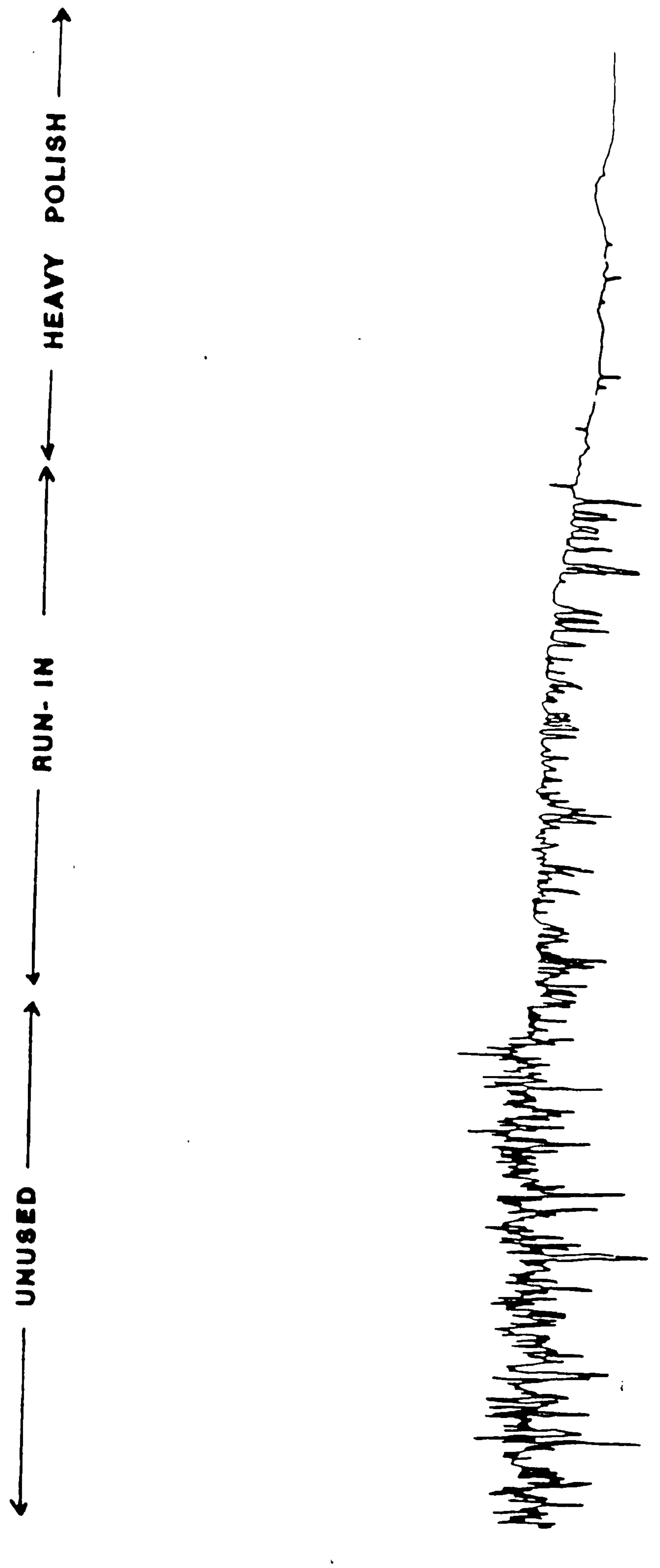


Fig.35 C.L.A. trace through an unworn and bore polished area of the liner

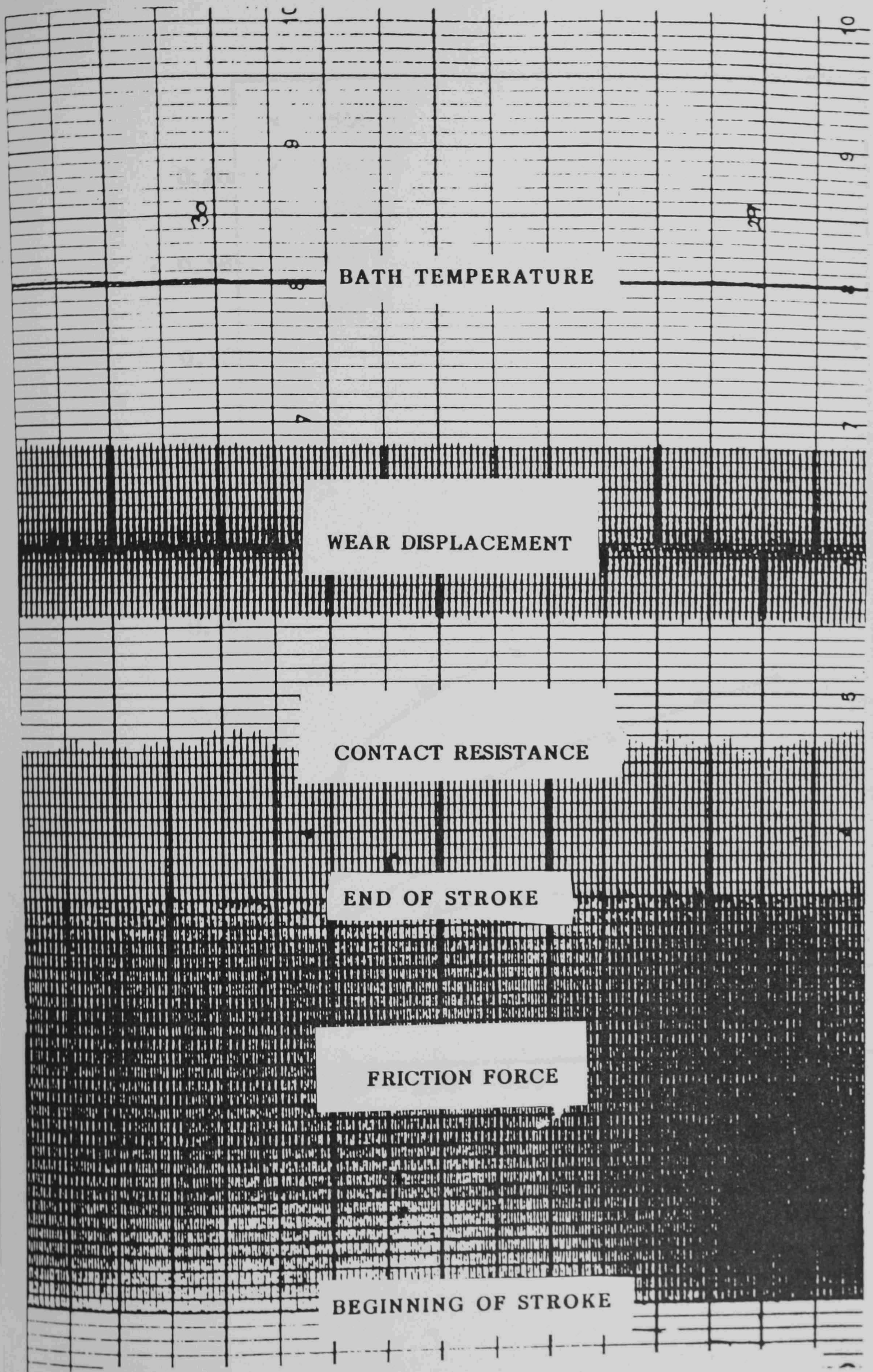


Fig.36 Section of four channel chart from reciprocating tribo test machine.

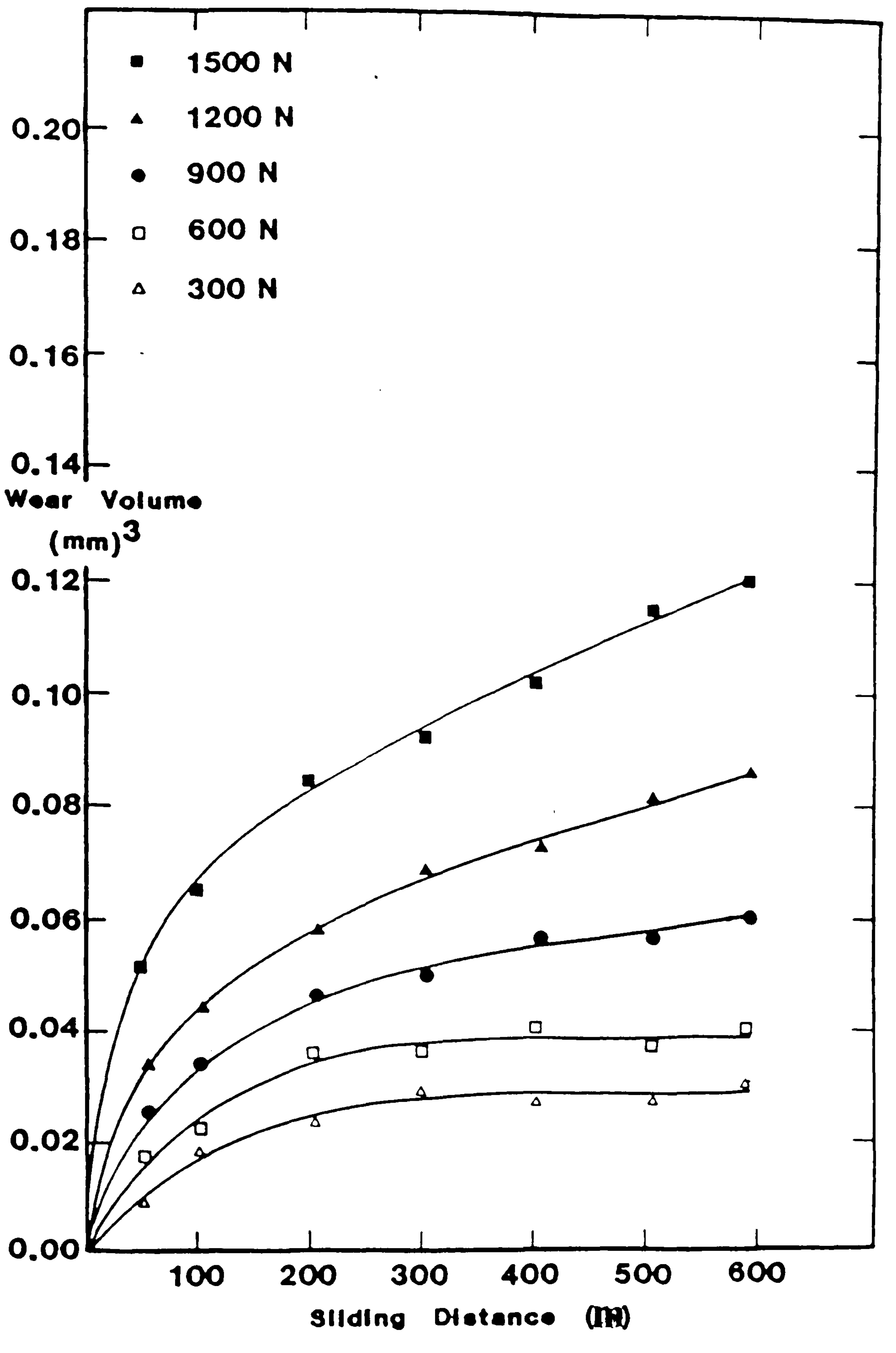


Fig.37 Wear for RL47 at 80°C.

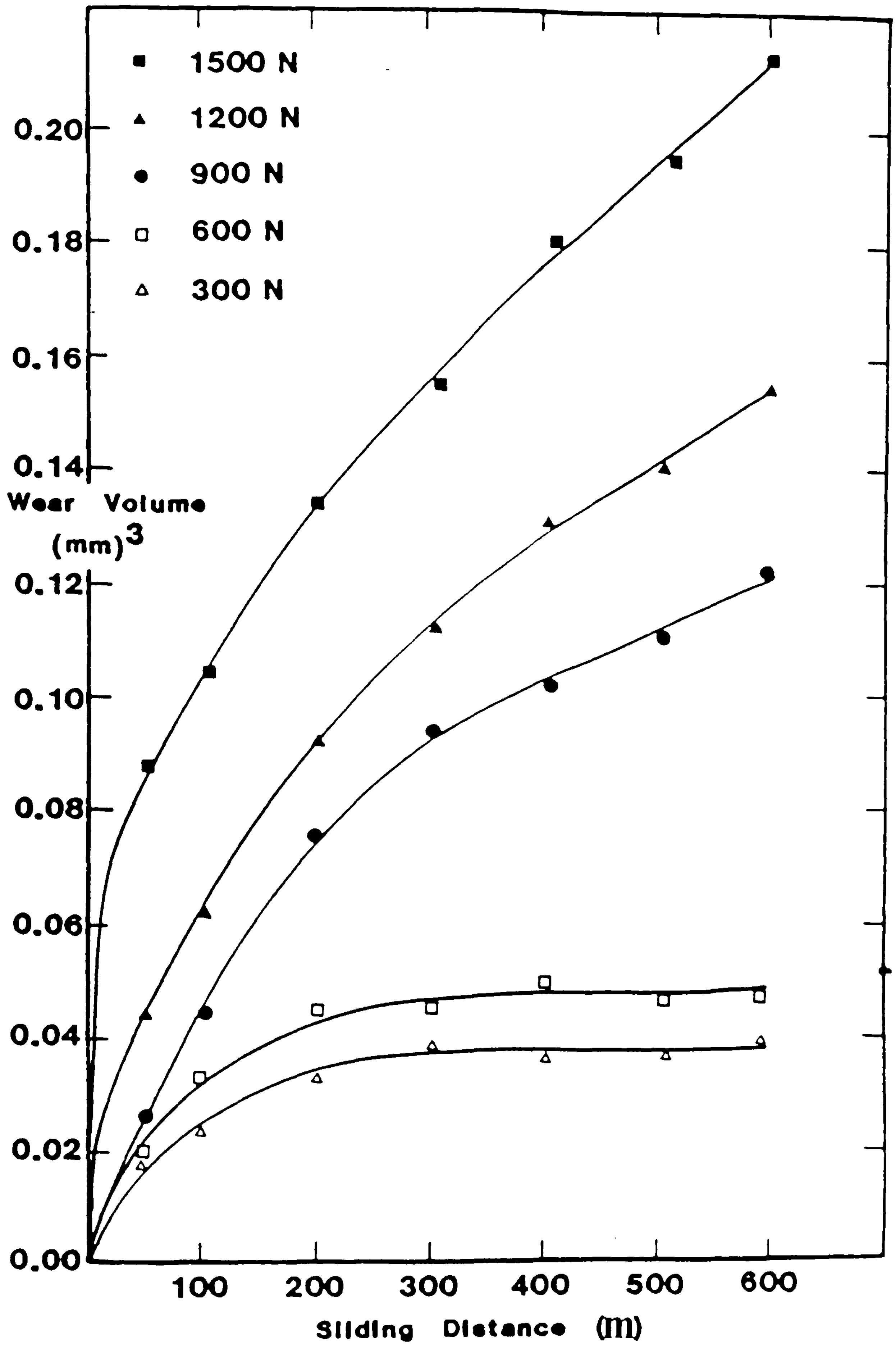


Fig.38 Wear for RL47 at 150^{o}C.

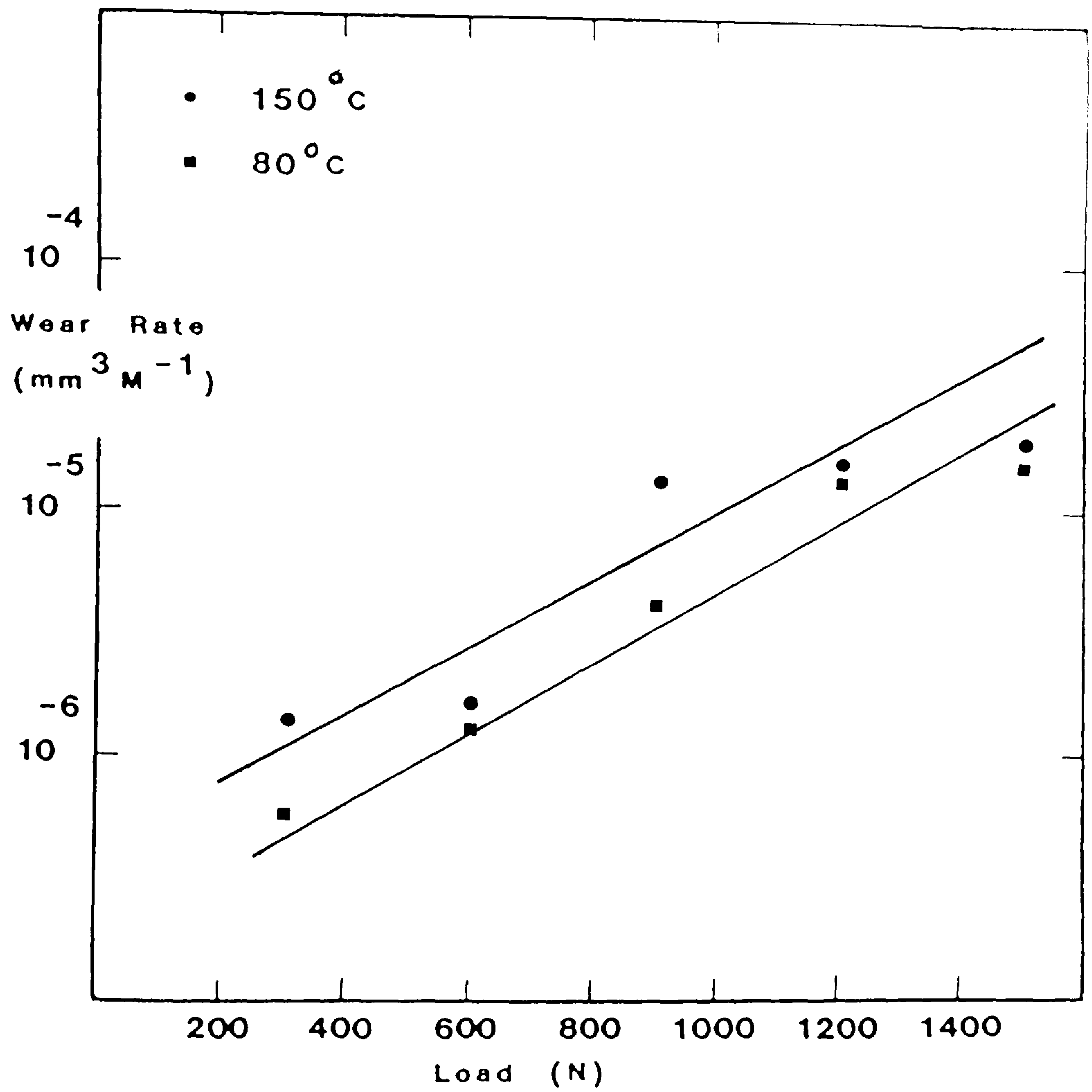


Fig.39 Equilibrium wear rates for RL47.

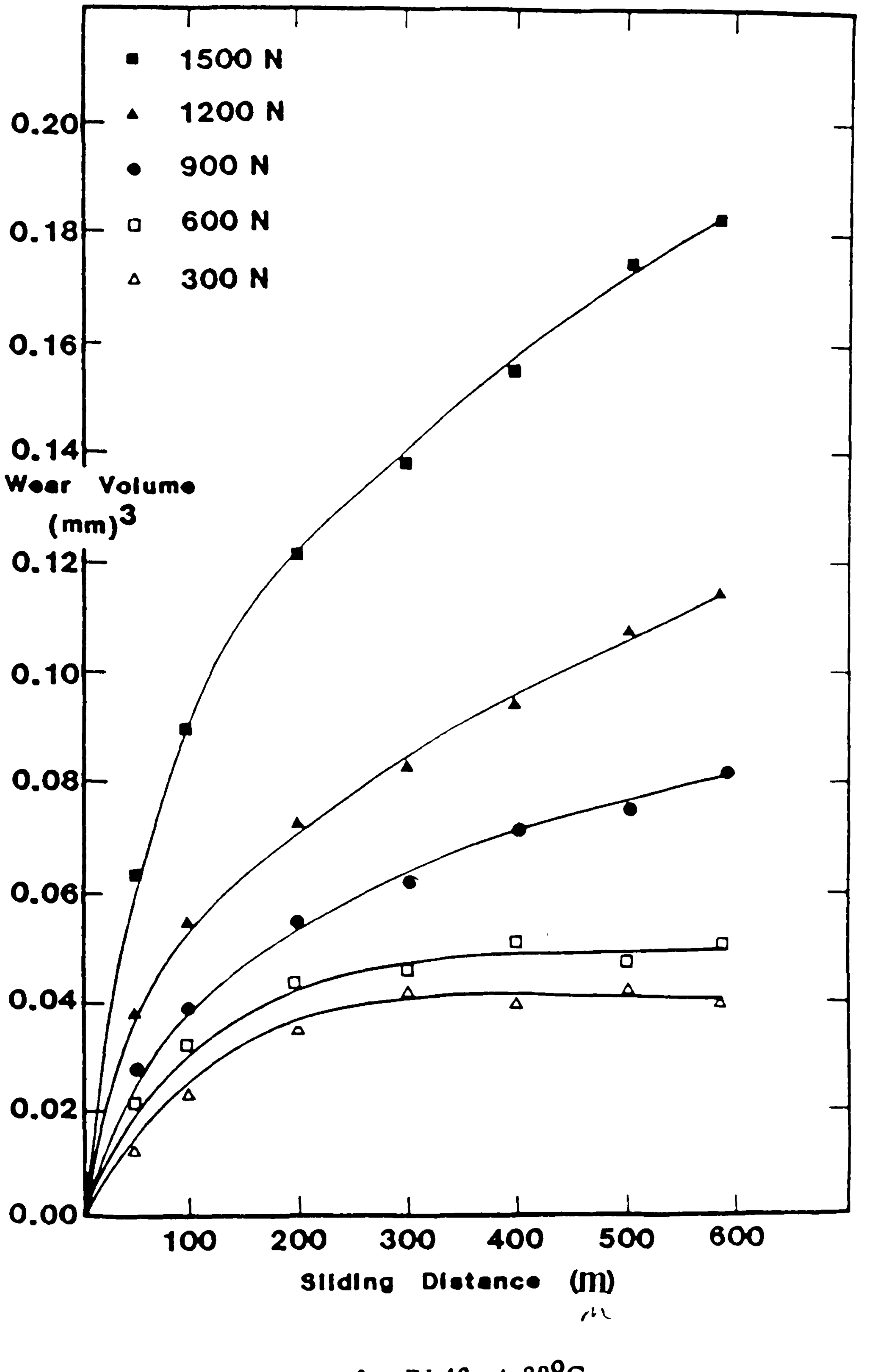


Fig.40 Wear for RL48 at $80^{o}C$.

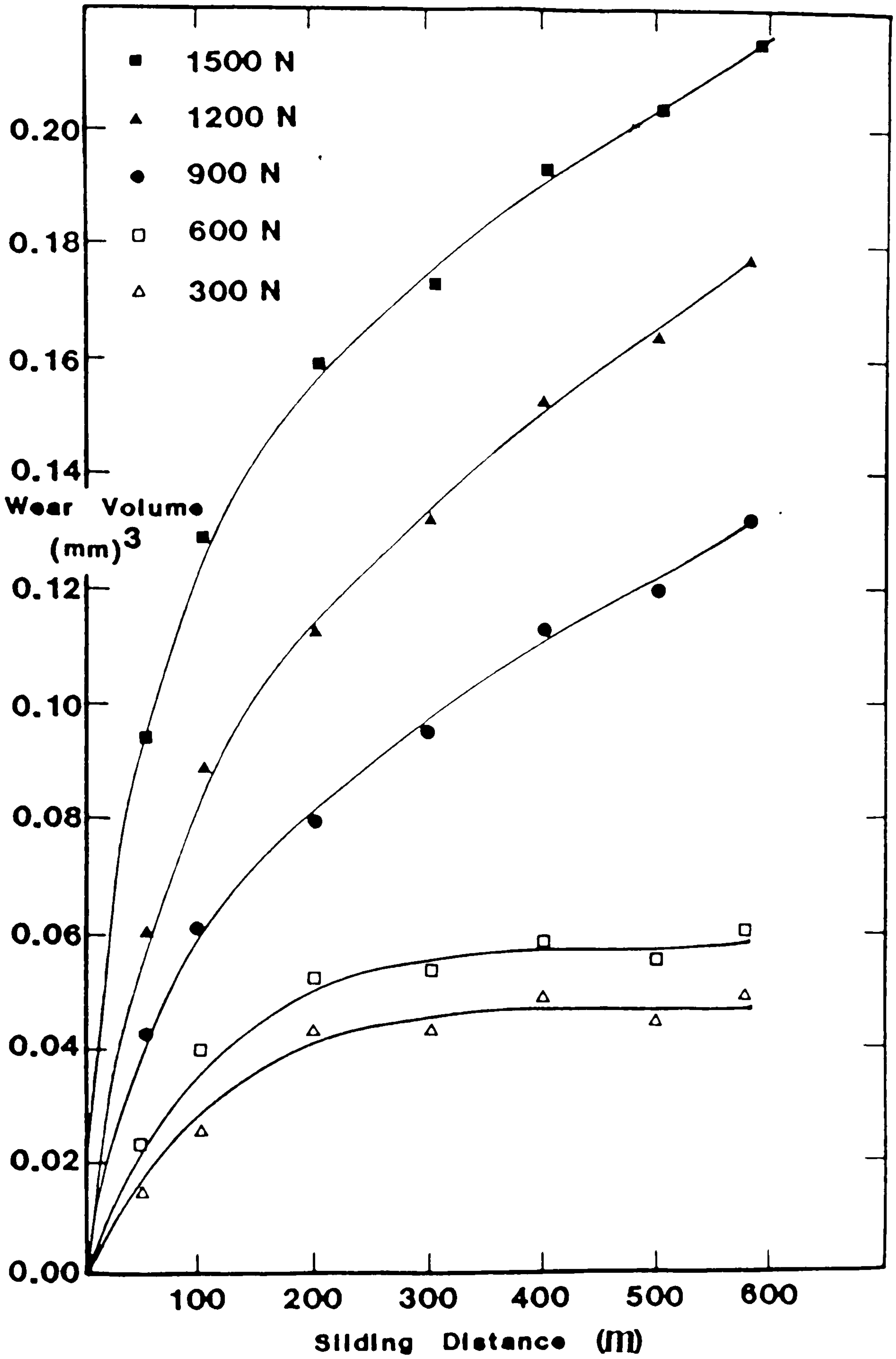


Fig.41 Wear for RL48 at 150^{o}C.

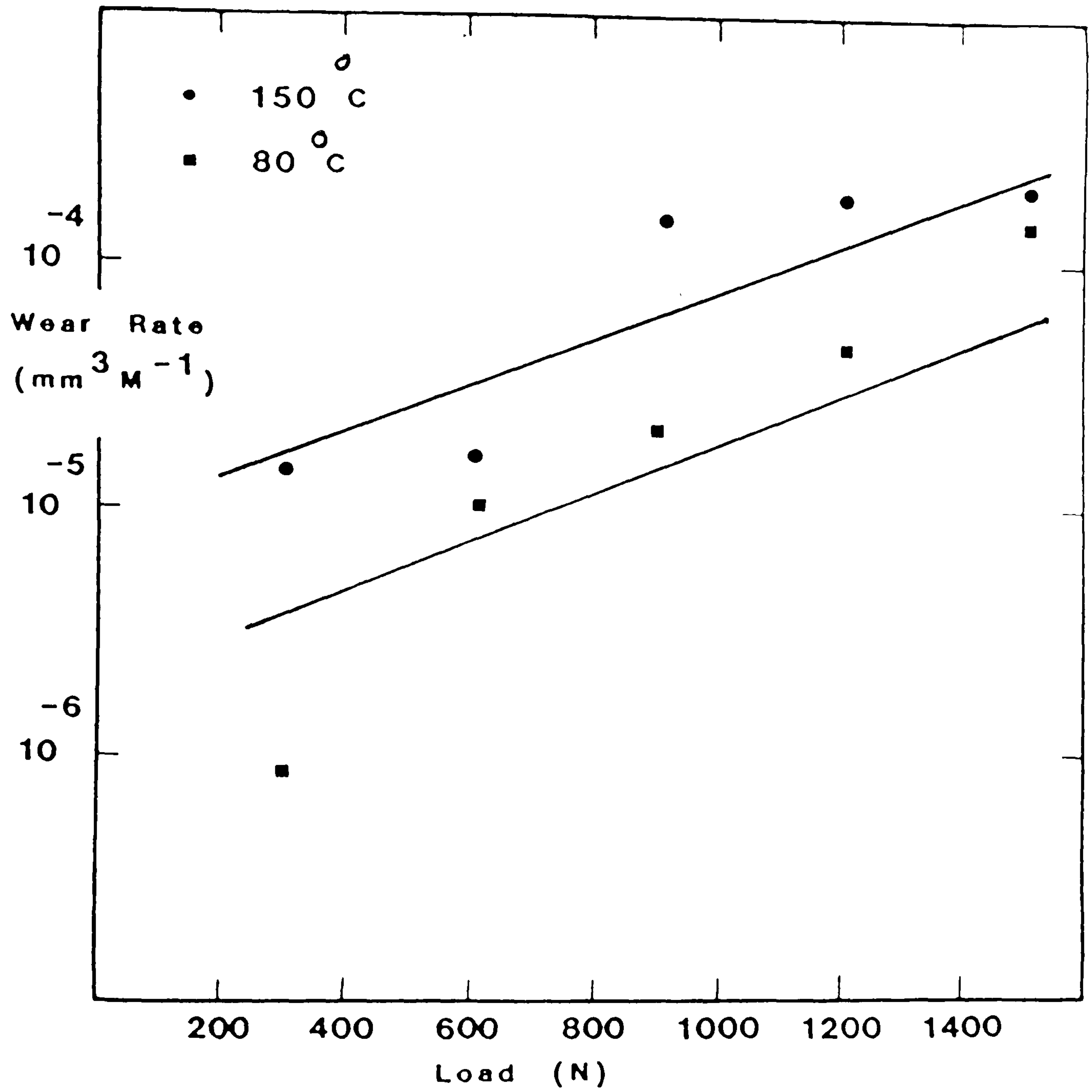


Fig.42 Equilibrium wear rates for RL48.

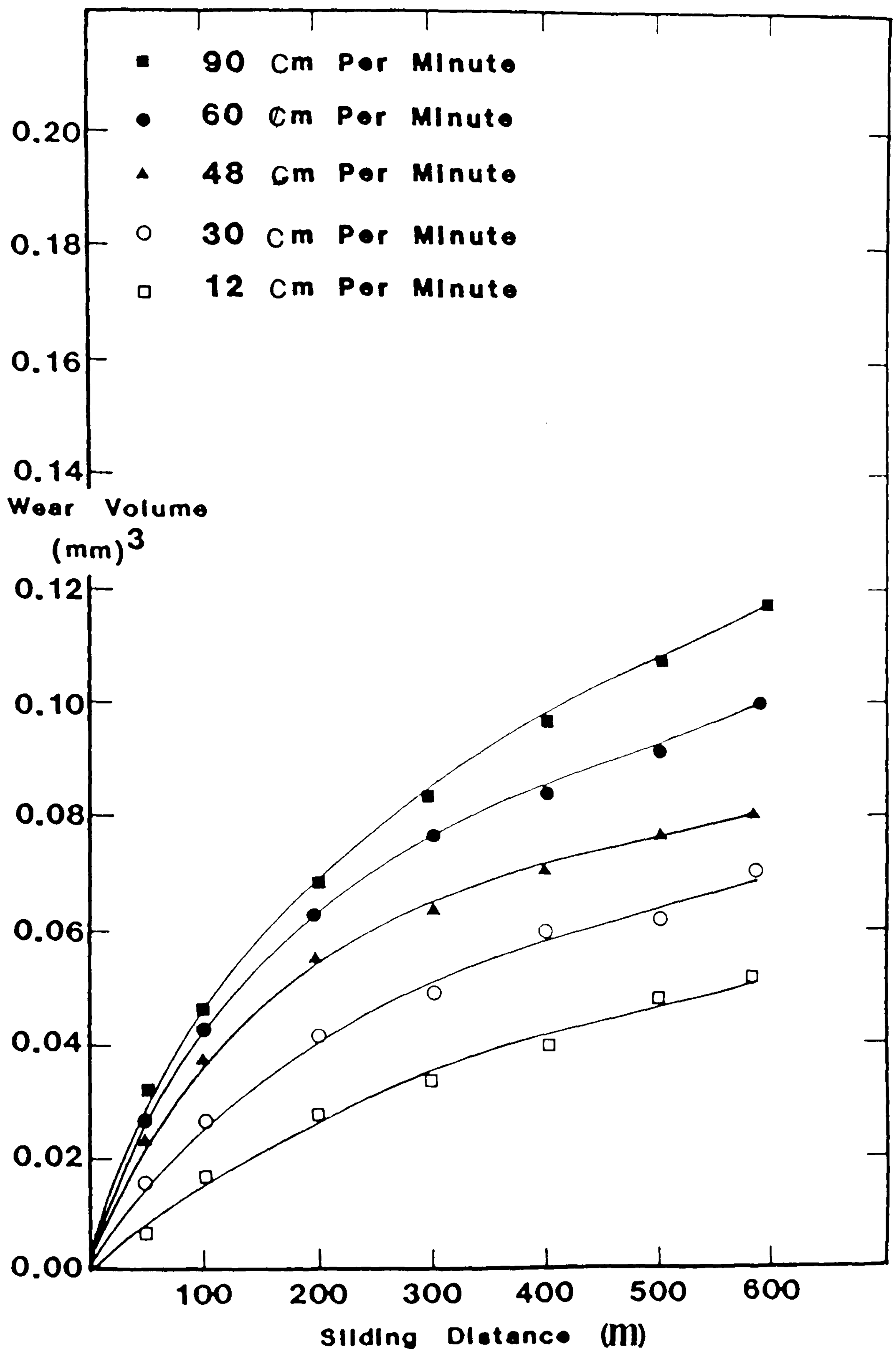


Fig.43 Wear for different speeds.

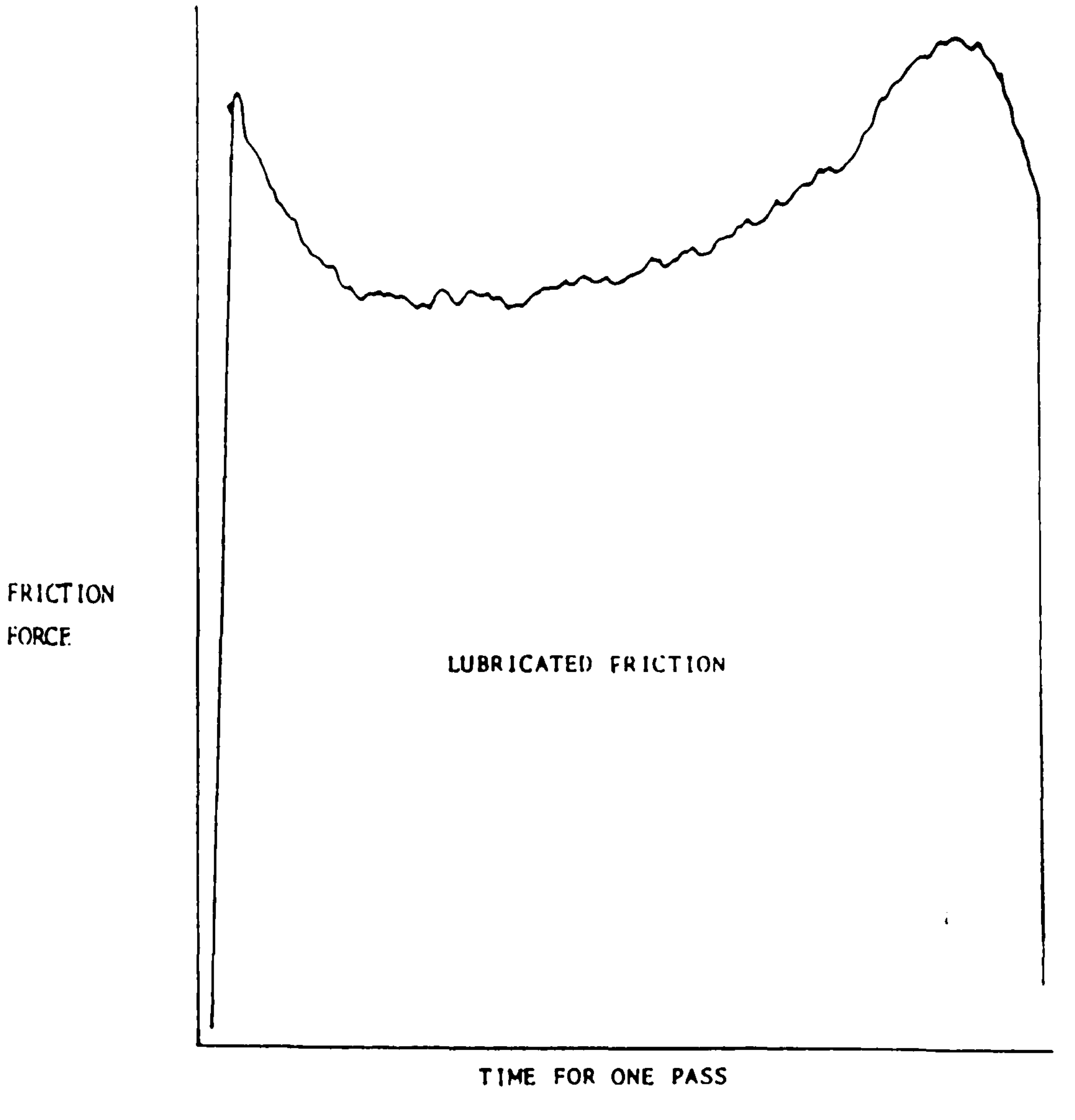


Fig.44 Osciloscope trace of friction force over one pass during equilibrium wear (load = 900N).

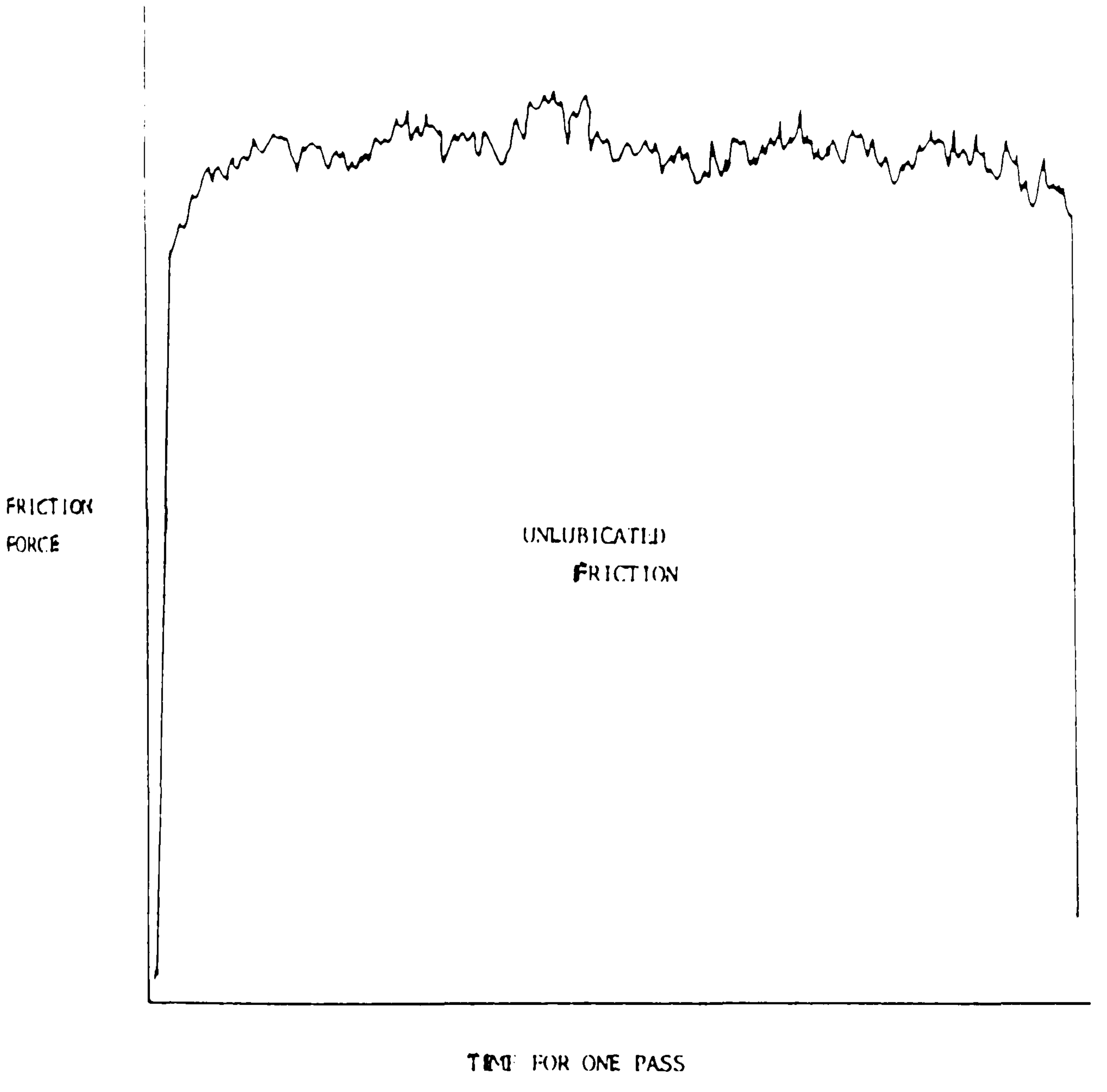


Fig.45 Oscilloscope trace of dry friction force over one pass during equilibrium wear (load = 10N).

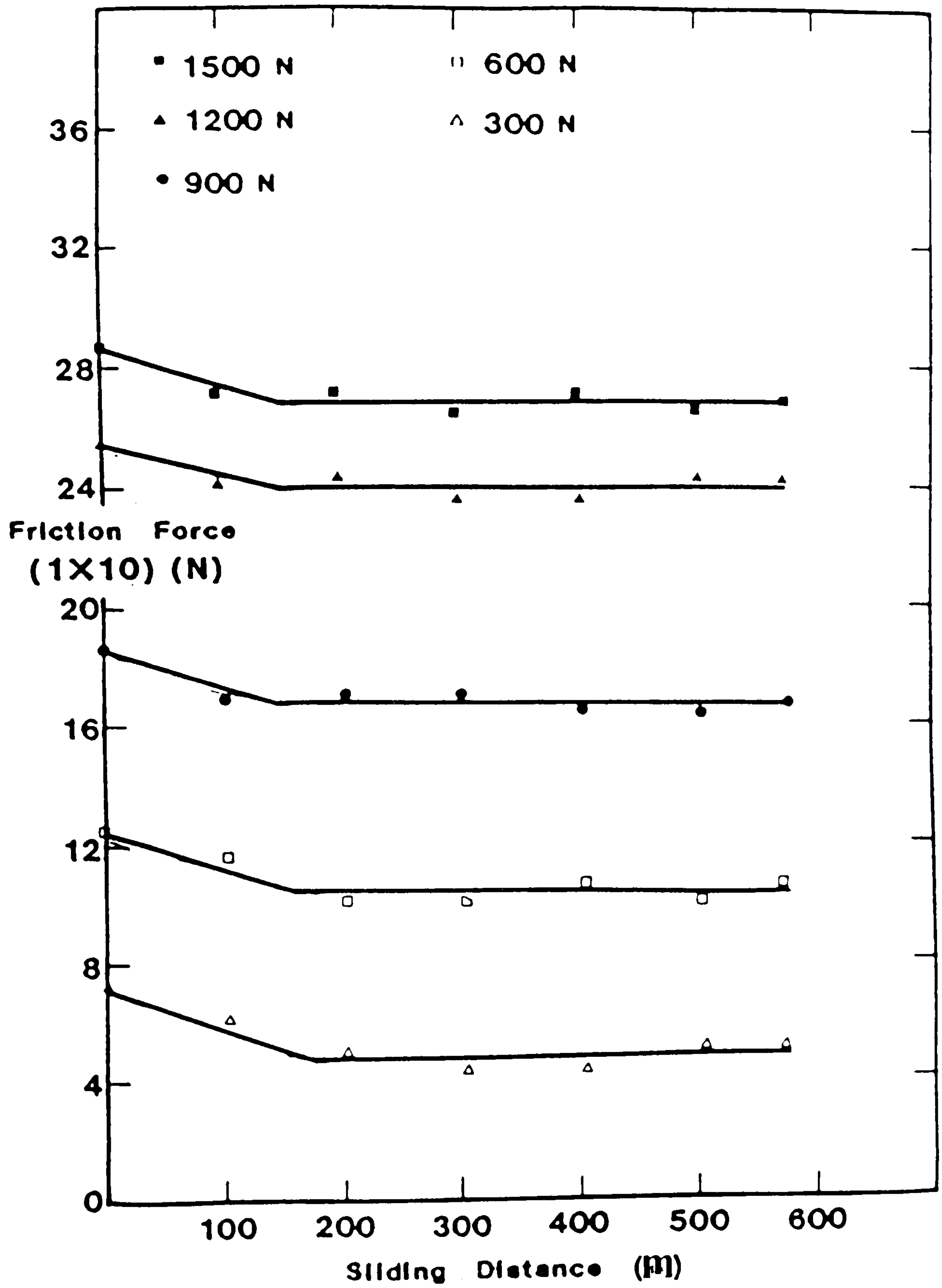


Fig.46 Friction force versus sliding distance for RL47 lubricant at 80°C.

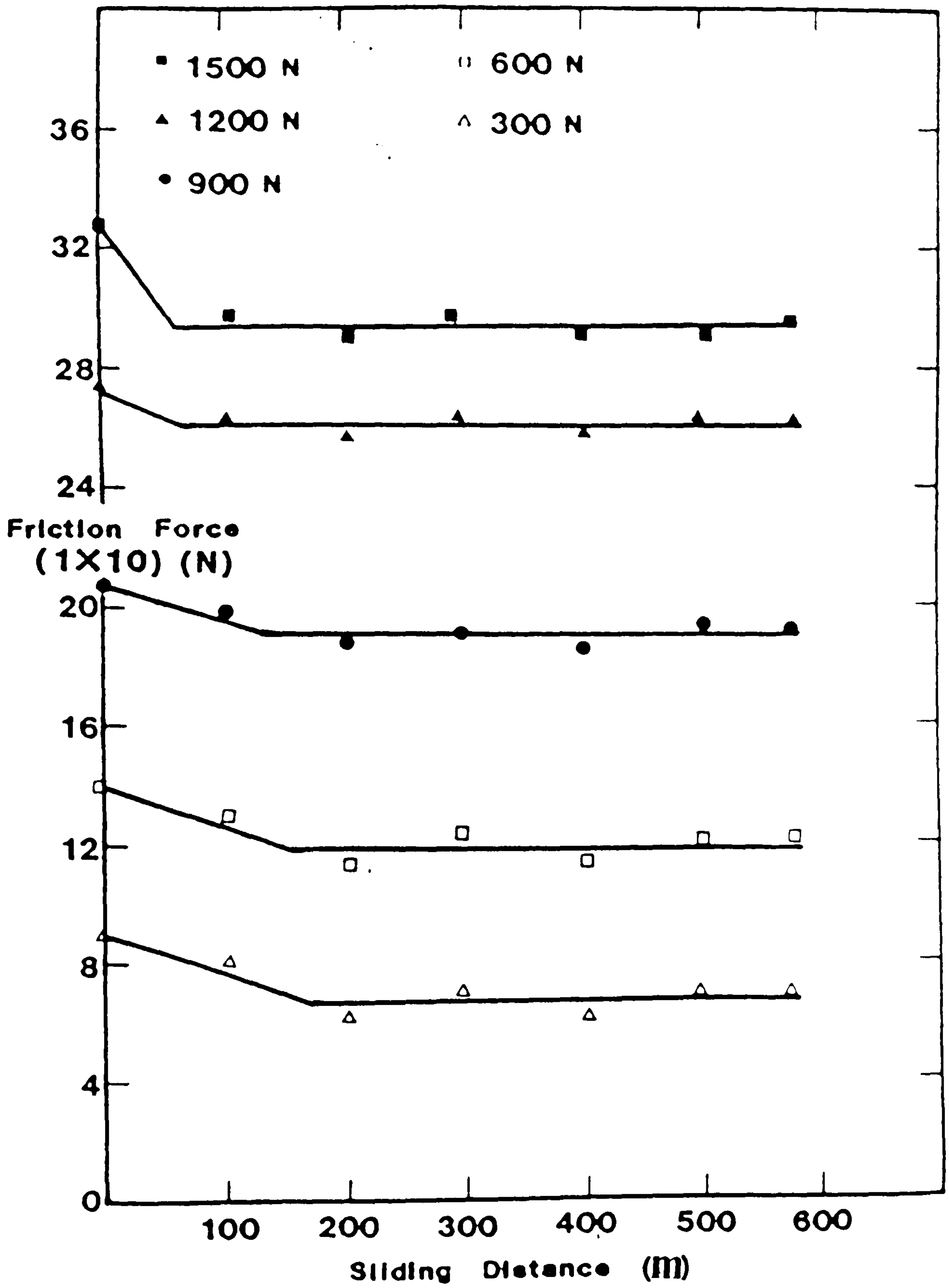


Fig.47 Friction force versus sliding distance for RL47 lubricant at 150°C.

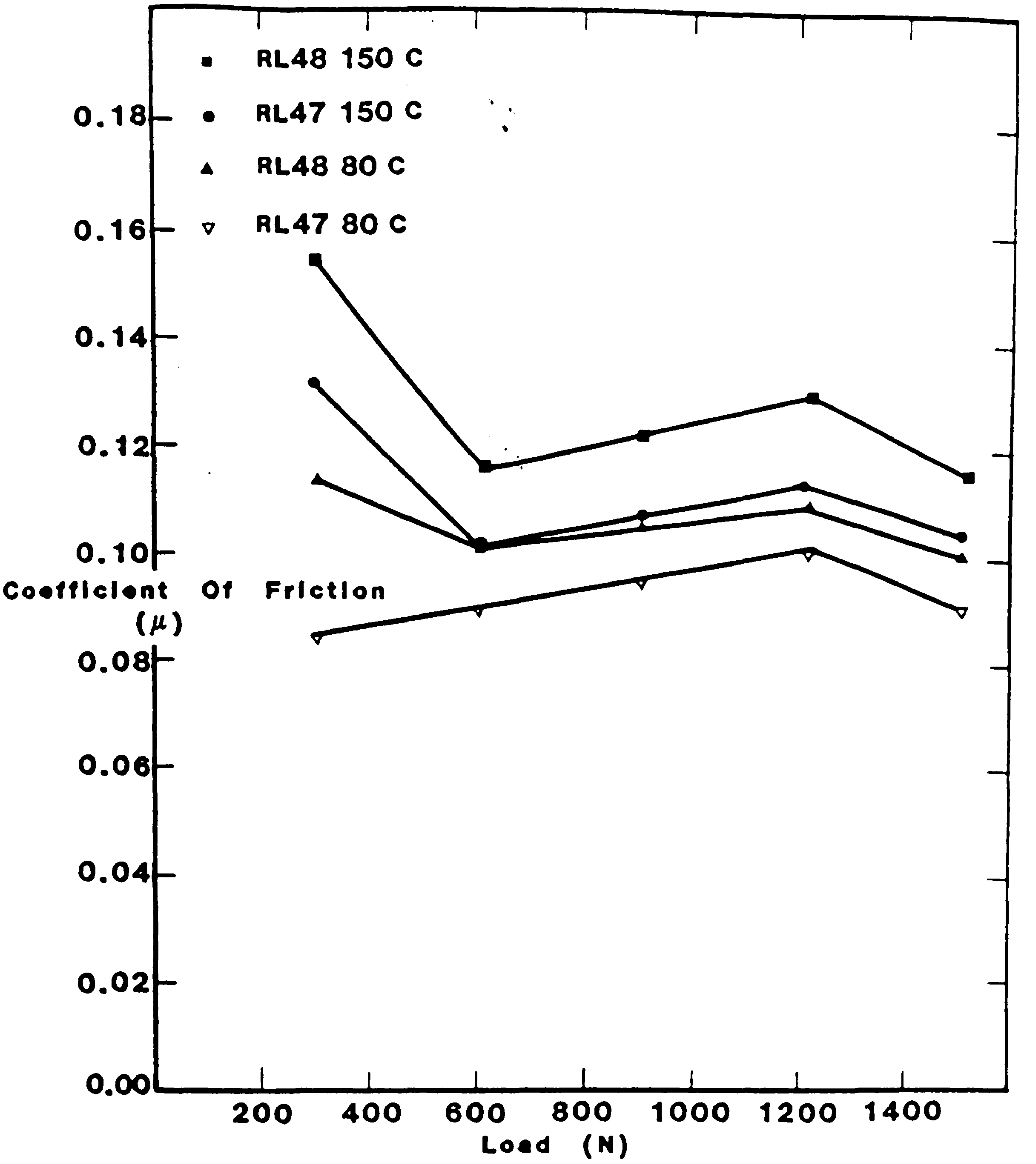


Fig.48 Coefficient of friction for RL47 and RL48.

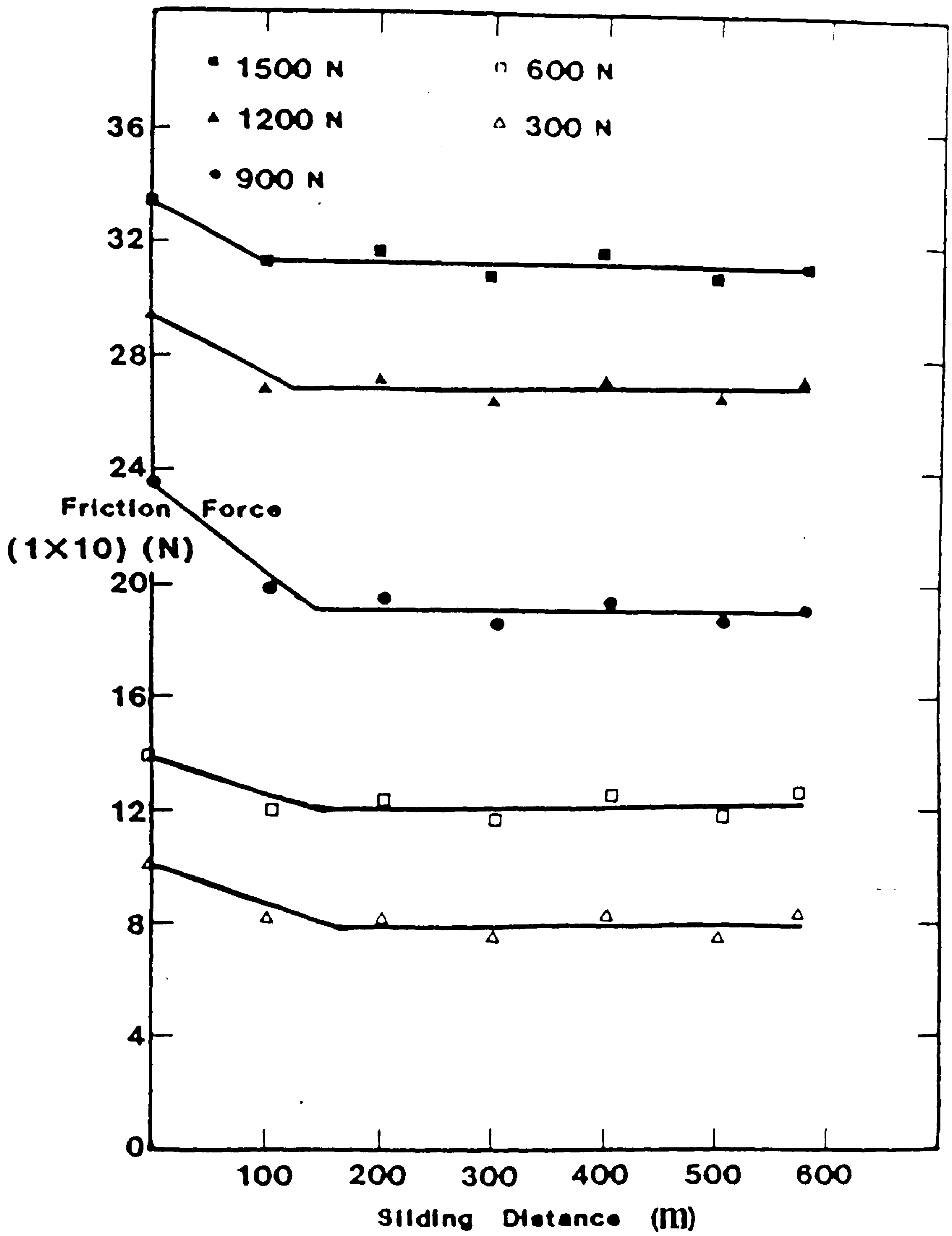


Fig.49 Friction force versus sliding distance for RL48 at 80°C.

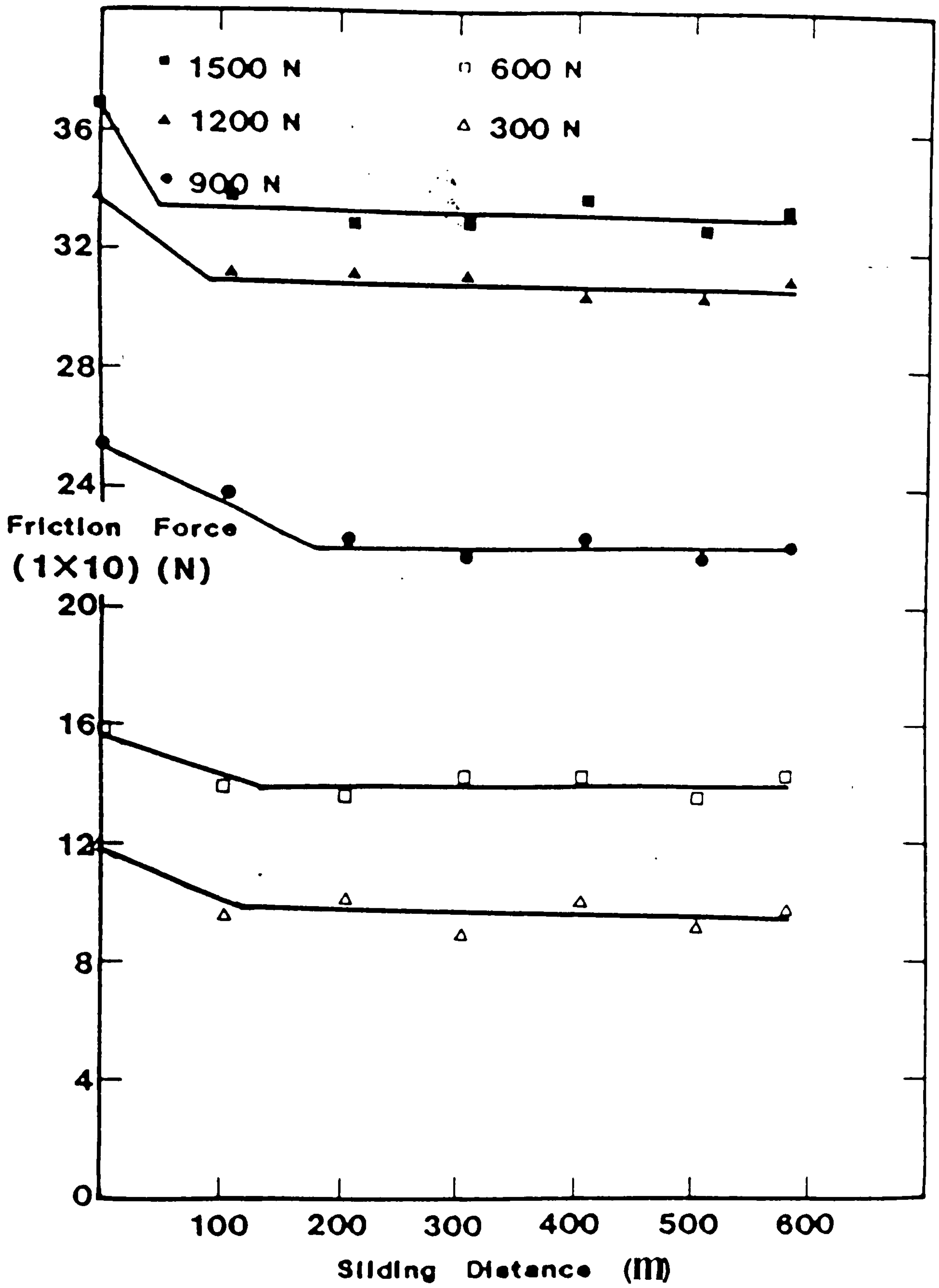


Fig.50 Friction force versus sliding for RL48 at 150°C.

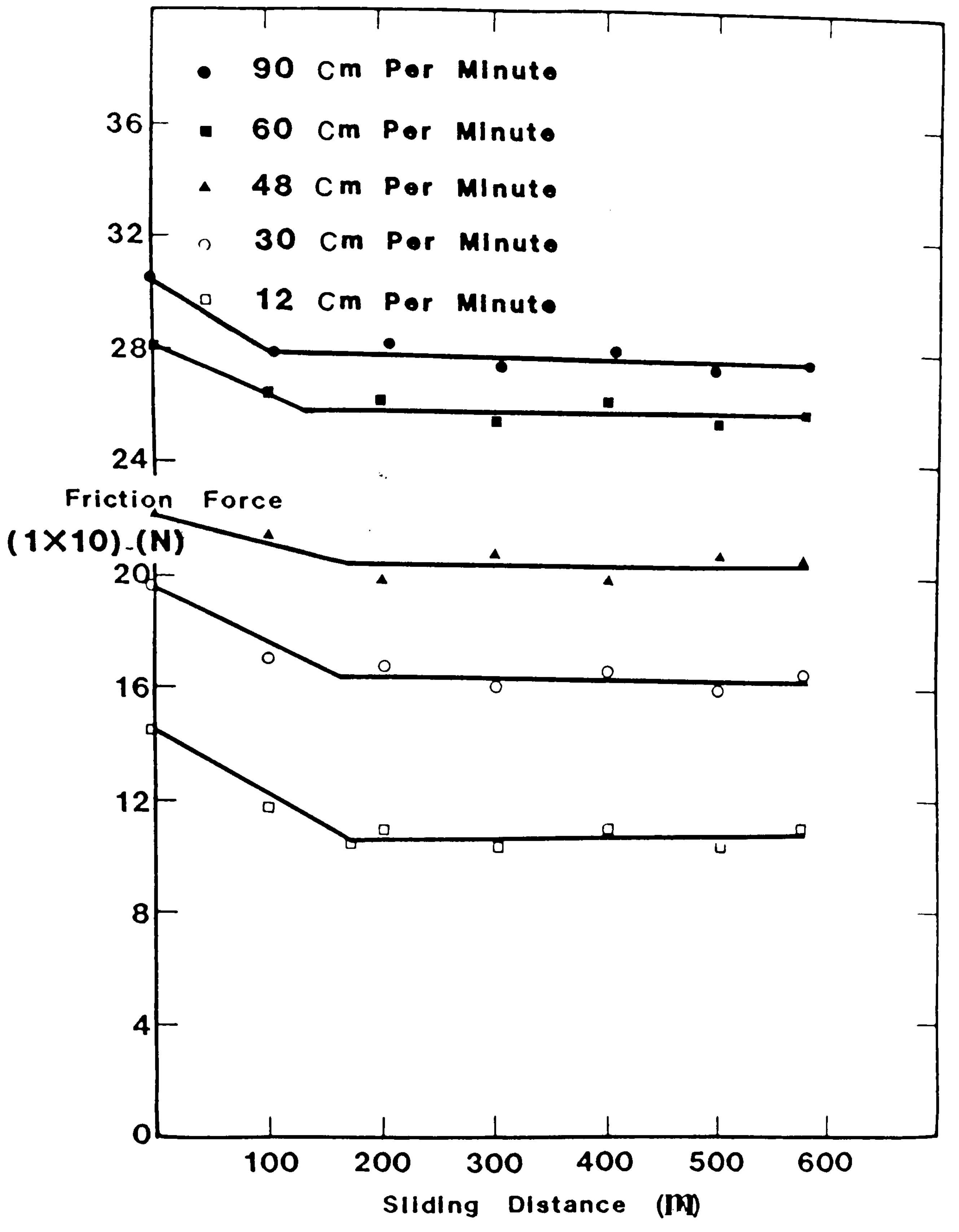


Fig.51 Friction force versus sliding distance for RL48 at 80°C, at different speeds, (load = 900N)

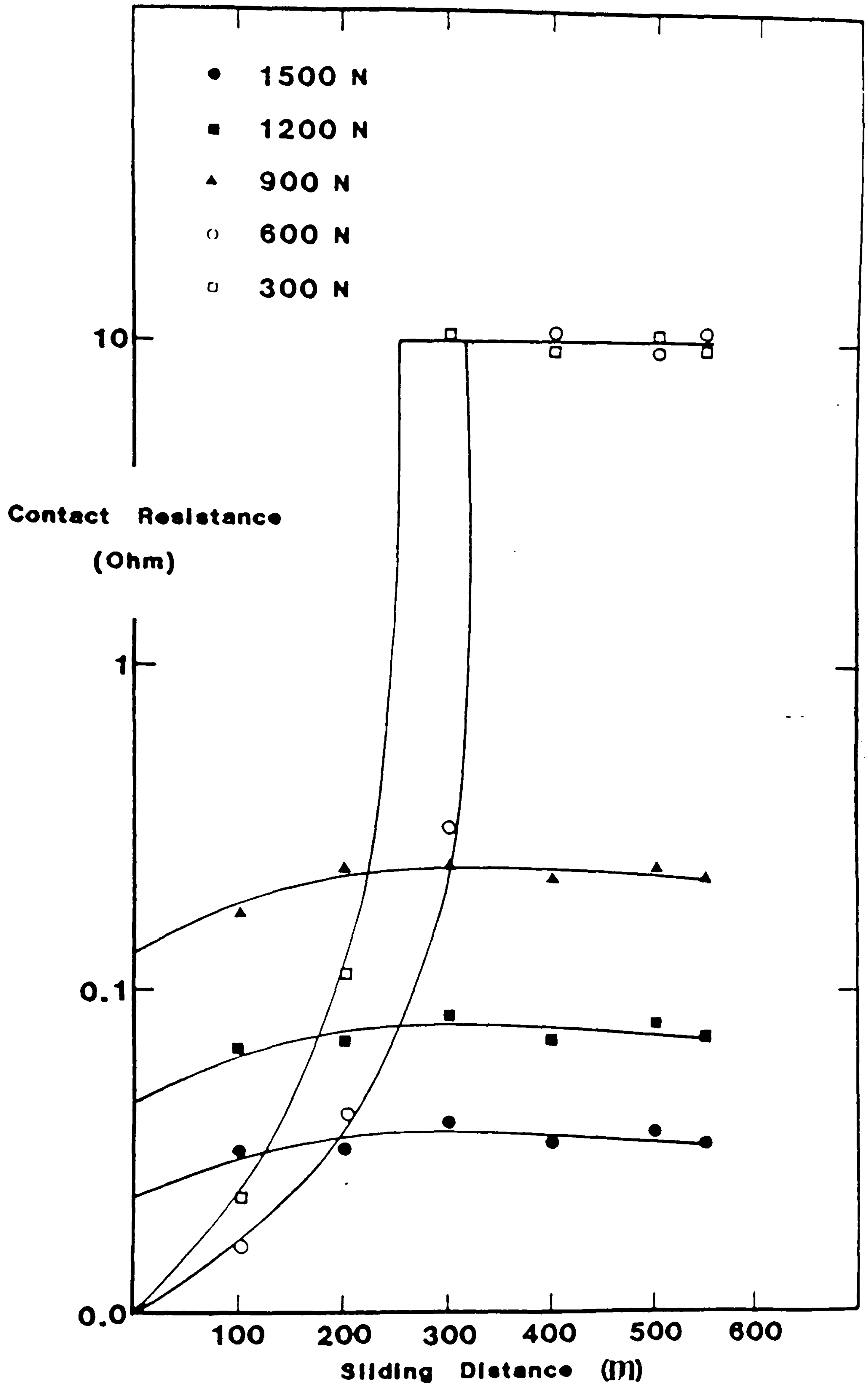


Fig.52 Contact resistance versus sliding distance for RL47 at 80^{o}C

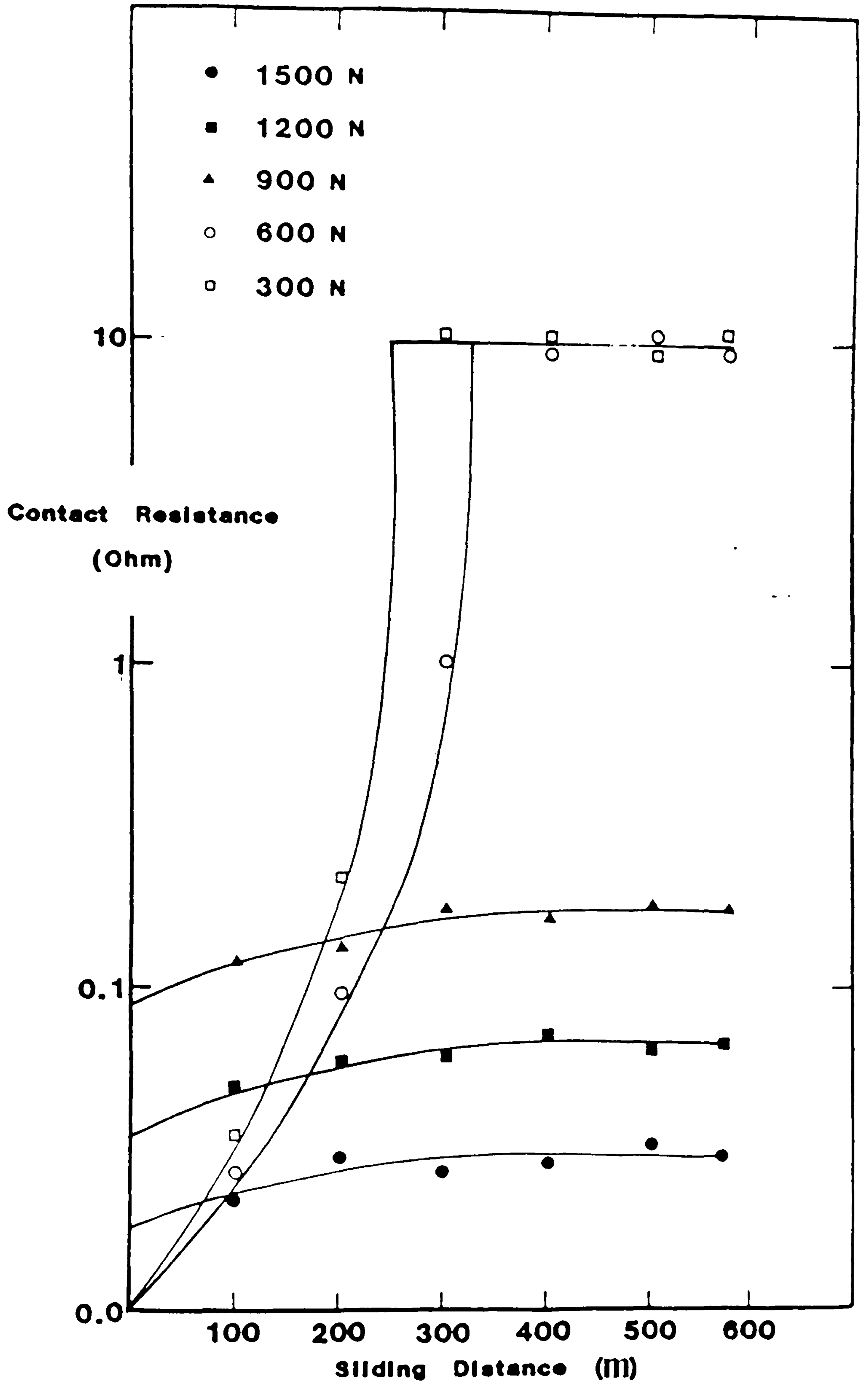


Fig.53 Contact resistance versus sliding distance for RL47 at 150°.

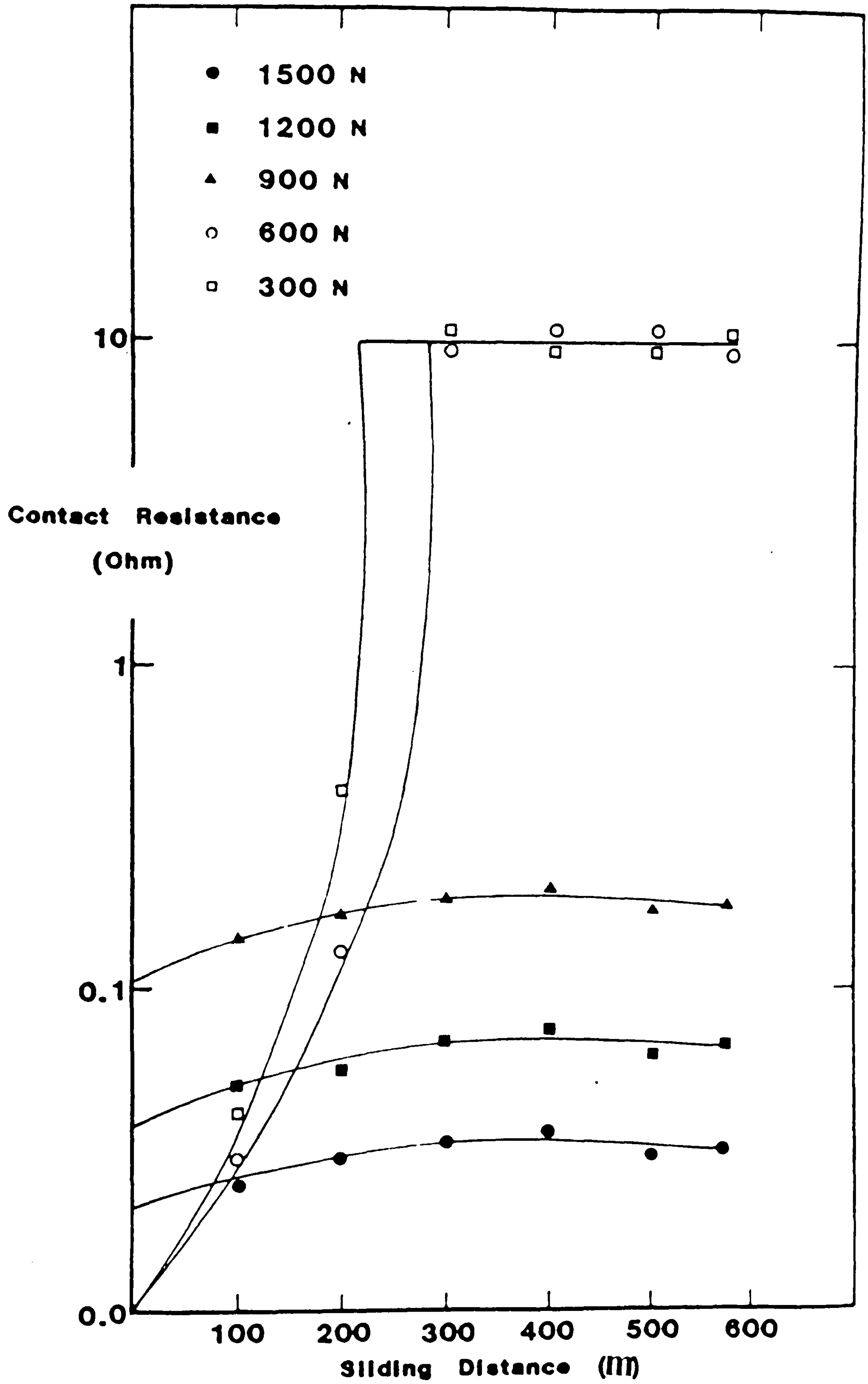


Fig.54 Contact resistance versus sliding distance for RL48 at 80°C

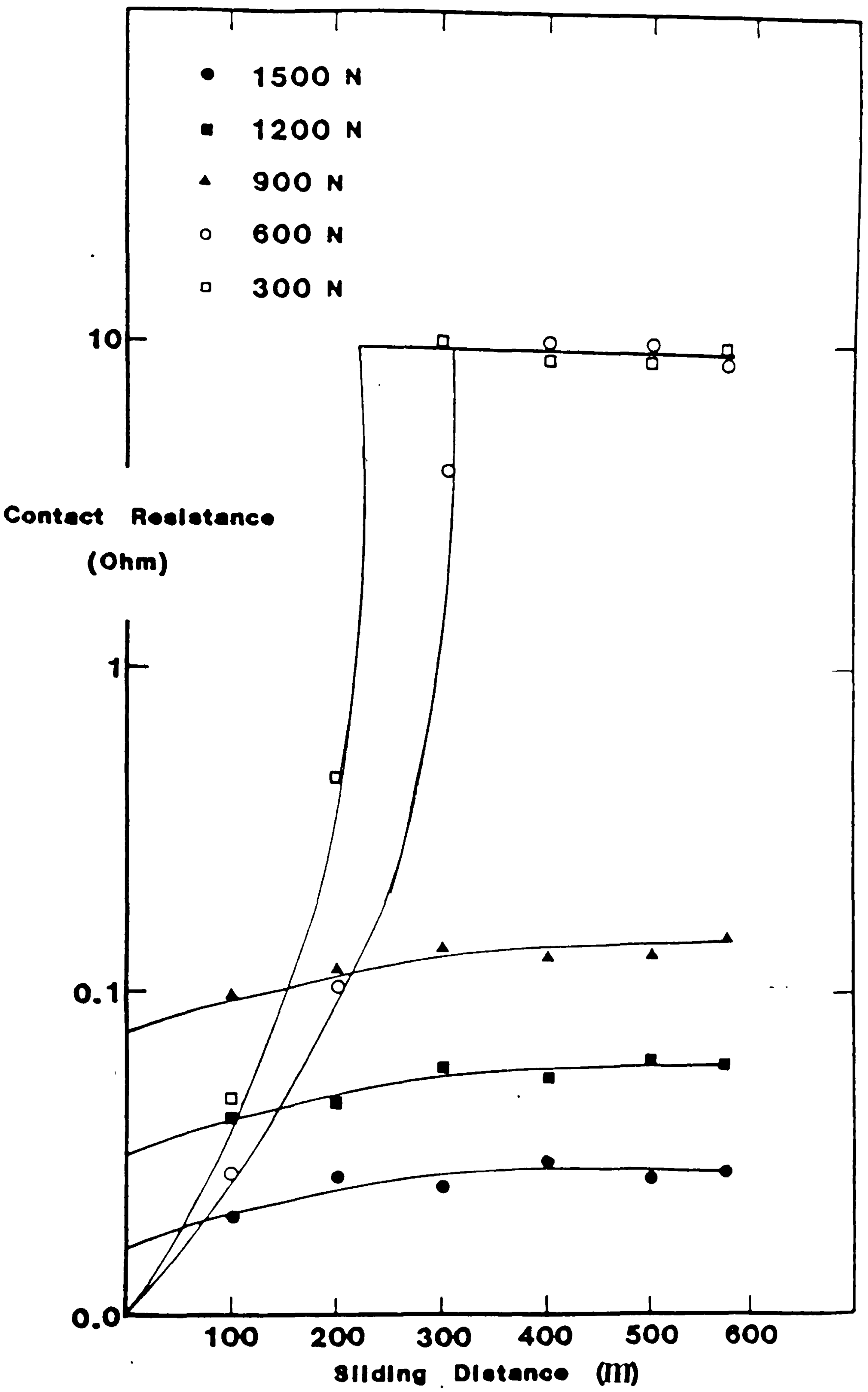


Fig.55 Contact resistance versus sliding distance for RL48 at 150°C

Direction of sliding.

For all of the following photographs, wear direction is marked with an arrow as above.

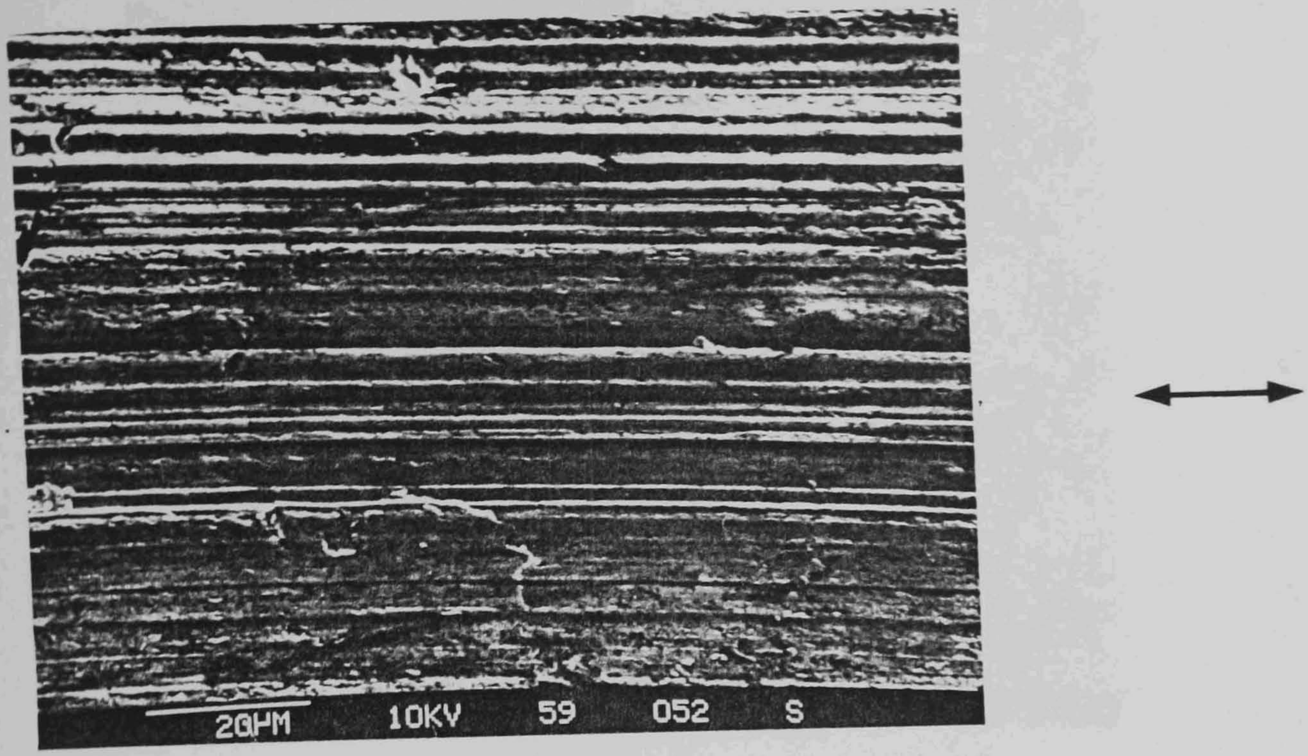


Fig.56 SEM appearance of abrasive wear for worn flat using RL47.

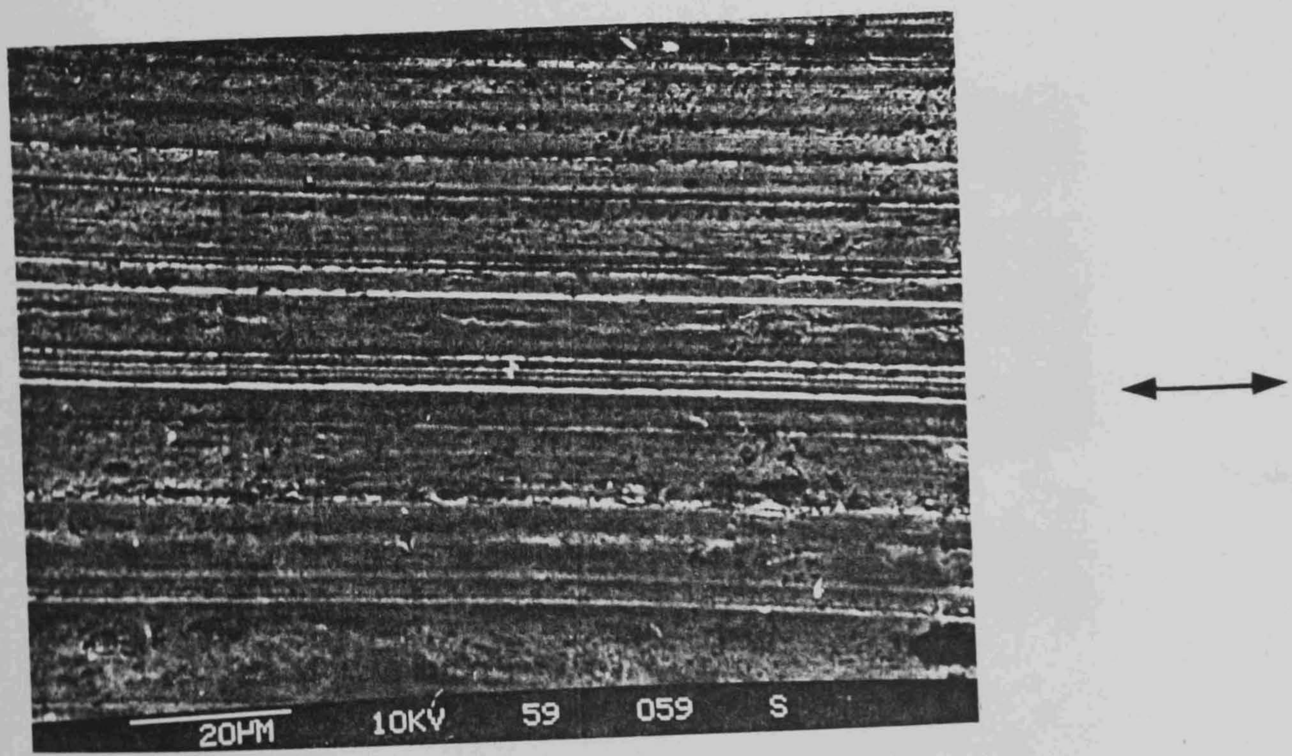


Fig.57 SEM appearance of abrasive wear for worn flat using RL48.

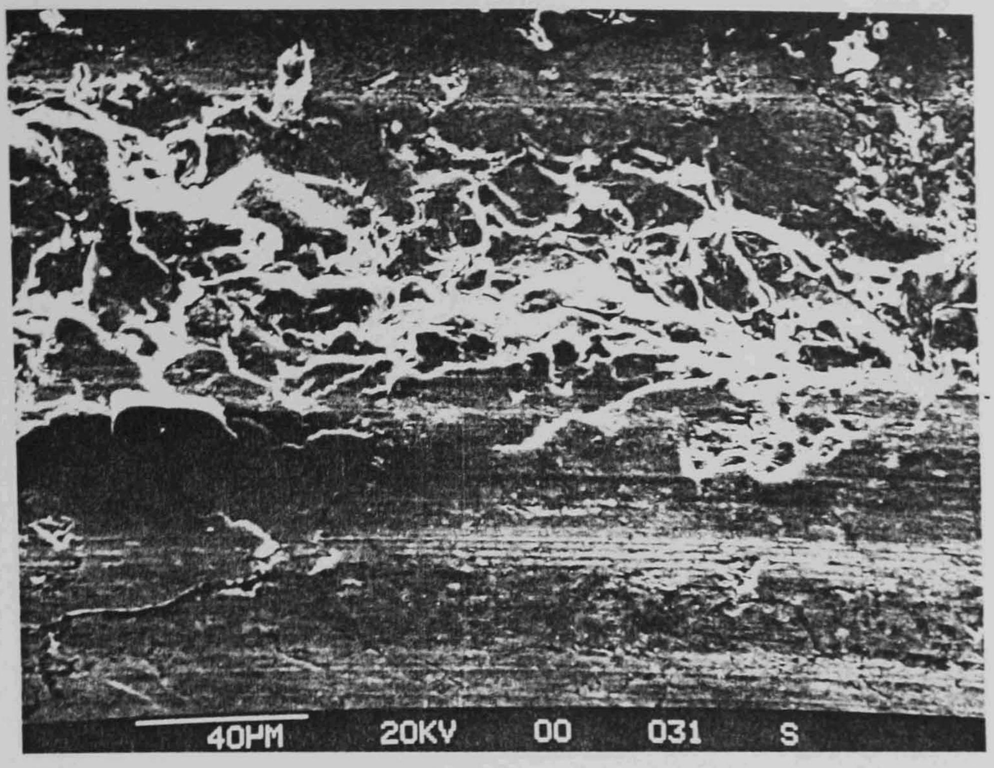


Fig.58 Delamination wear for worn flat using RL47.

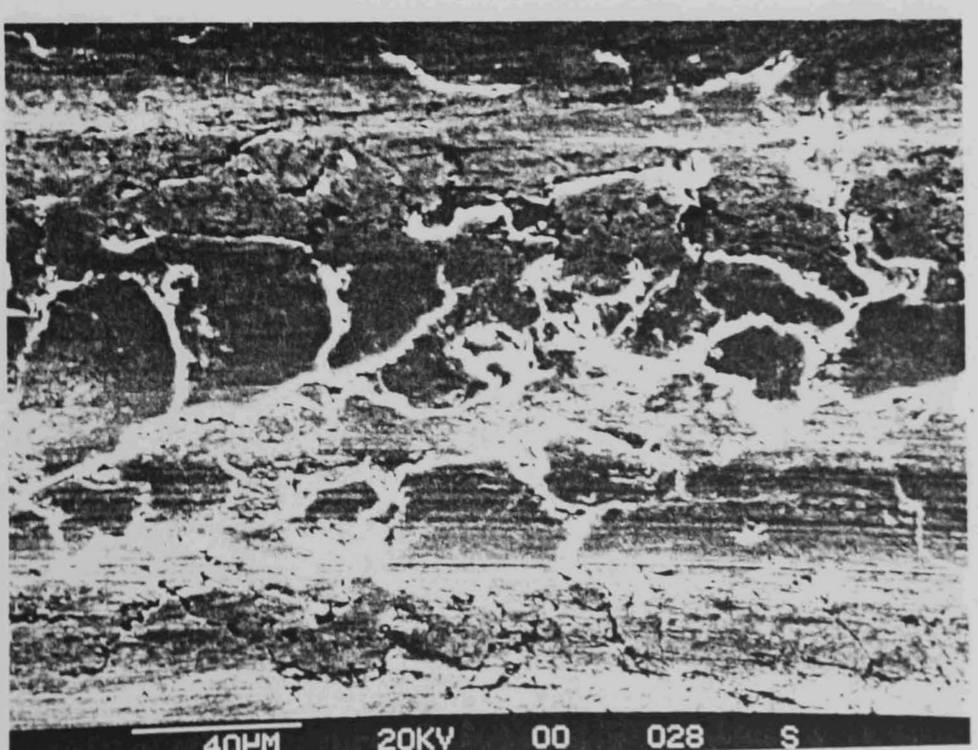


Fig.59 Delamination wear for worn flat using RL48.

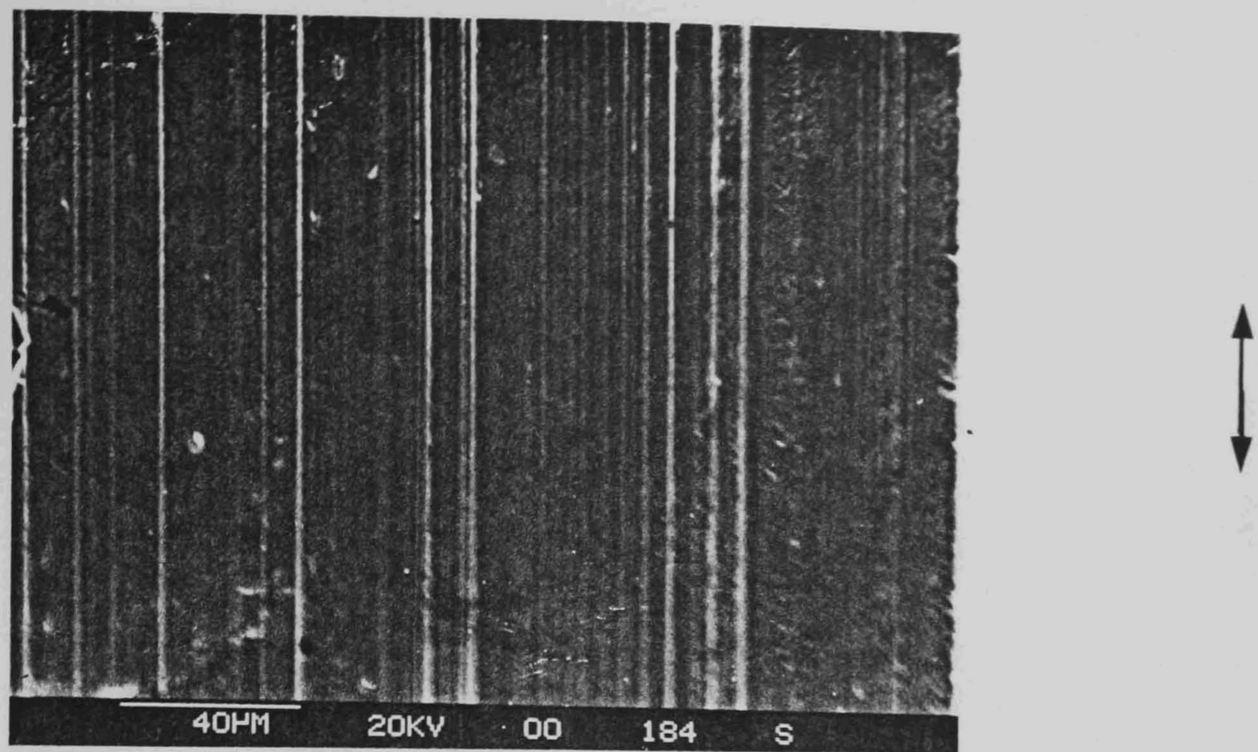


Fig.60 Abrasive wear for worn pin using RL47.

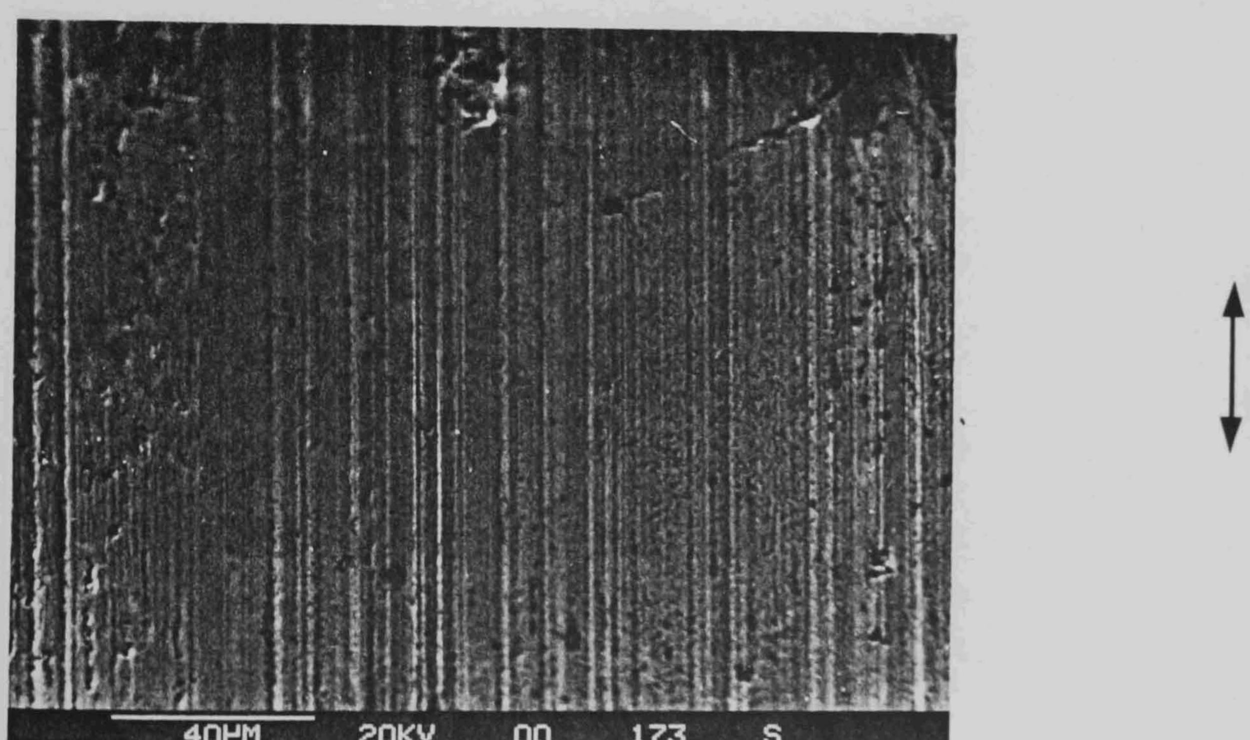


Fig.61 Abrasive wear for worn pin using RL48

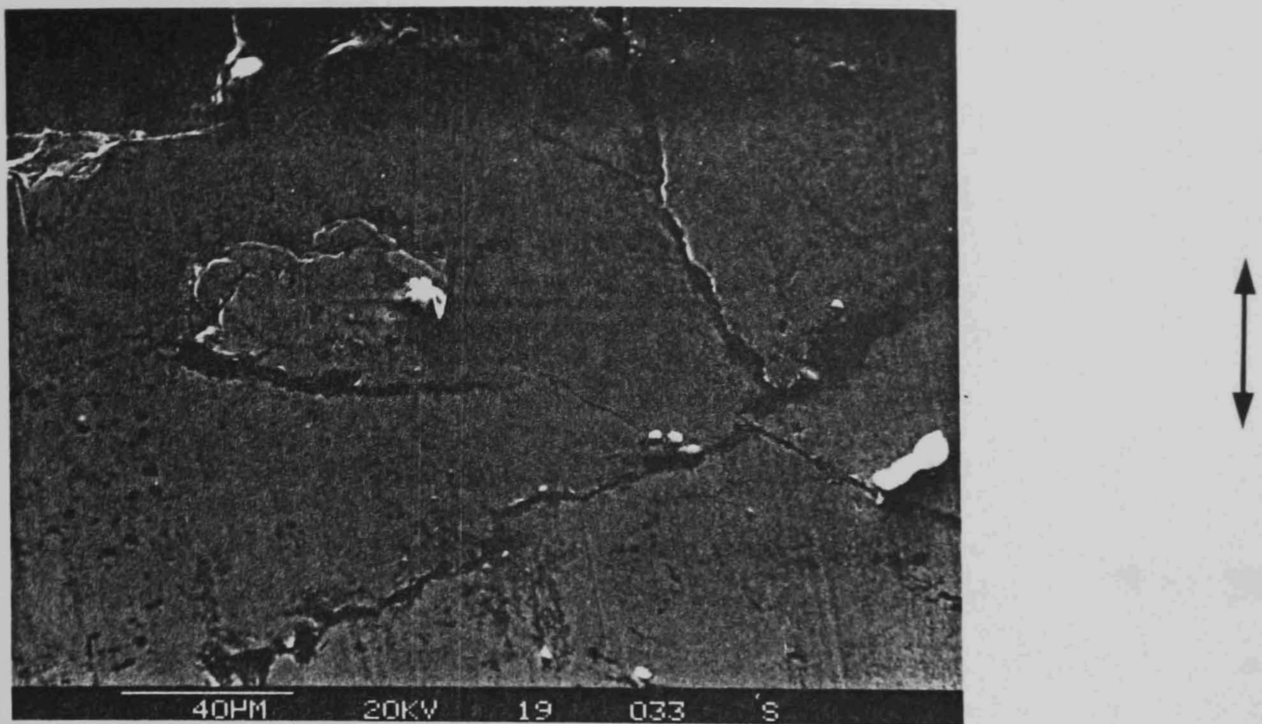


Fig.62 Delamination wear for worn pin using RL47

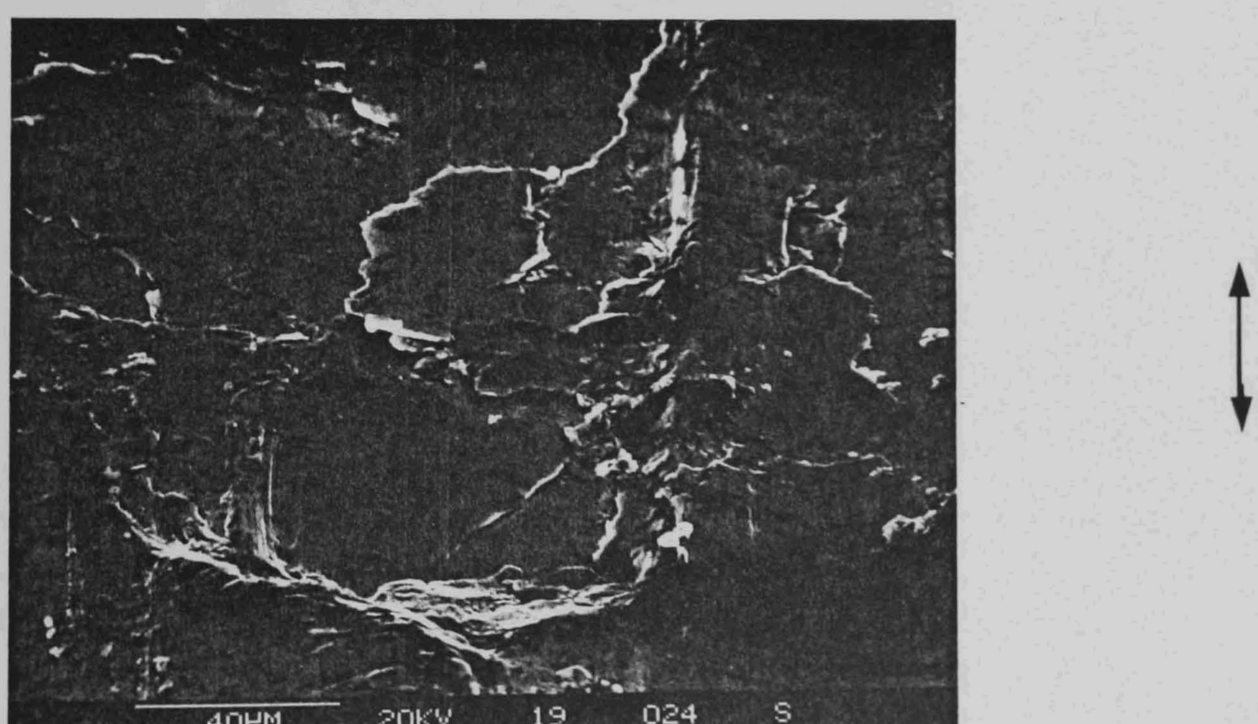


Fig.63 Delamination wear for worn pin using RL48.

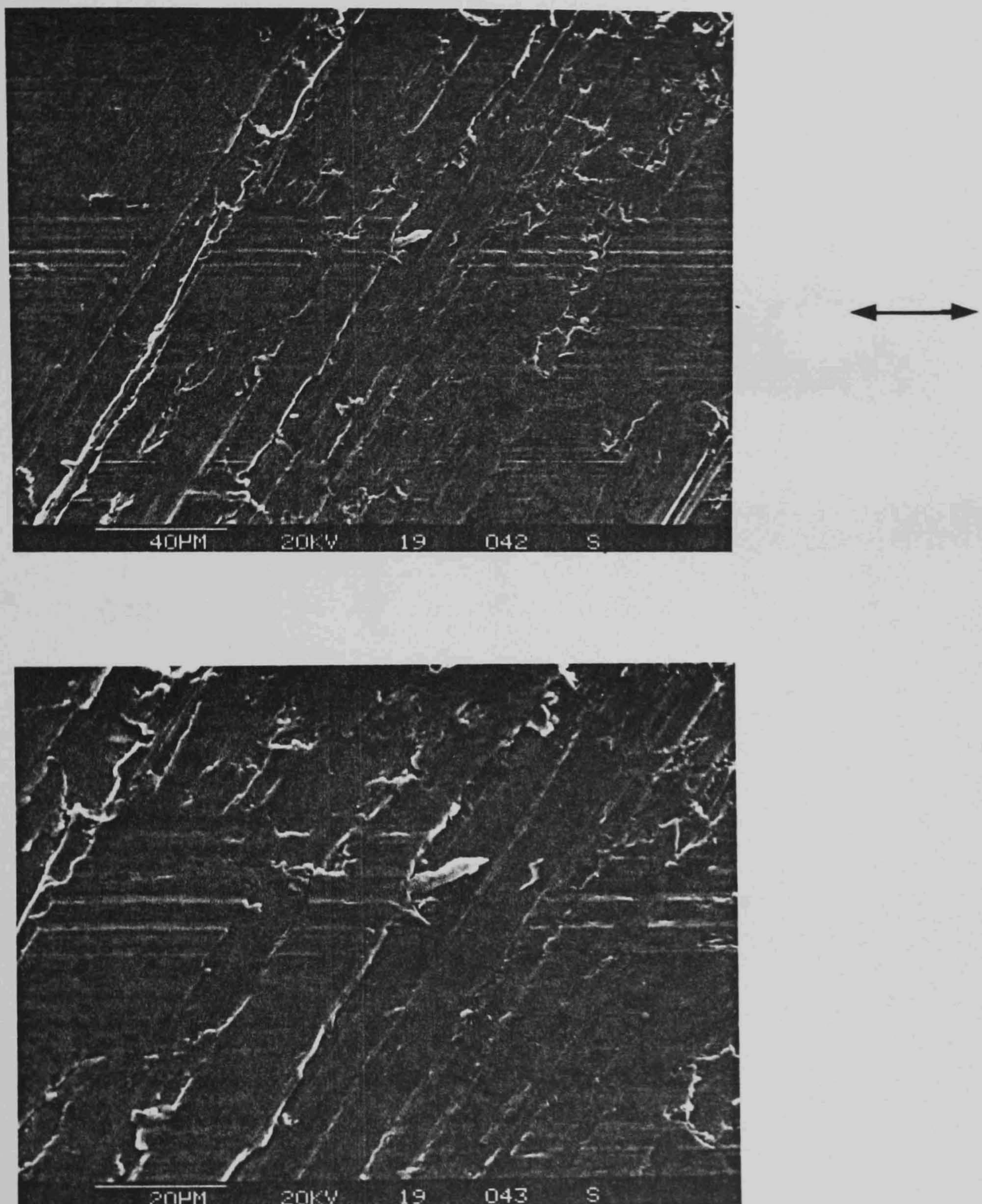


Fig.64 SEM appearance for worn flat at low loads (300N).

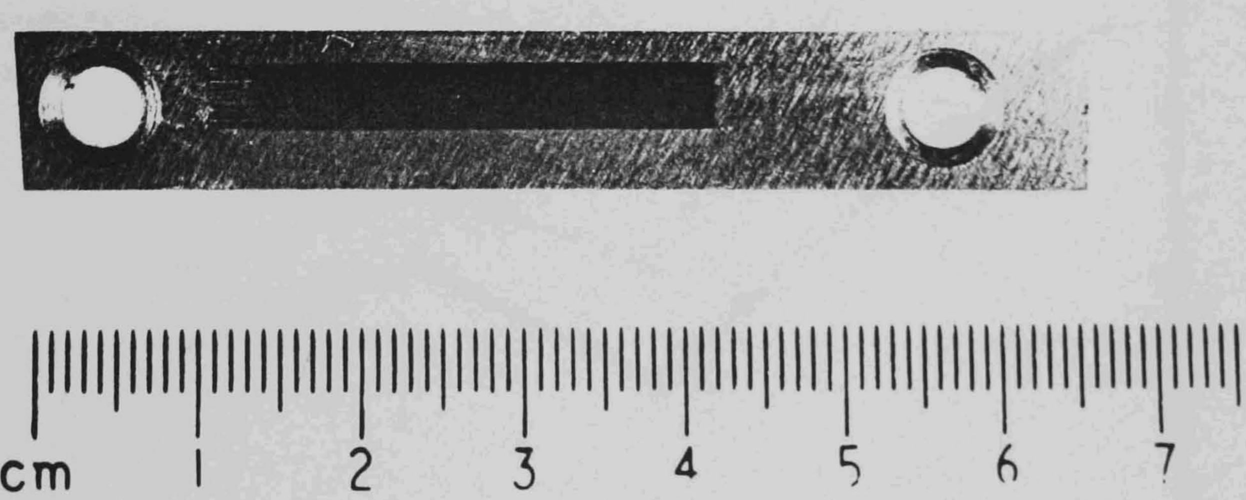


Fig.65 The wear track of the worn flat.

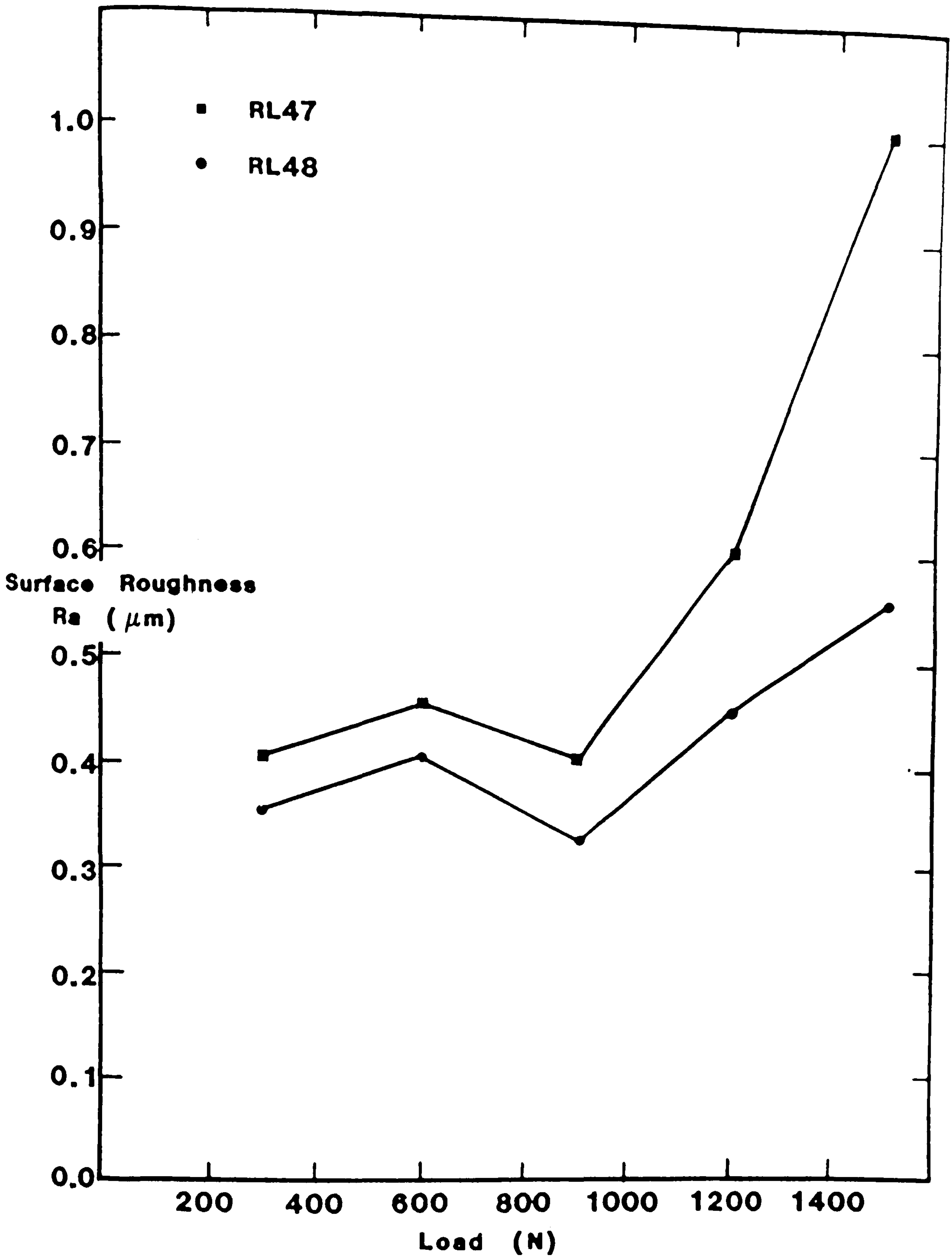


Fig.66 Surface roughness measurement (Ra) for the flats, for both oils, at 80°C.

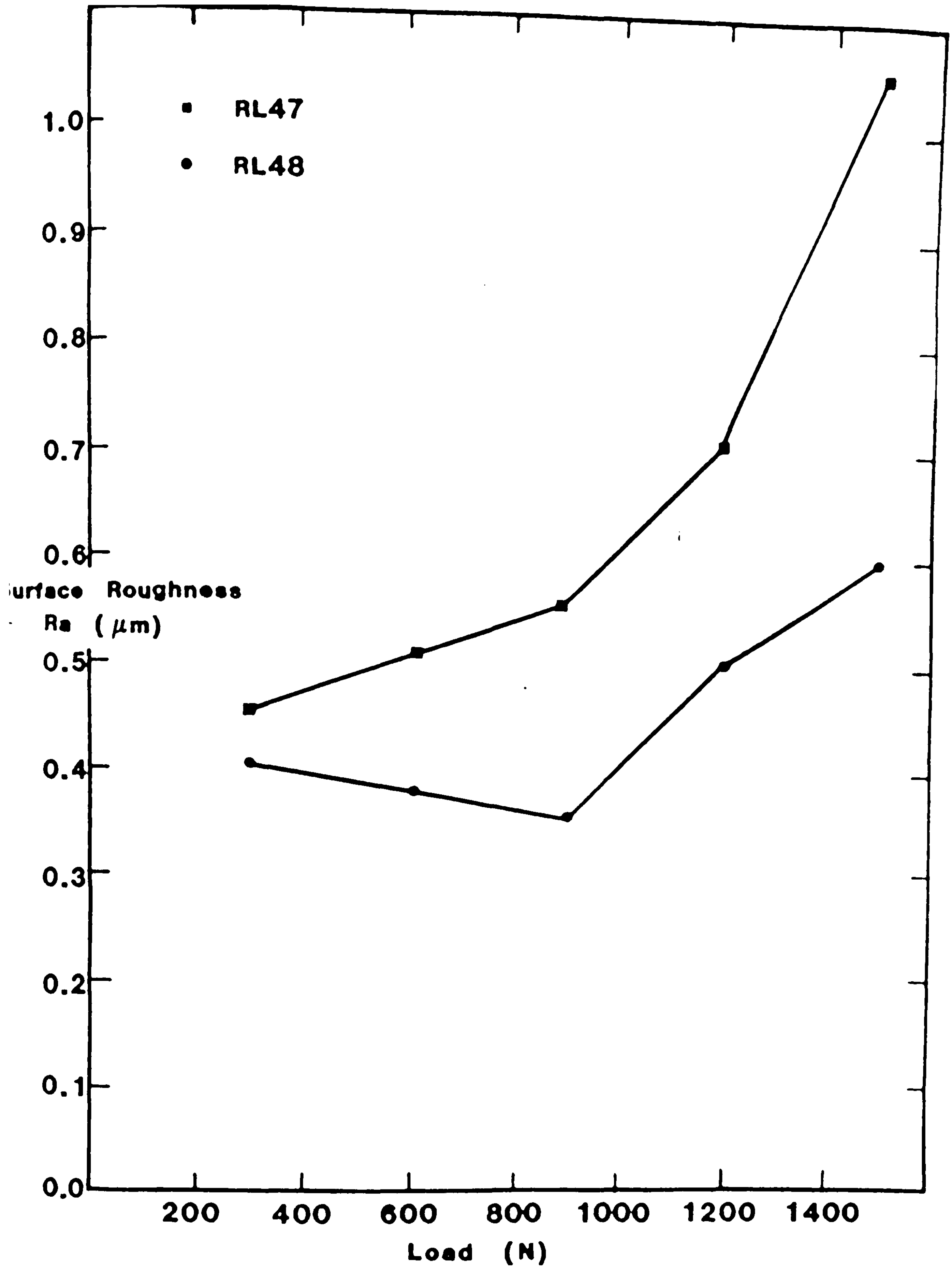


Fig.67 Surface roughness measurement (Ra) for the flats, for both oils, at 150°C.

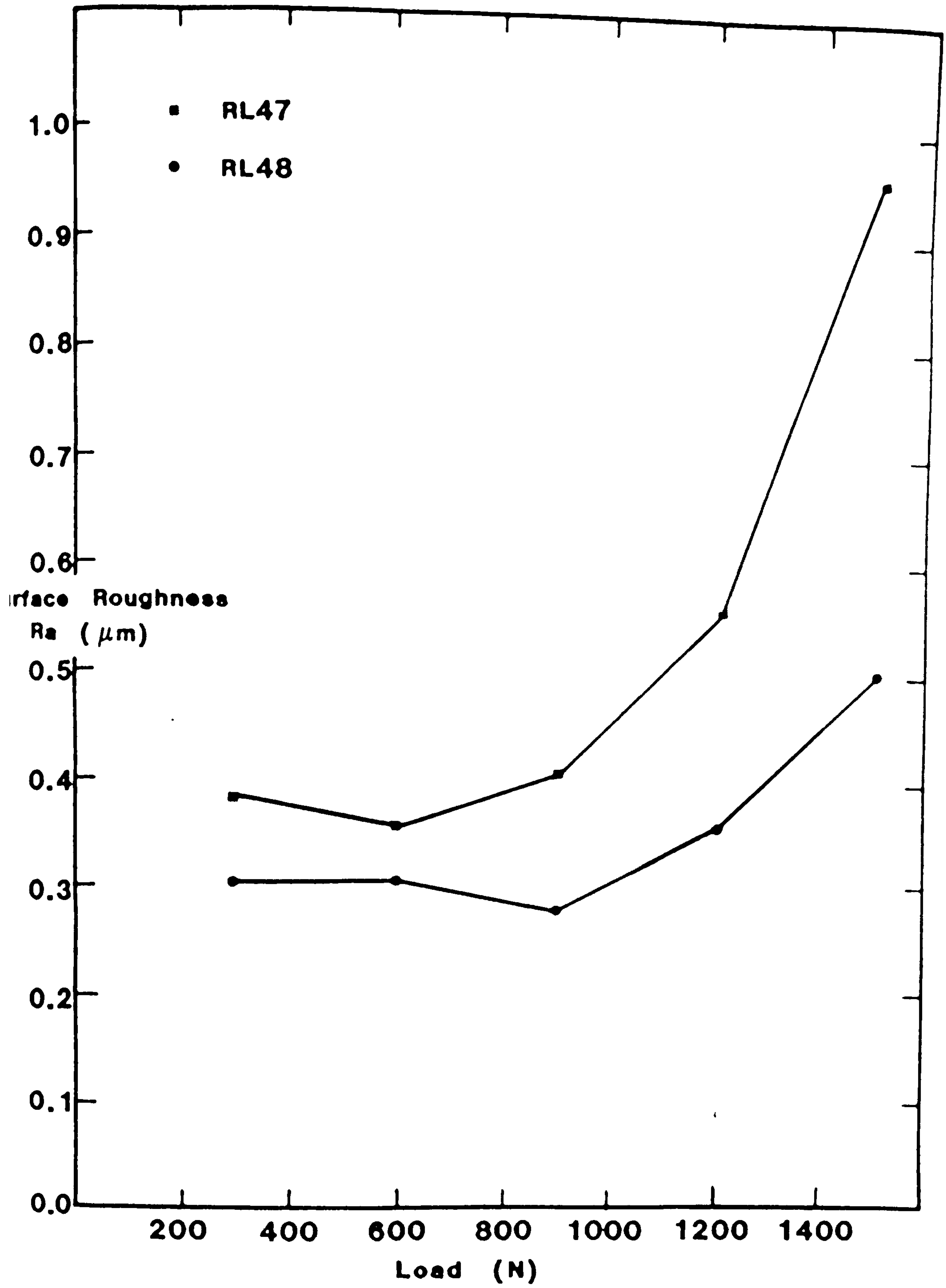


Fig.68 Surface roughness measurement (Ra) for the pins, for both oils, at 80^{o}C.

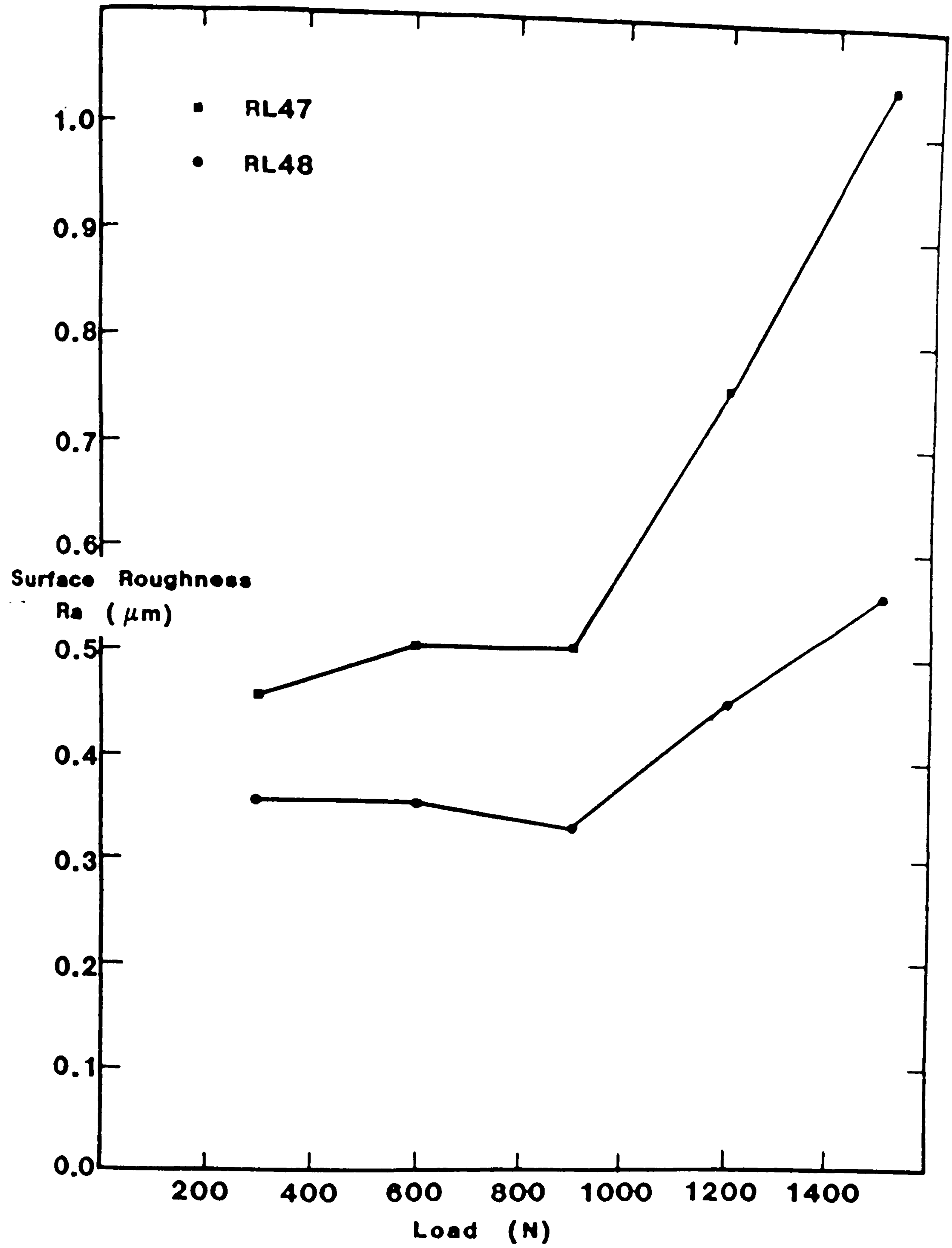


Fig.69 Surface roughness measurement (Ra) for the pins, for both oils, at 150°C.

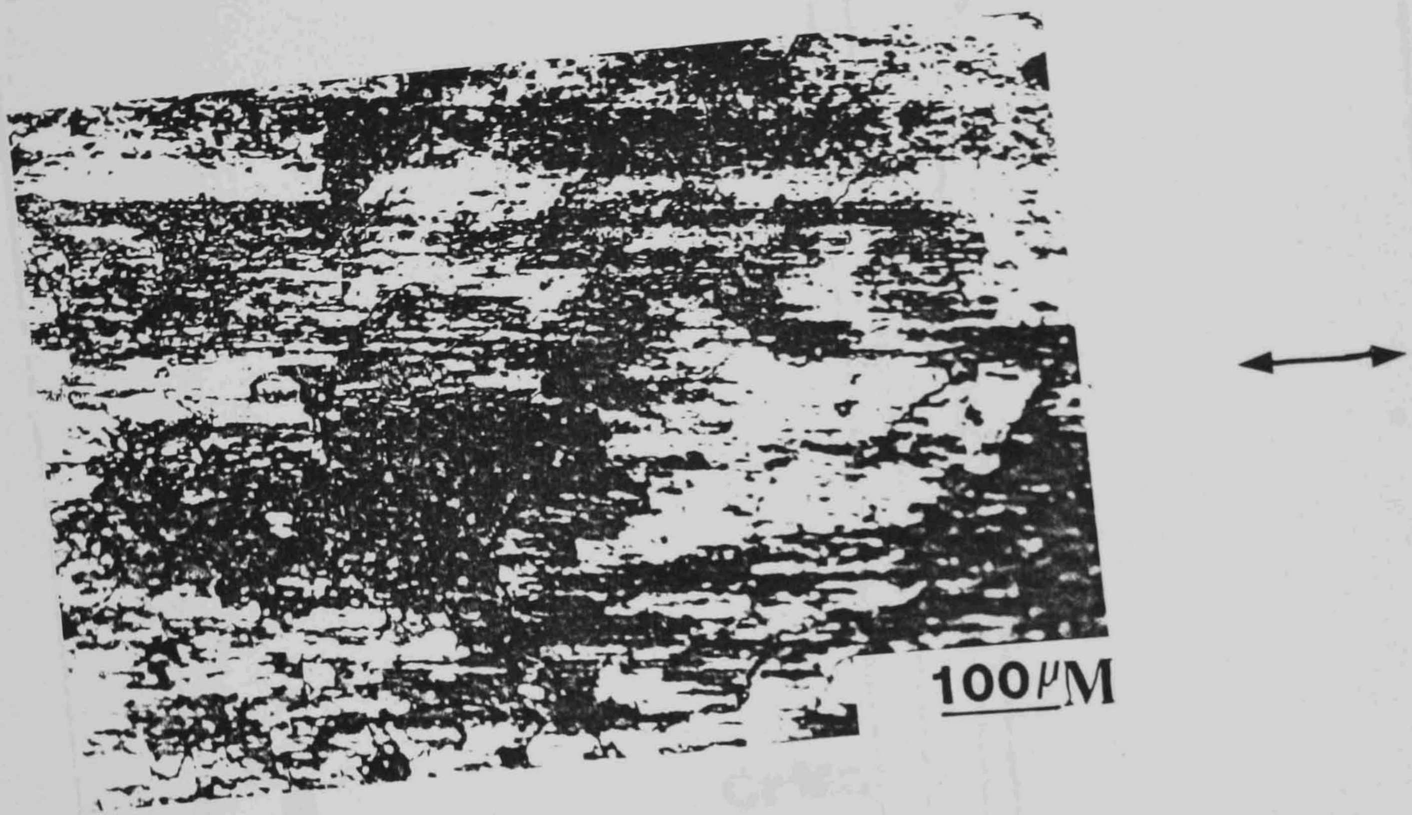


Fig.70 Deposit on the wear track for the worn flat.

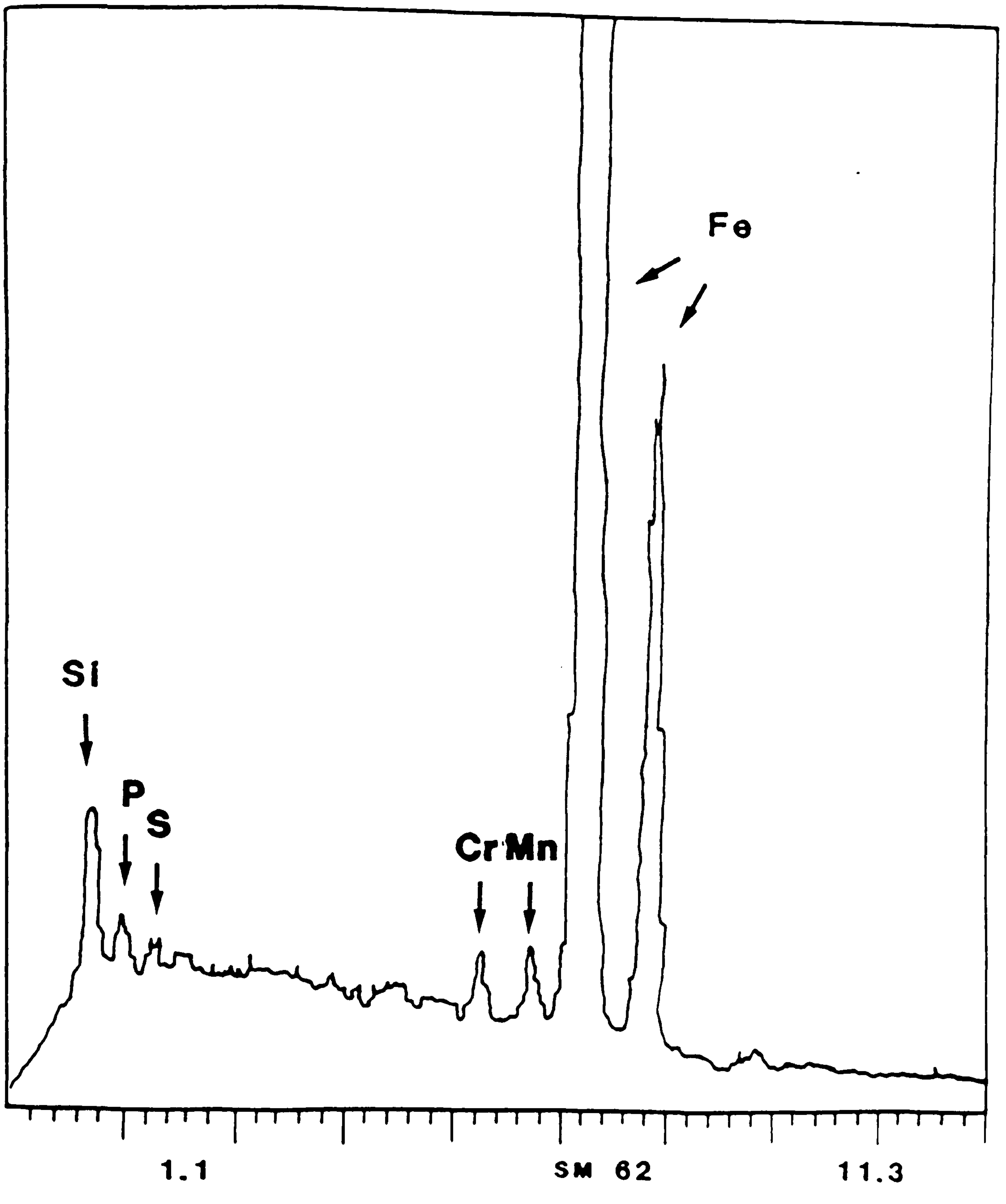


Fig.71 E.p.M.A. for the unused flat.

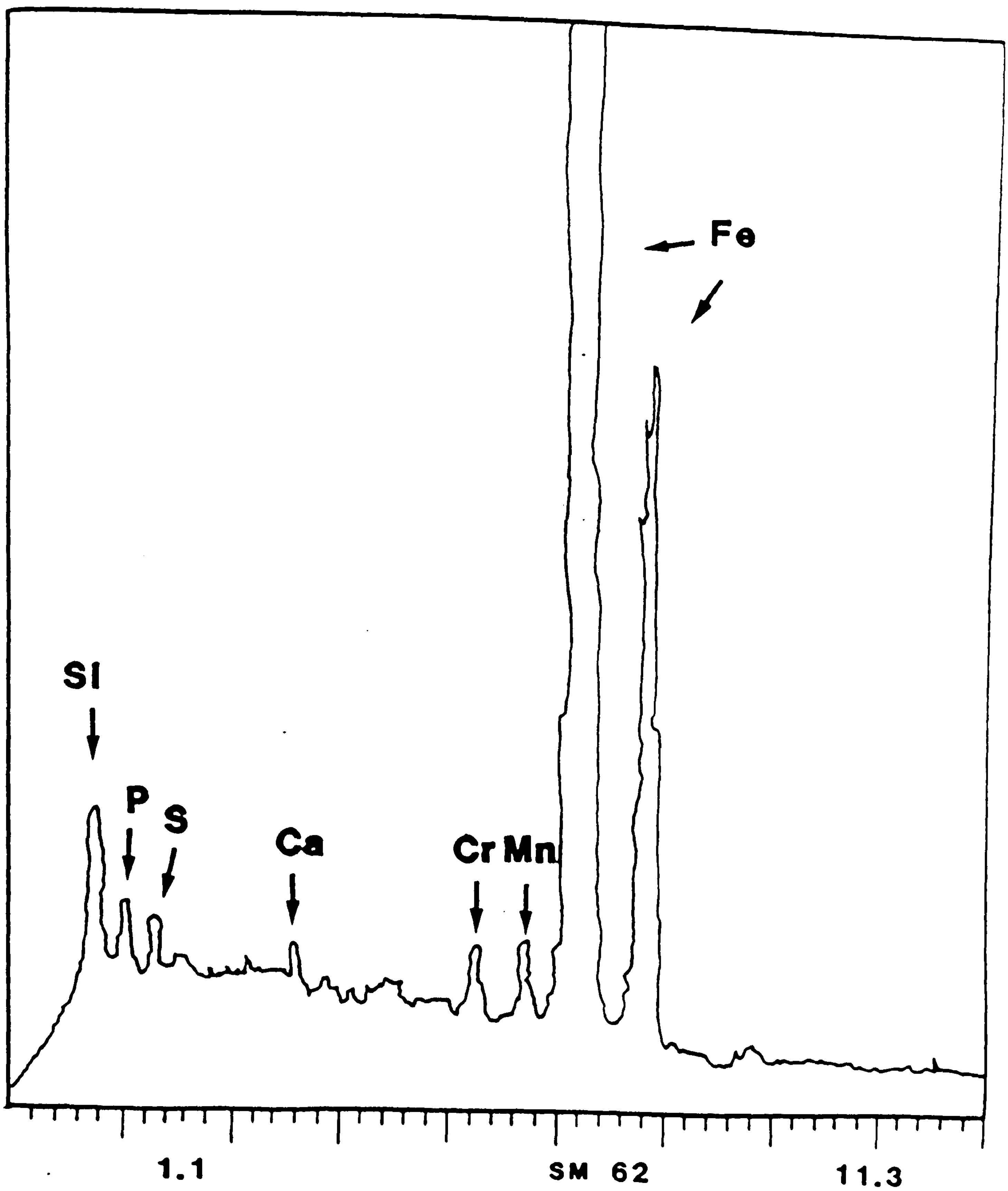


Fig.72 E.p.M.A. for the flat worn using RL47 at 80°C, (load = 900N).

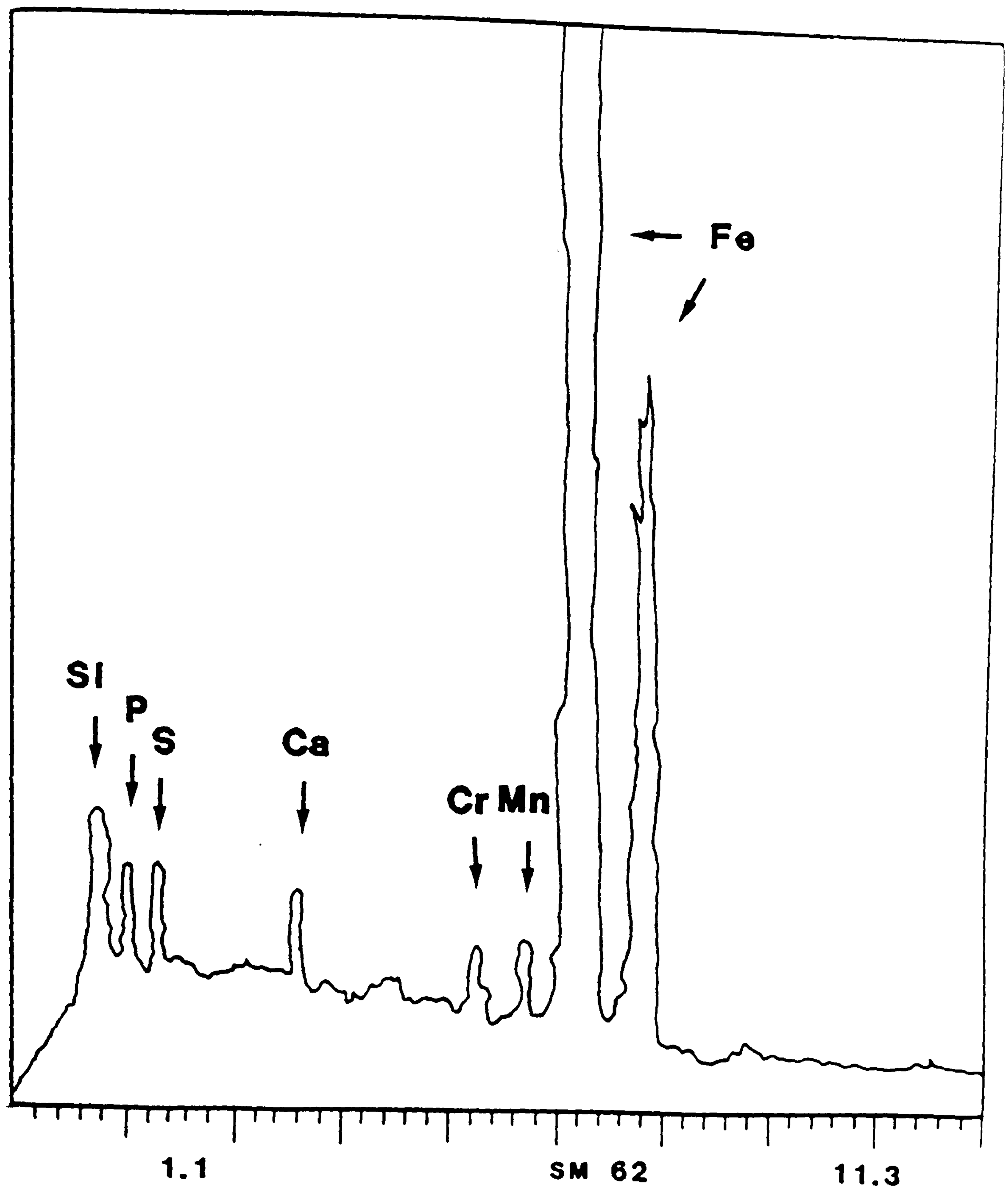


Fig.73 E.p.M.A. for the flat worn using RL48 at 80°C, (load = 900N).

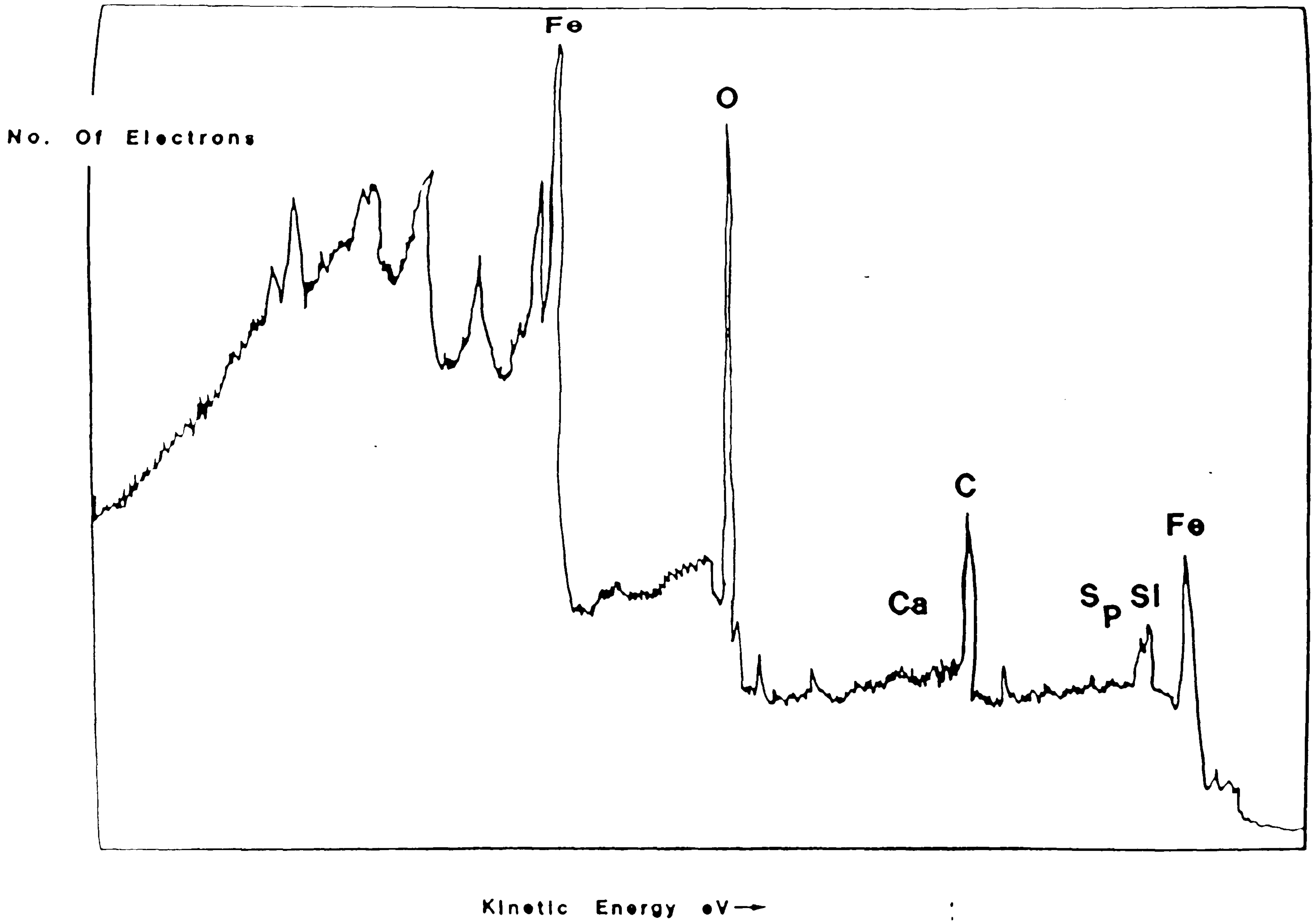
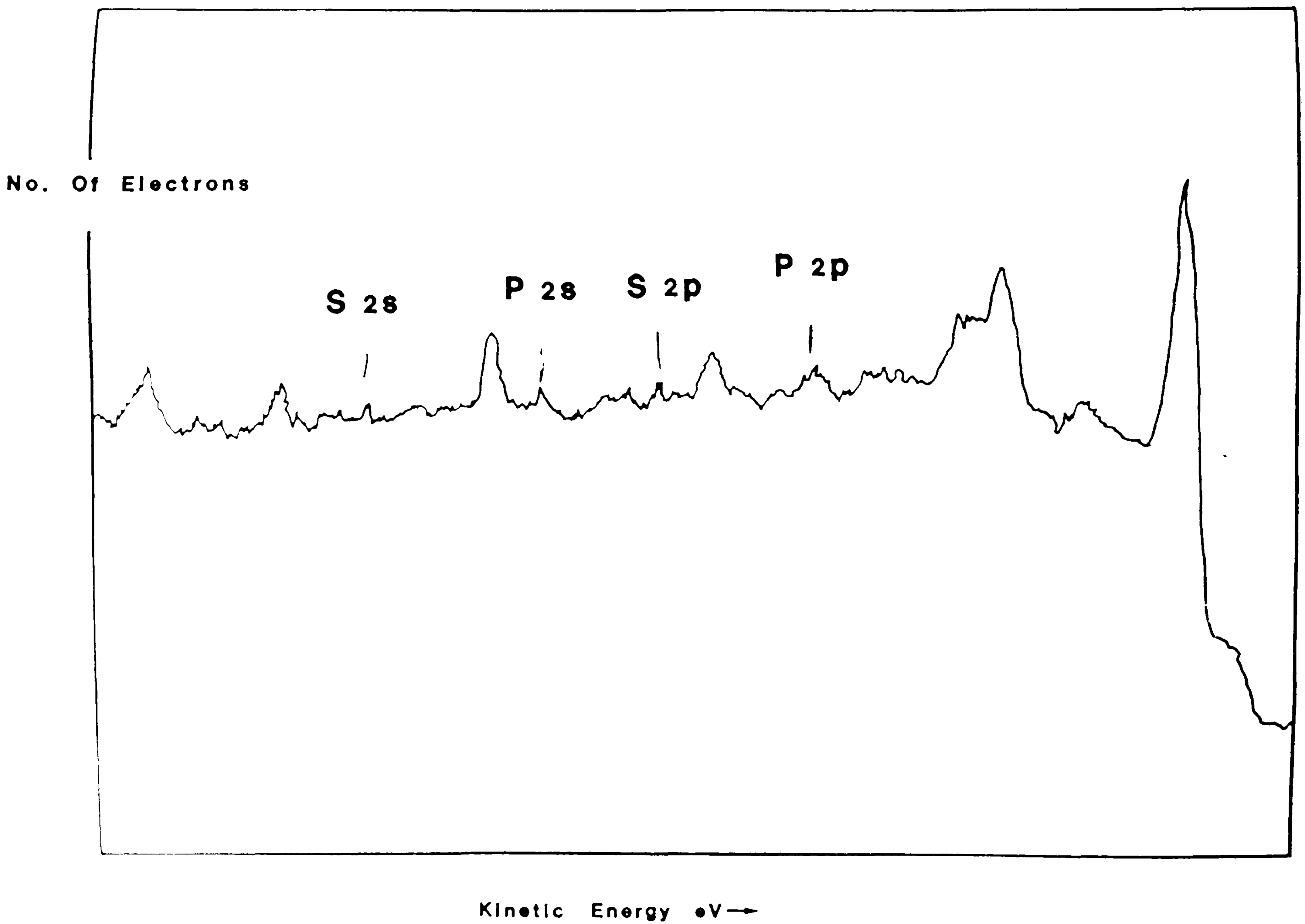
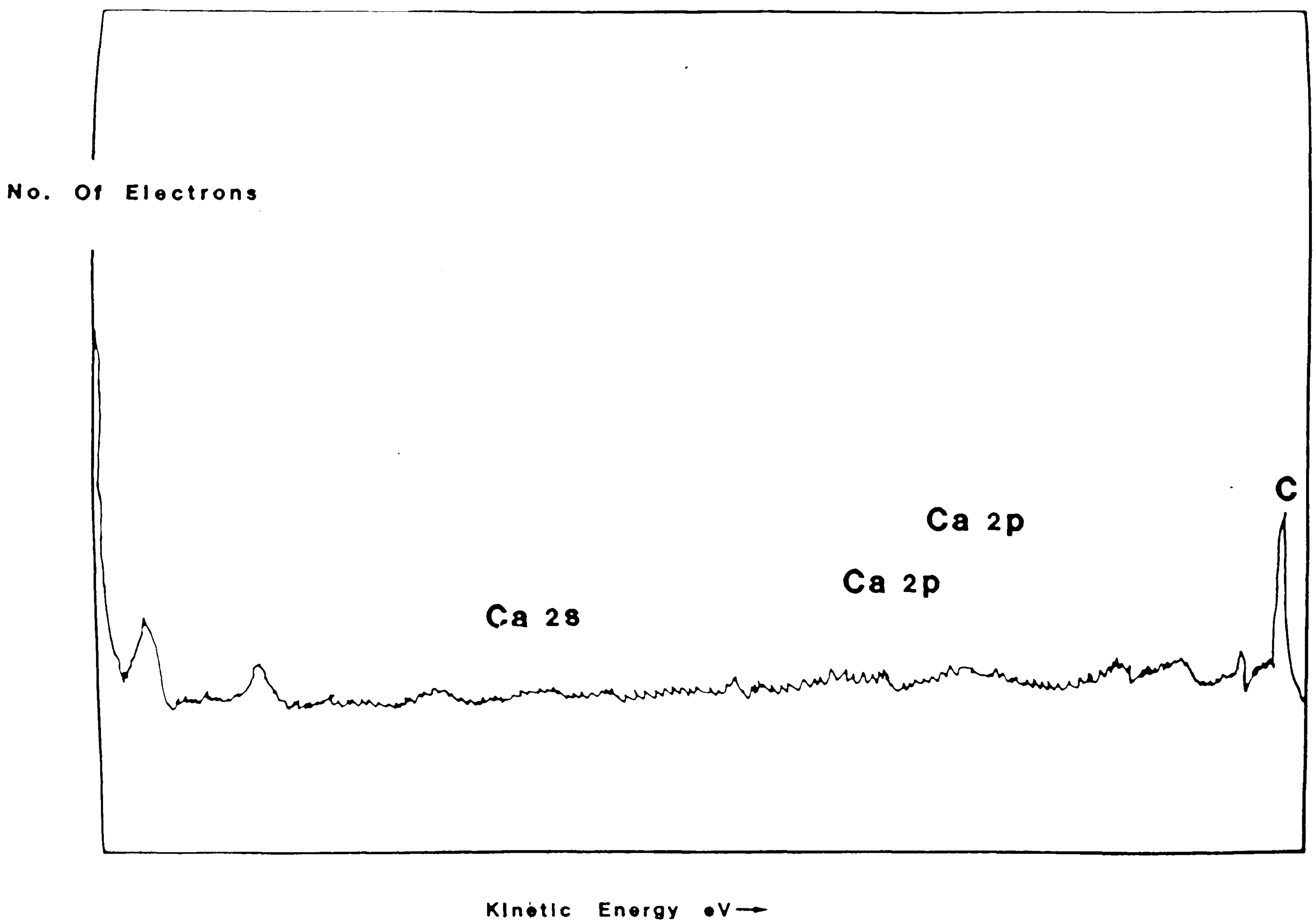


Fig.74 Energy spectrum for the unused flat.


No. Of Electrons
S 2s
P 2s
S 2p
P 2p
Kinetic Energy eV→


No. Of Electrons
C
Ca 2p
Ca 2p
Ca 2s
Kinetic Energy eV →

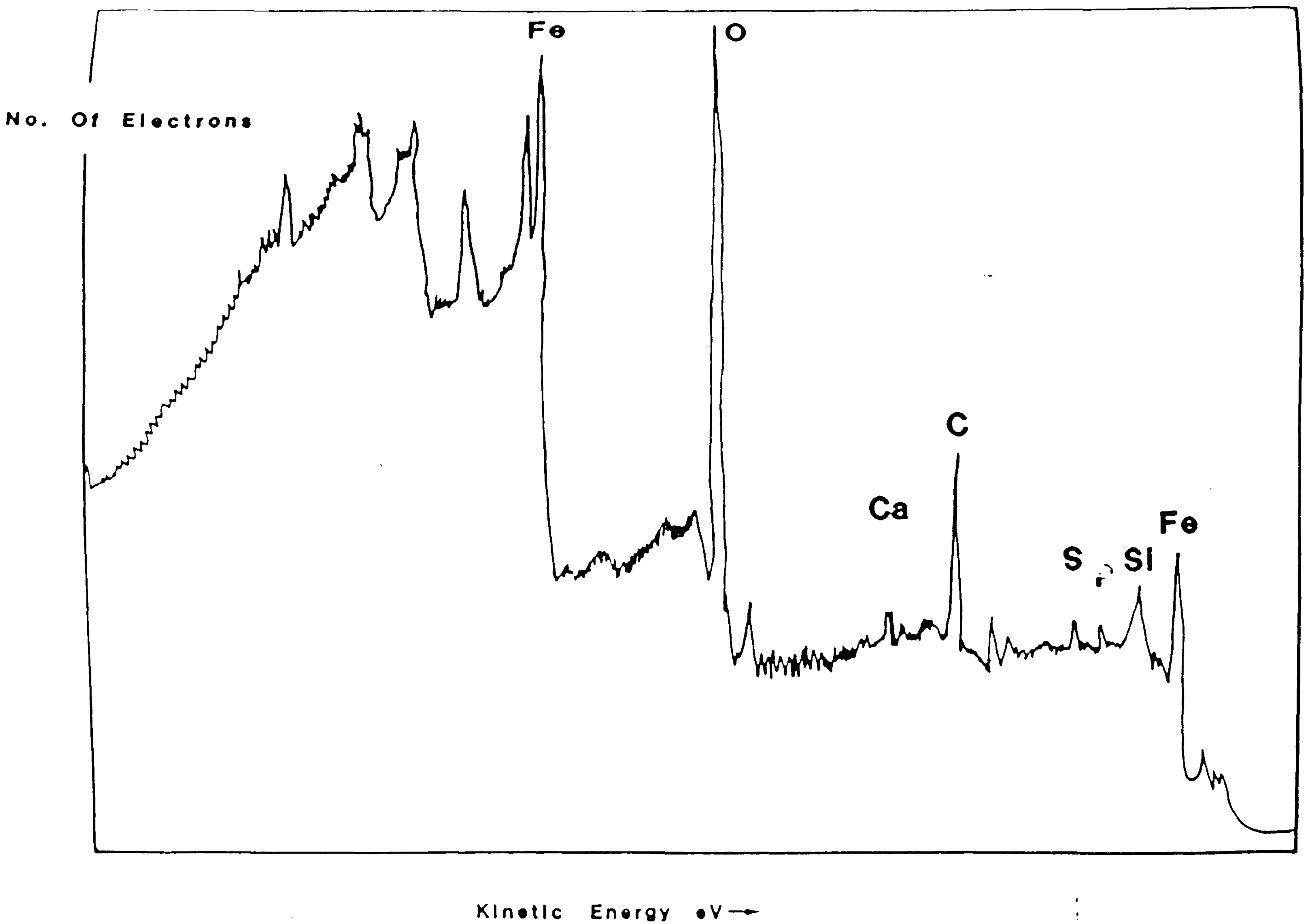
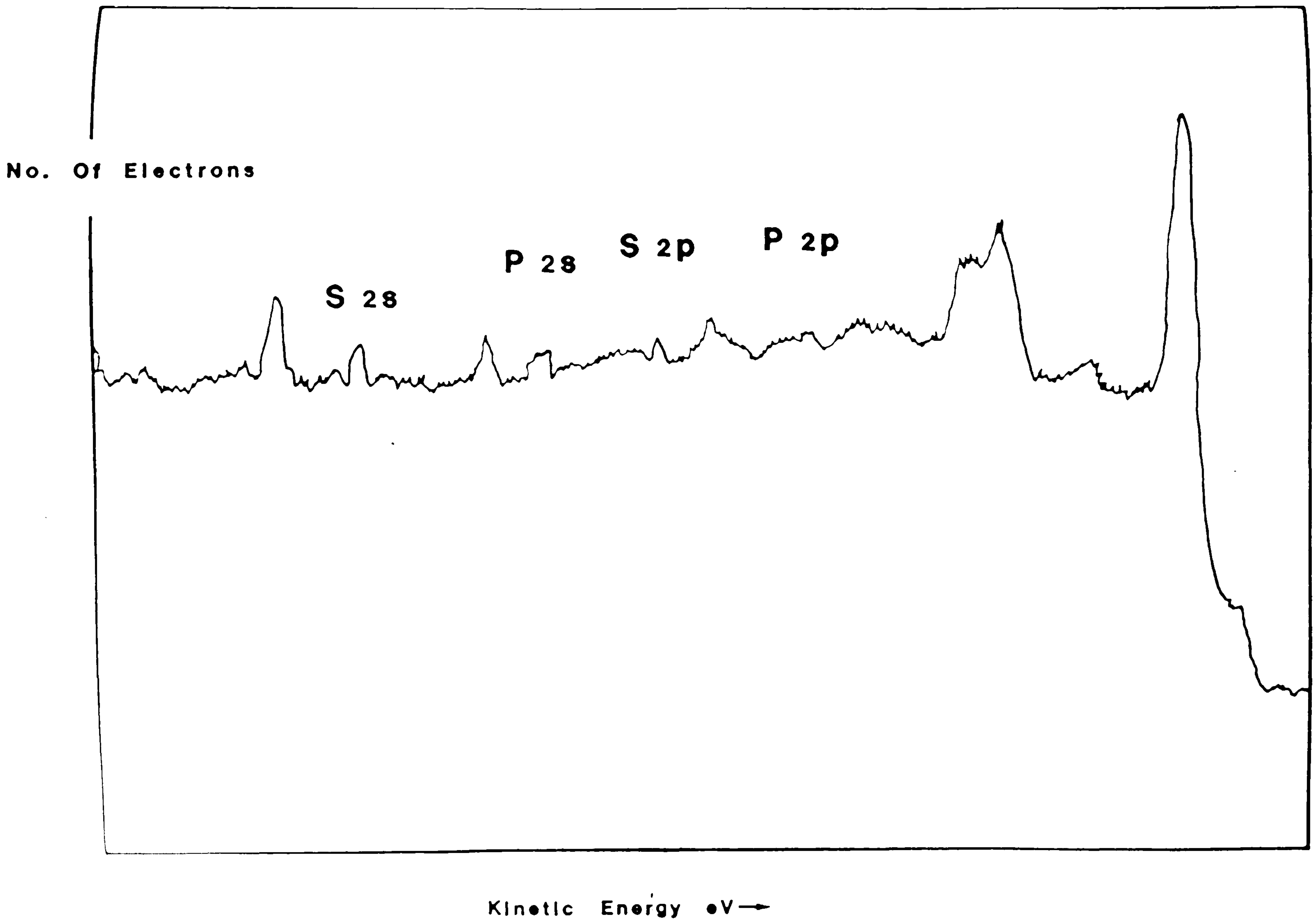
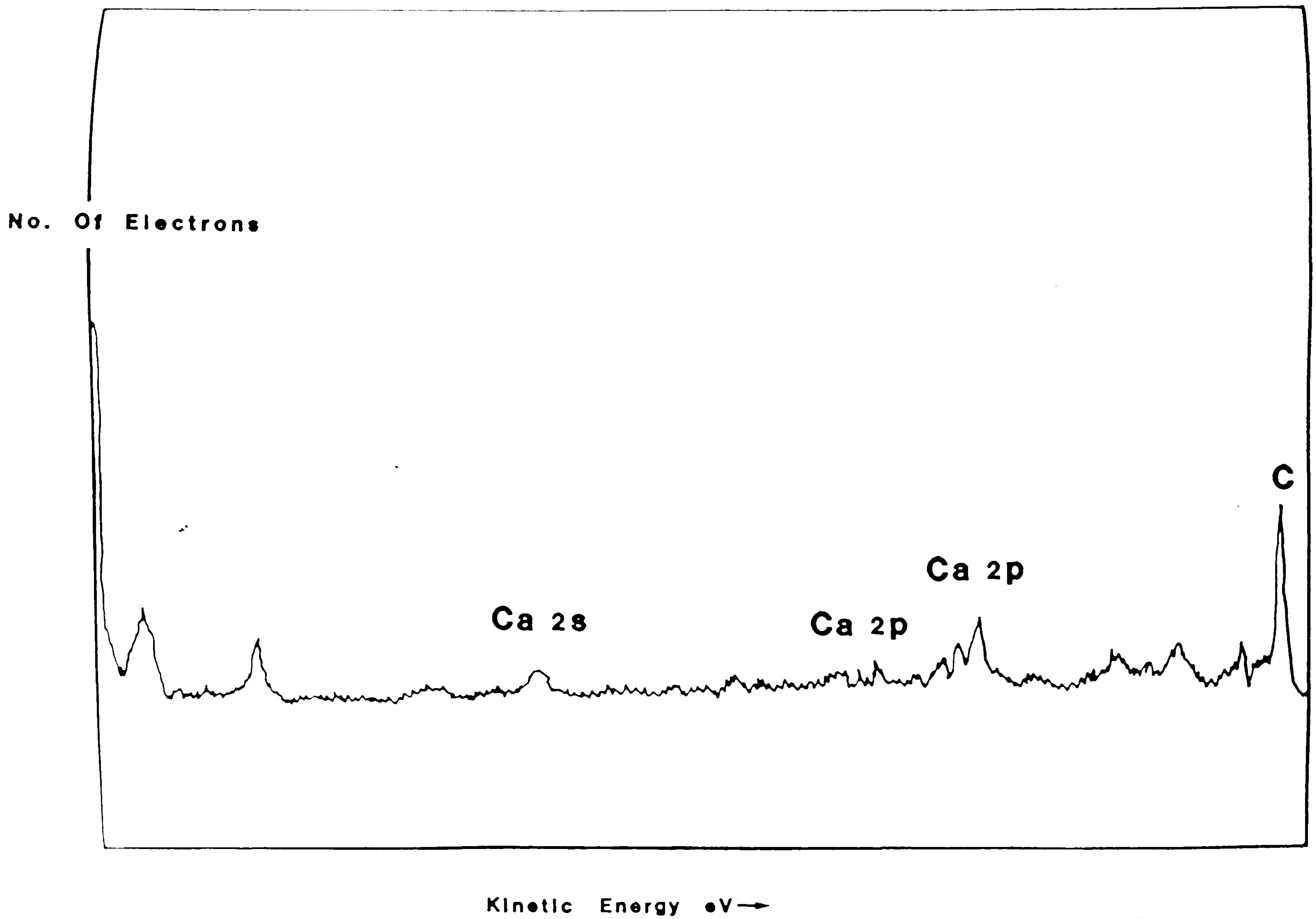


Fig.75 Energy spectrum for the flat worn using RL47 at 80°C, (load = 900N).


No. Of Electrons
S 2s
P 2s
S 2p
P 2p
Kinetic Energy eV


No. Of Electrons
C
Ca 2p
Ca 2p
Ca 2s
Kinetic Energy eV→

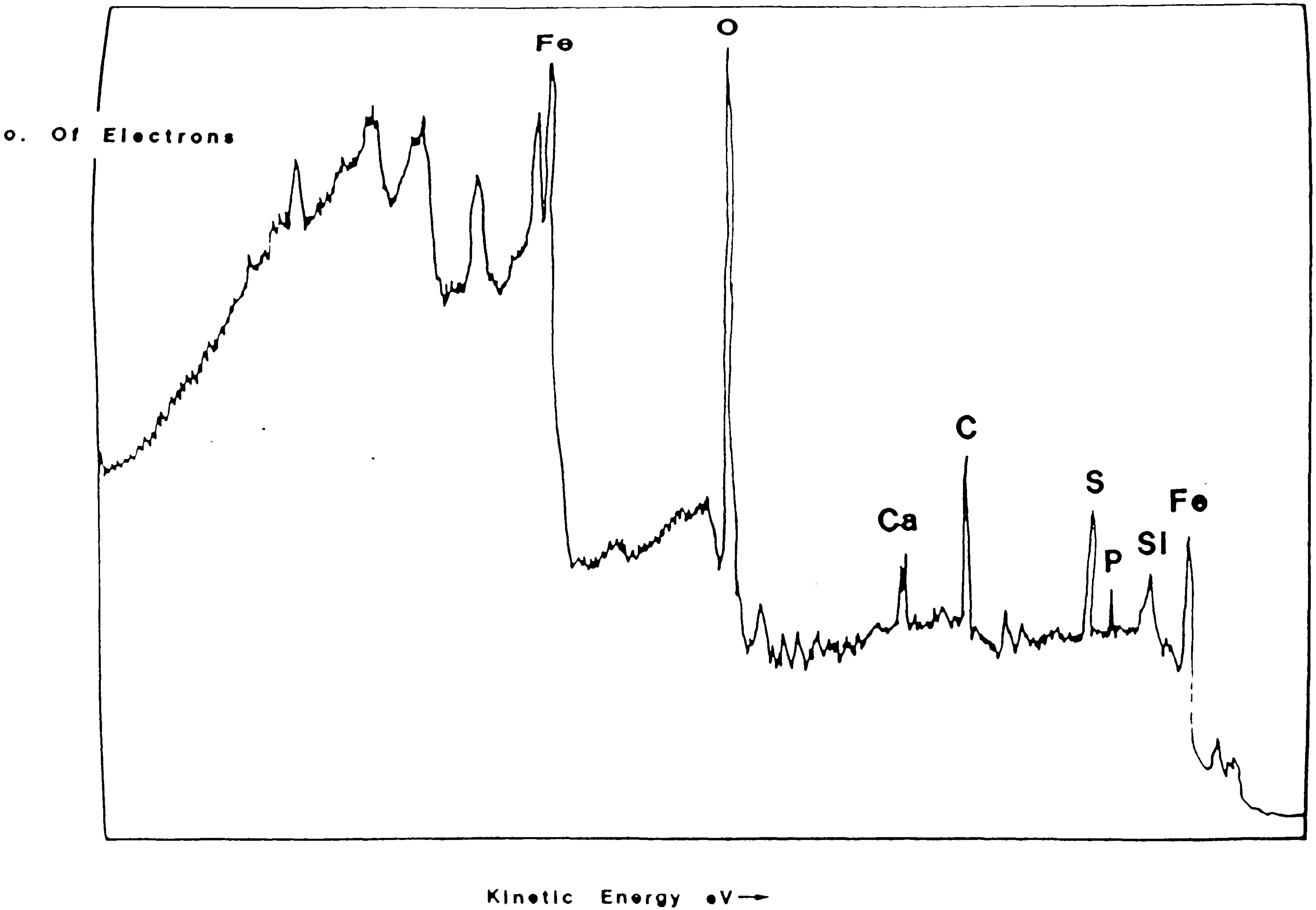
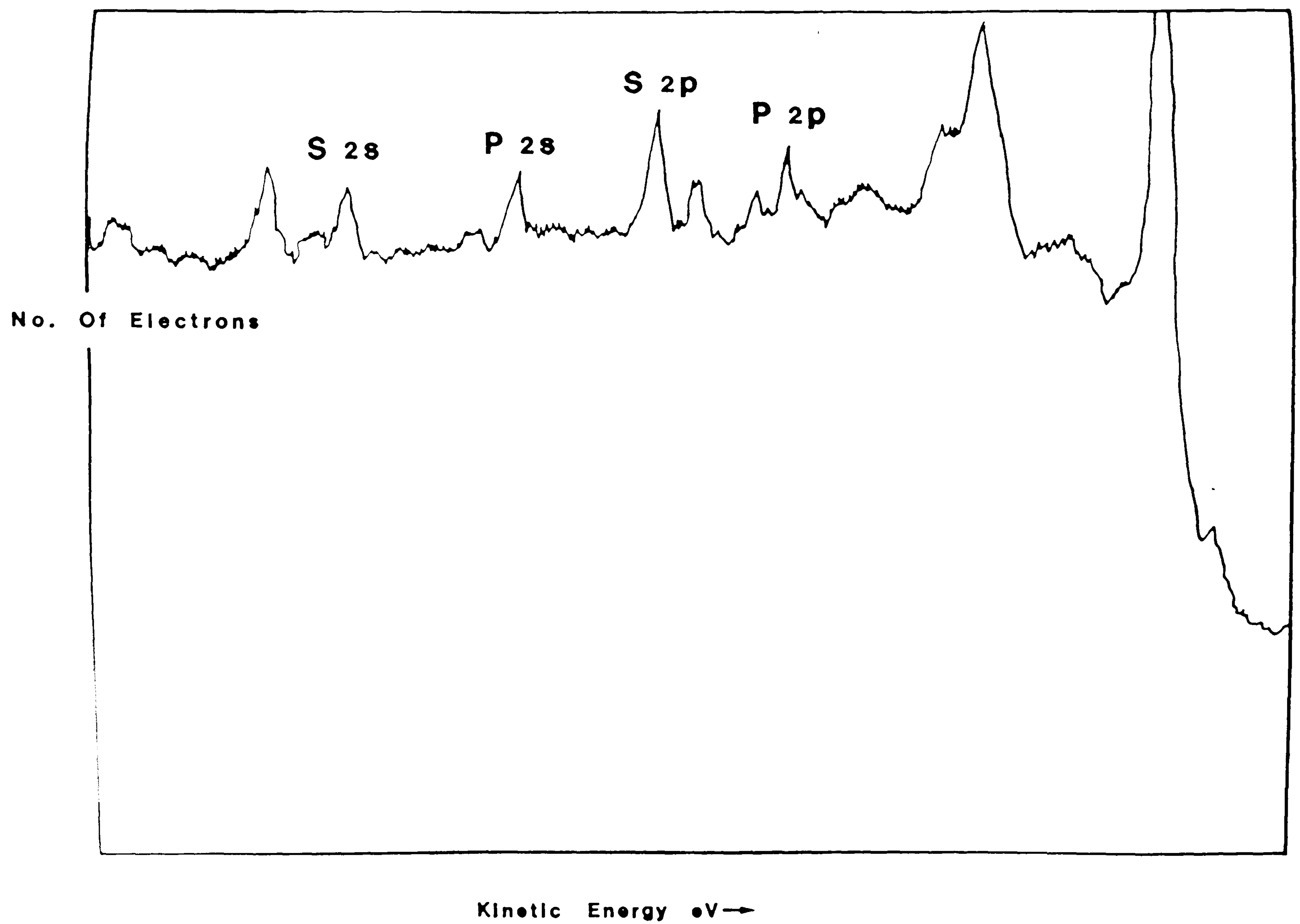
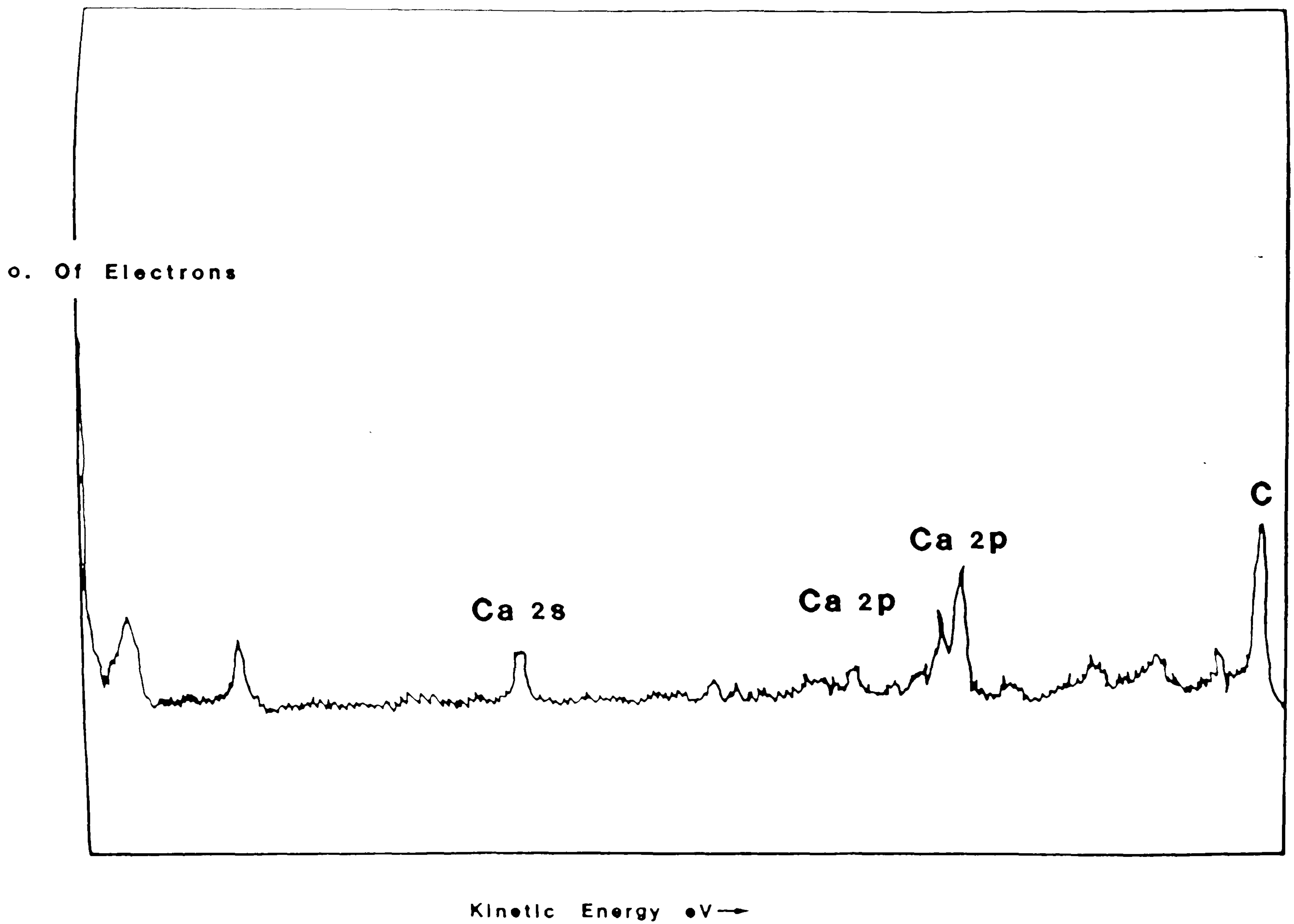


Fig.76 Energy spectrum for the flat worn using RL48 at 80°C, (load = 900N).


S 2s
P 2s
S 2p
P 2p
No. Of Electrons
Kinetic Energy eV→


o. Of Electrons
C
Ca 2p
Ca 2p
Ca 2s
Kinetic Energy eV→

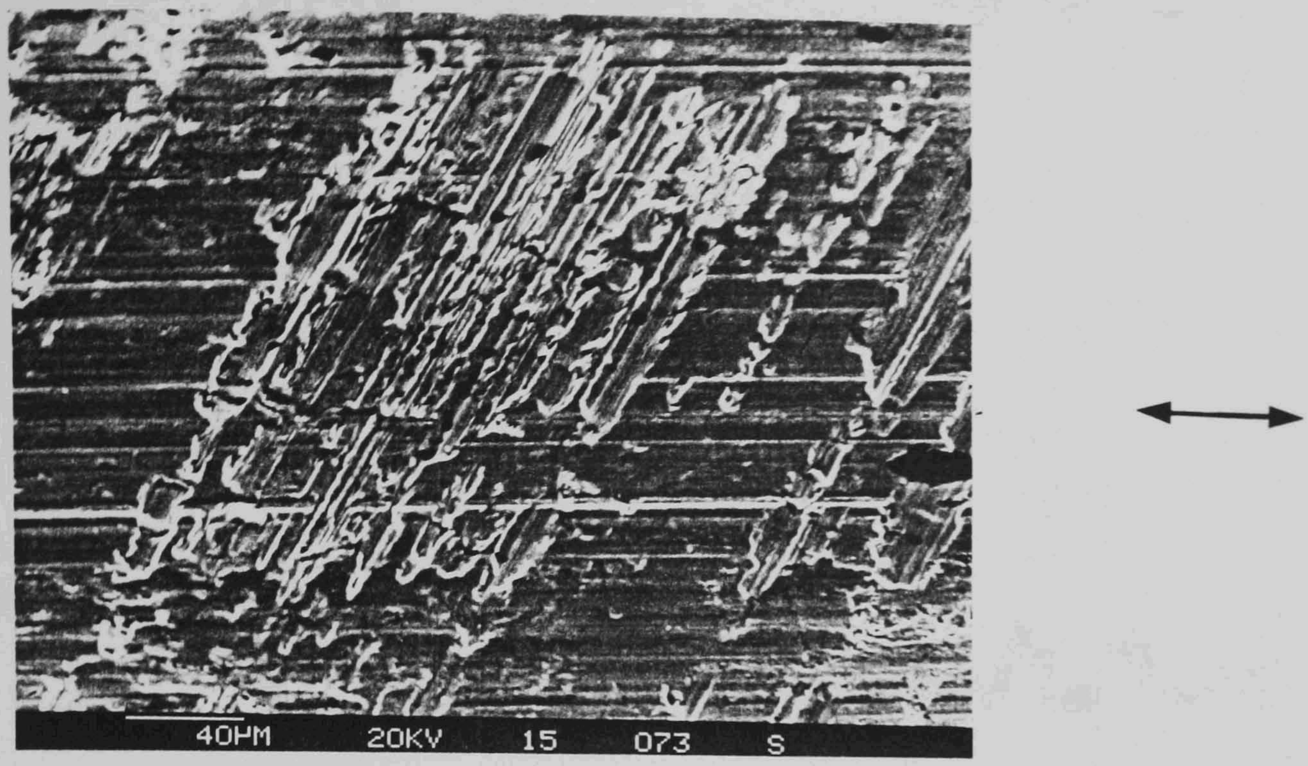


Fig.77 SEM appearance in the middle of the wear track, for the flat worn using RL48 at $80^{o}C$, (load = 600N).

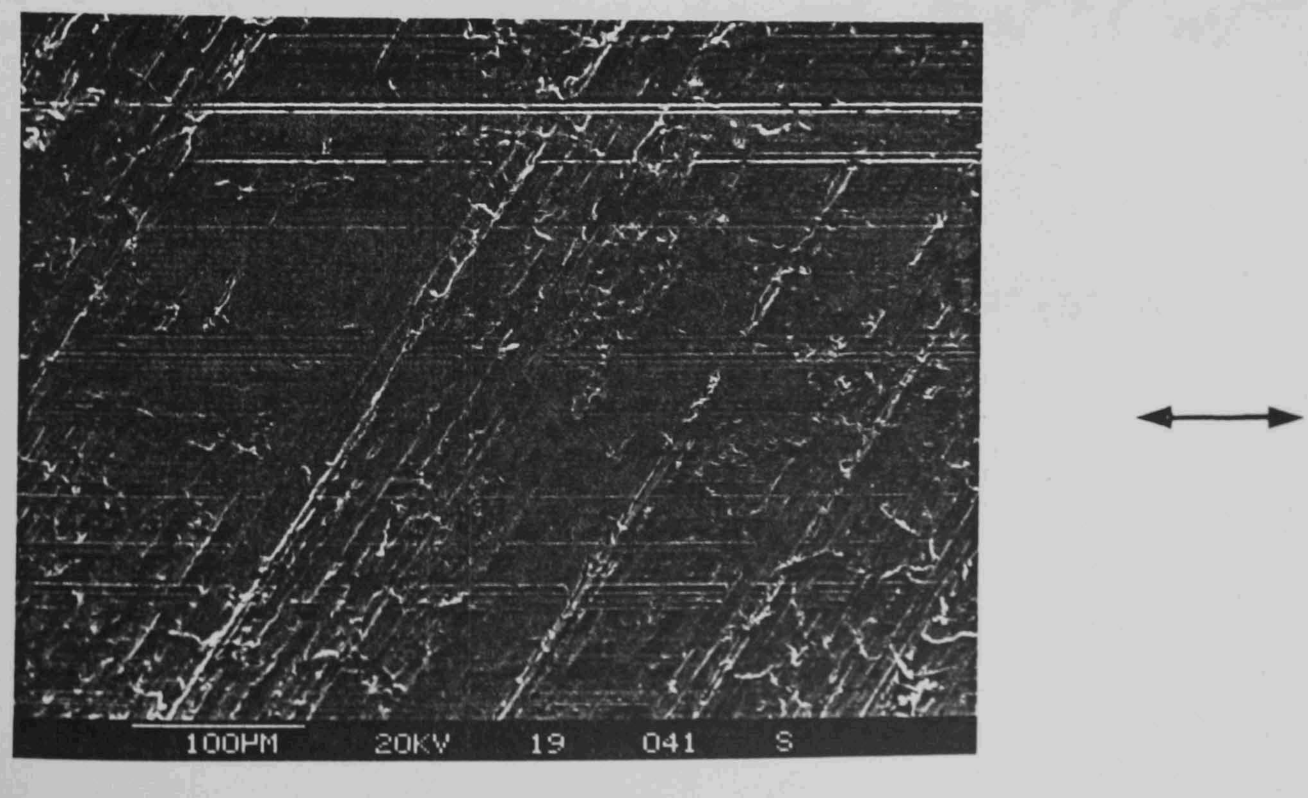


Fig.78 SEM appearance at the end of the wear track, for the flat worn using RL48 at $80^{o}C$, (load = 600N).

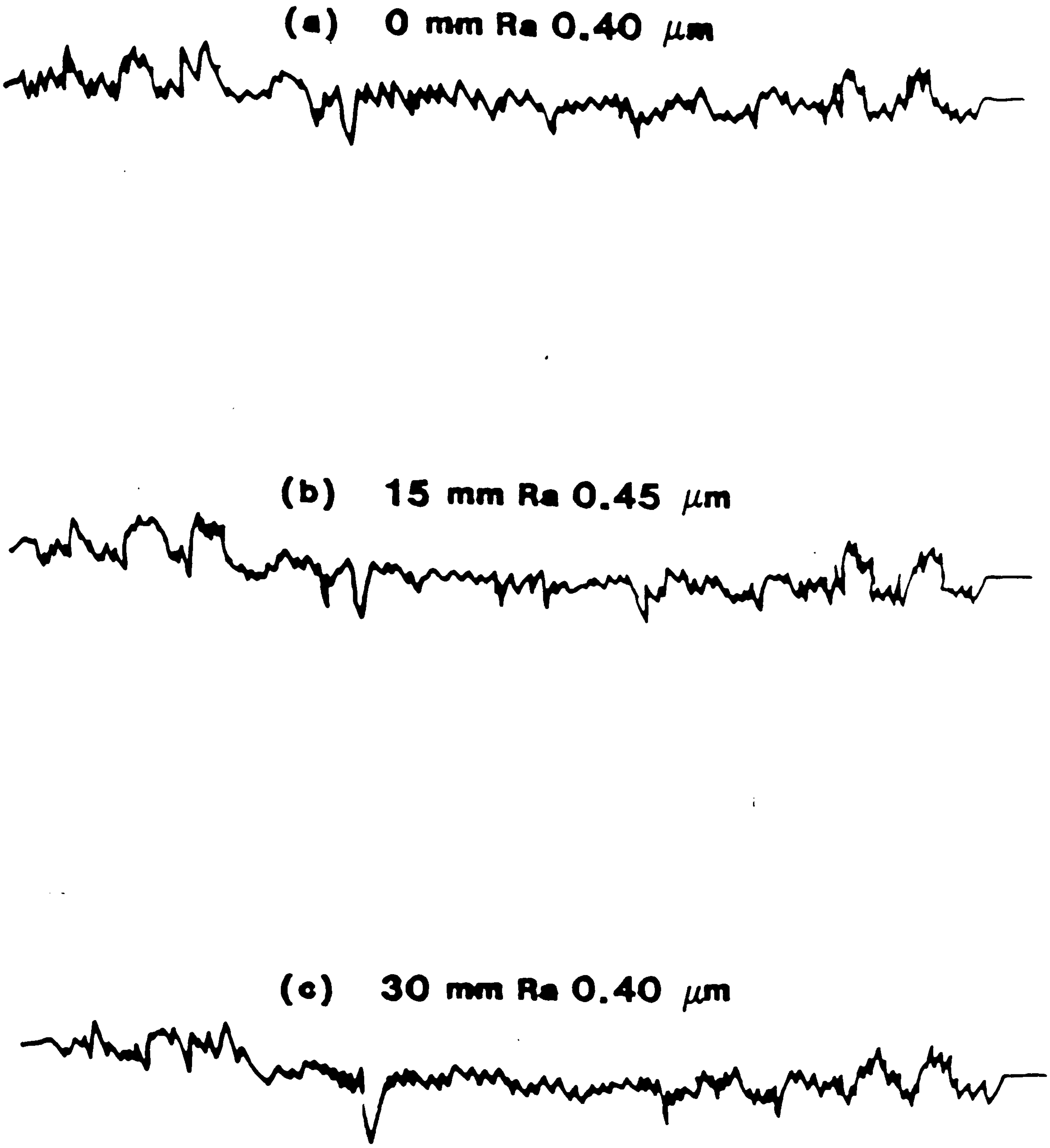


Fig.79 C.L.A. values for the worn flat, for Figures 77 and 78.

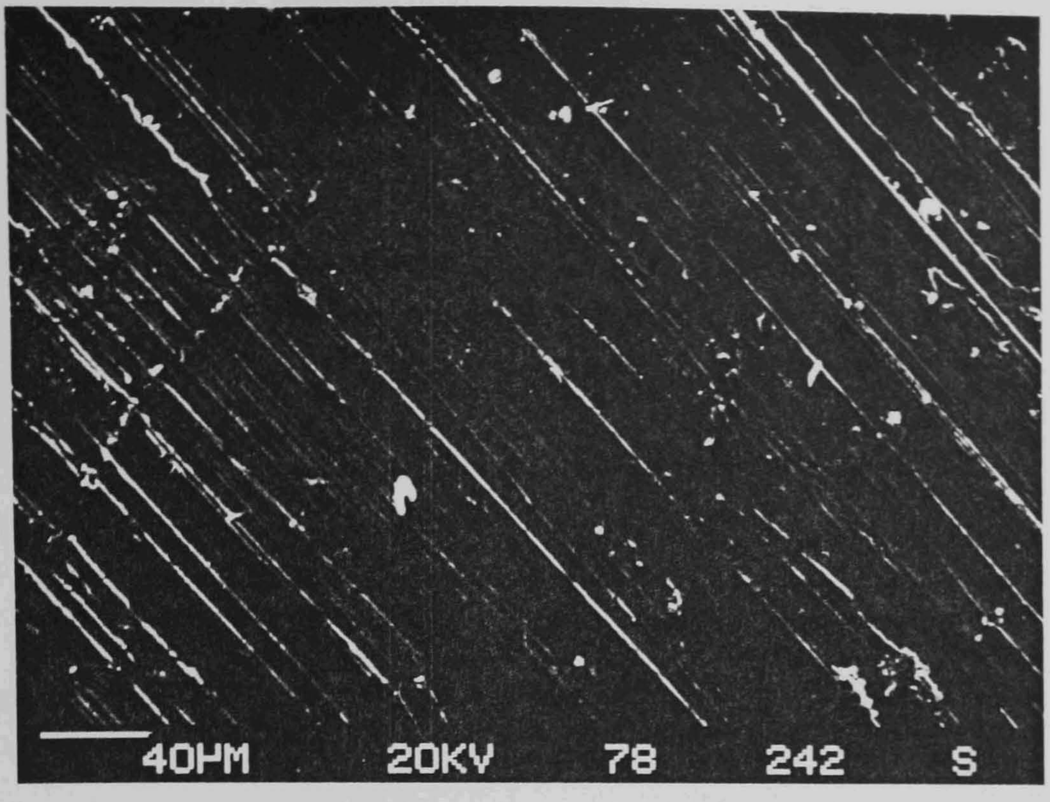


Fig.80 SEM appearance for the unused pin.

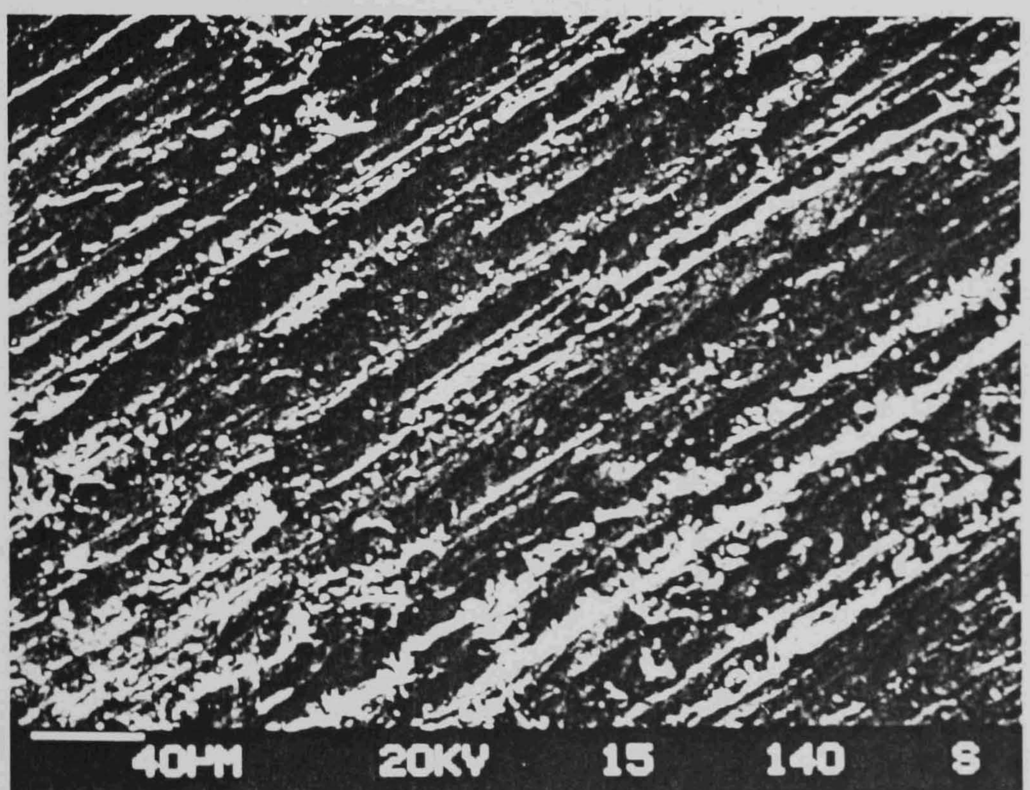


Fig.81 SEM appearance for the unused flat.

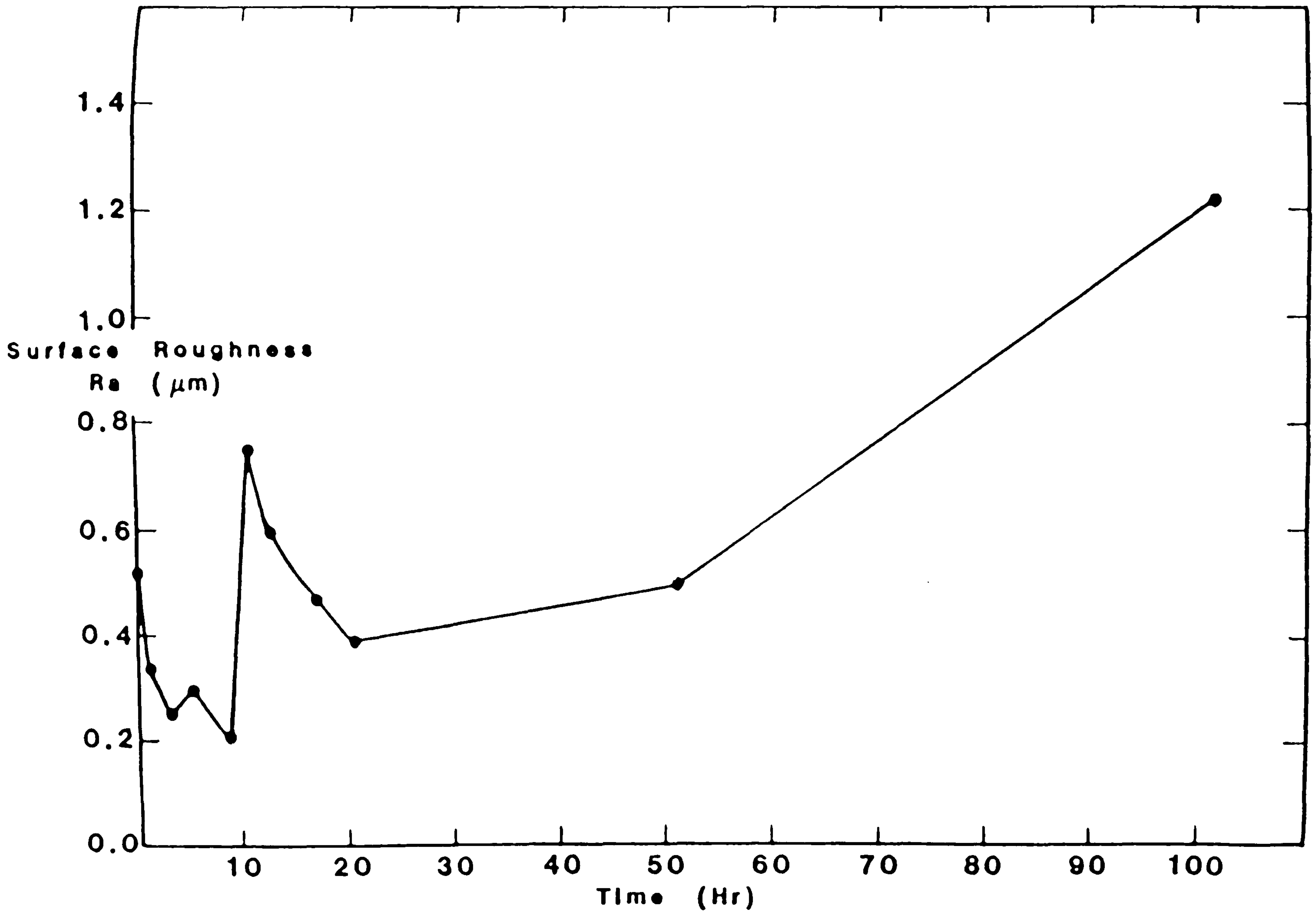


Fig.82 Surface roughness measurement (Ra) for the pin worn using RL48 for different test periods, at 80^{o}C, (load = 900N).

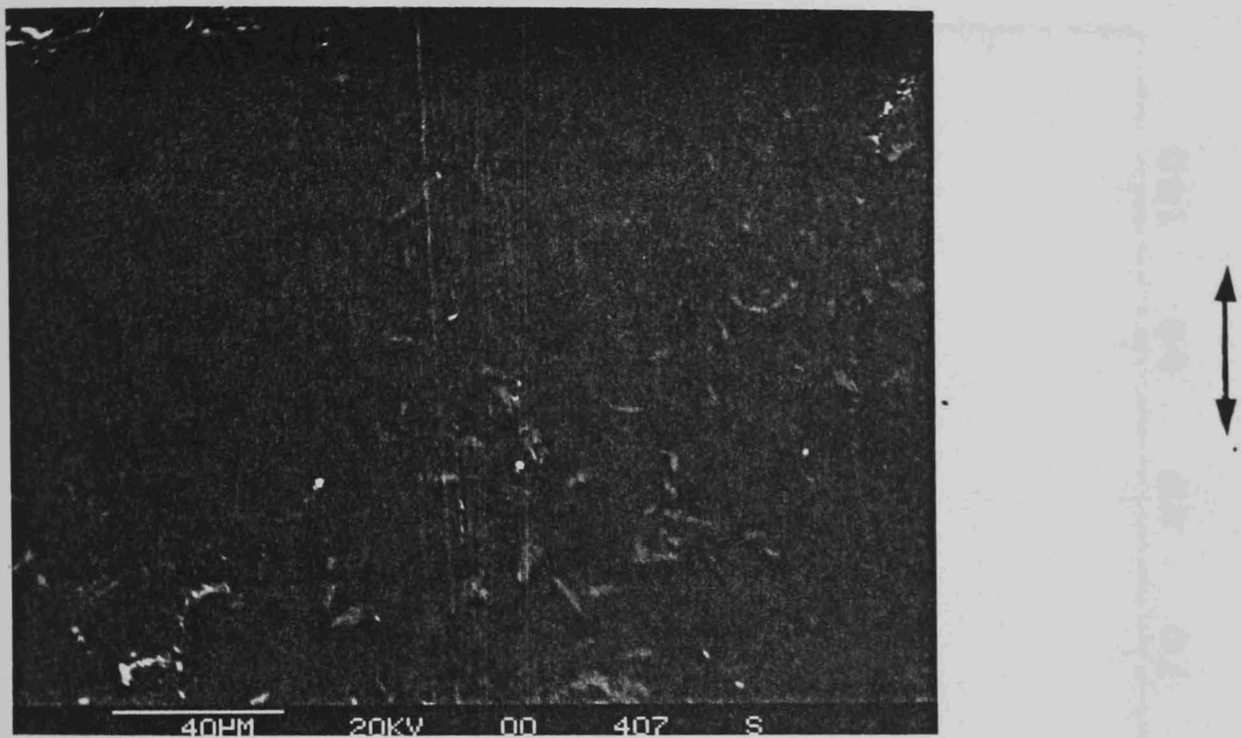


Fig.83 SEM appearance for the pin worn after 1 hour test using RL48 at 80°C, (load 900N).

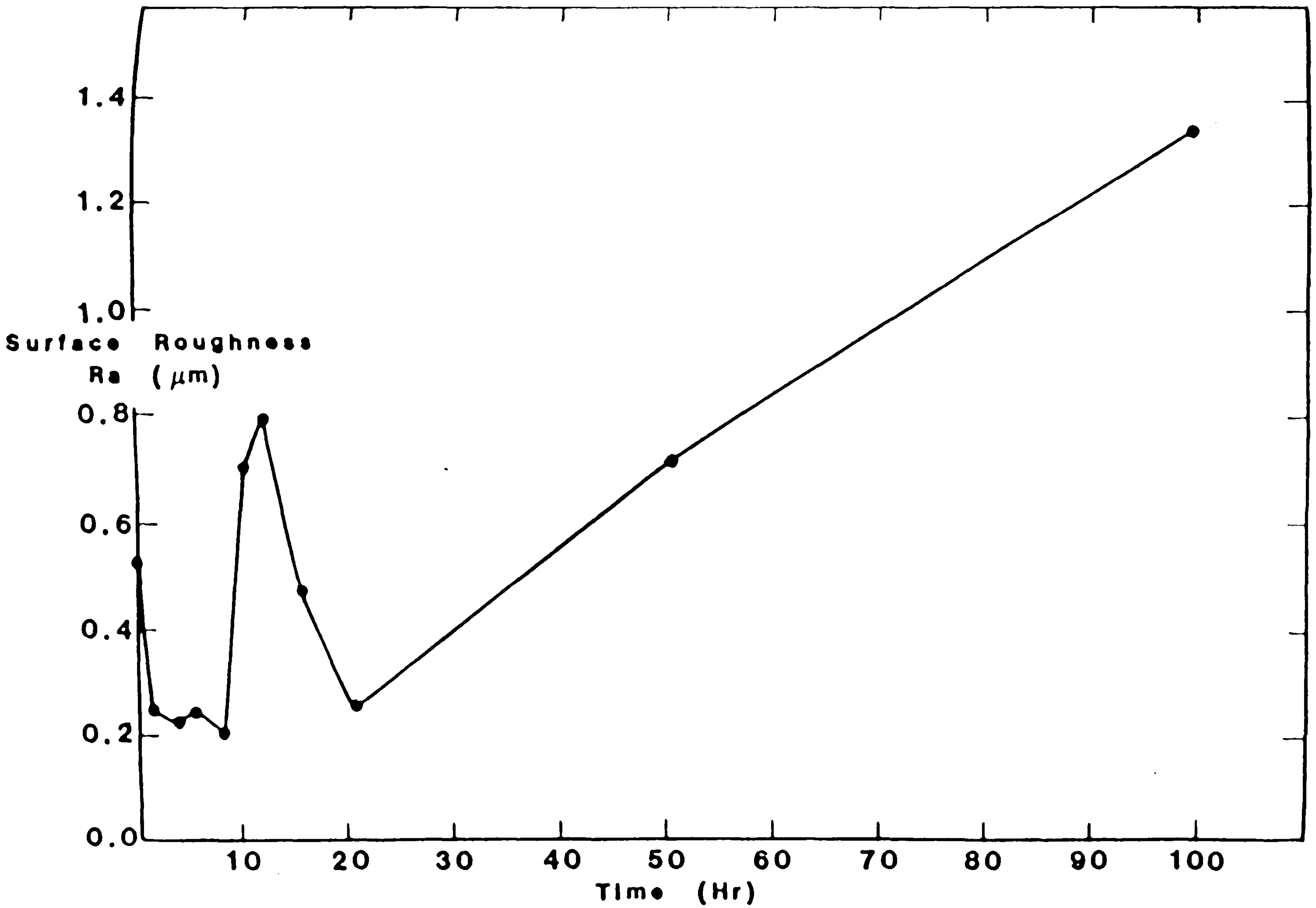


Fig.84 Surface roughness measurement (Ra) for the flat worn using RL48 for different test periods at 80^{o}C, (load = 900N).

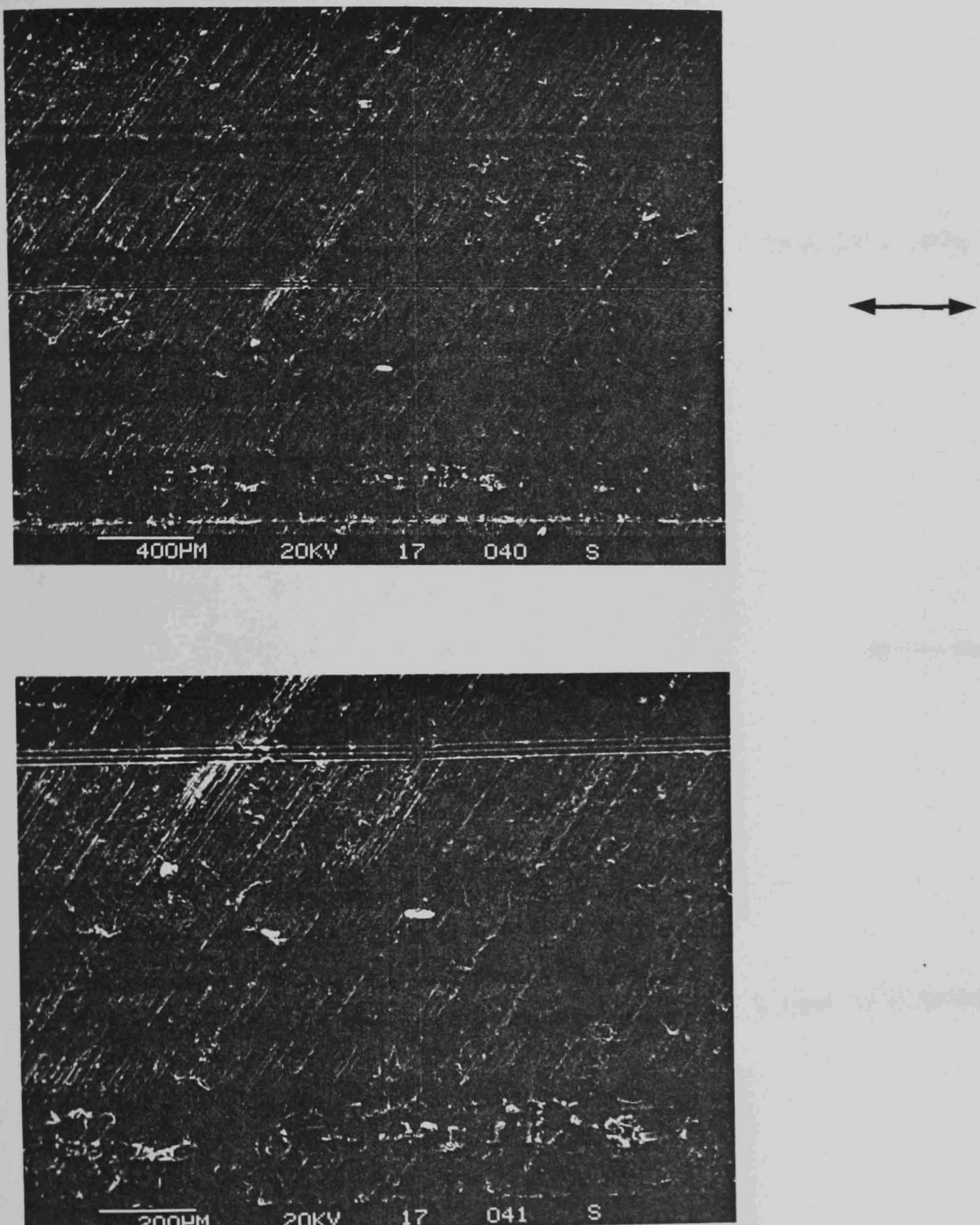


Fig.85 SEM appearance for the flat worn after 1 hour test using RL48 at 80°C, (load 900N).

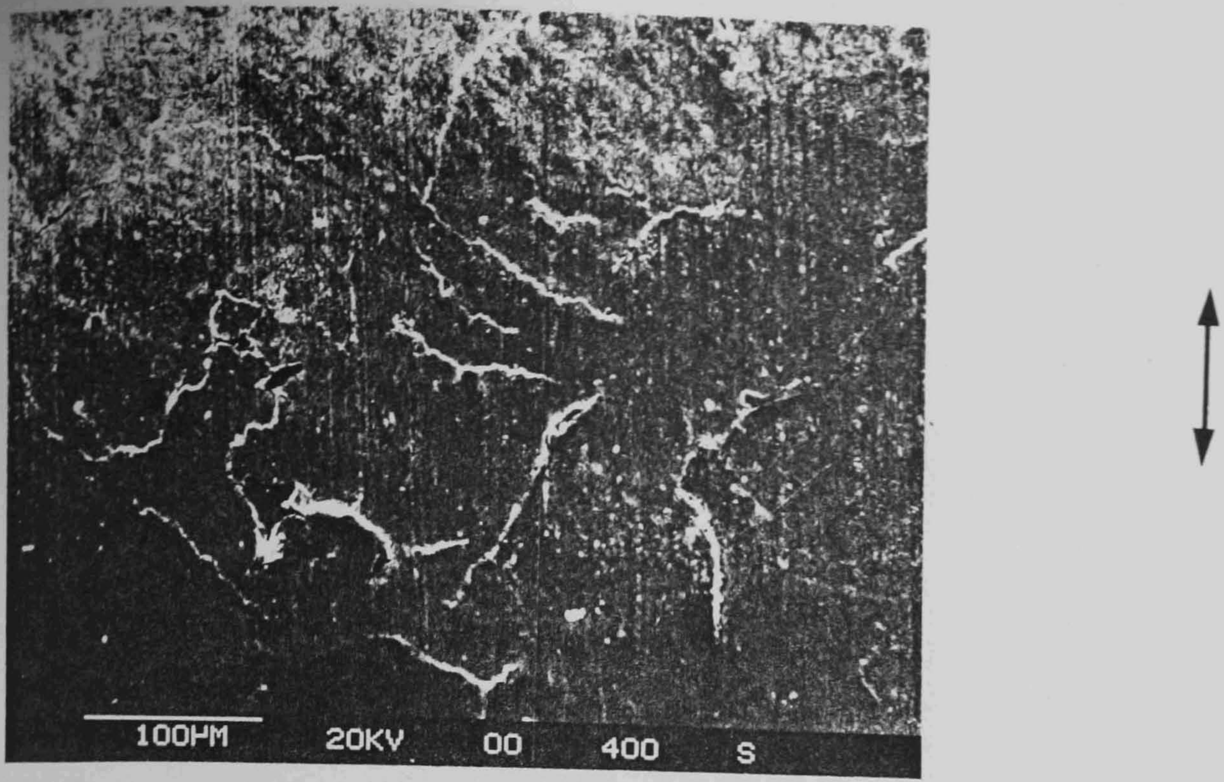


Fig.86 SEM appearance for the pin worn after 3 hour test using RL48 at 80^{o}C, (load 900N).

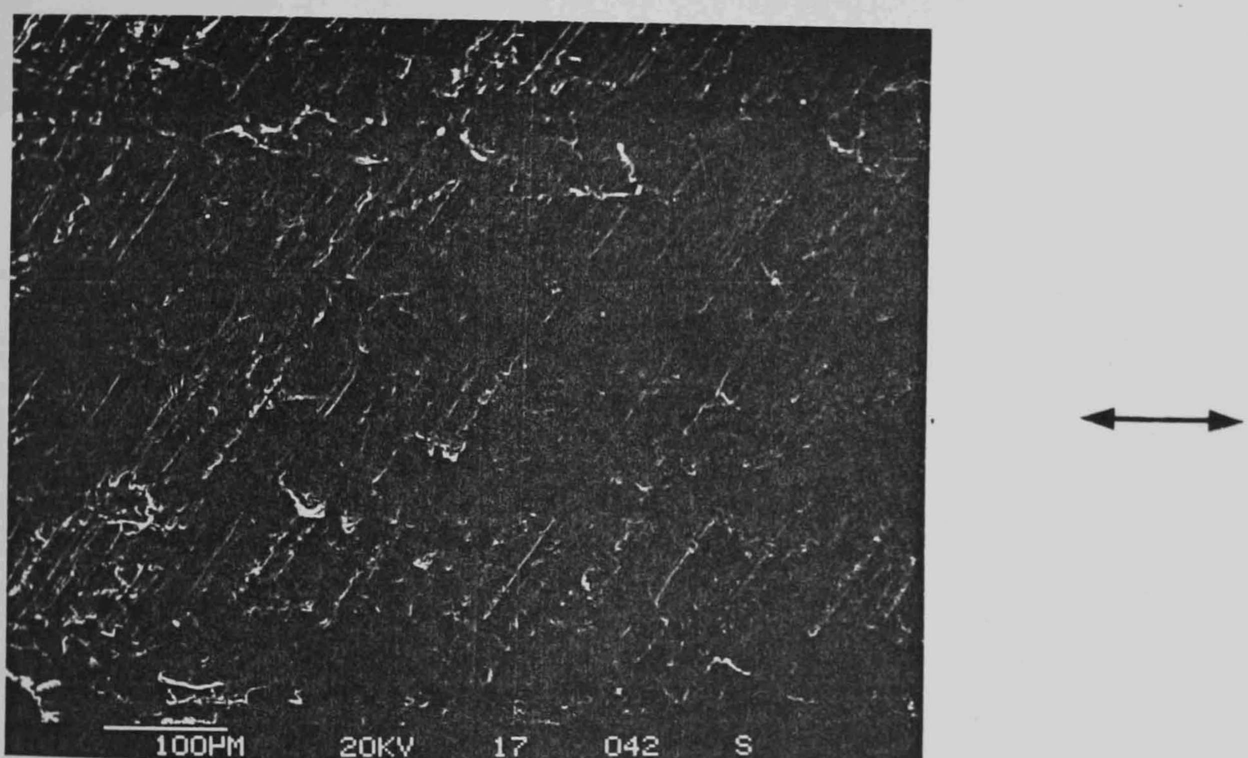


Fig.87 SEM appearance for the flat worn after 3 hour test using RL48 at 80^{o}C, (load 900N).

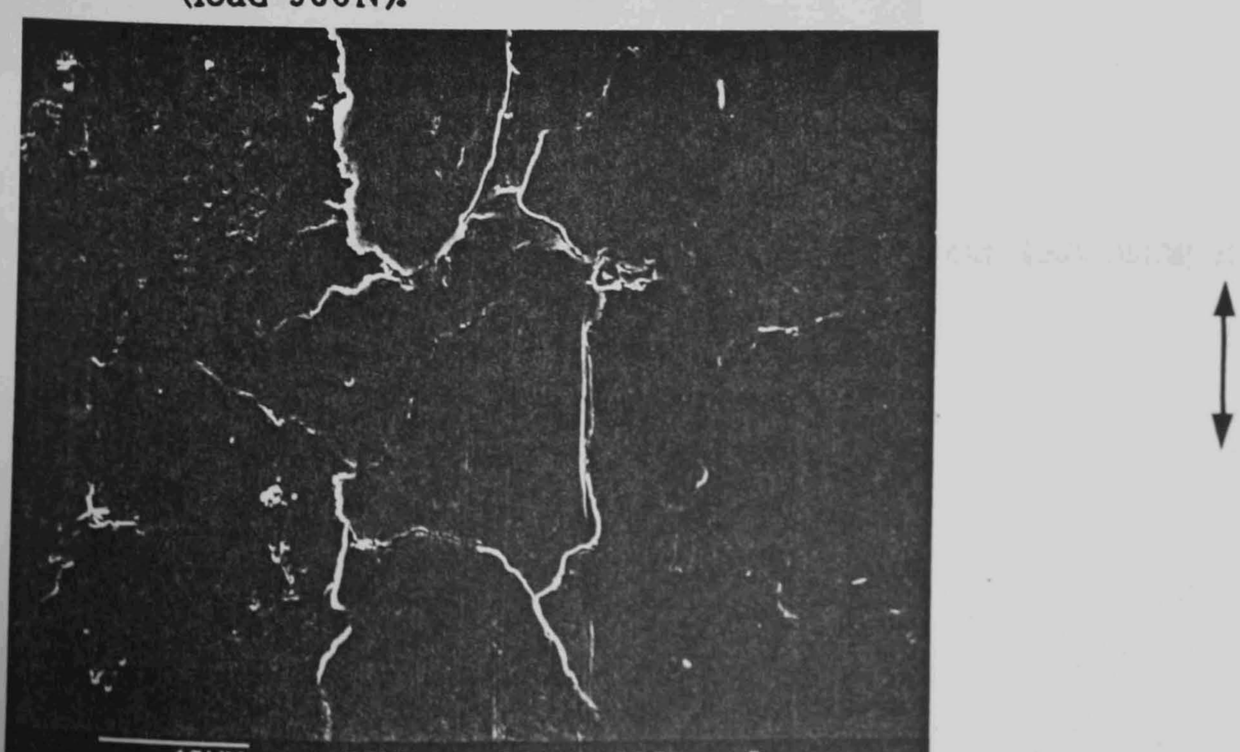


Fig.88 SEM appearance for the pin worn after 5 hour test using RL48 at 80^{o}C,

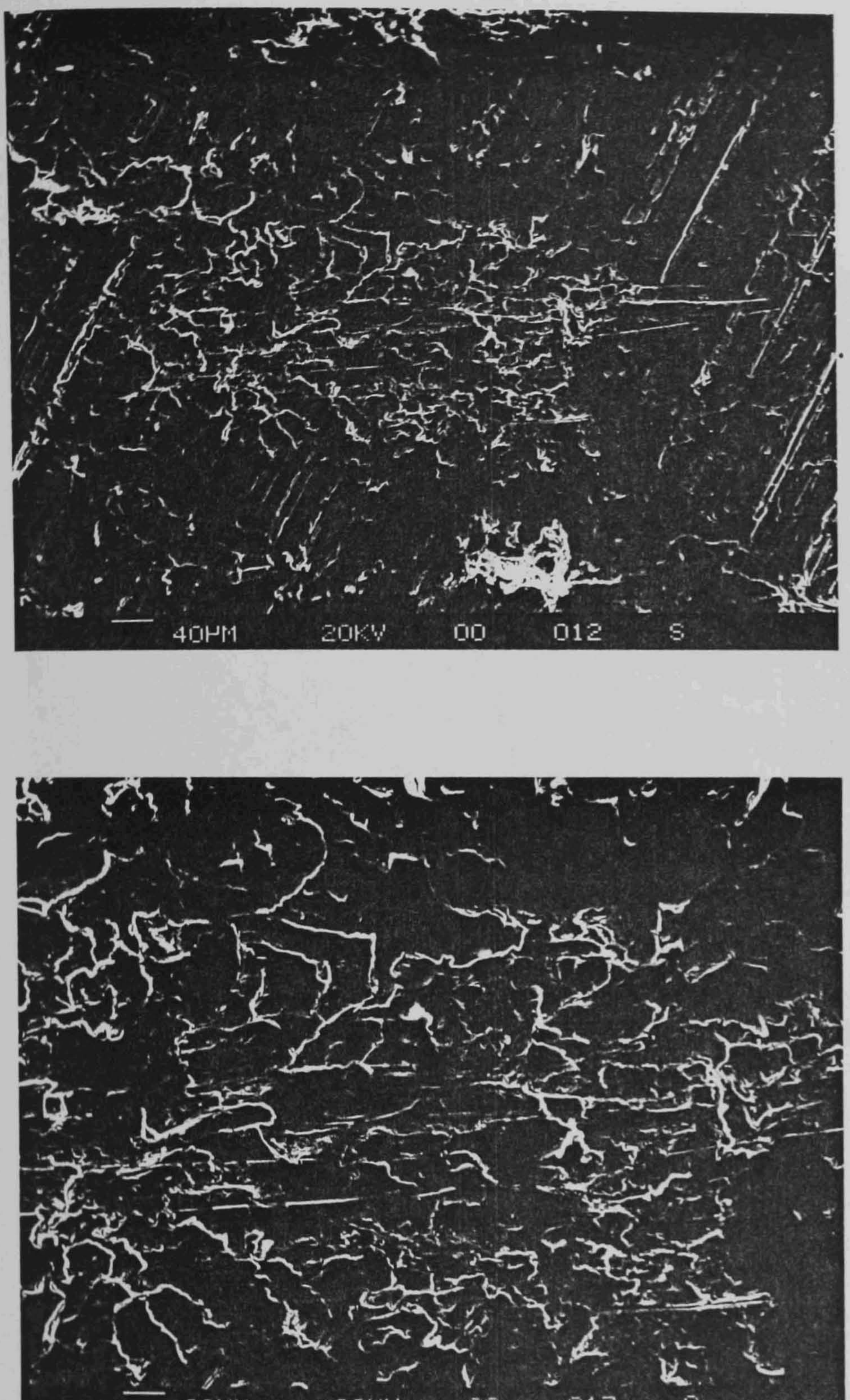


Fig.89 SEM appearance for the flat worn after 5 hour test using RL48 at 80^{o}C, (load 900N).

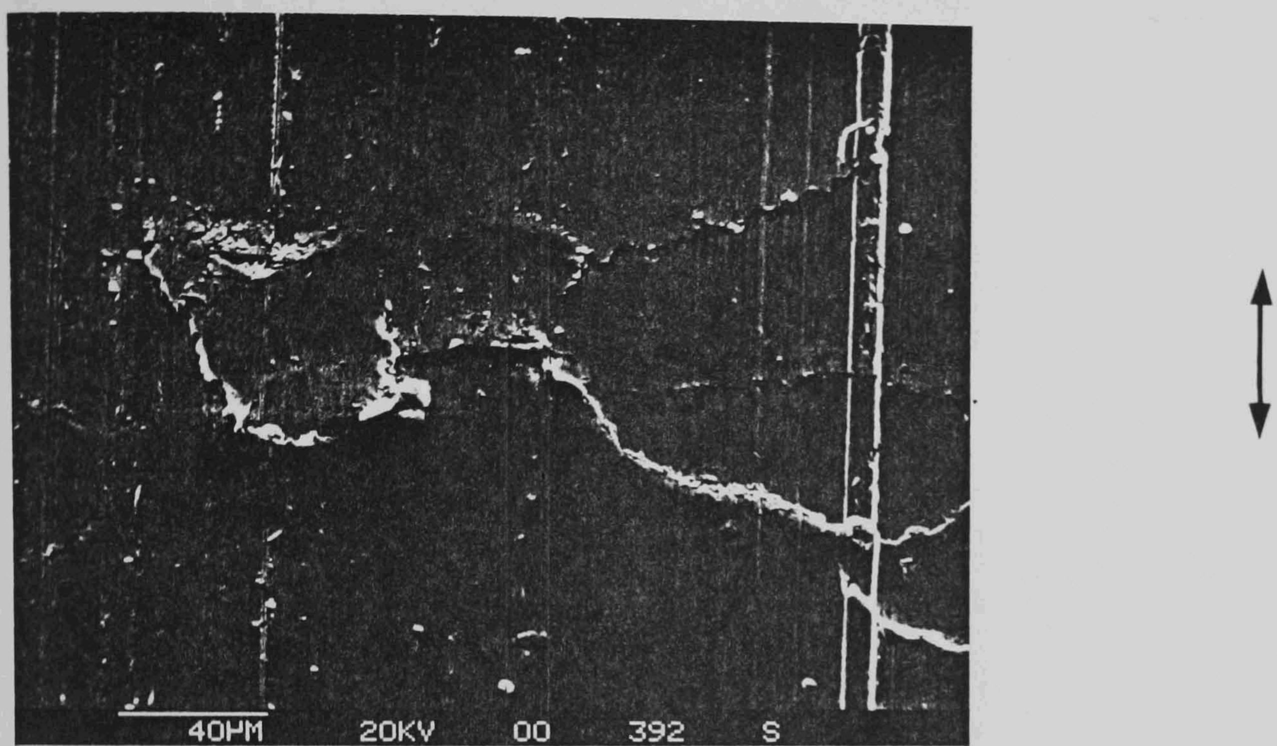


Fig.90 SEM appearance for the pin worn after 8 hour test using RL48 at 80^{o}C, (load 900N).

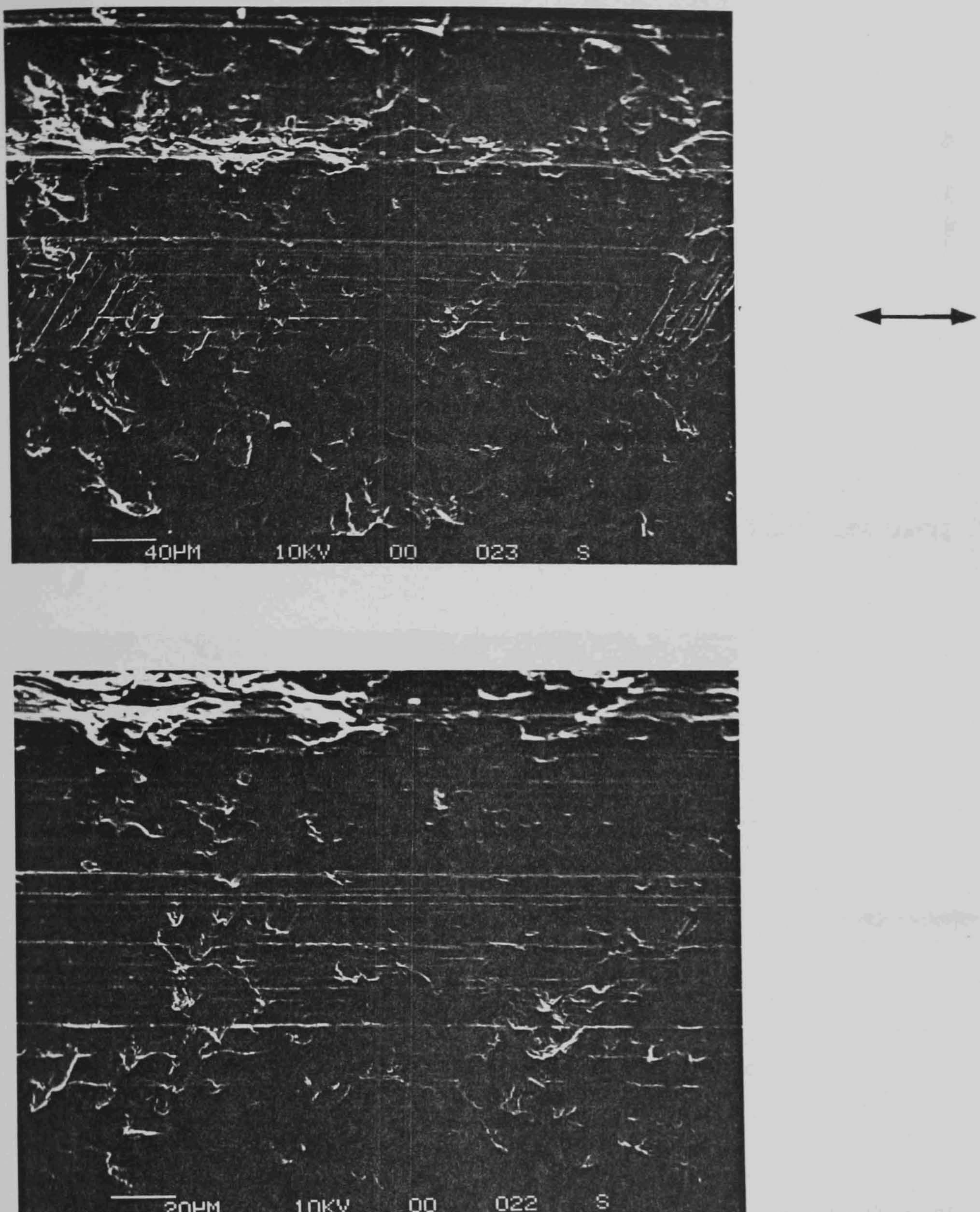


Fig.91 SEM appearance for the flat worn after 8 hour test using RL48 at $80^{o}C$, (load 900N).

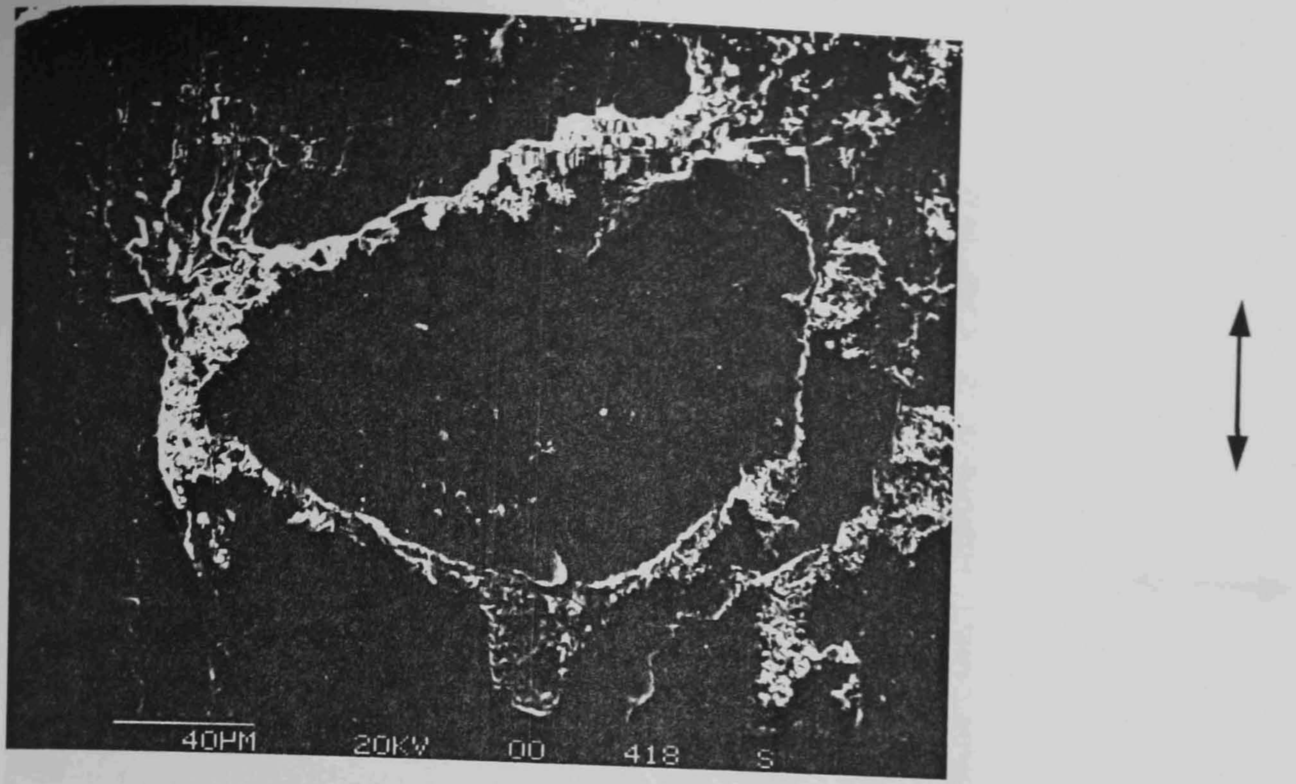


Fig.92 SEM appearance for the pin worn after 10 hour test using RL48 at 80^{o}C, (load 900N).

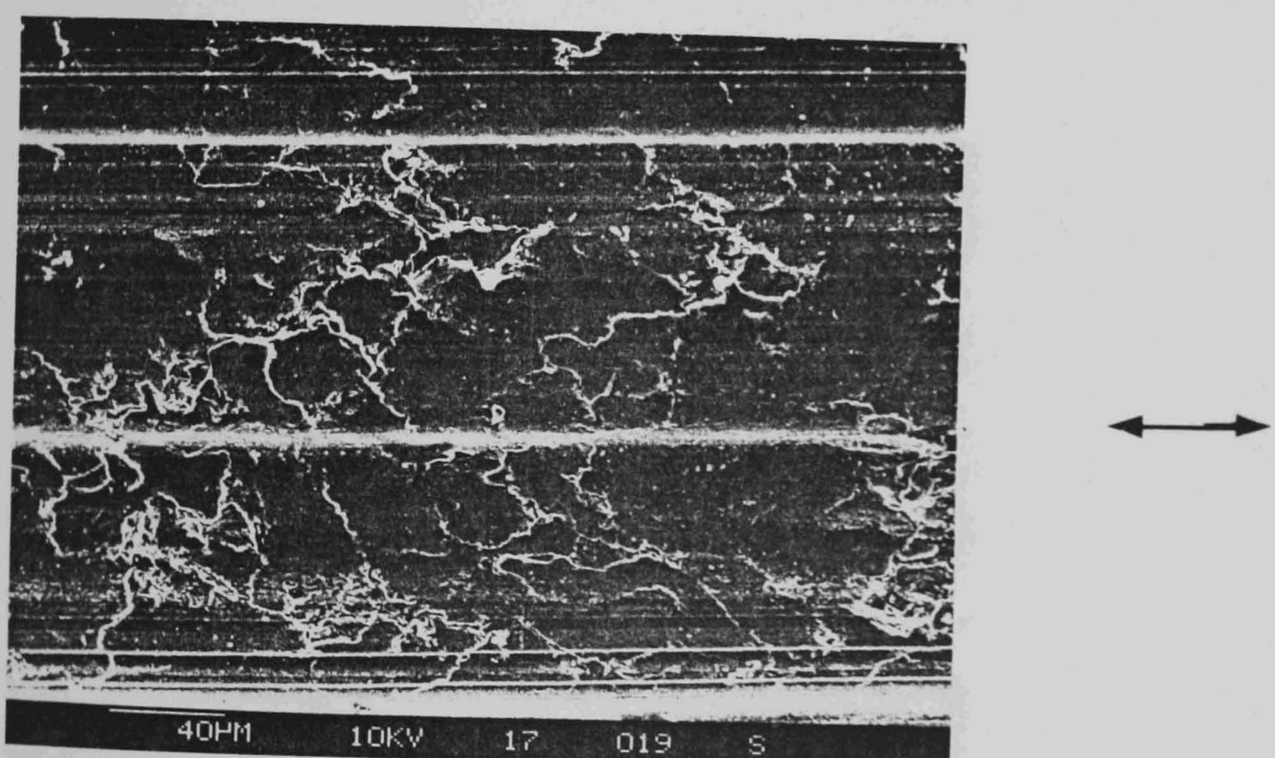


Fig.93 SEM appearance for the flat worn after 10 hour test using RL48 at 80^{o}C, (load 900N).

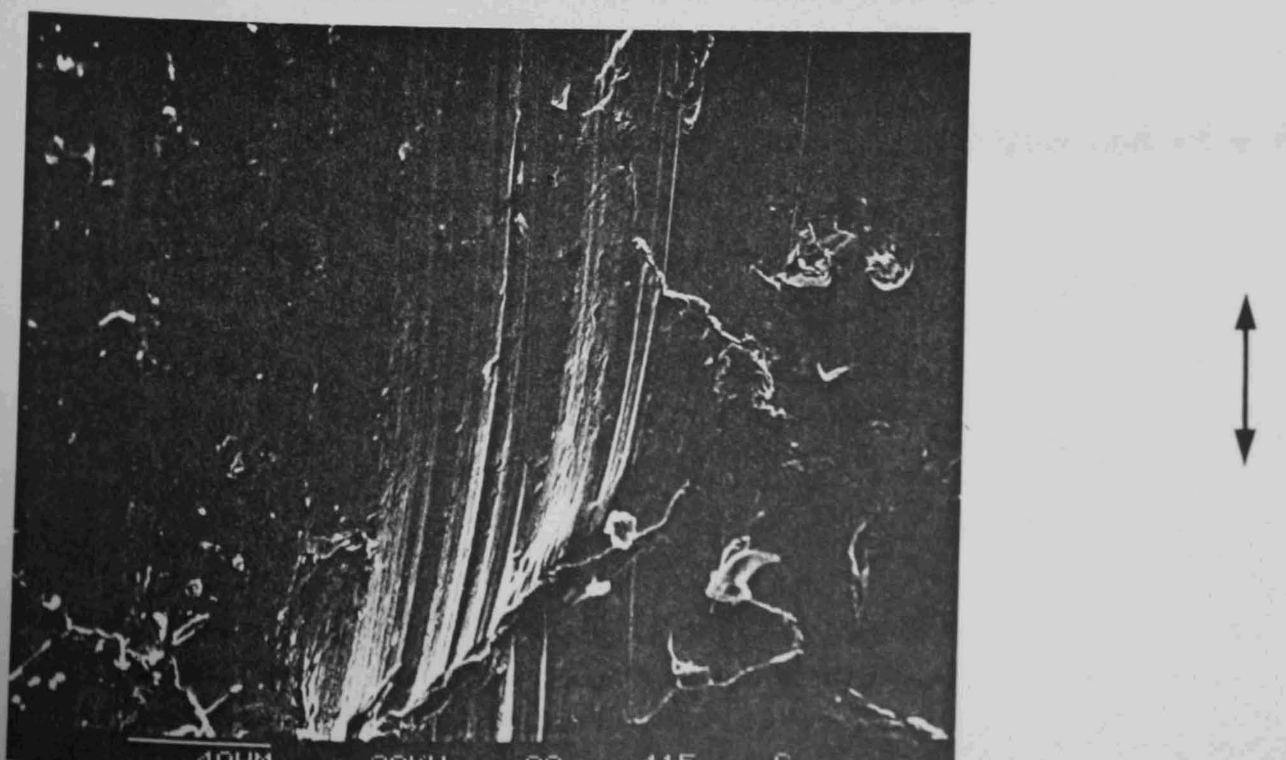


Fig.94 SEM appearance for the pin worn after 20 hour test using RL48 at 80^{o}C,

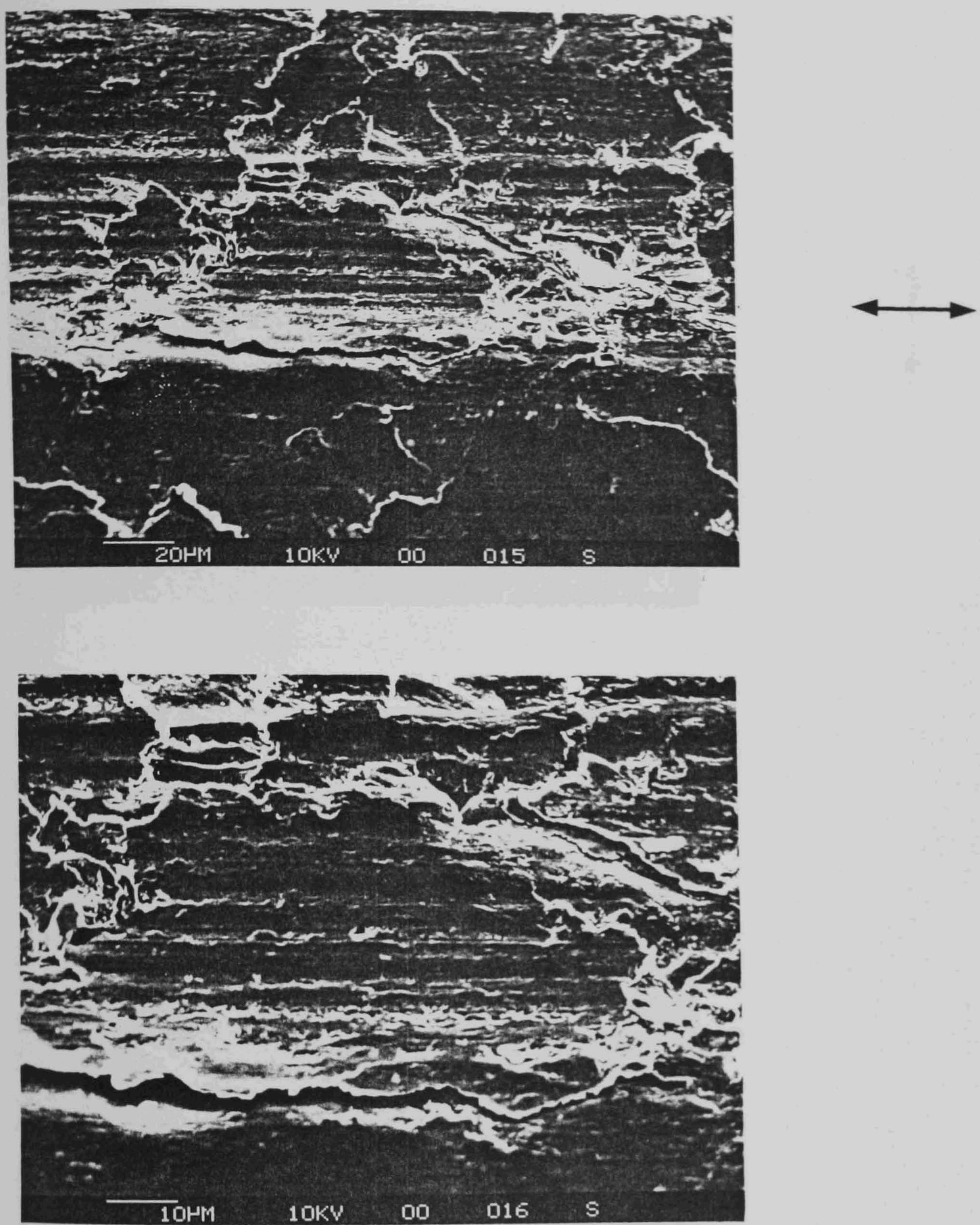
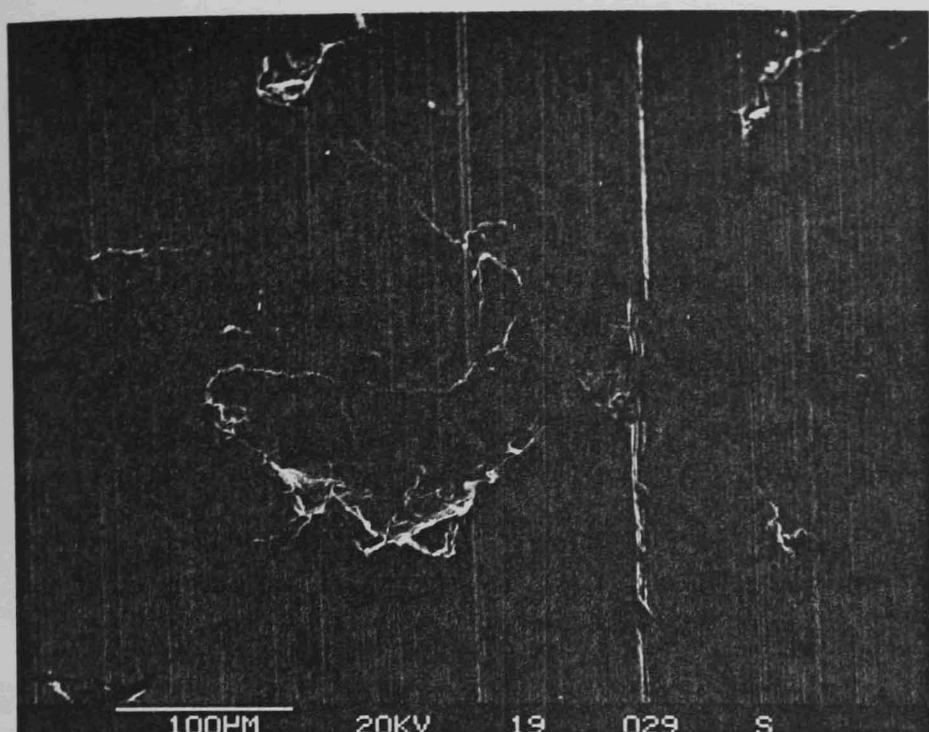


Fig.95 SEM appearance for the flat worn after 20 hour test using RL48 at $80^{o}C$, (load 900N).



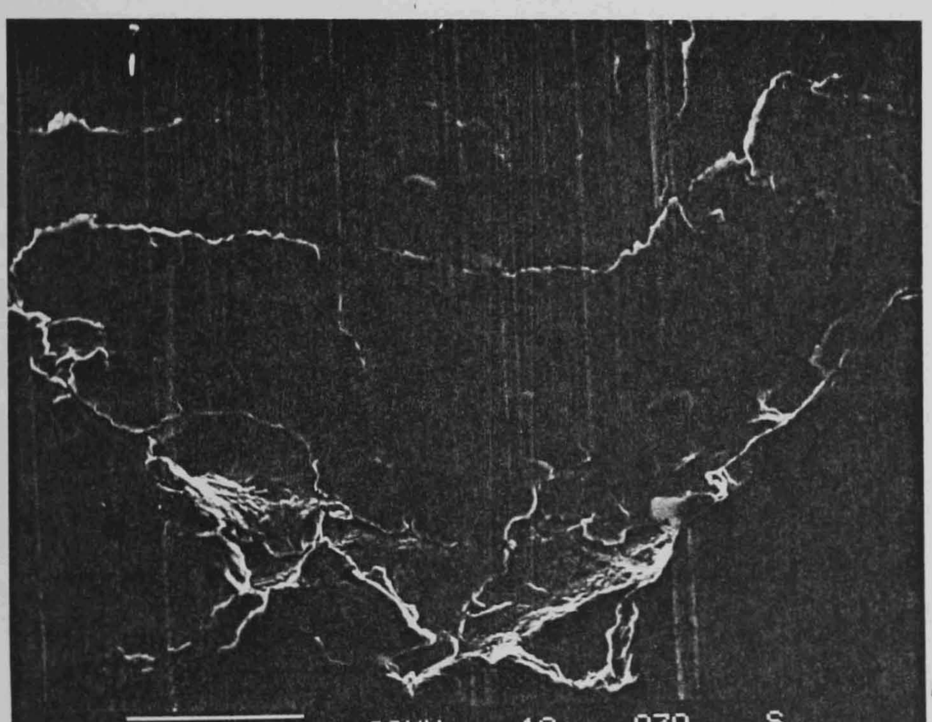


Fig.96 SEM appearance for the pin worn after 100 hour test using RL48 at 80^{o}C, (load 900N).

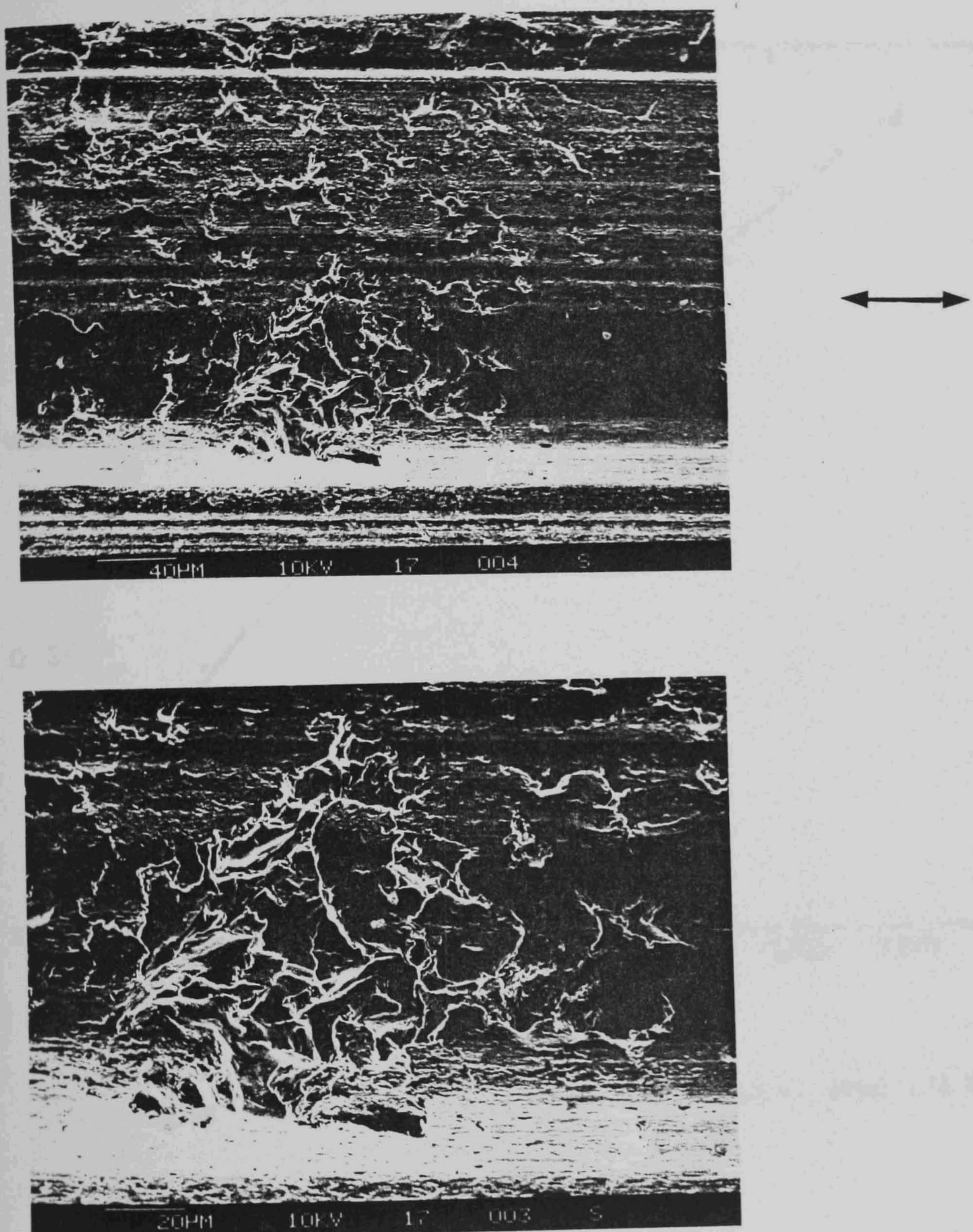


Fig.97 SEM appearance for the flat worn after 100 hour test using RL48 at 80°C, (load 900N).

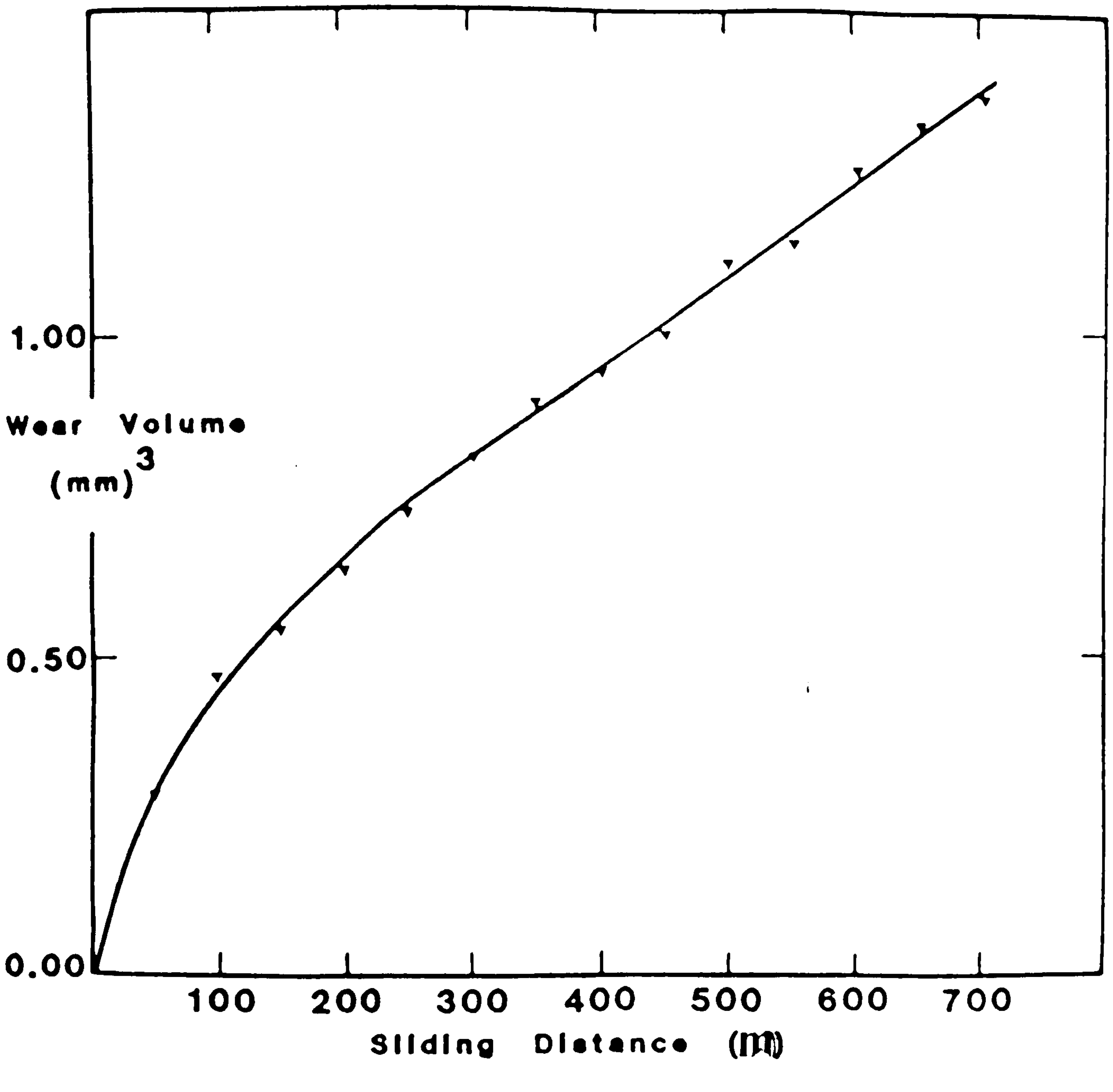


Fig.98 Wear for RL48 at 80°C, (load = 900N), after 100 hour test.

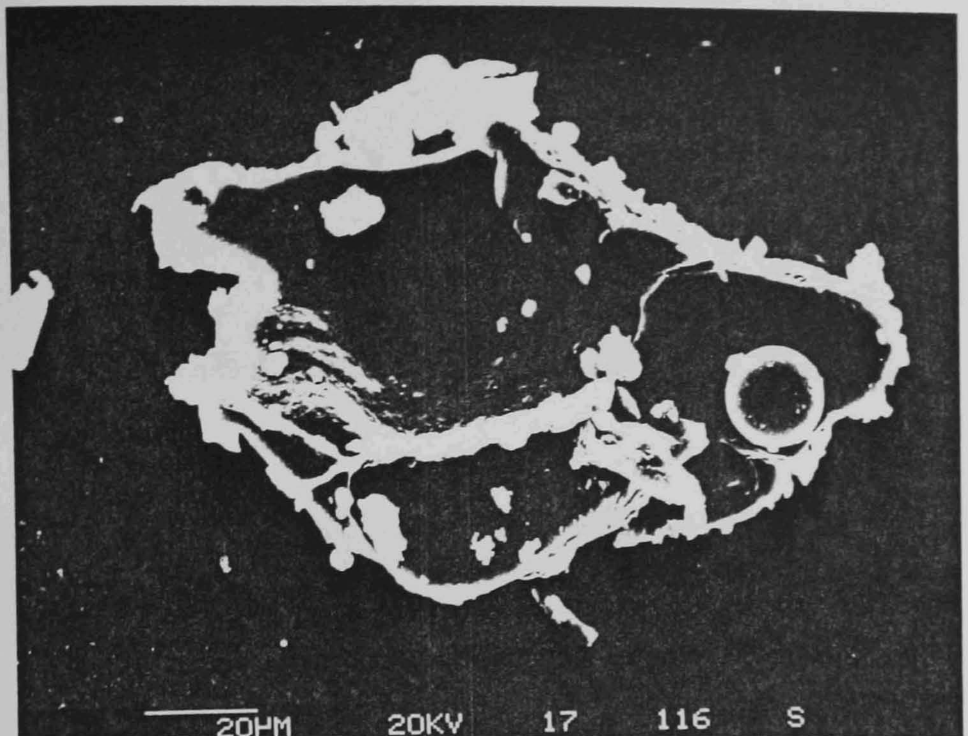


Fig.99 Debris produced from RL47 test.

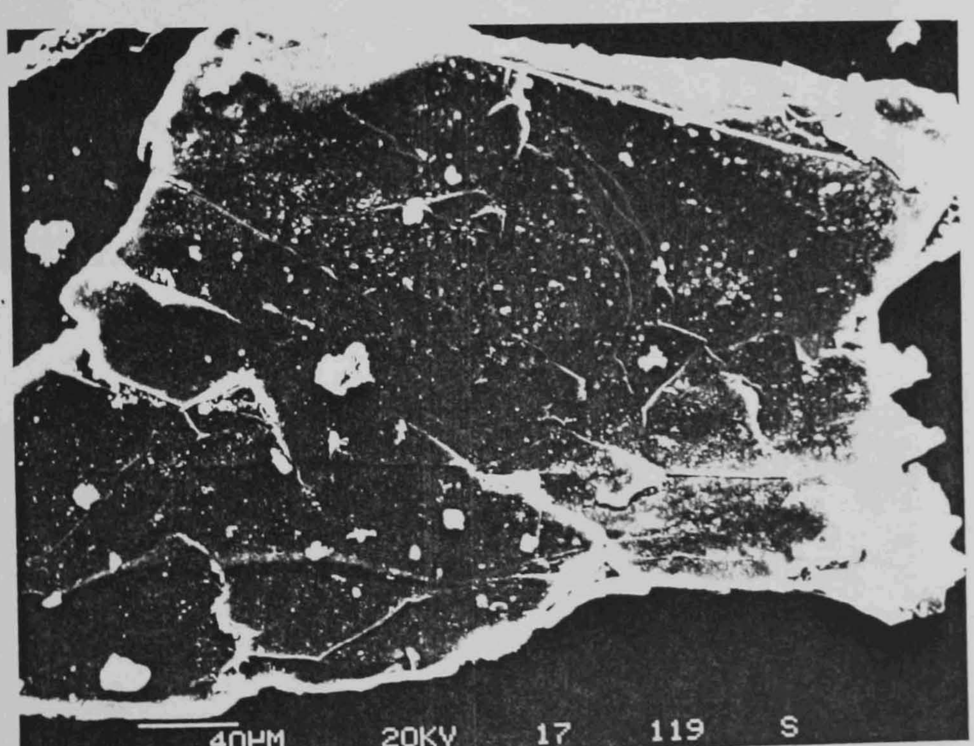


Fig.100 Debris produced from RL48 test.

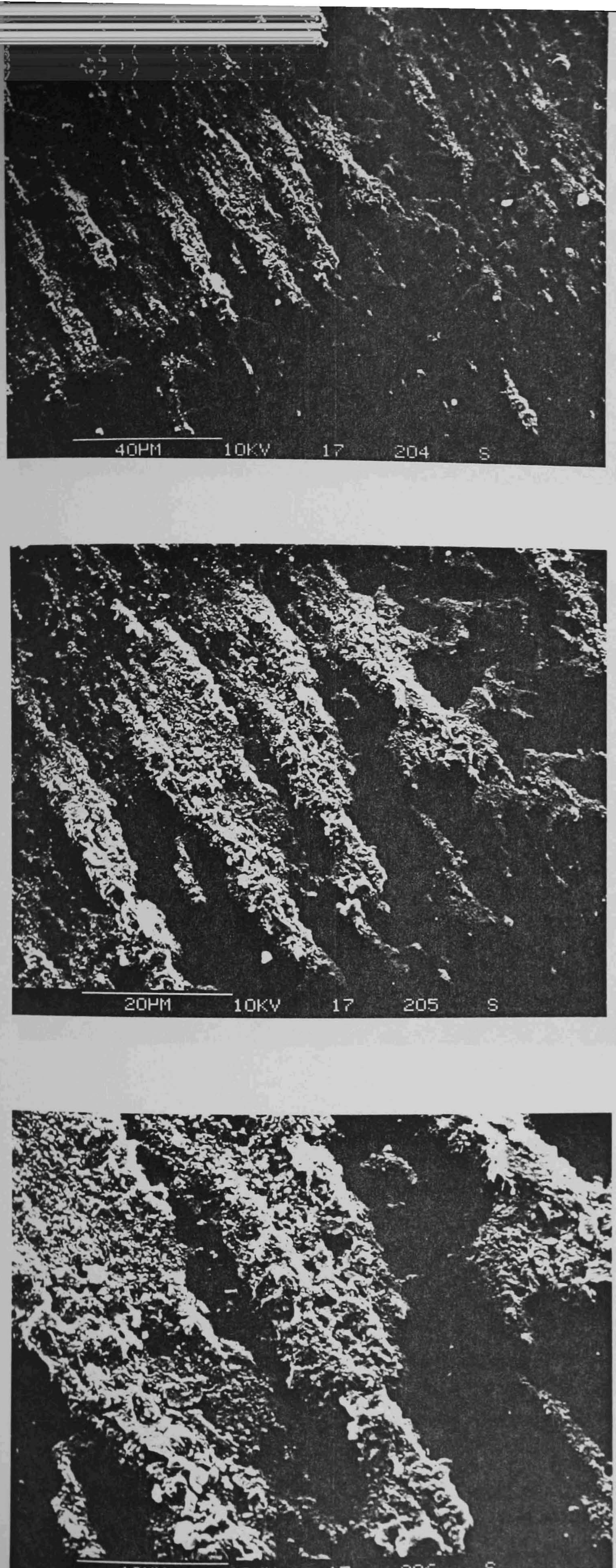
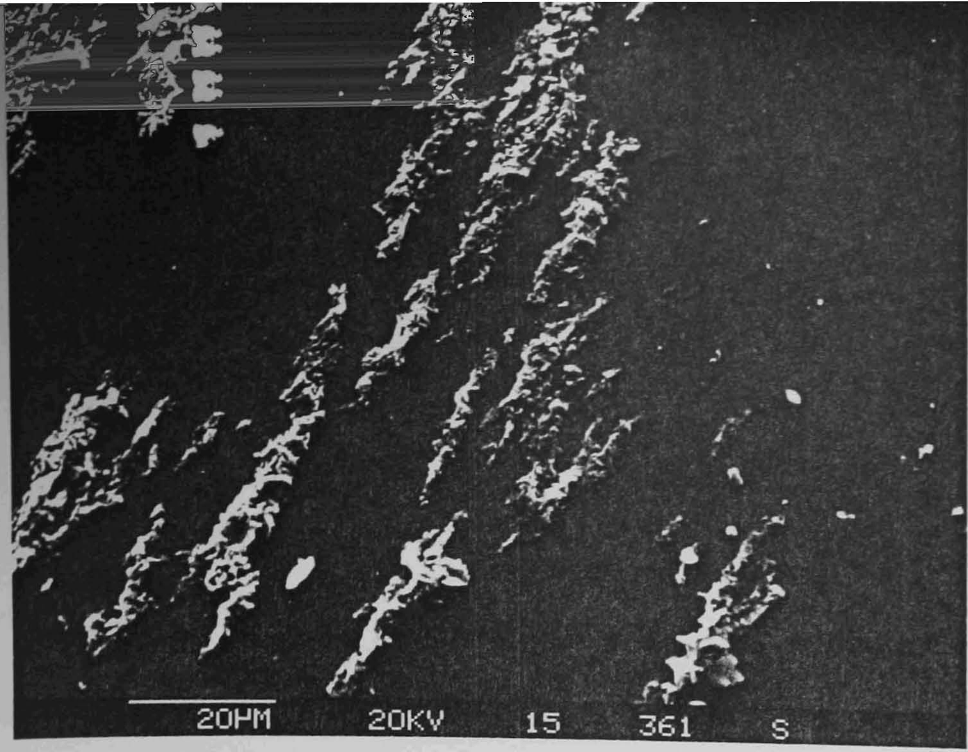
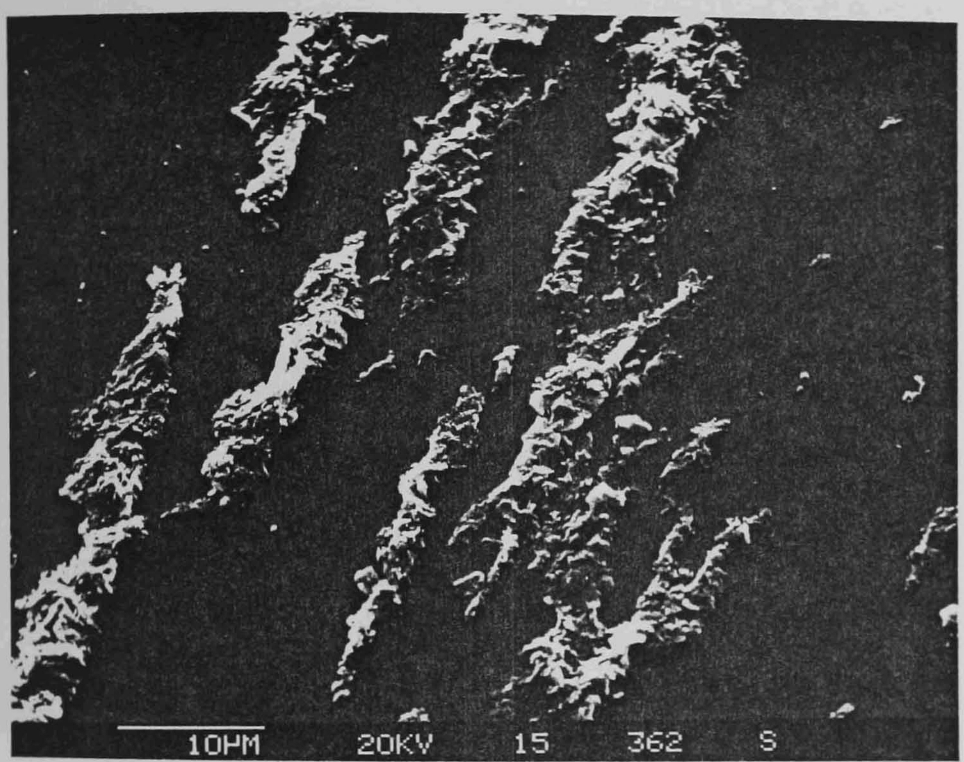


Fig.101 Debris produced after 1 hour test using RL48 at 80°C, (load = 900N).





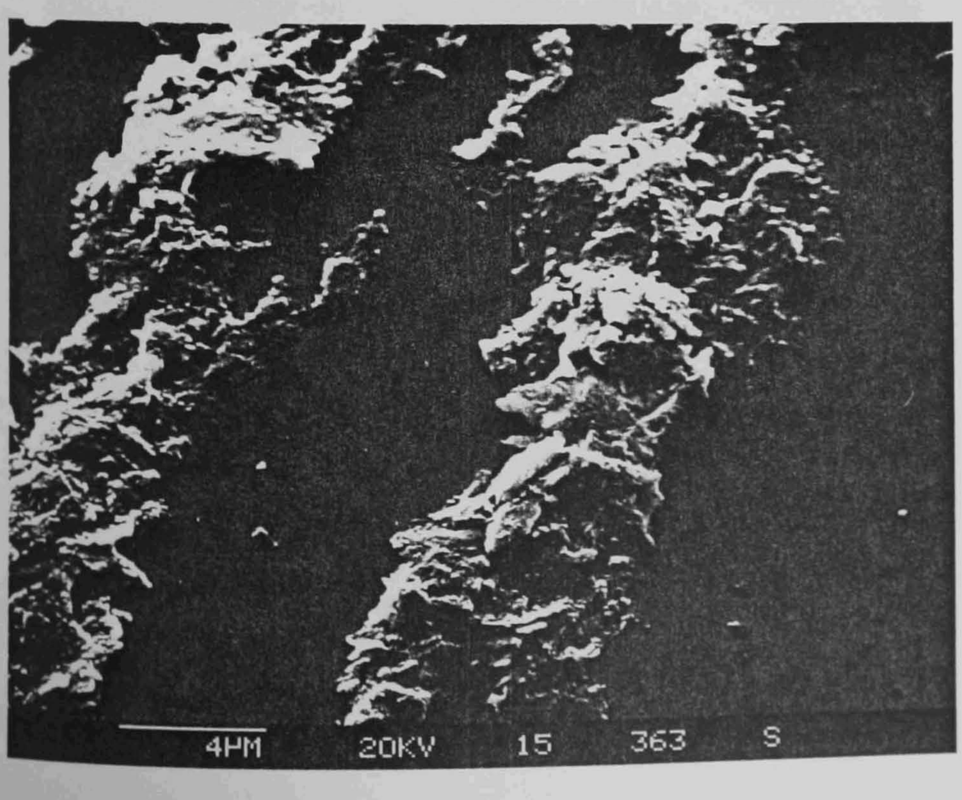
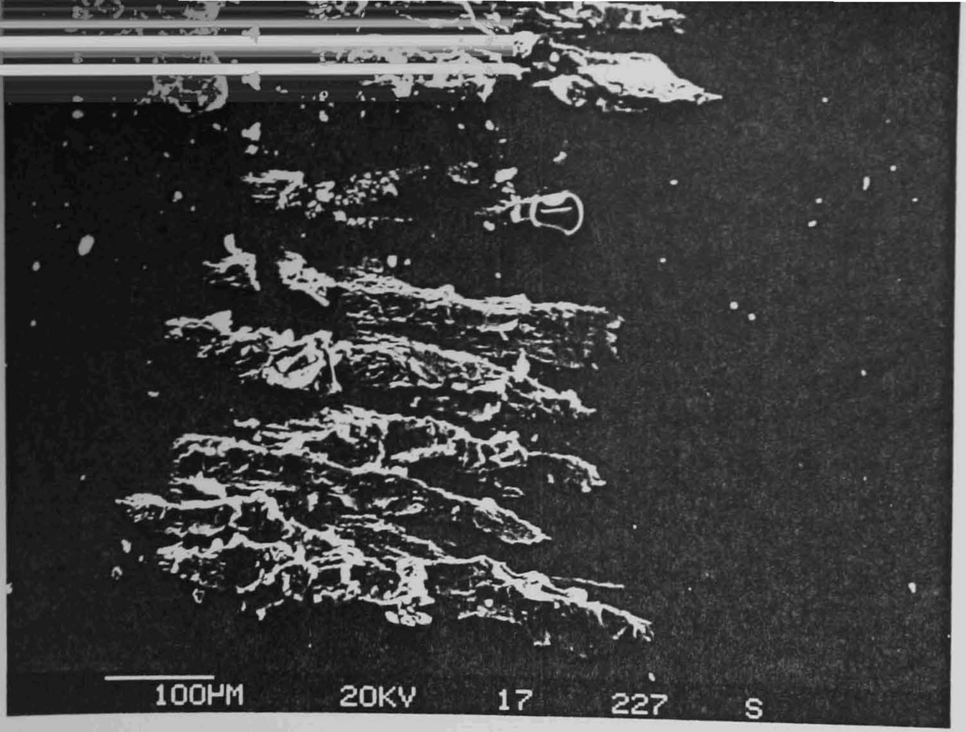
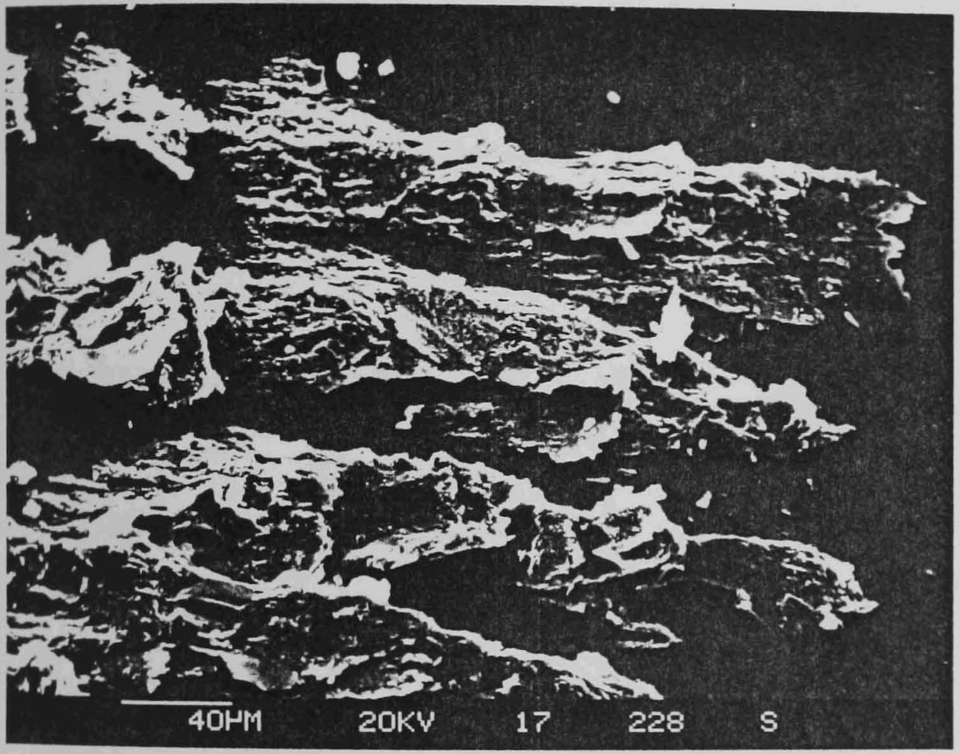


Fig.102 Debris produced after 8 hour test using RL48 at 80°C, (load = 900N).





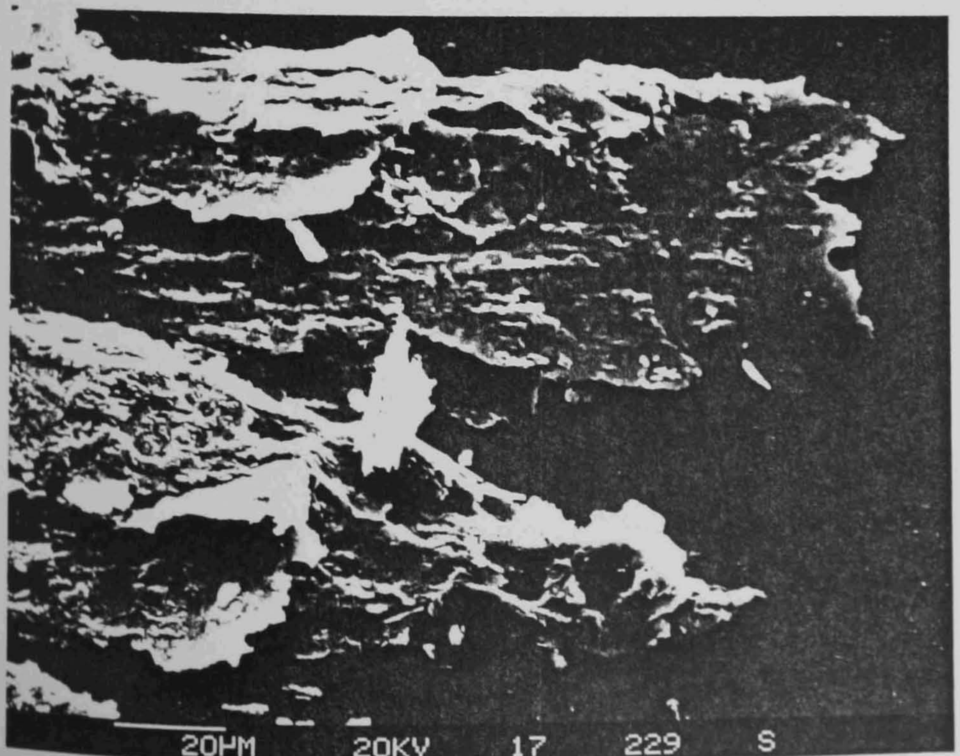


Fig.103 Debris produced after 100 hour test using RL48 at 80°C, (load = 900N).

METADI® II, manufactured diamond

Fig.104 Samples of 1/4μm diamond paste used in the bore polished test.

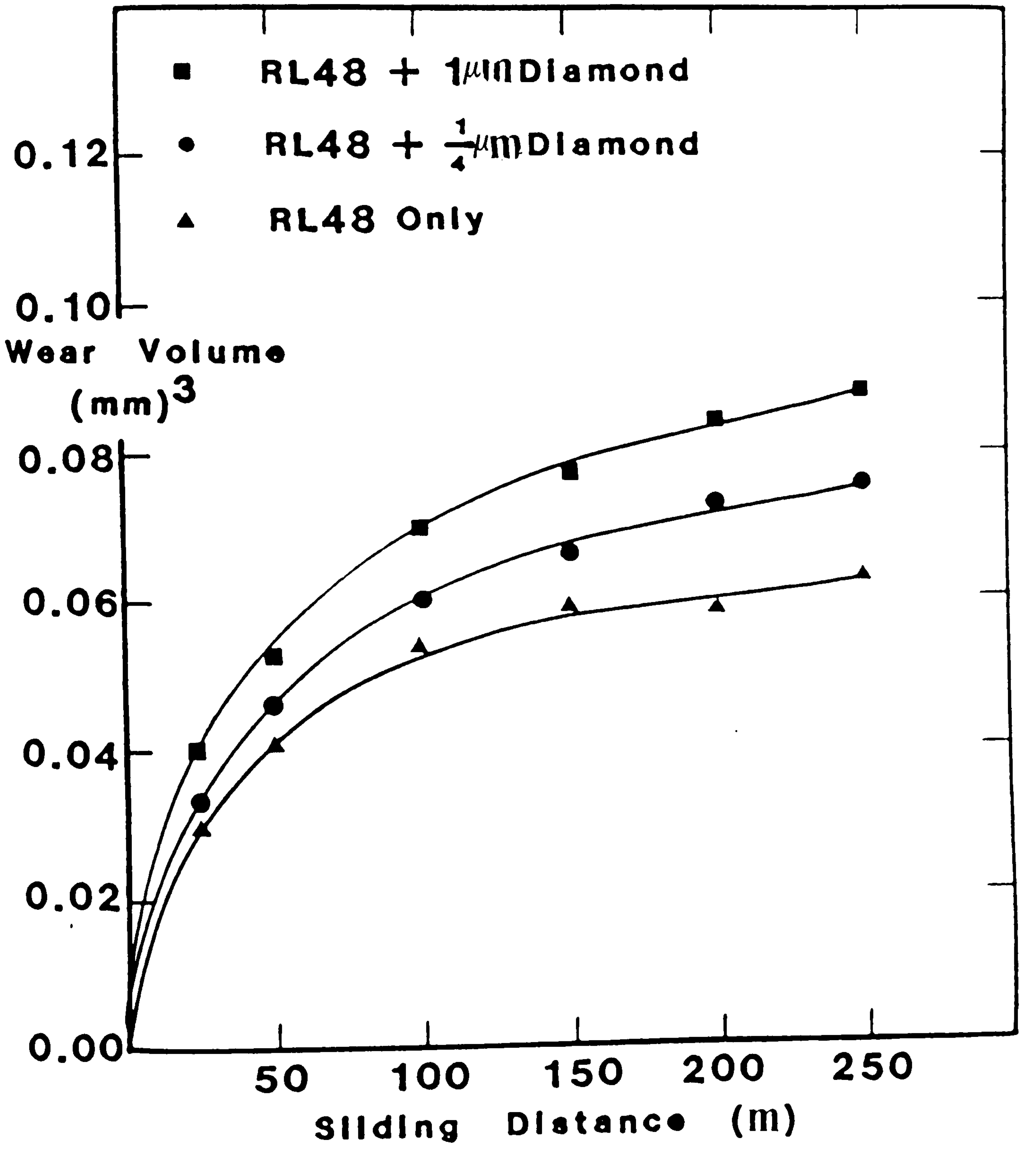


Fig.105 Wear volume versus sliding distance.

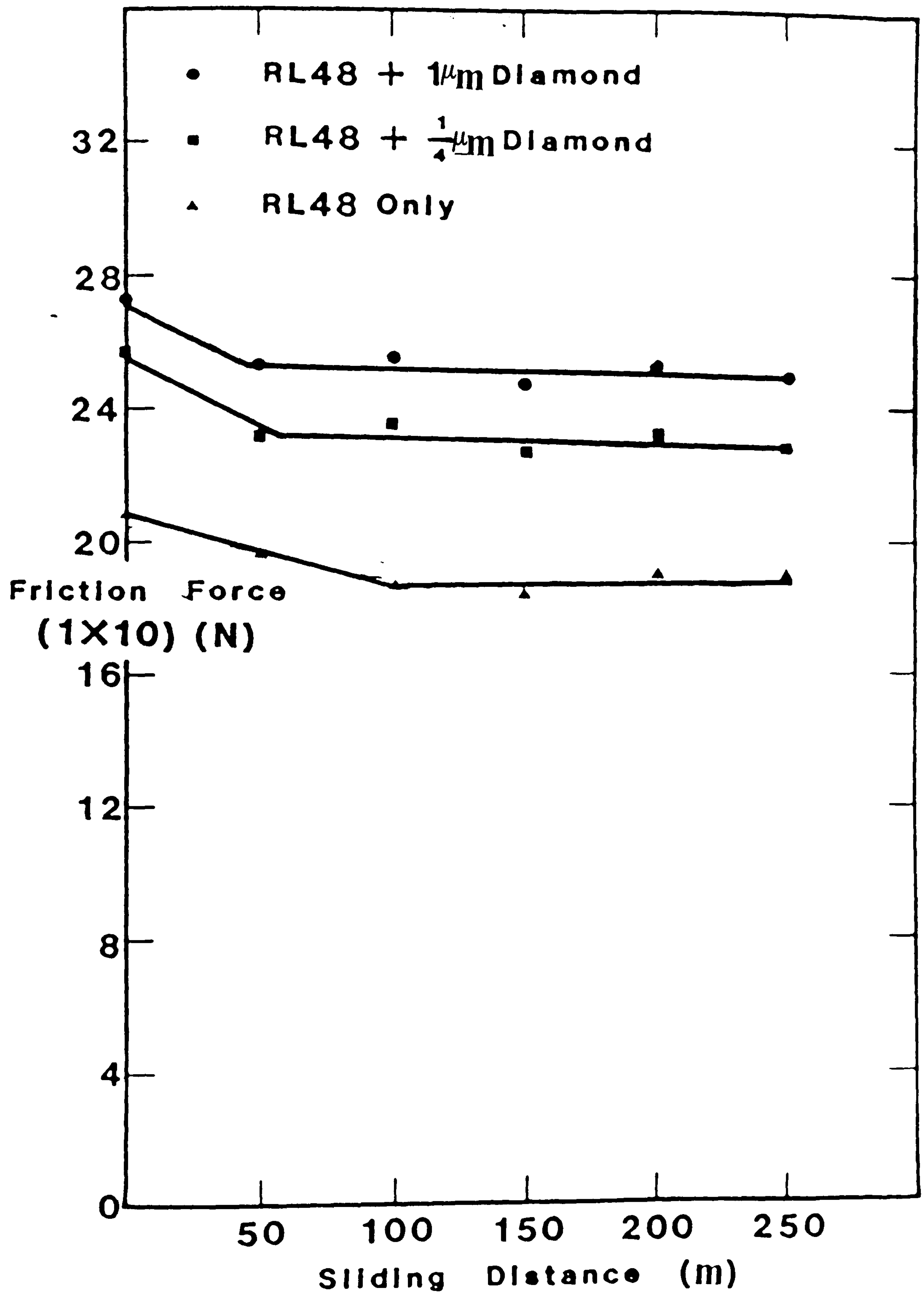


Fig.106 Friction force versus sliding distance.

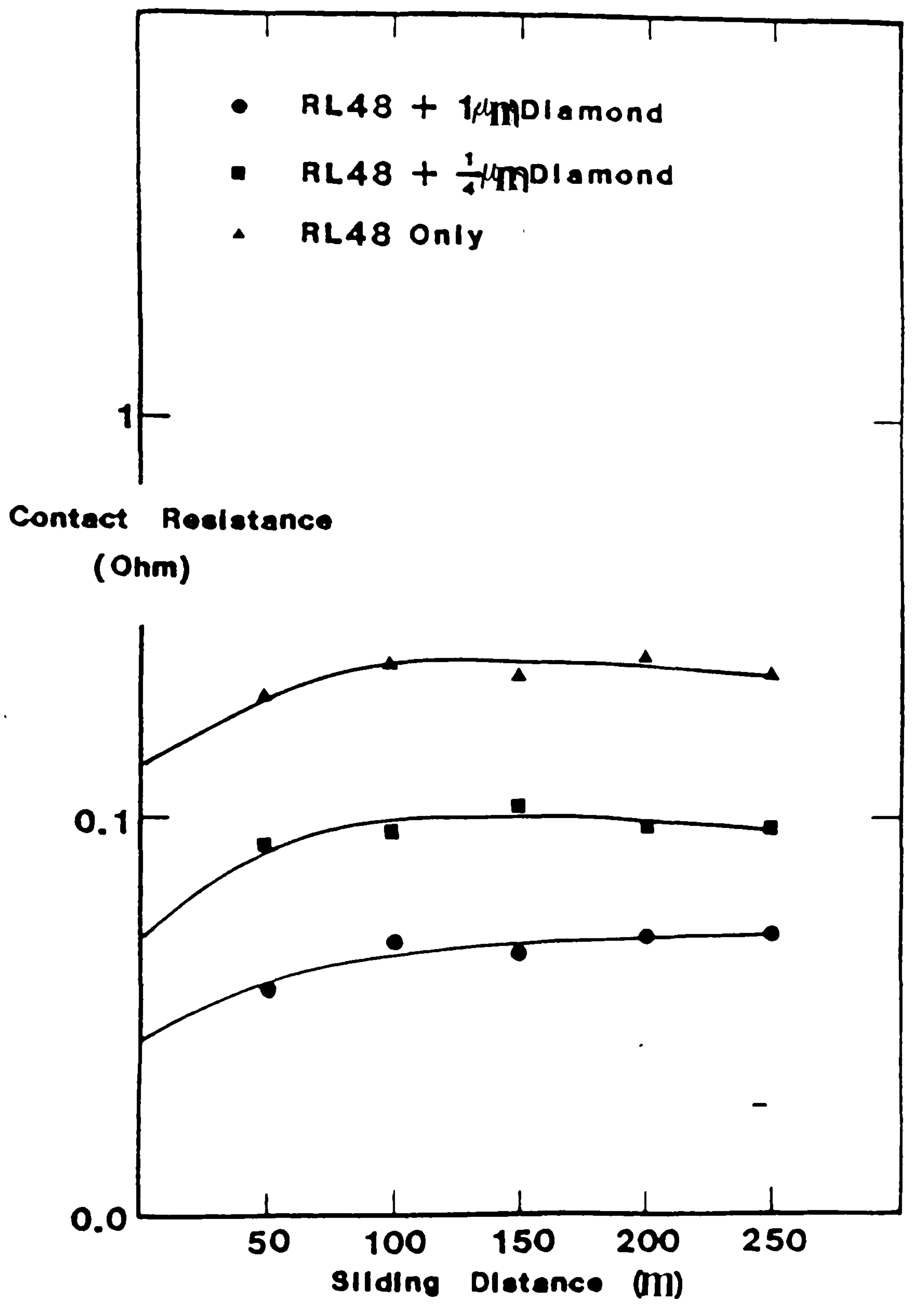


Fig.107 Contact resistance versus sliding distance.

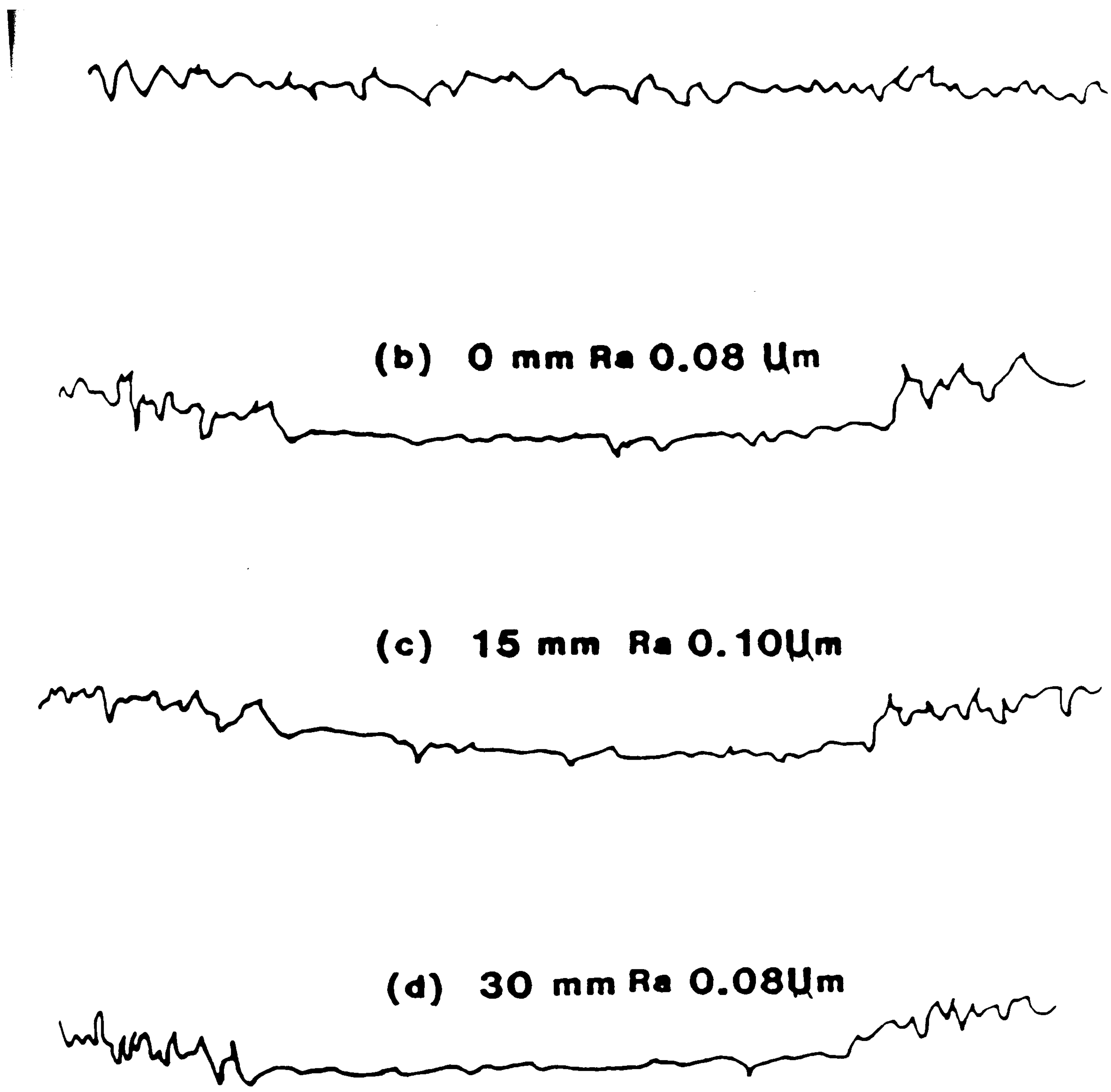


Fig.108 C.L.A. values for polished flat using RL48 + 1/4μ diamond paste.

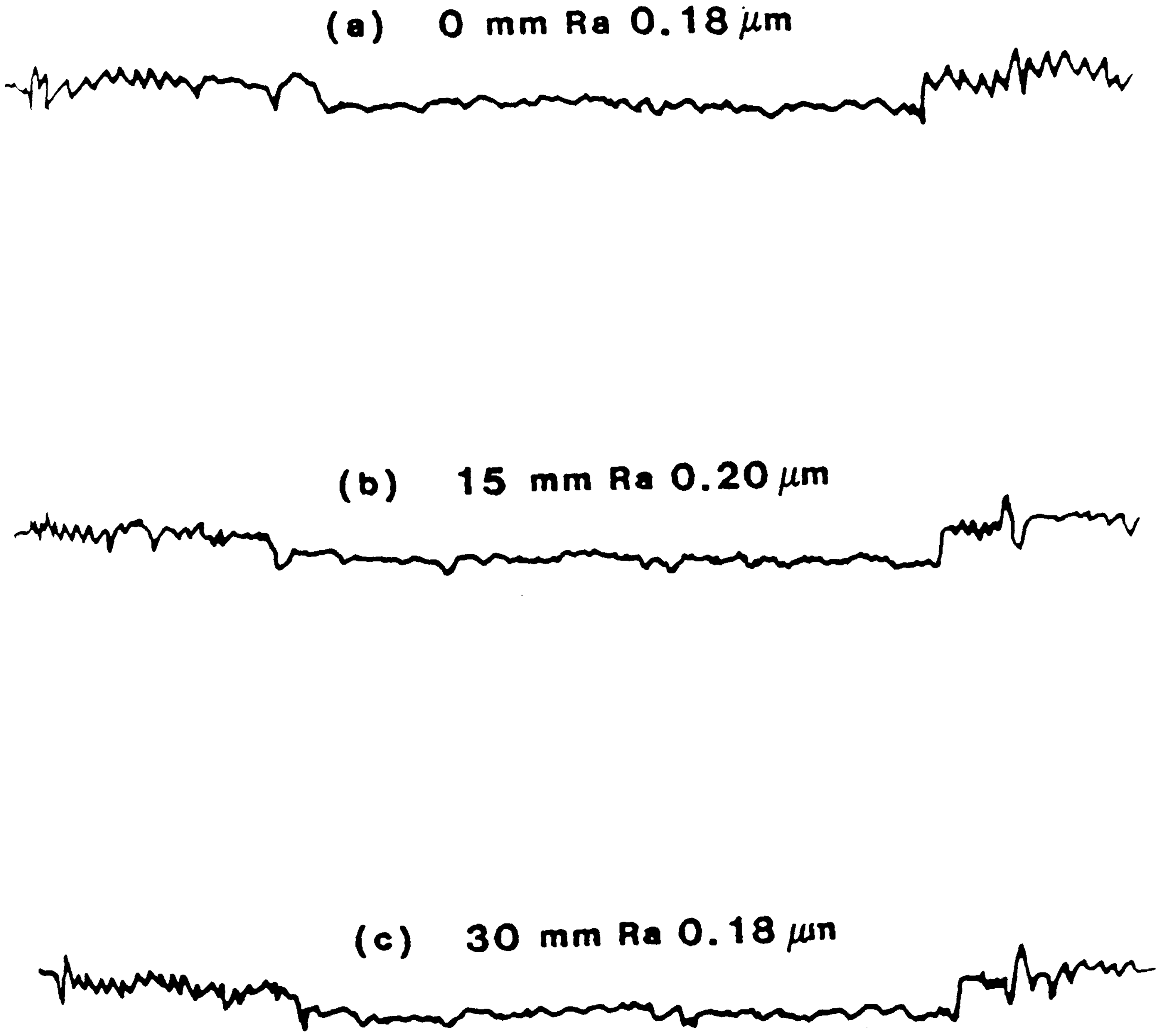


Fig.109 C.L.A. values for worn flat using RL48 only.

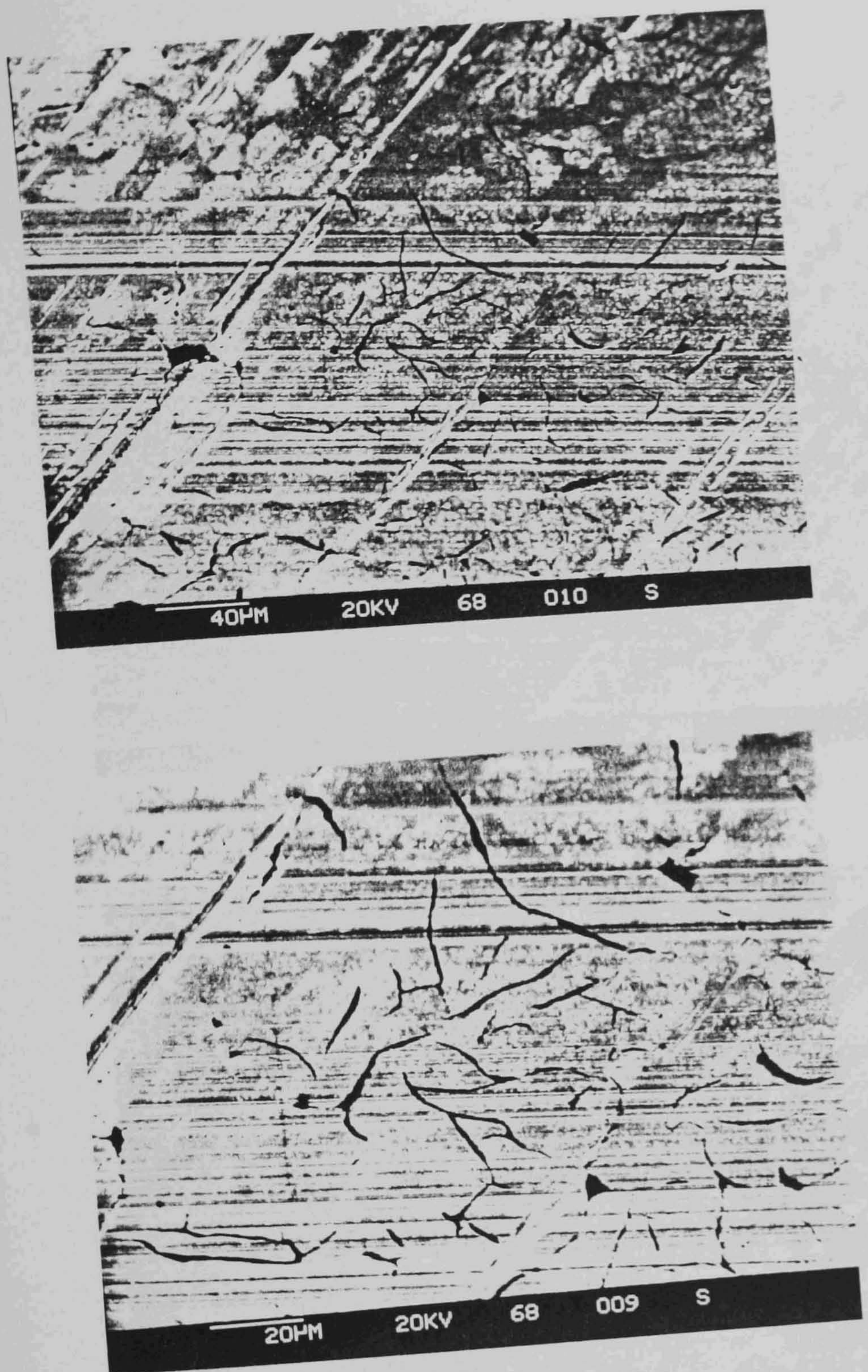


Fig.110 Bore polishing produced using diamond paste with RL48 at 80^{o}C, (load = 900N).

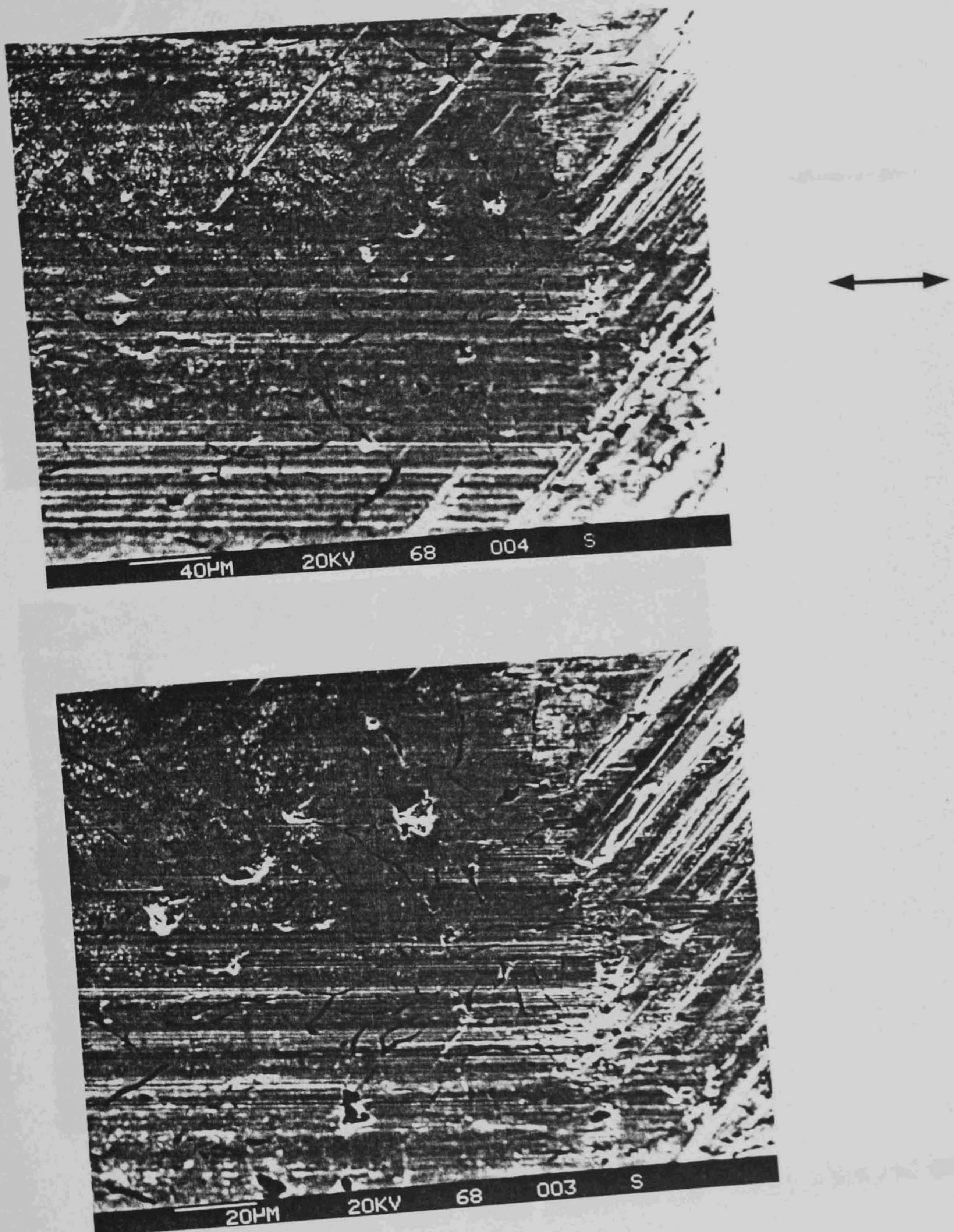


Fig.111 End of wear track for polished flat using diamond paste.

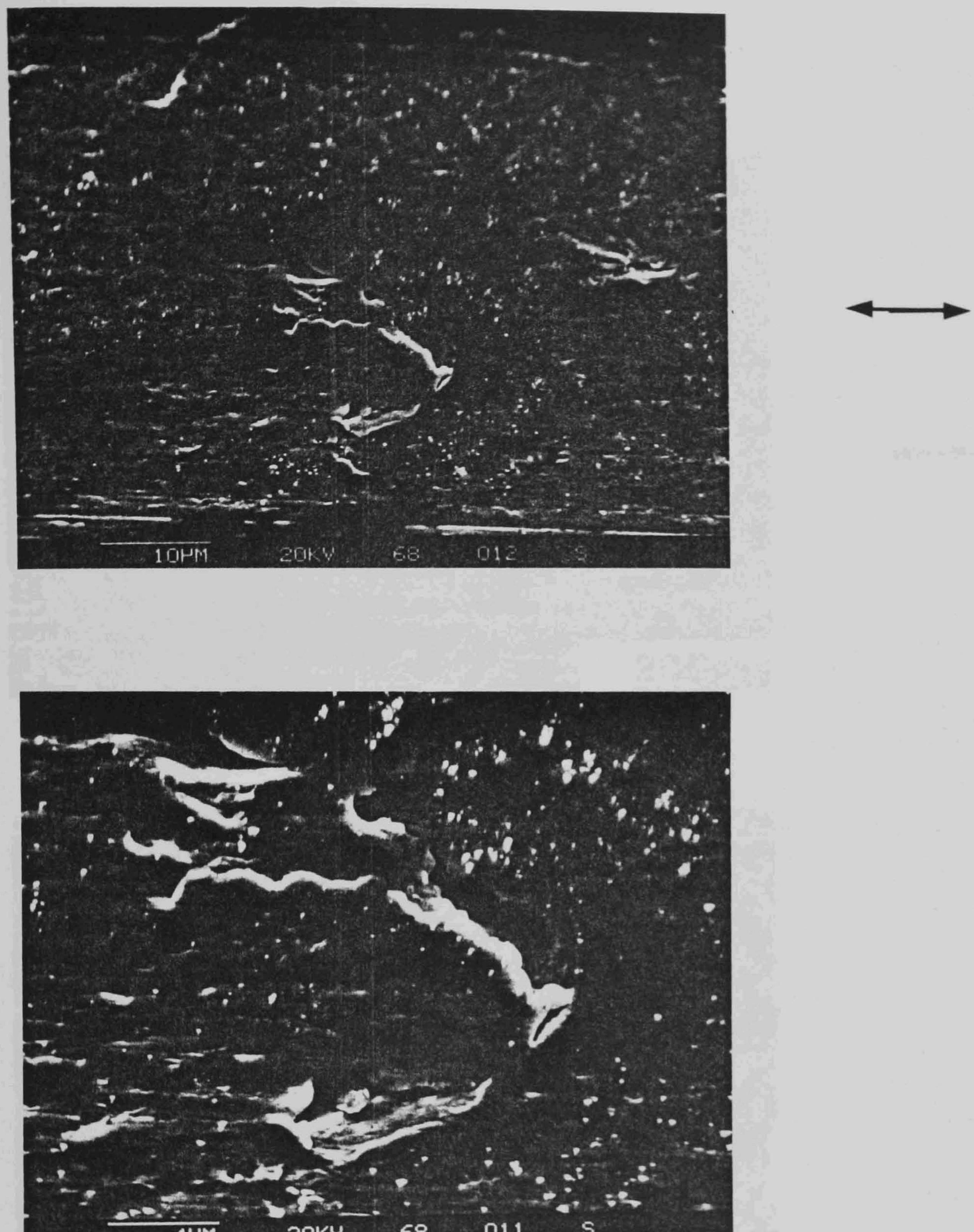
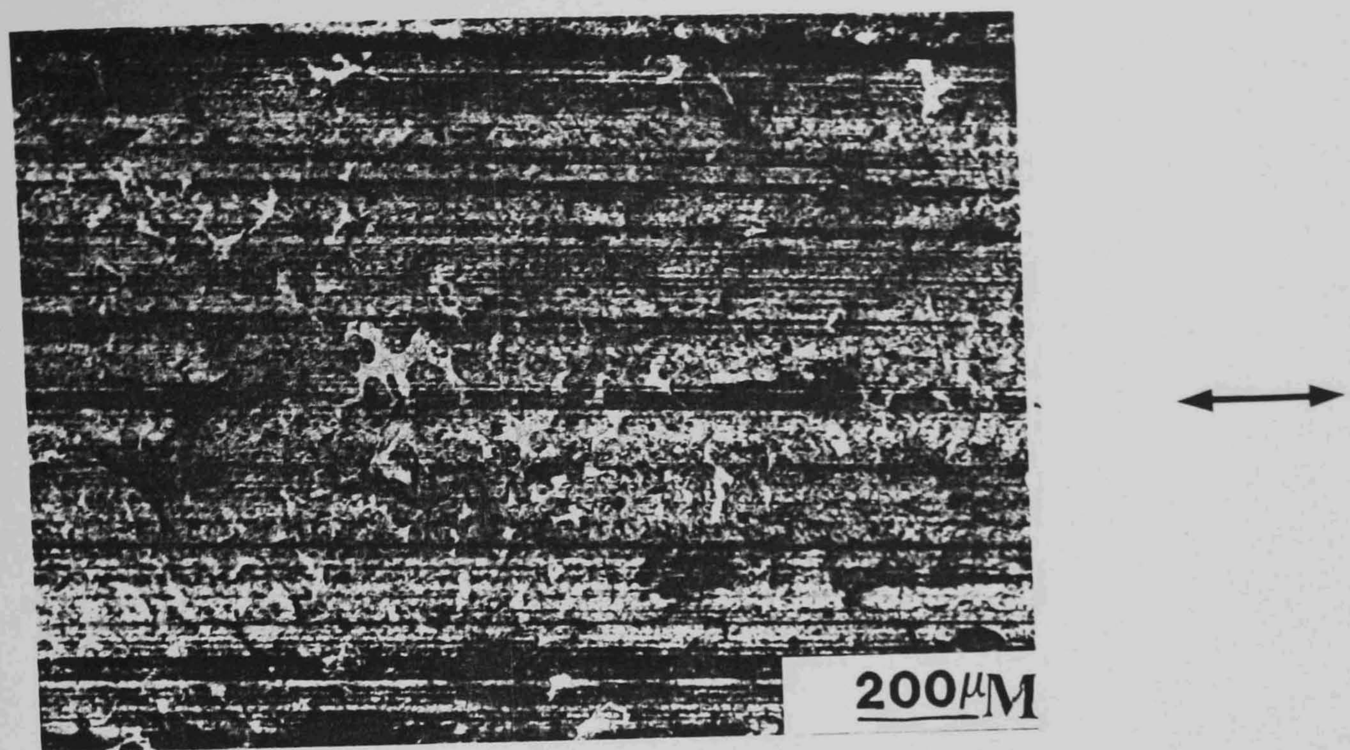


Fig.112 Delamination wear from polished flat using diamond paste.



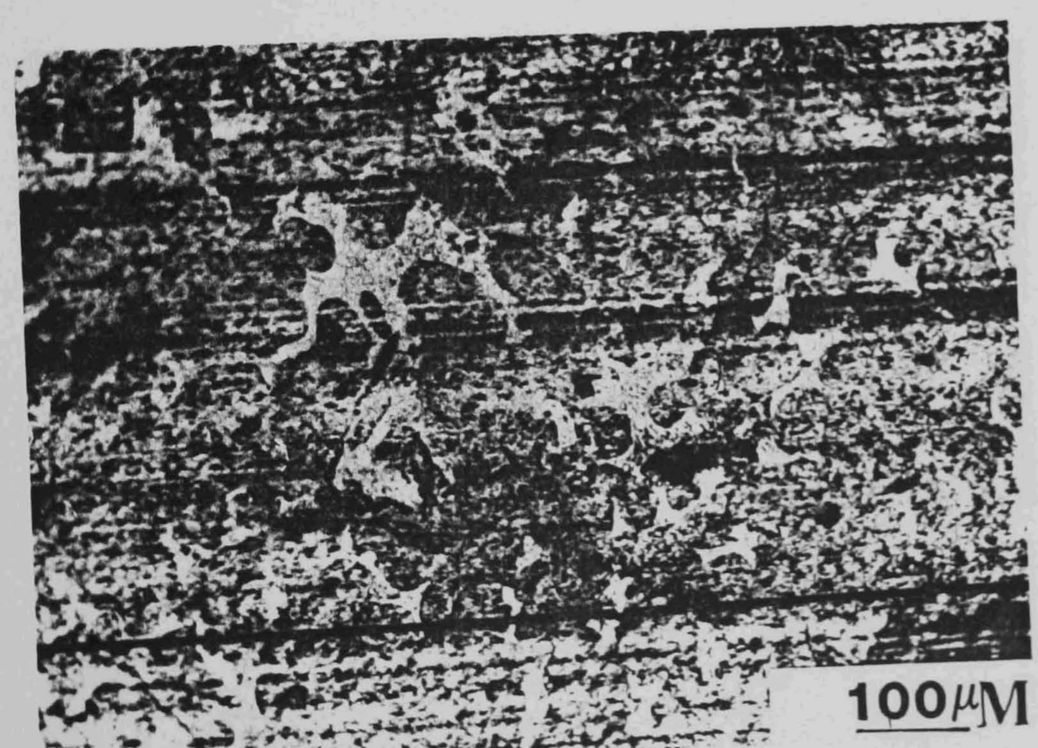


Fig.113 Differential abrasive wear of the polished flat using diamond paste.

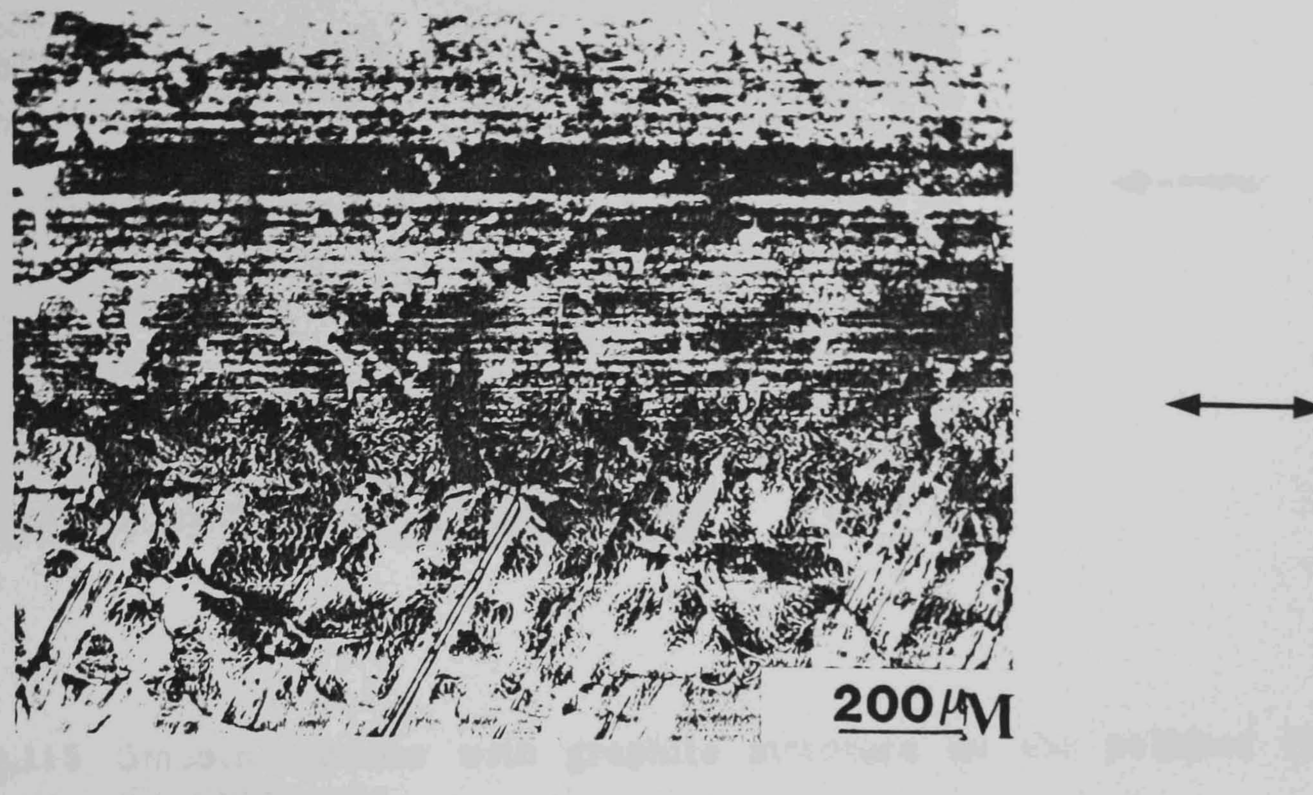


Fig.114 Mottled area or thumb print structure of the polished flat using diamond paste.

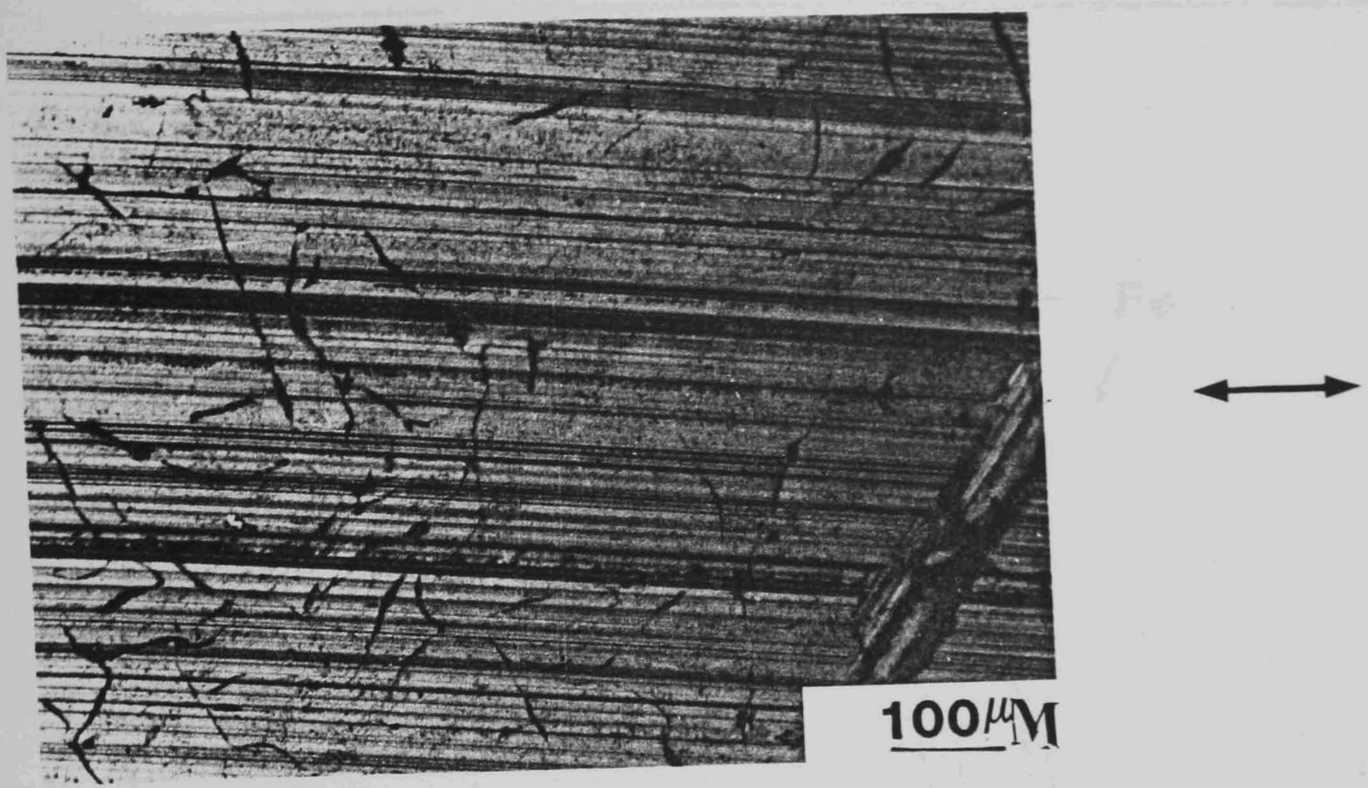


Fig.115 Smooth surface with graphite structure on the polished flat using diamond paste.

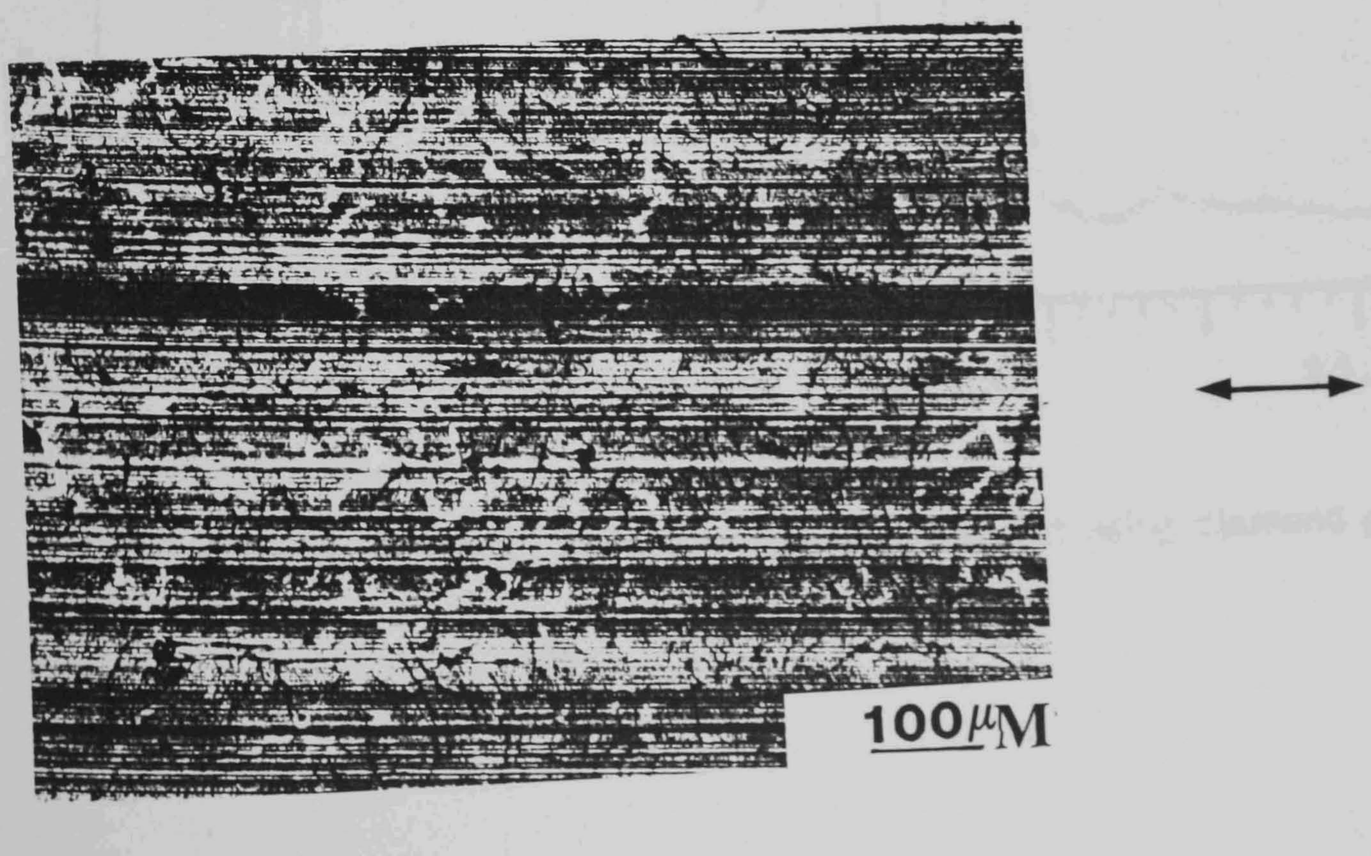


Fig.116 Etched surface on the polished flat using diamond paste

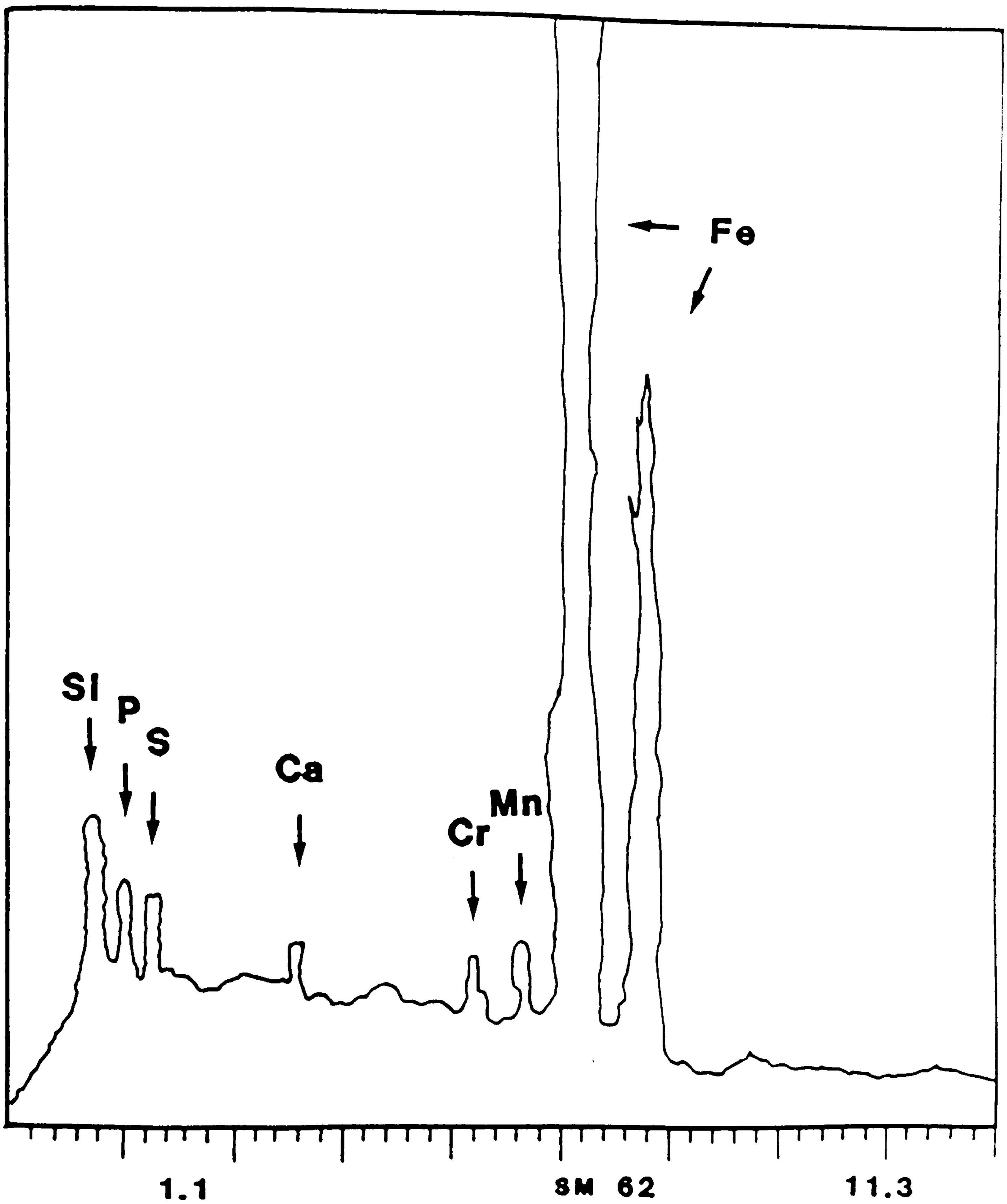


Fig.117 E.p.M.A. surface on the polished flat using diamond paste.

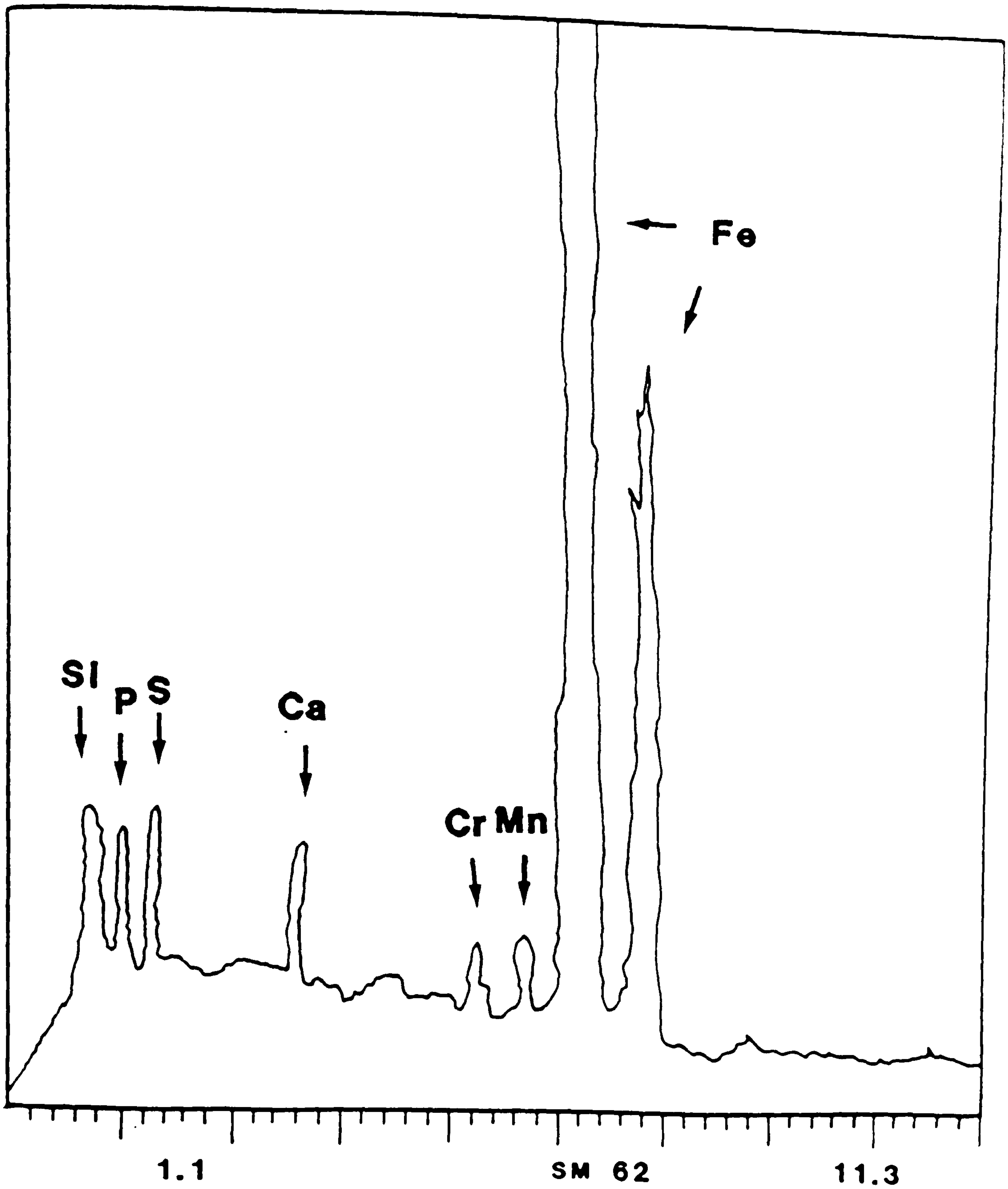


Fig.118 E.p.M.A. for the flat worn using RL48 + 1/4 μ diamond paste at 80°C (load = 900N).

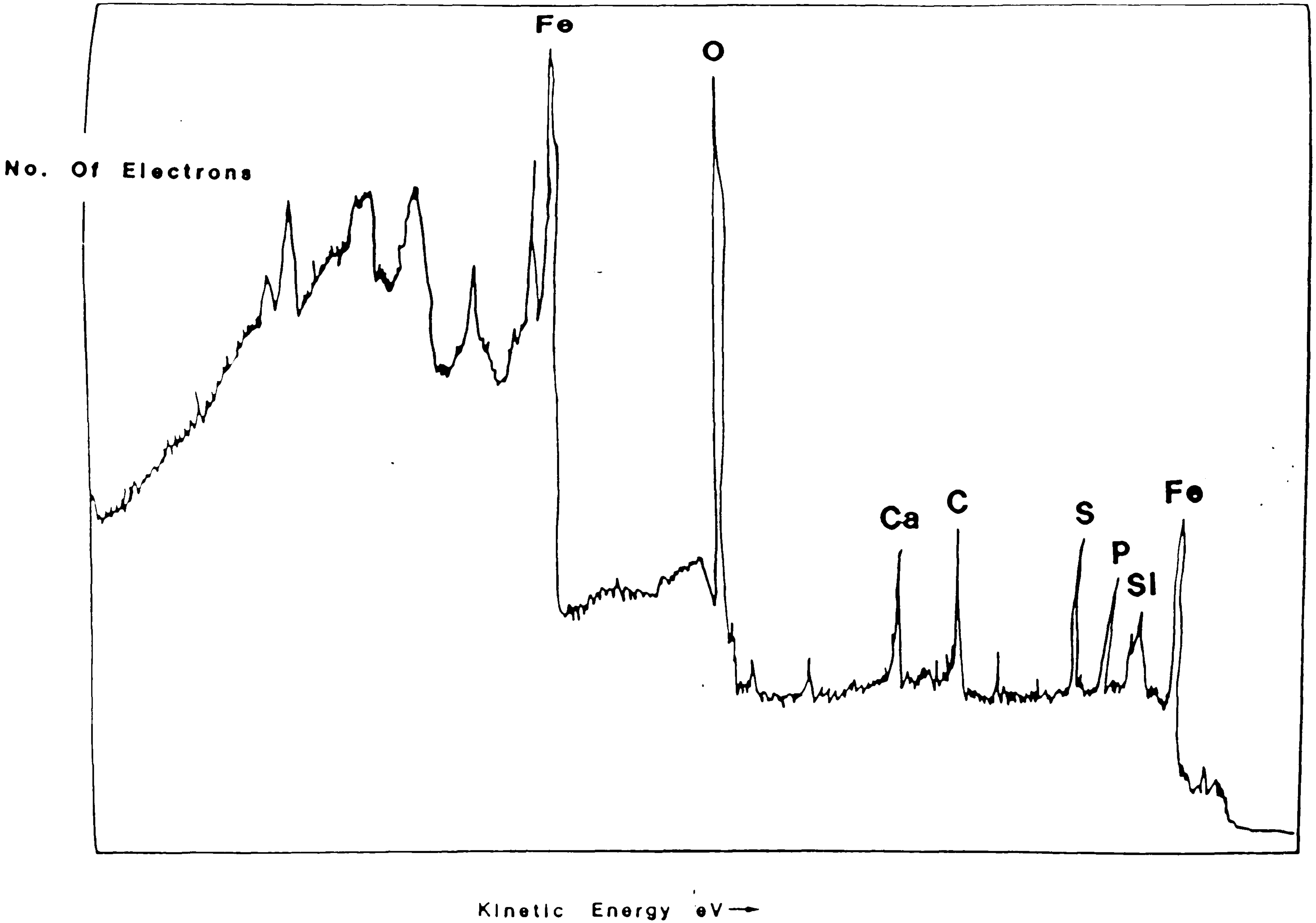
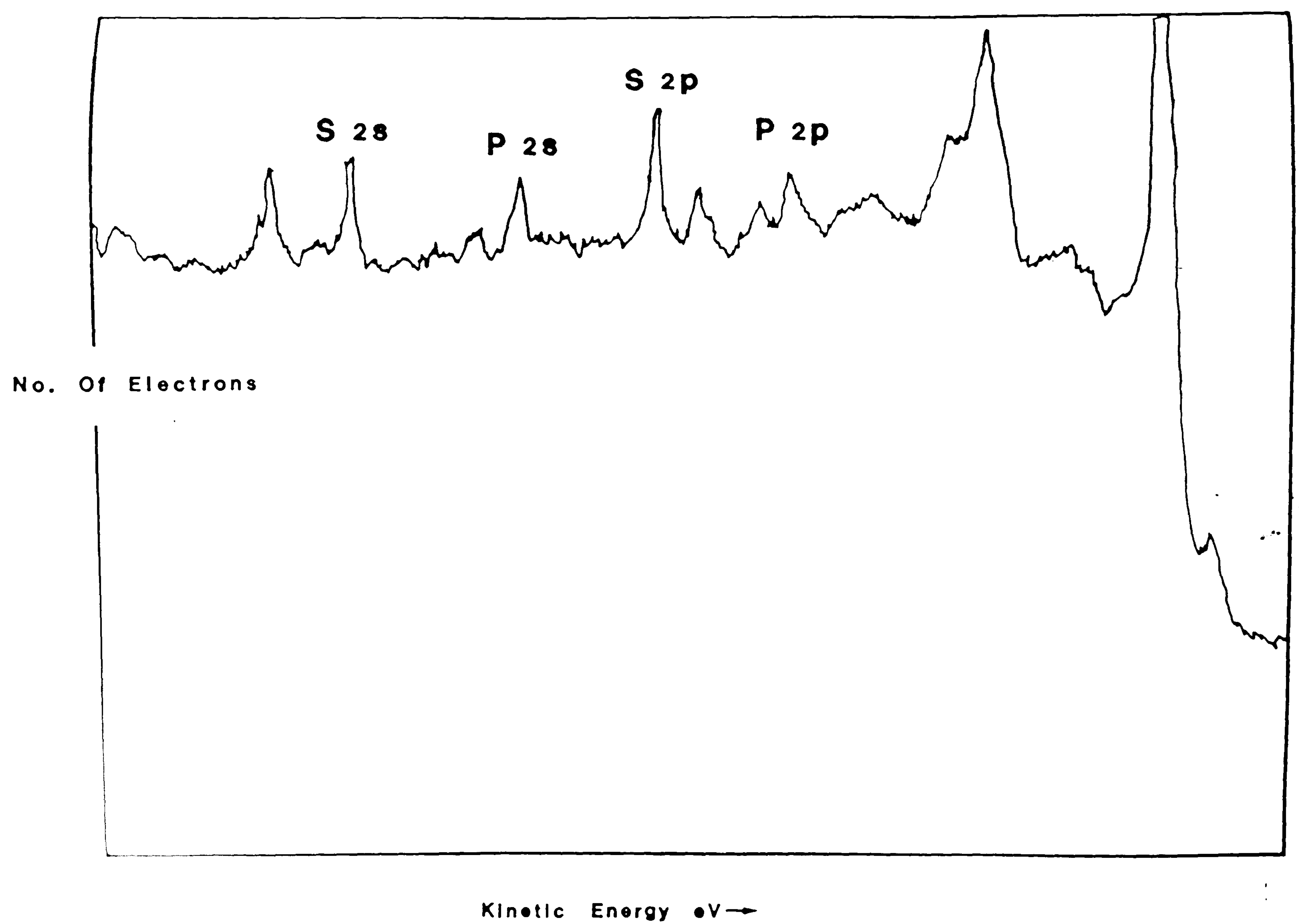
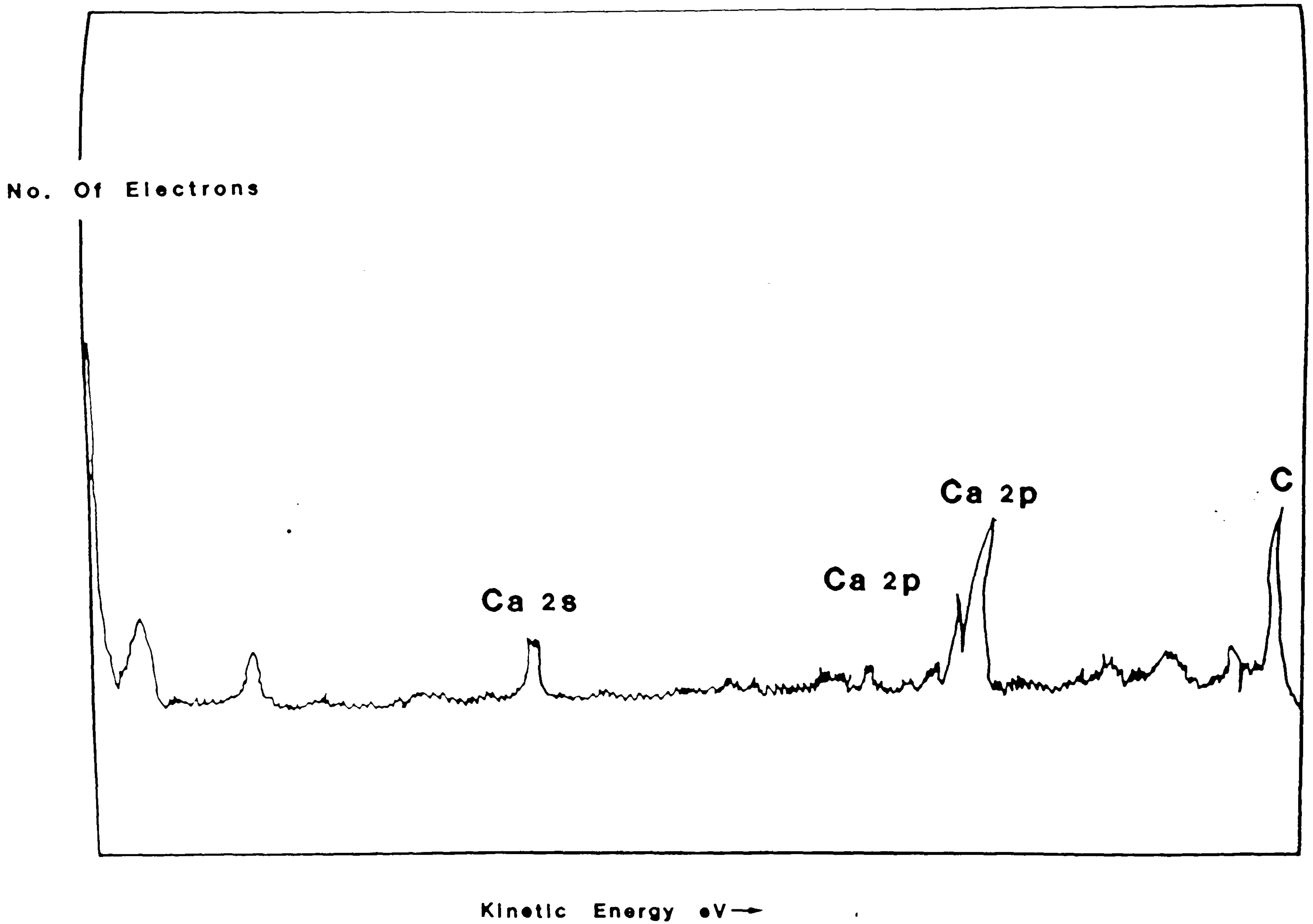


Fig.119 Energy spectrum for the flat worn using RL48 + 1/4 μ diamond paste at $80^{o}C$, (load = 900N).


S 2s
P 2s
S 2p
P 2p
No. Of Electrons
Kinetic Energy eV →


No. Of Electrons
Ca 2s
Ca 2p
Ca 2p
C
Kinetic Energy eV→

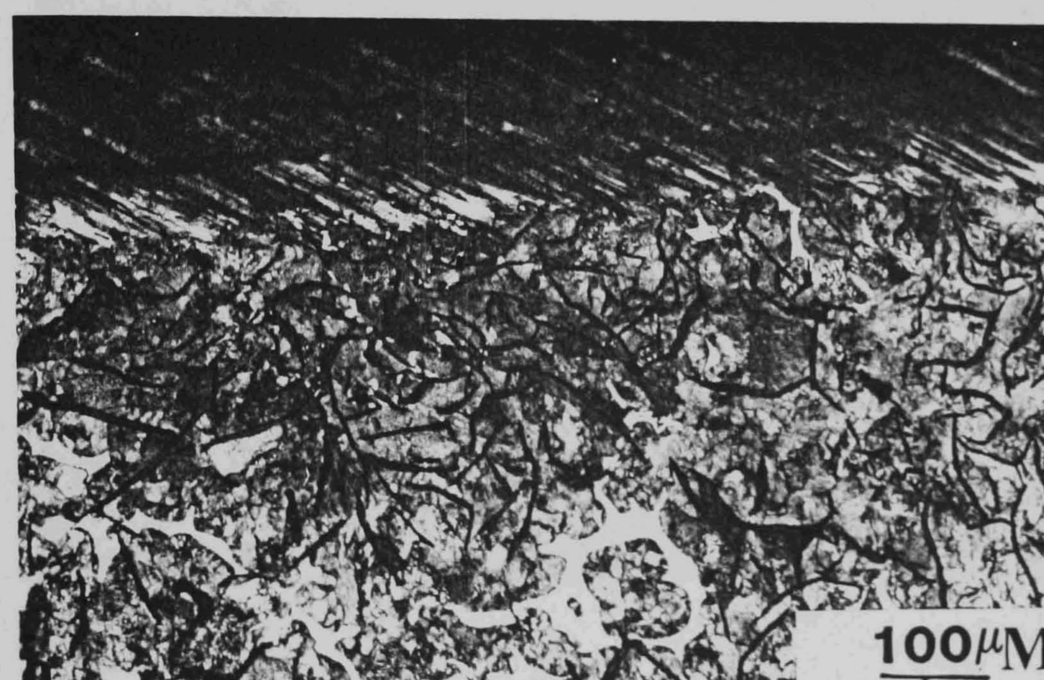


Fig.1 20 Taper section through
a) Unused flat
b) Etched flat.

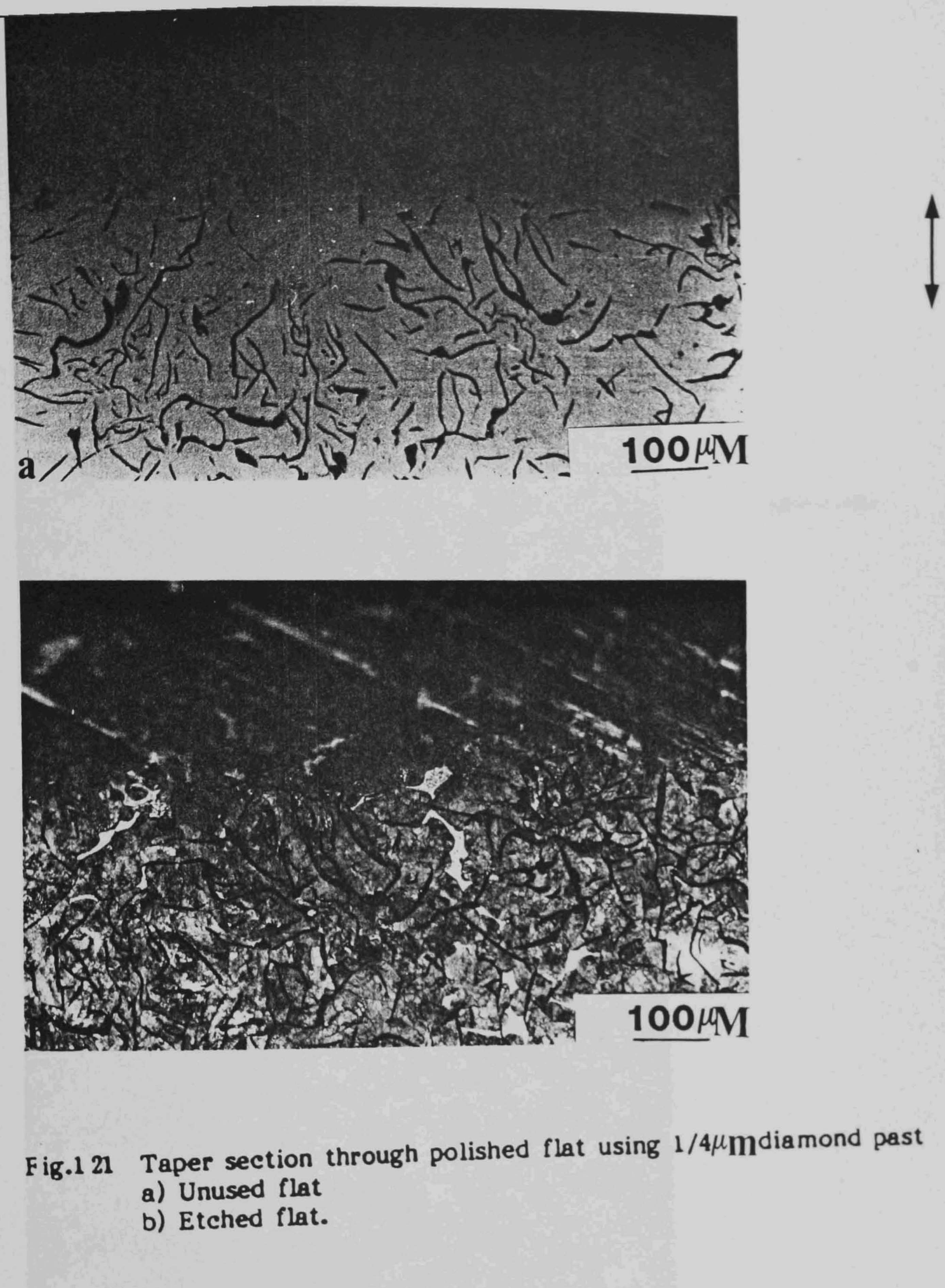
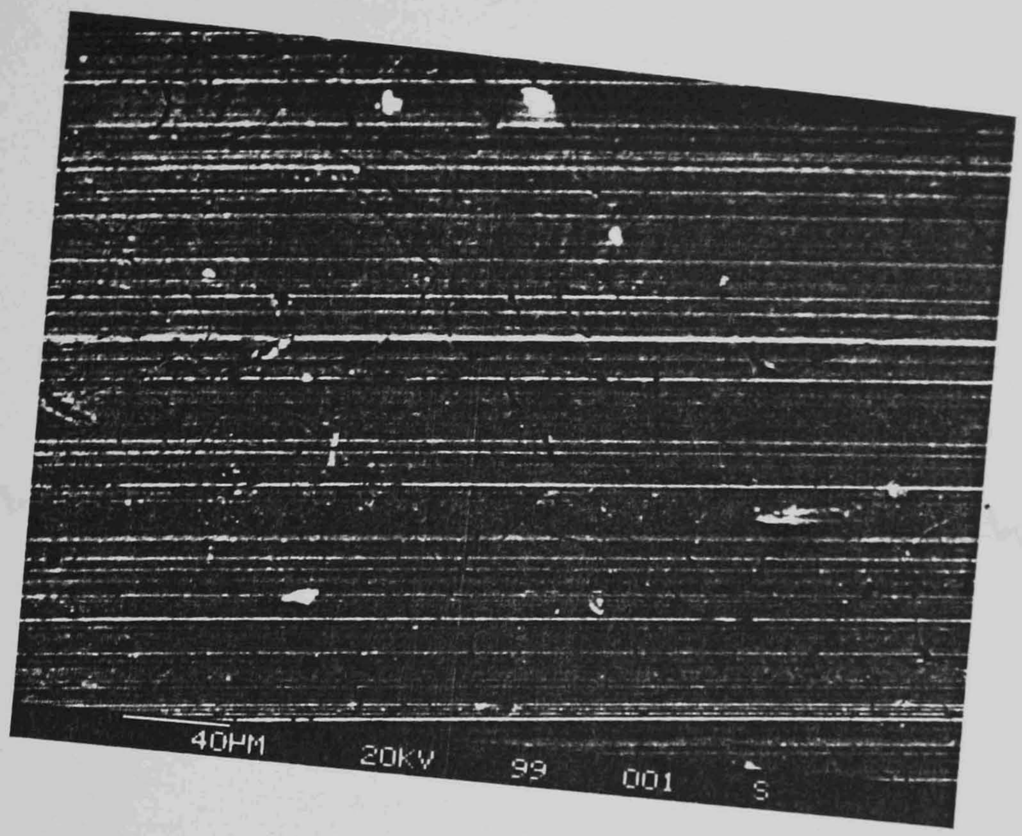


Fig.121 Taper section through polished flat using 1/4μm diamond past
a) Unused flat
b) Etched flat.



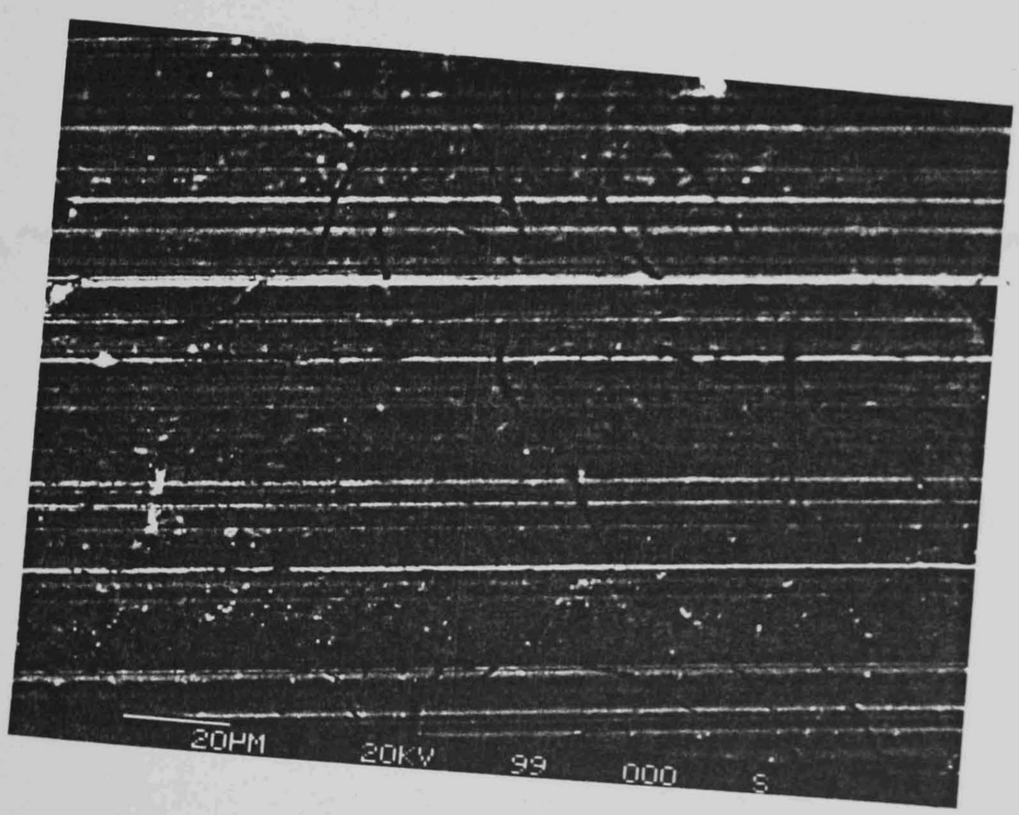


Fig.1 22 SEM appearance of the flat worn using RL48 + 1μm diamond paste at 80^{o}C, (load = 900N).

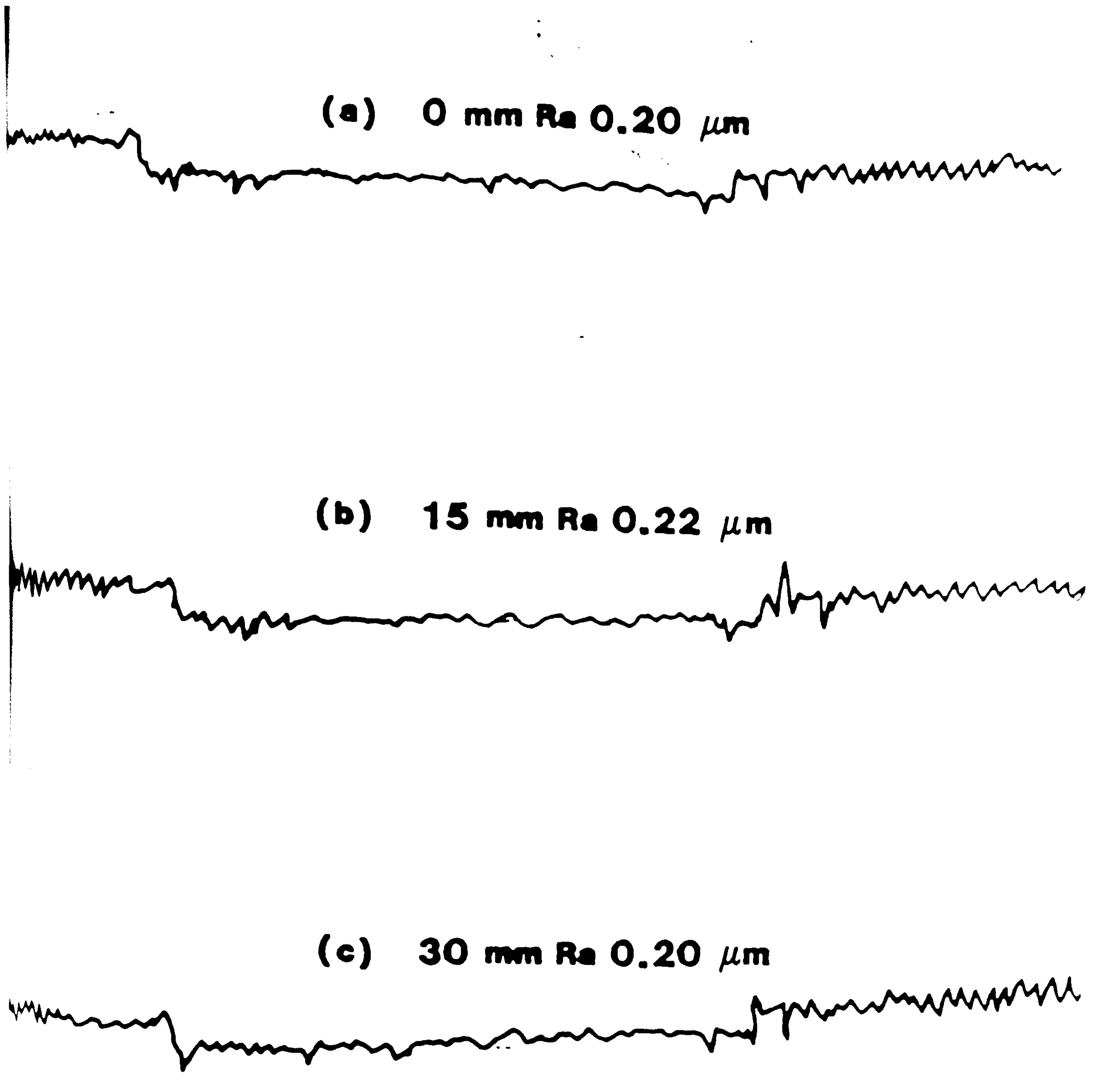


Fig.1 23 C.L.A. values for Fig.1 21.

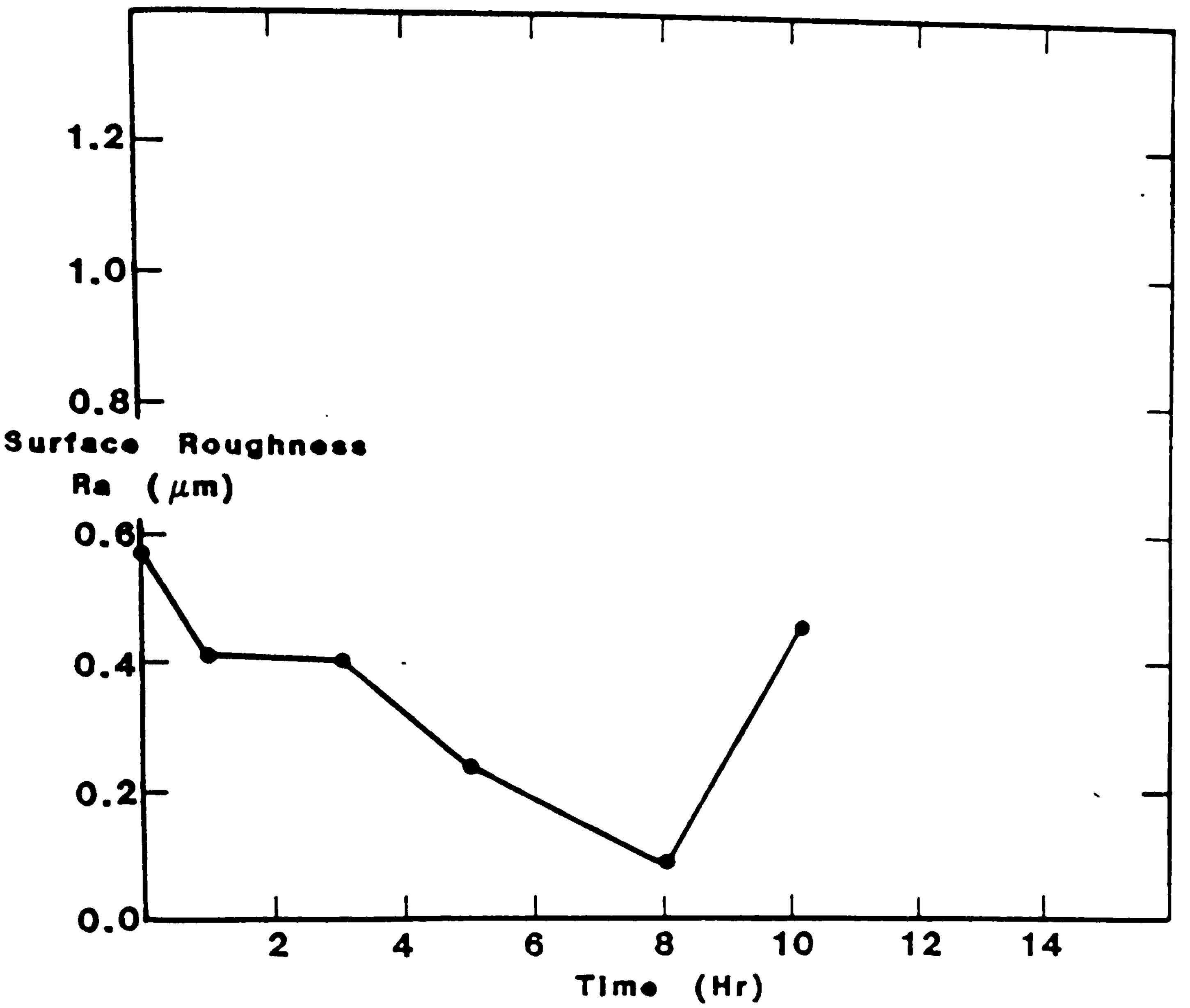


Fig.1 24 C.L.A. values for different test periods for the flat using RL48 + 1/4 μm diamond paste at 80°C, (load = 900N).

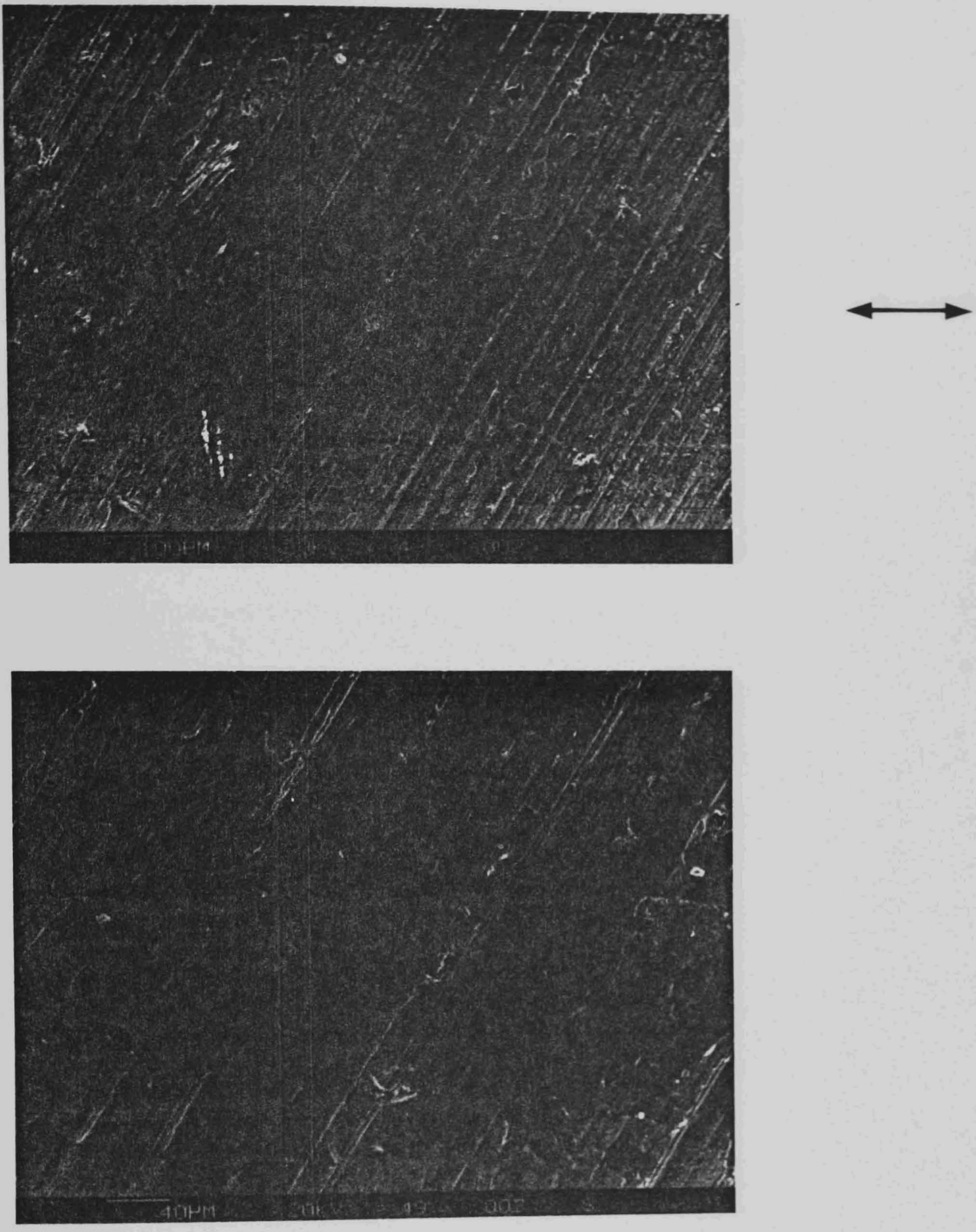
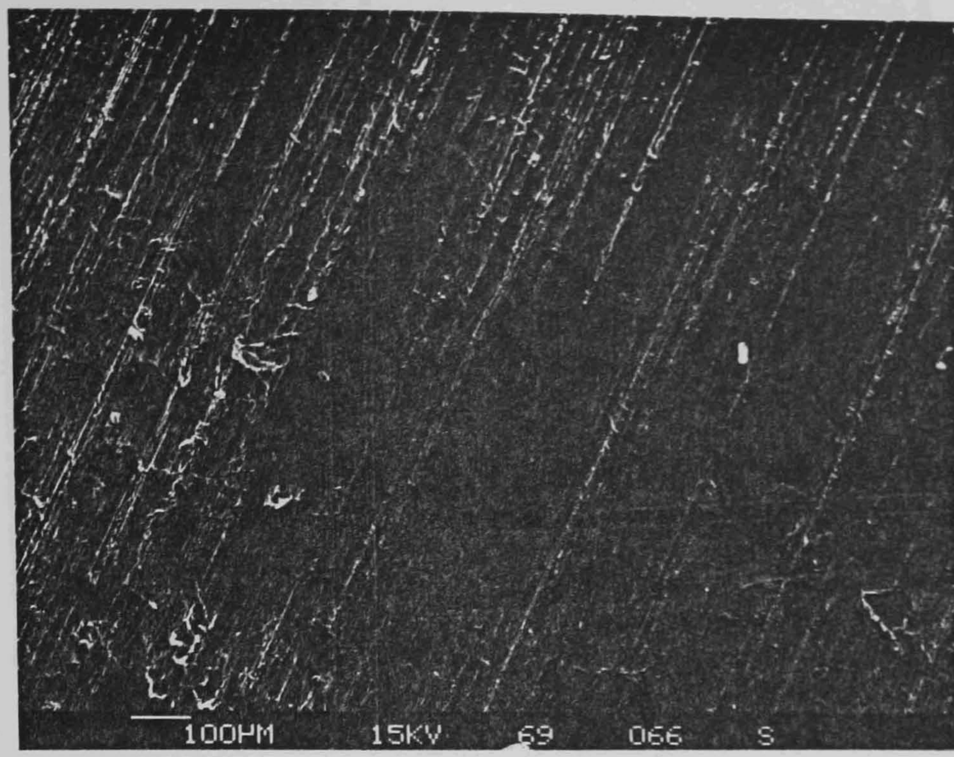


Fig.1 25 SEM appearance of the flat after 1 hour test using RL48 + 1/4 μm diamond paste at $80^{o}C$, (load = 900N).



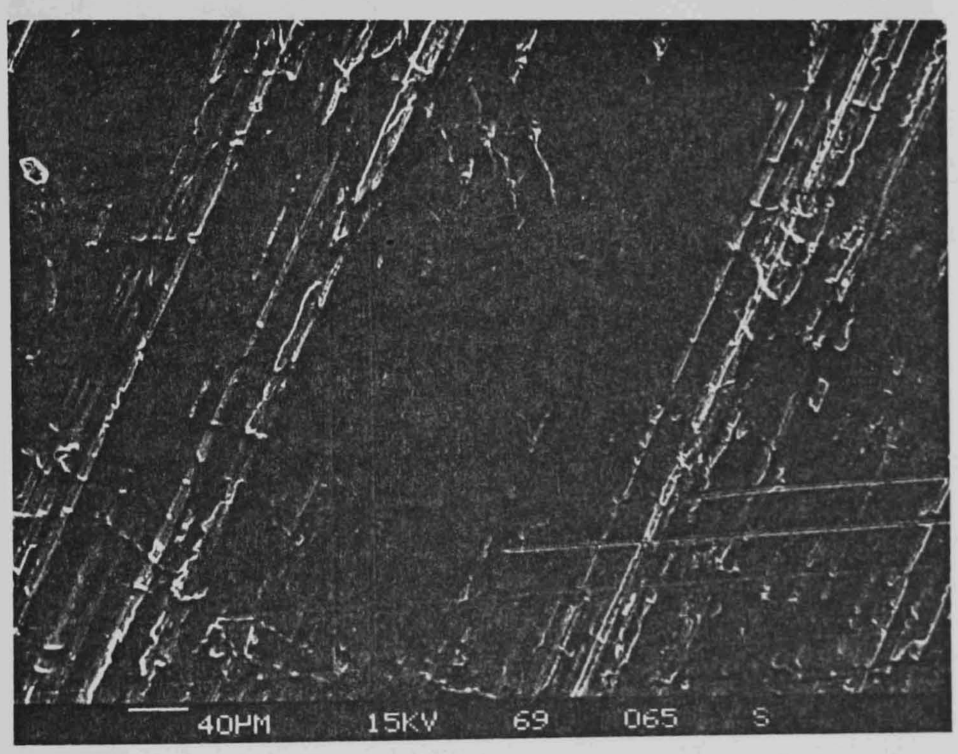
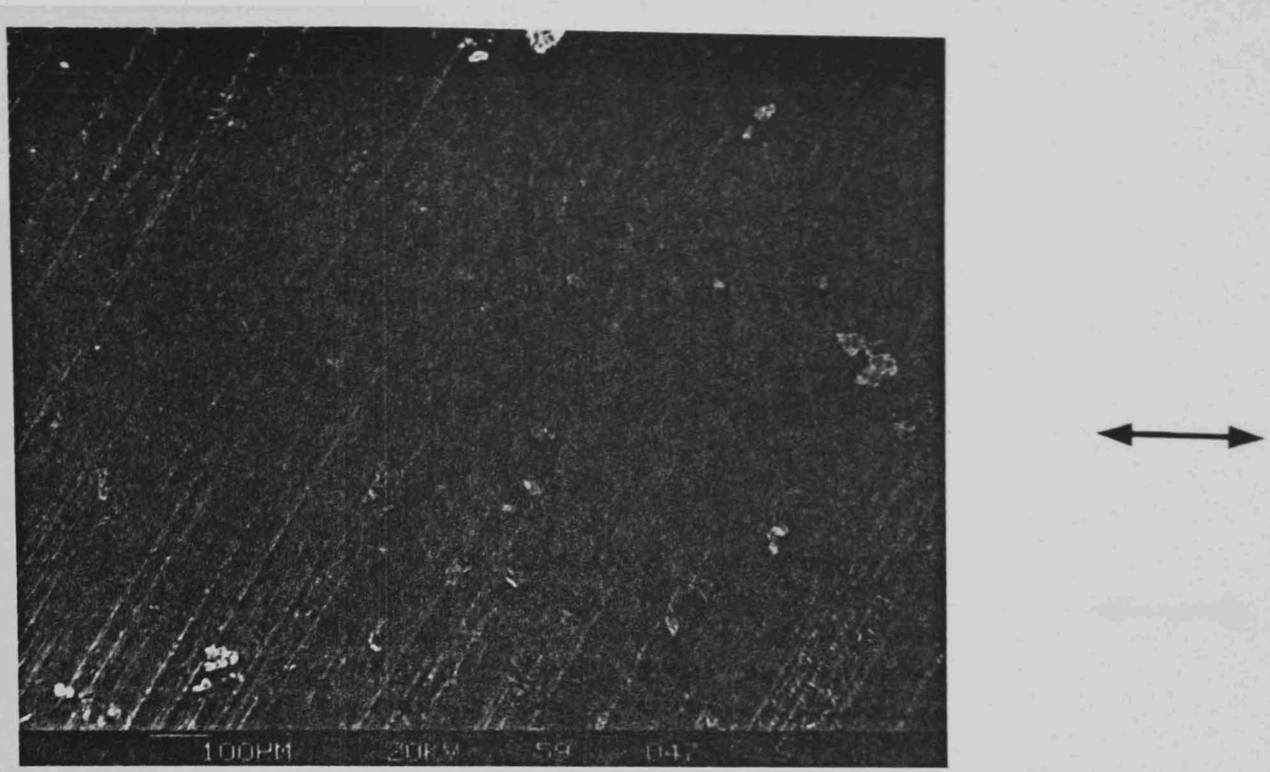


Fig.1 26 SEM appearance of the flat after 3 hour test using RL48 + 1/4 μm diamond paste at 80°C, (load = 900N).



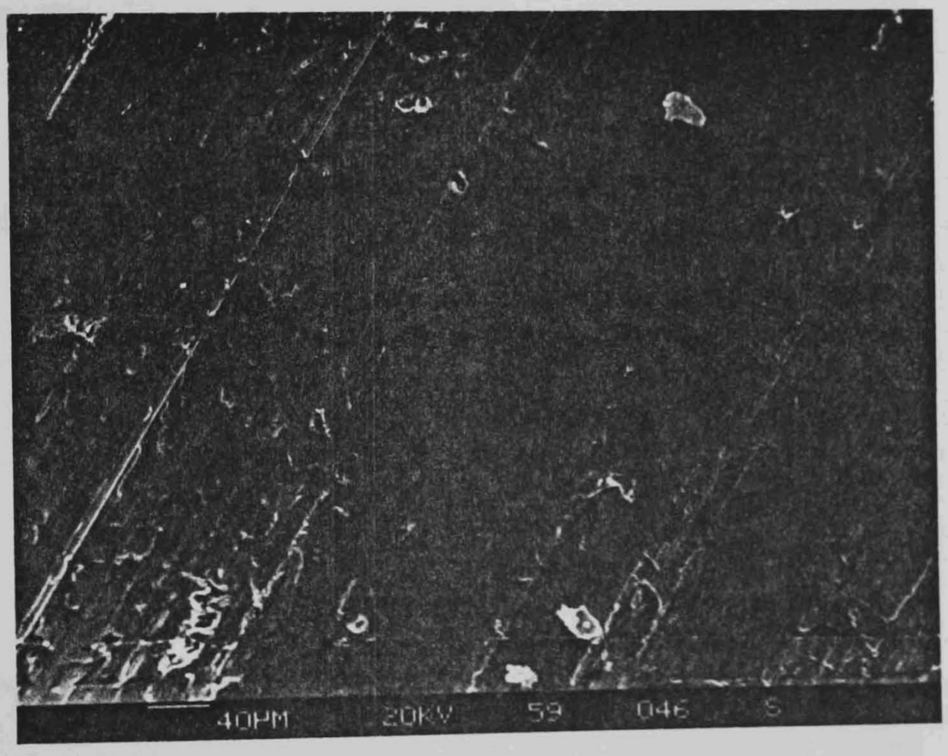


Fig.1 27 SEM appearance of the flat after 5 hour test using RL48 + 1/4 μm diamond paste at 80^{o}C, (load = 900N).

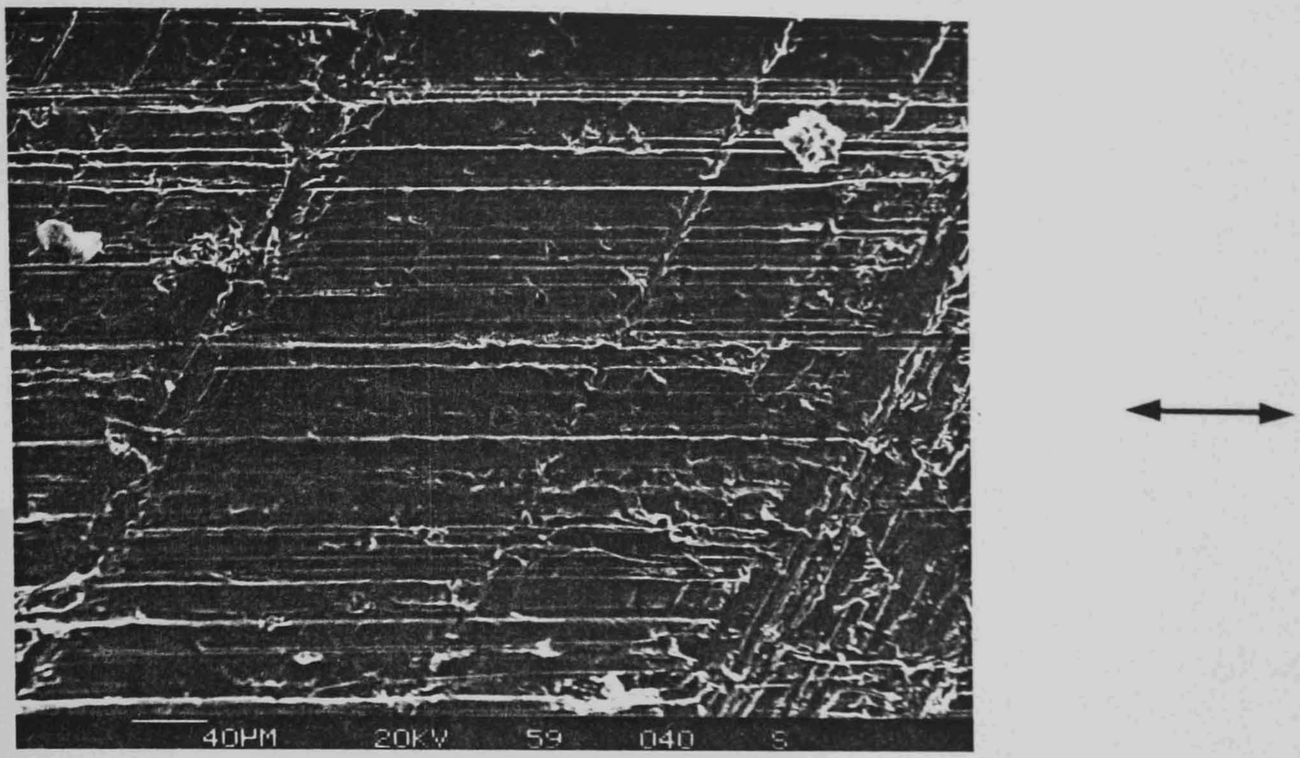


Fig.1 28 SEM appearance of the flat after 8 hour test using RL48 + 1/4μm diamond paste at 80^{o}C, (load = 900N).

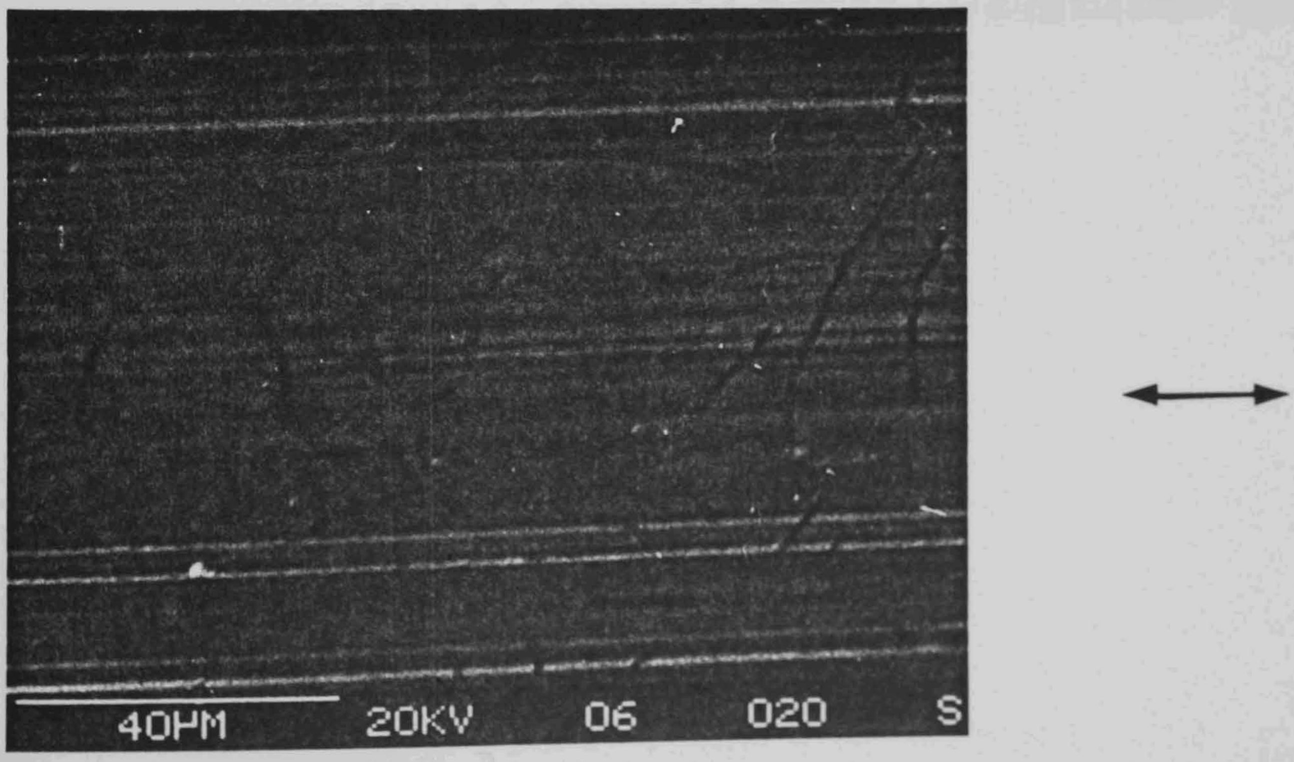


Fig.1 29 SEM appearance of the flat after 10 hour test using RL48 + 1/4μm diamond paste at 80^{o}C, (load = 900N).

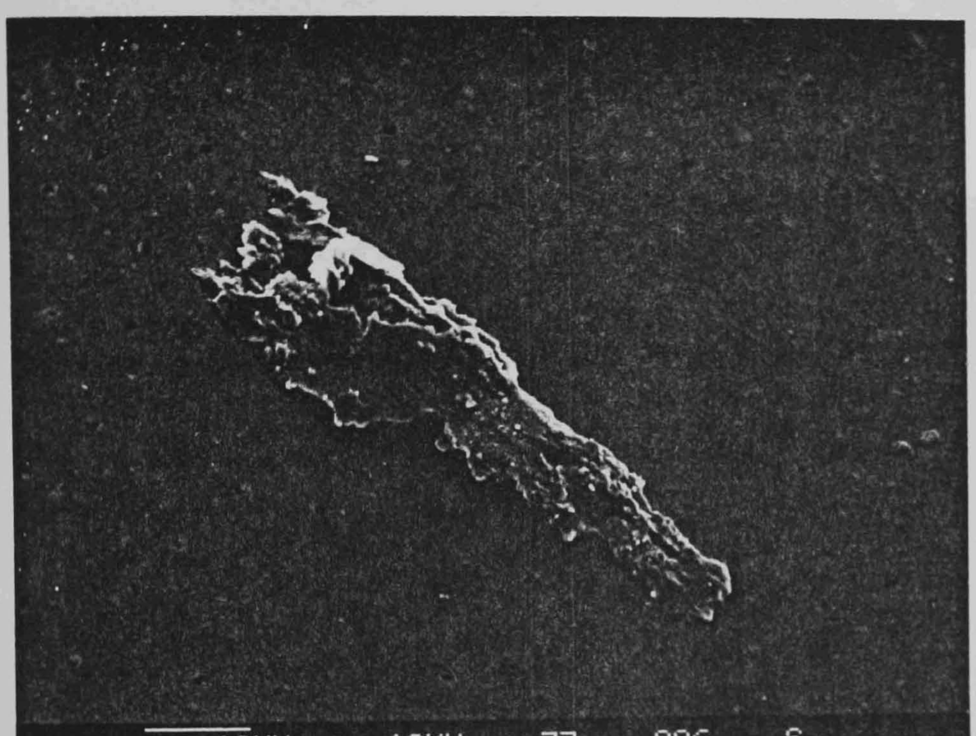
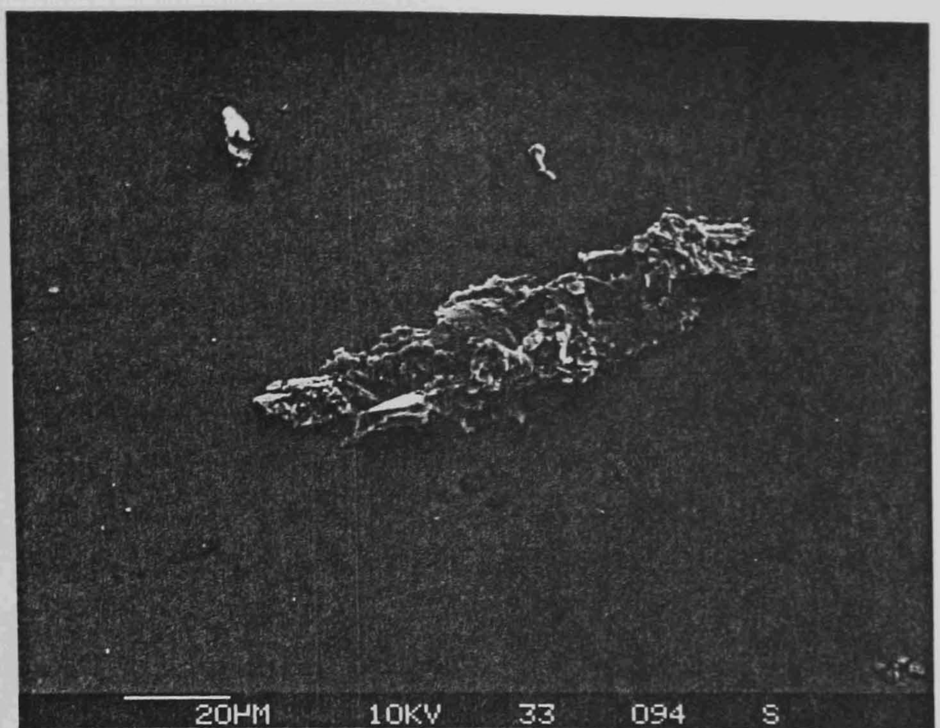


Fig.130 Debris produced after 1 hour test using RL48 + 1/4μm diamond paste at 80^{o}C, (load = 900N).



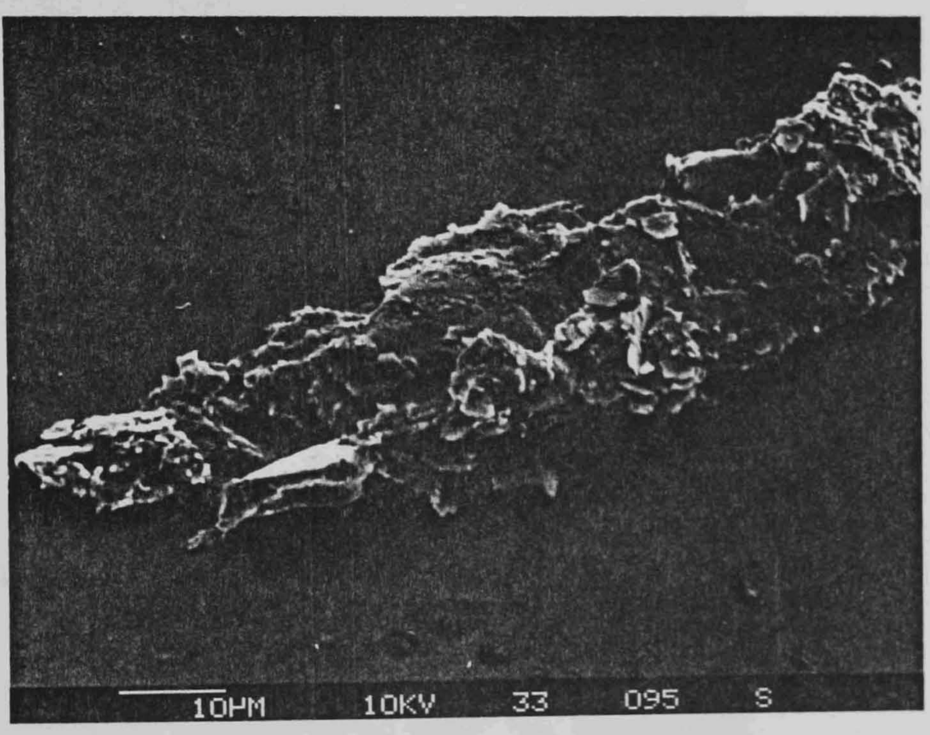


Fig.131 Debris produced after 5 hour test using RL48 + 1/4μm diamond paste at 80^{o}C, (load = 900N).

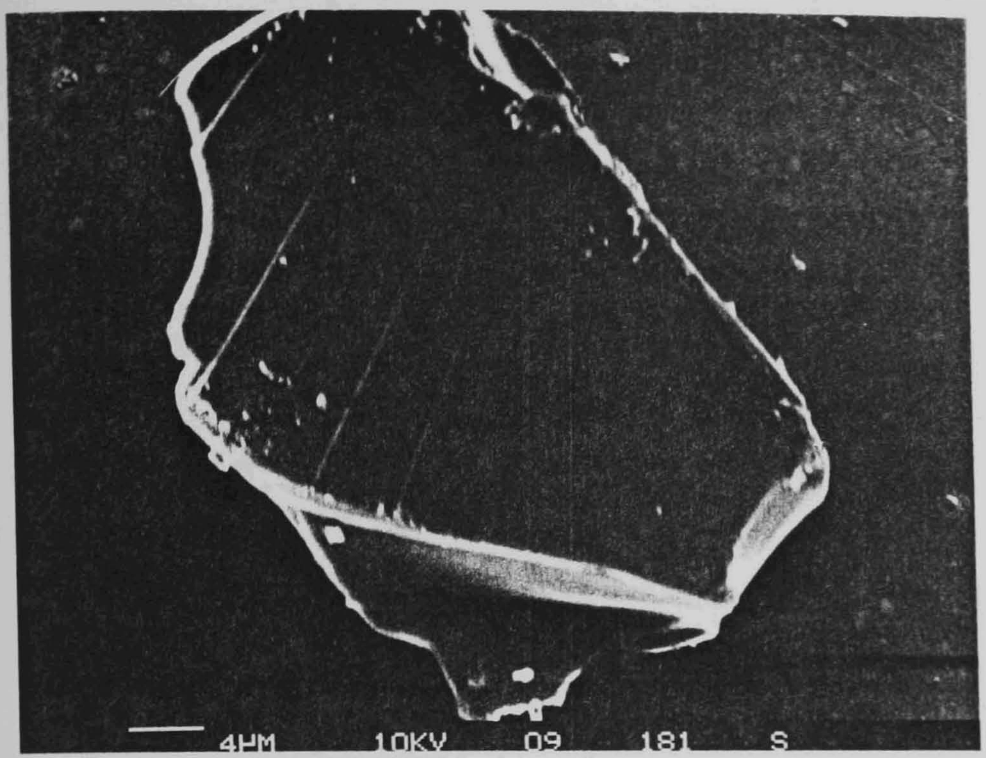


Fig.132 Debris produced after 8 hour test using RL48 + 1/4μm diamond paste at 80^{o}C, (load = 900N).

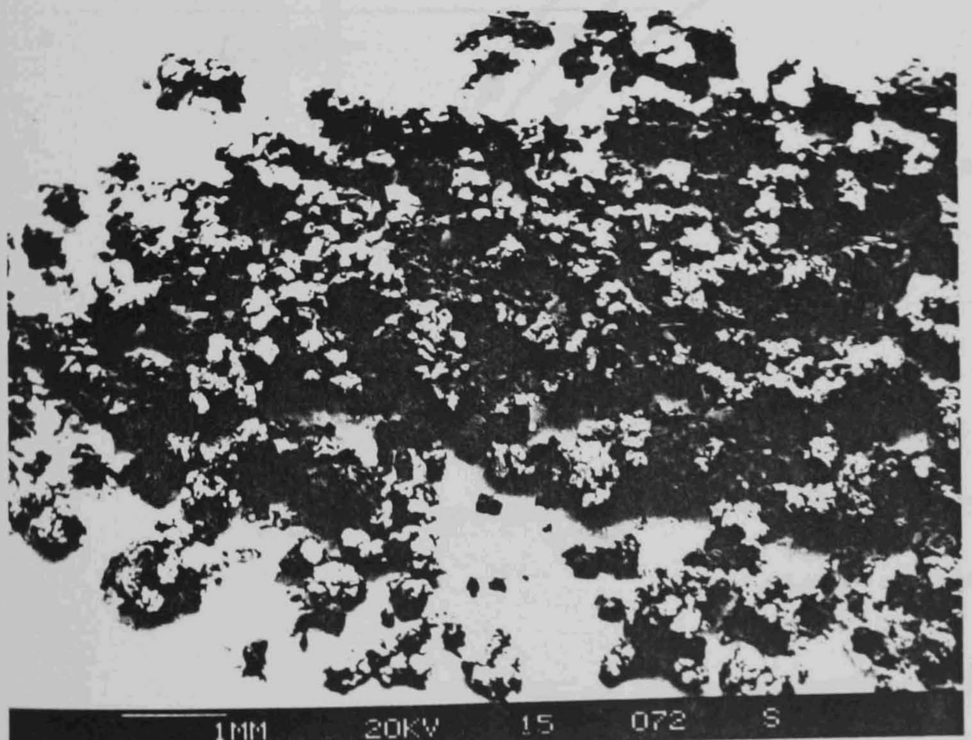


Fig.133 Sample of hard carbon used in the bore polishing test.

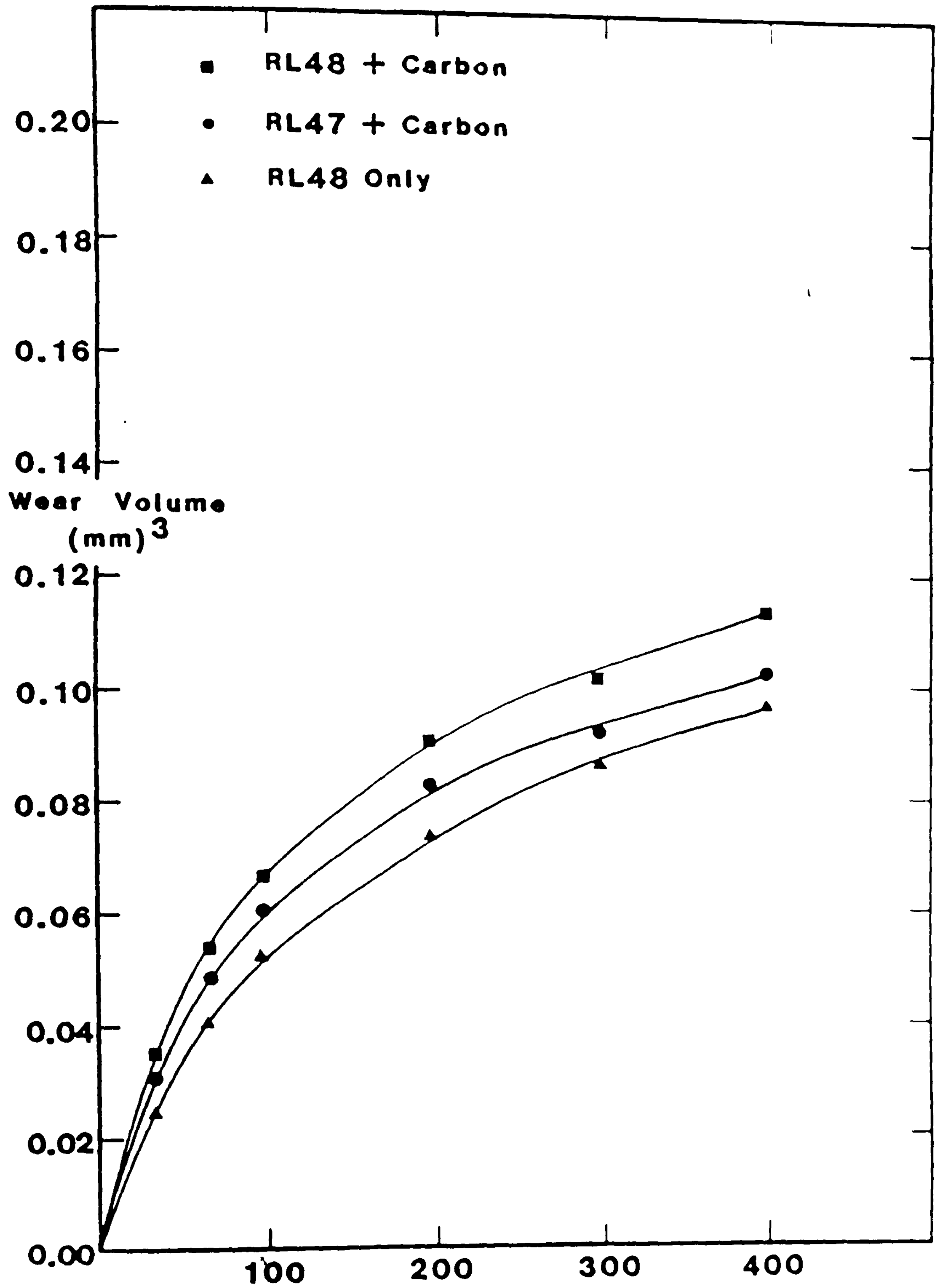


Fig.134 Wear volume versus sliding distance.

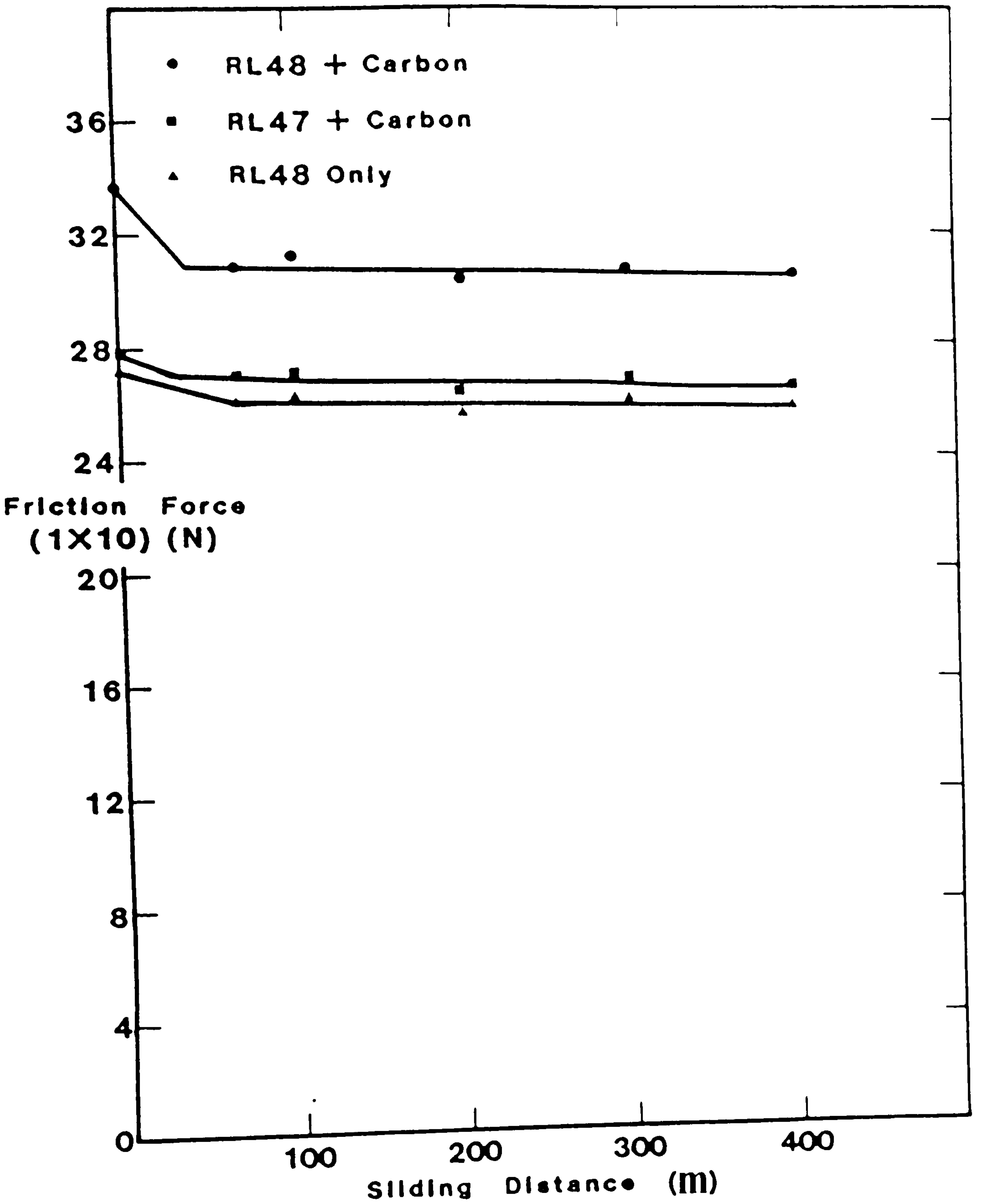


Fig.135 Friction force versus sliding distance.

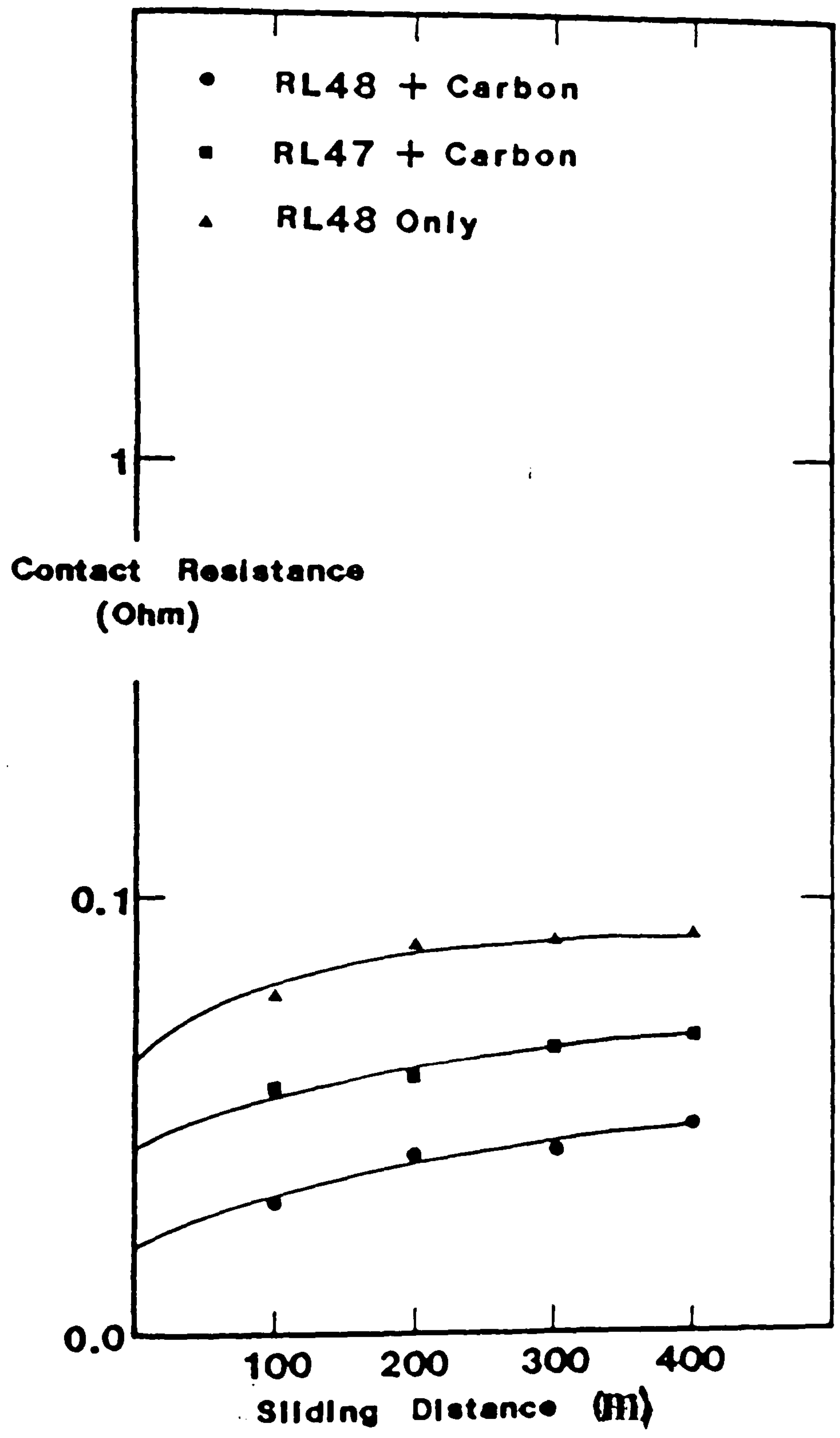


Fig.136 Contact resistance versus sliding distance.

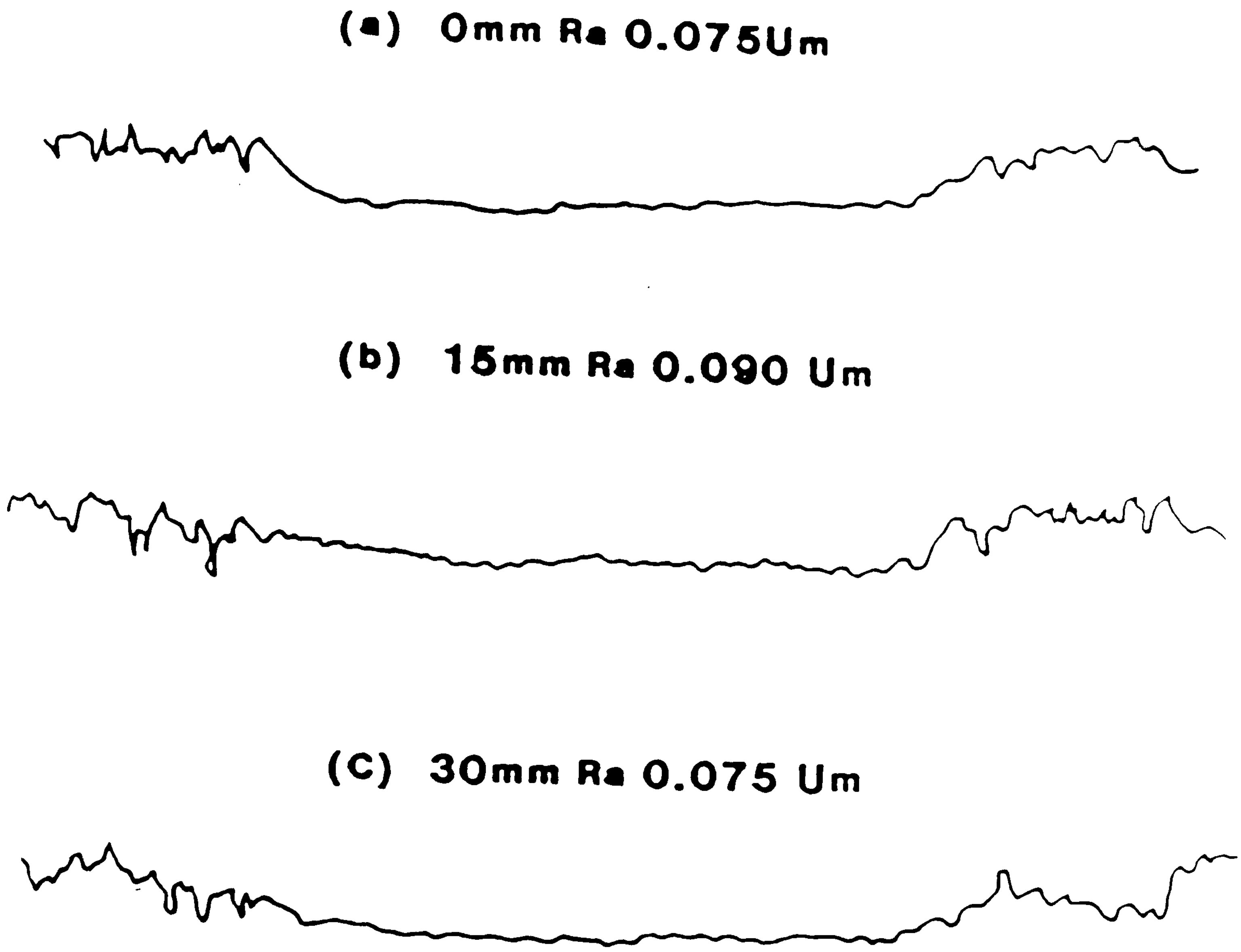


Fig.137 C.L.A. values for the polished flat using RL48 + carbon.

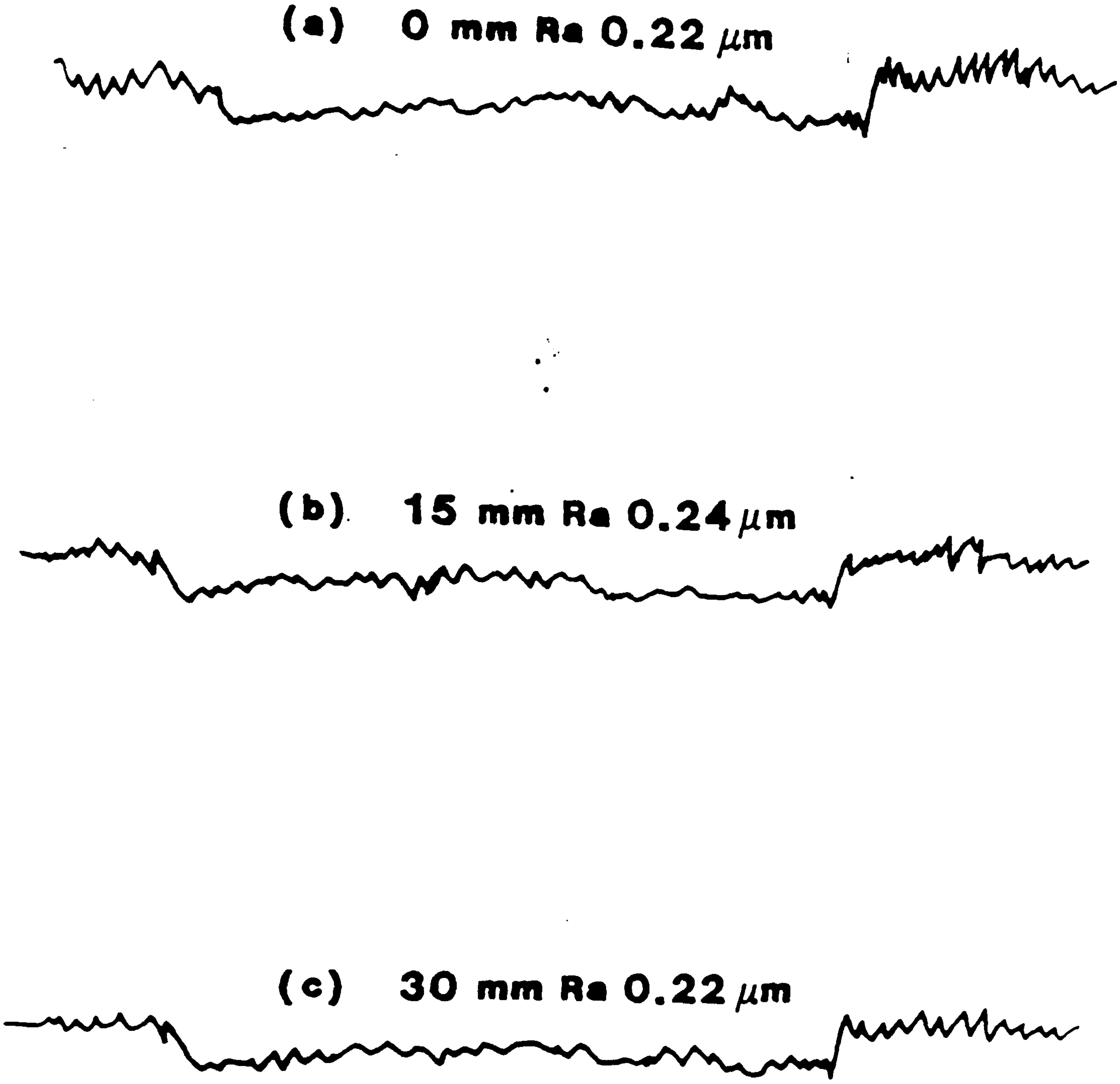


Fig.138 C.L.A. values for the flat using RL48 only.

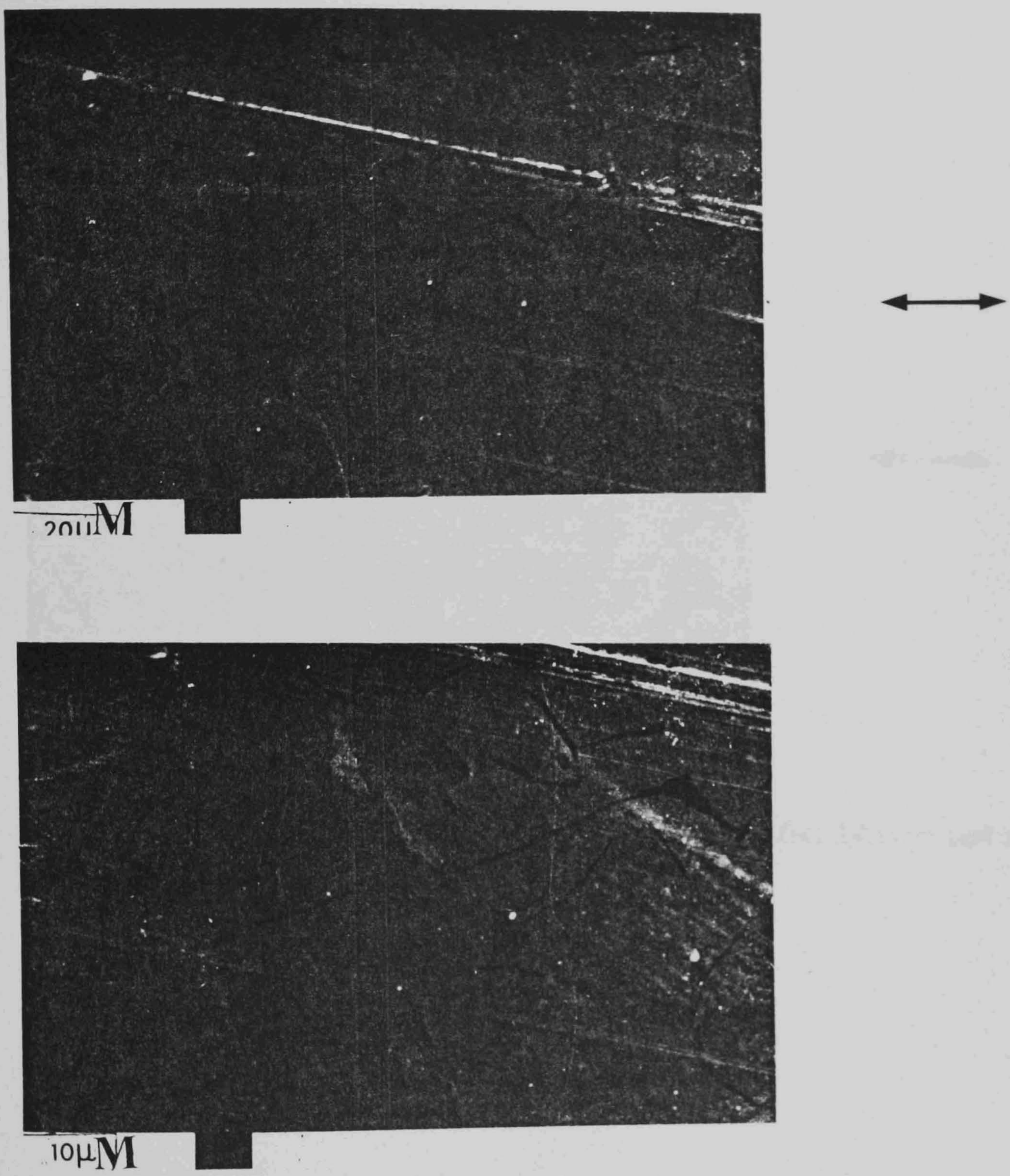


Fig.139 SEM appearance of the polished flat using RL48 + carbon after 14 hour test at $80^{o}C$, (load = 1 200N).

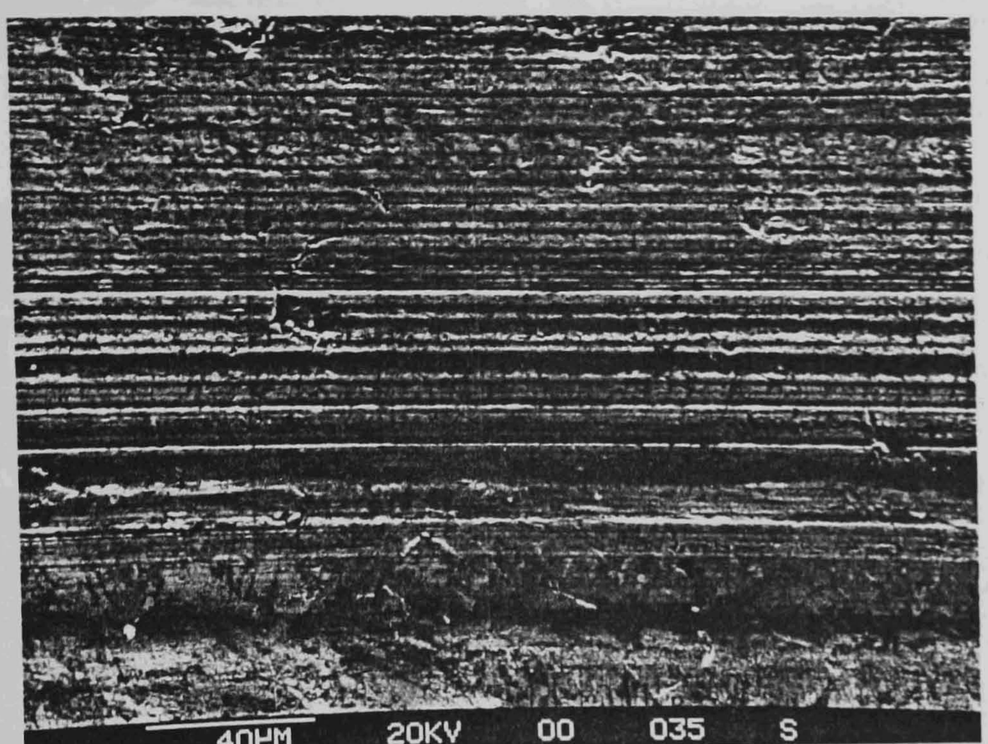


Fig.140 SEM appearance of the flat using RL48 only after 14 hour test at 80°C. (load = 1200N).

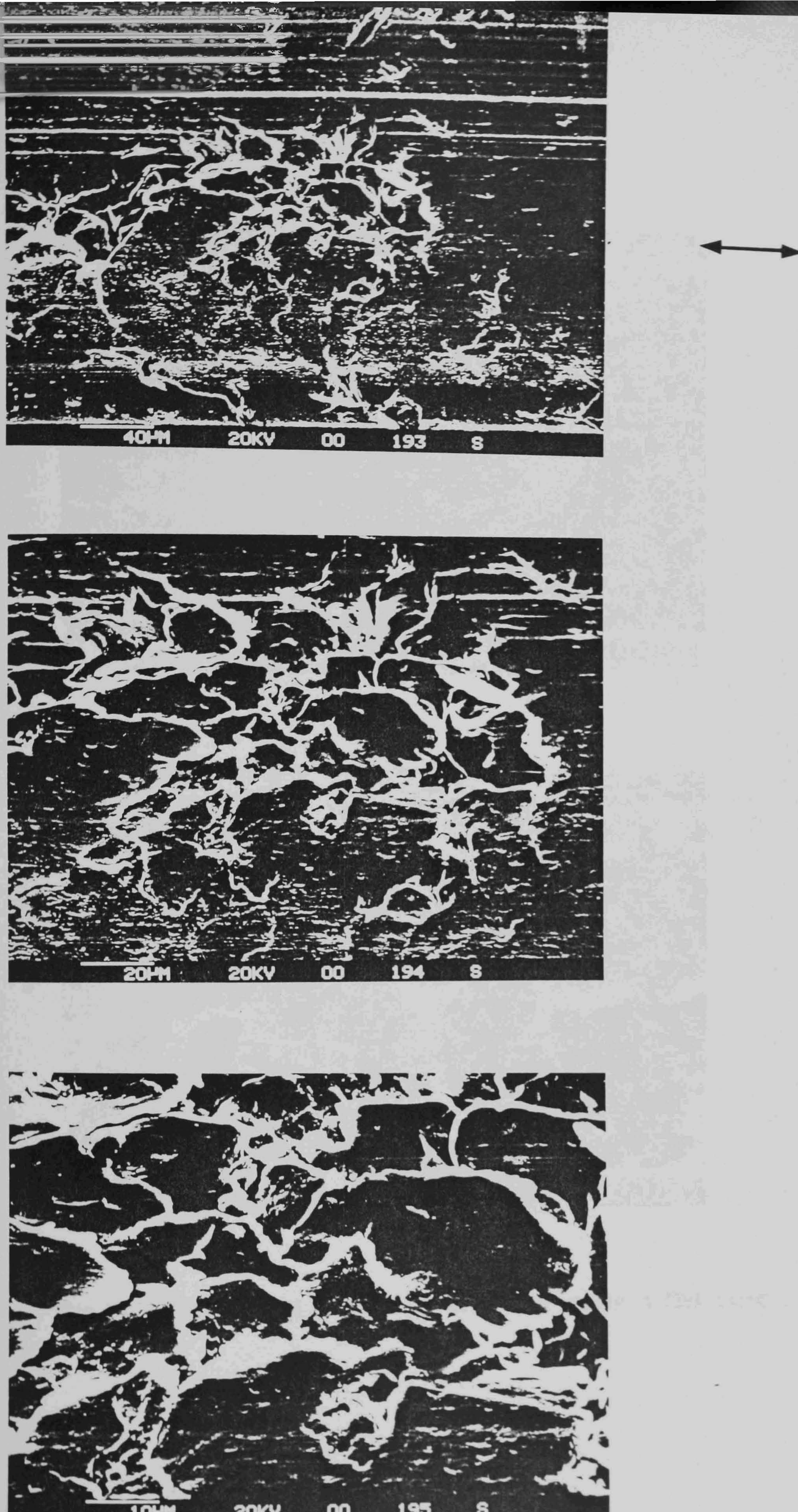


Fig.141 Delamination wear on the polished flat using RL48 + carbon.

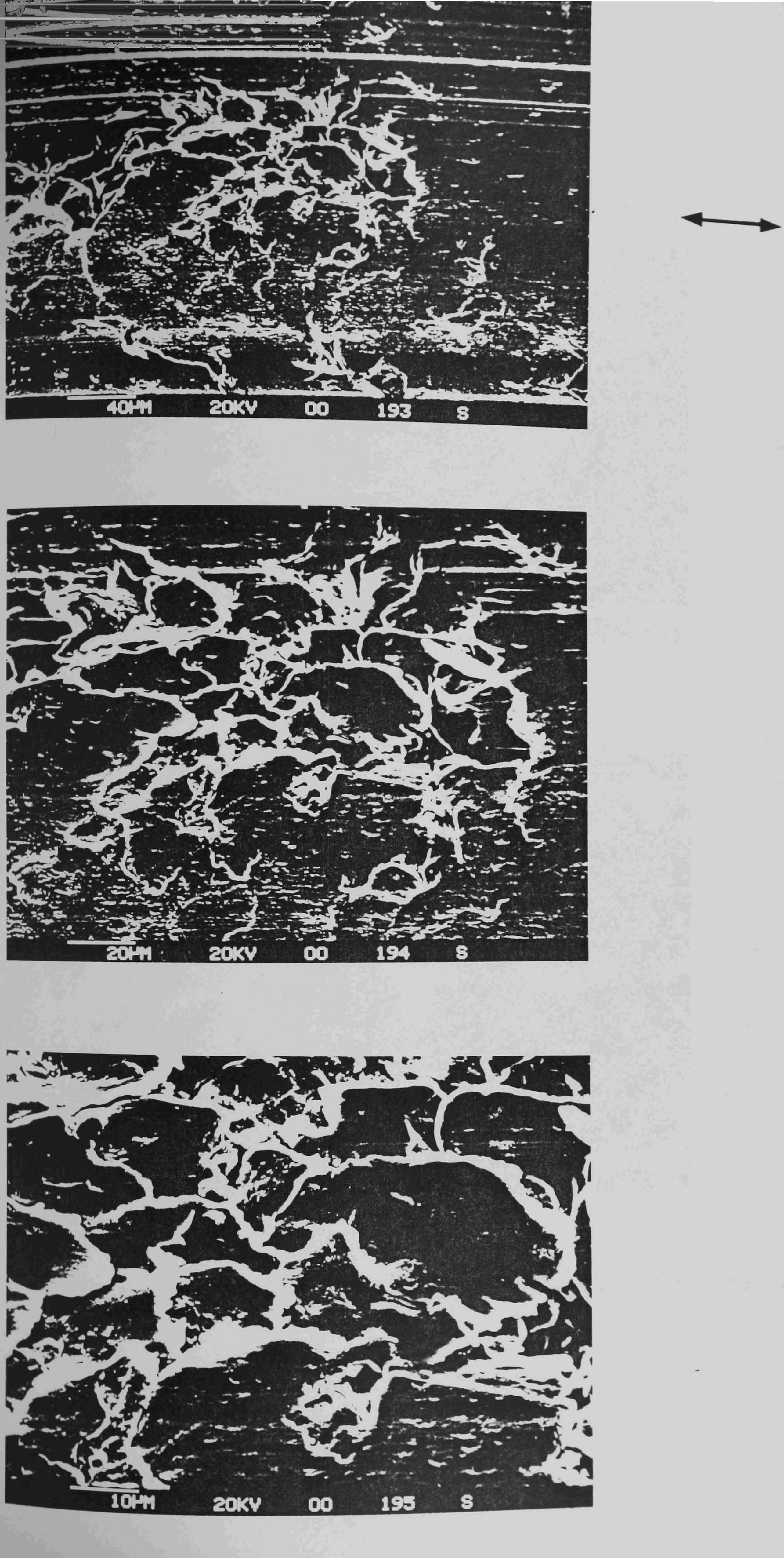


Fig.141 Delamination wear on the polished flat using RL48 + carbon.

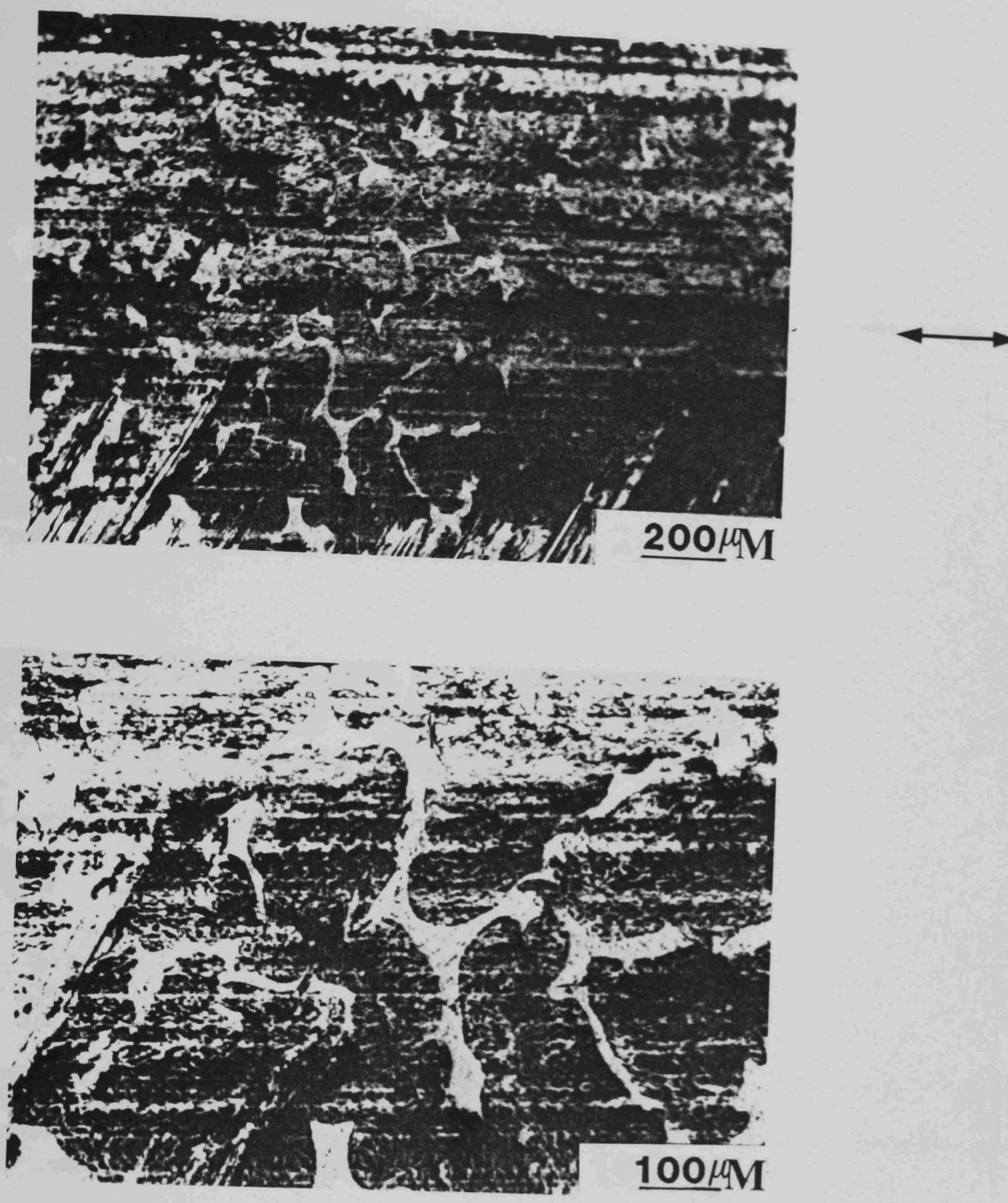


Fig.142 Differential abrasive wear on the polished flat using RL48 + carbon.

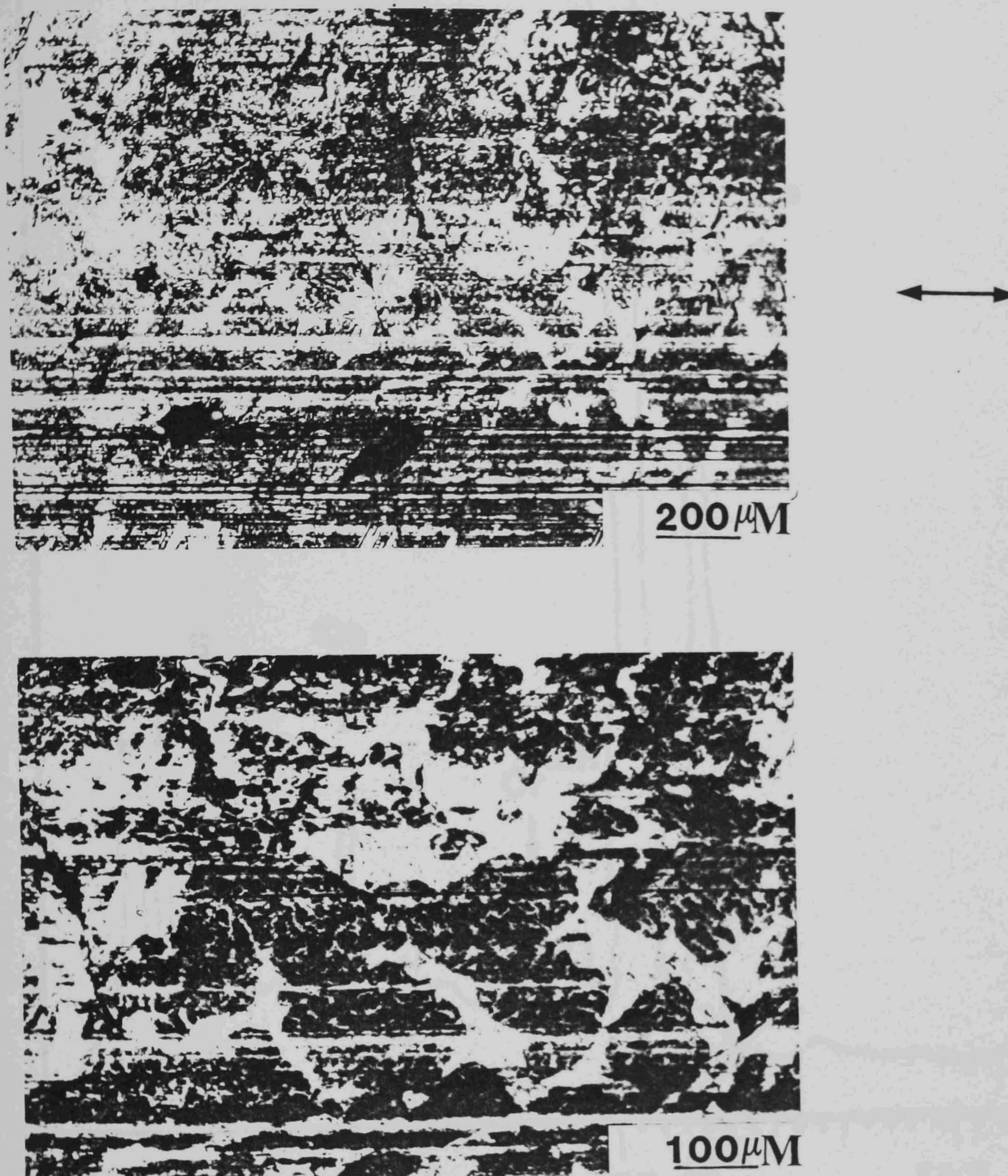


Fig.143 Mottled area or thumb print structure on the polished flat using RL48 + carbon.

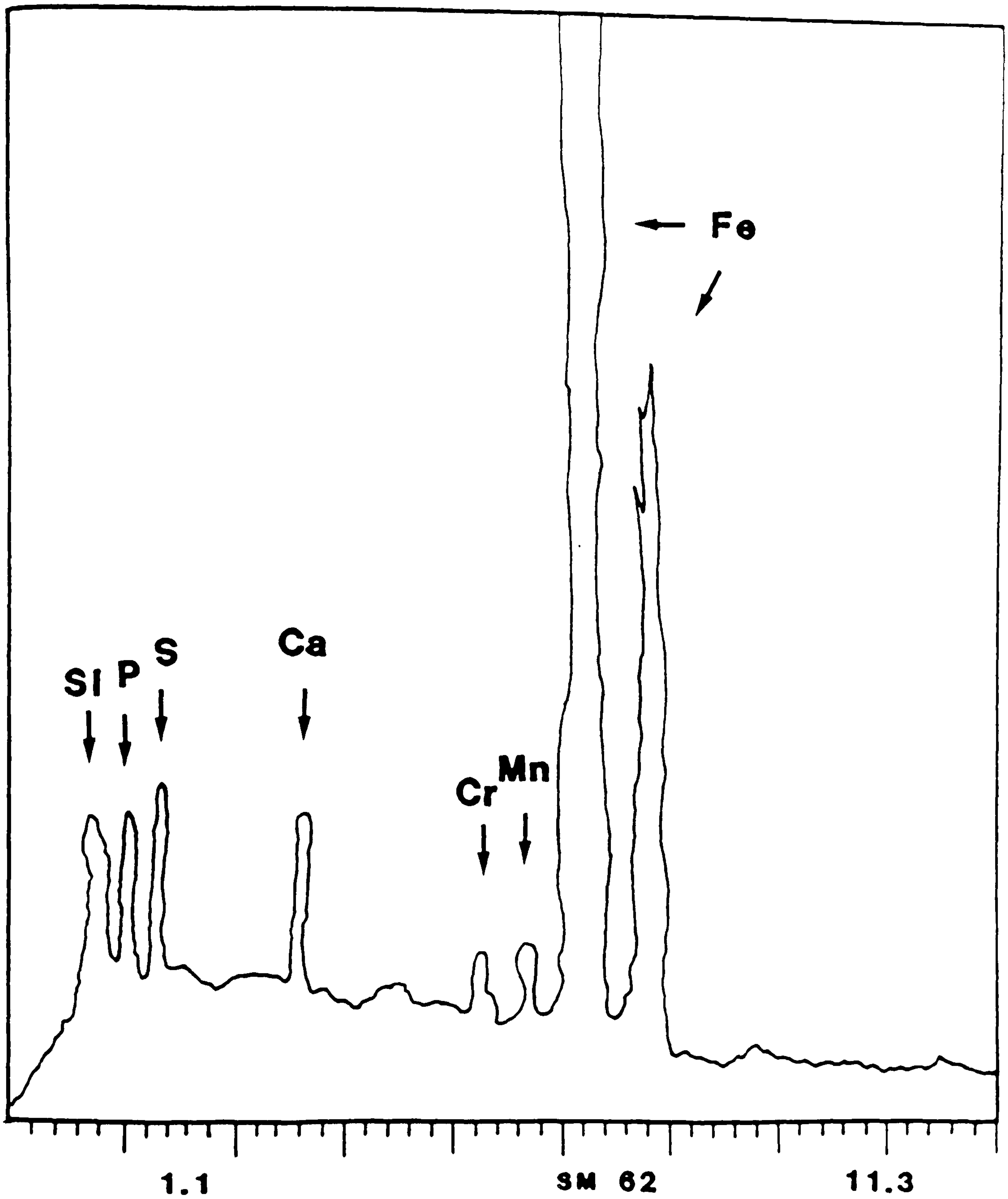


Fig.144 E.p.M.A. for the flat worn using RL48 + carbon at 80°C, (load = 1200N).

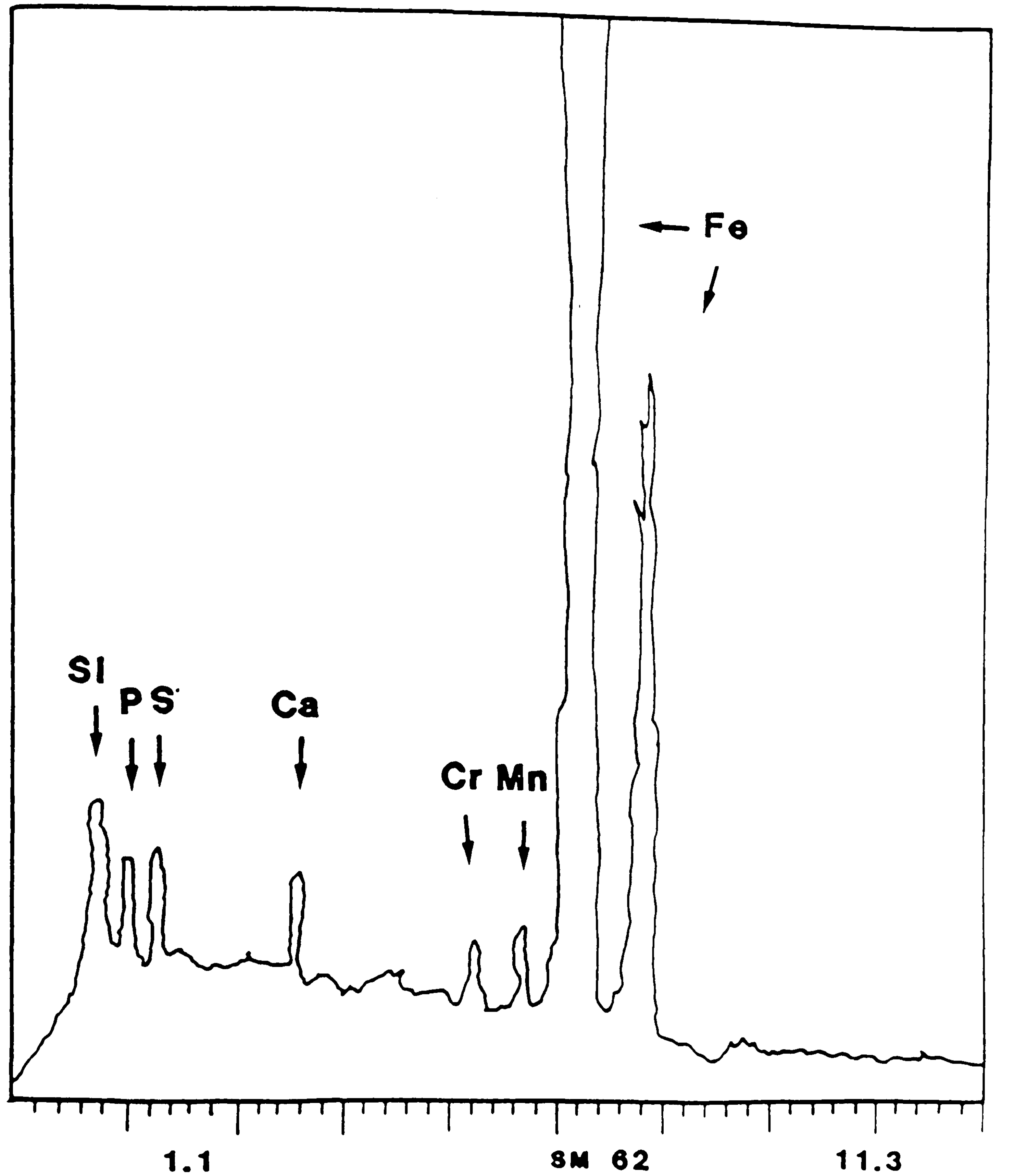


Fig.145 E.p.M.A. for the flat worn using RL48 only at 80^{o}C, (load = 1200N).

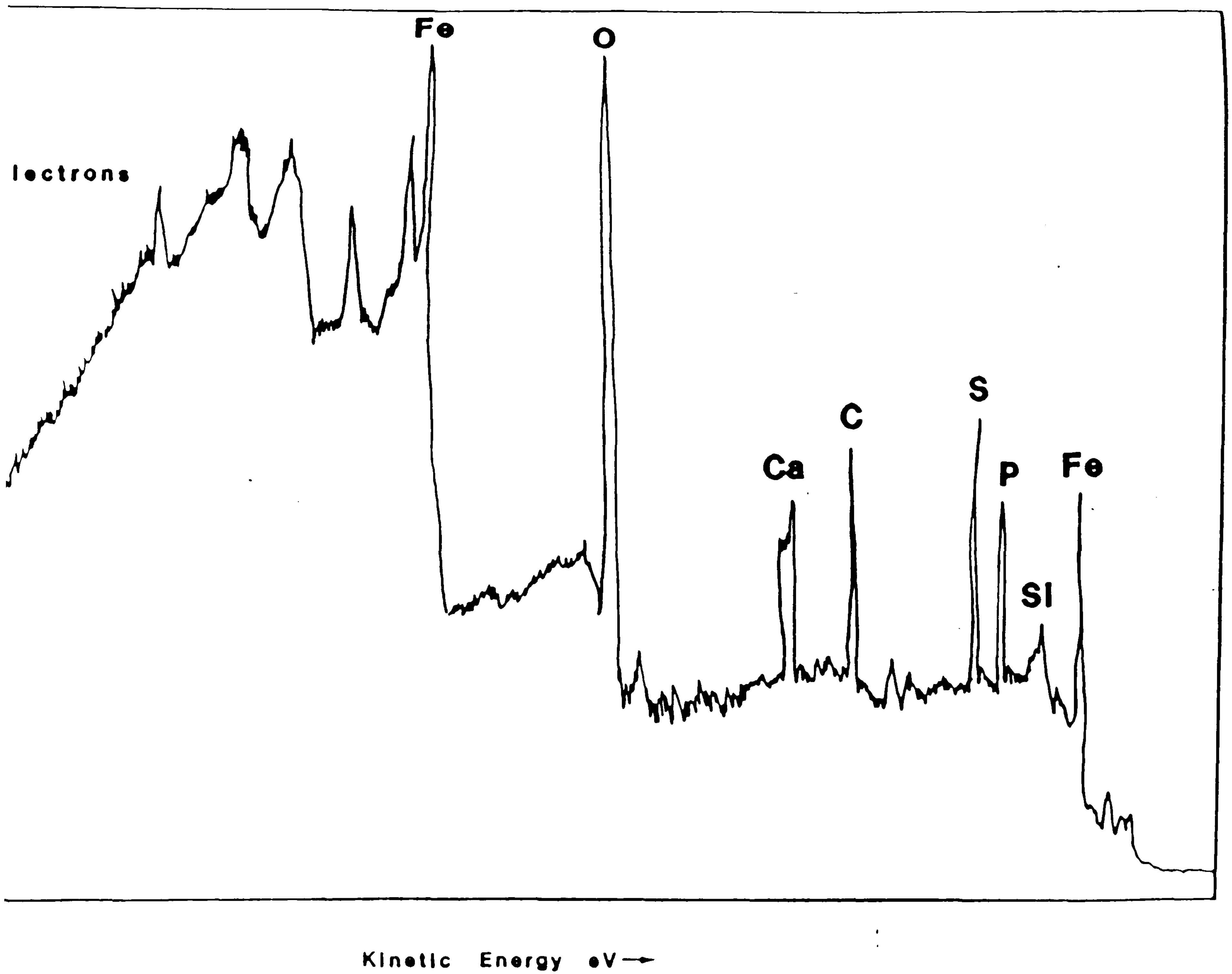
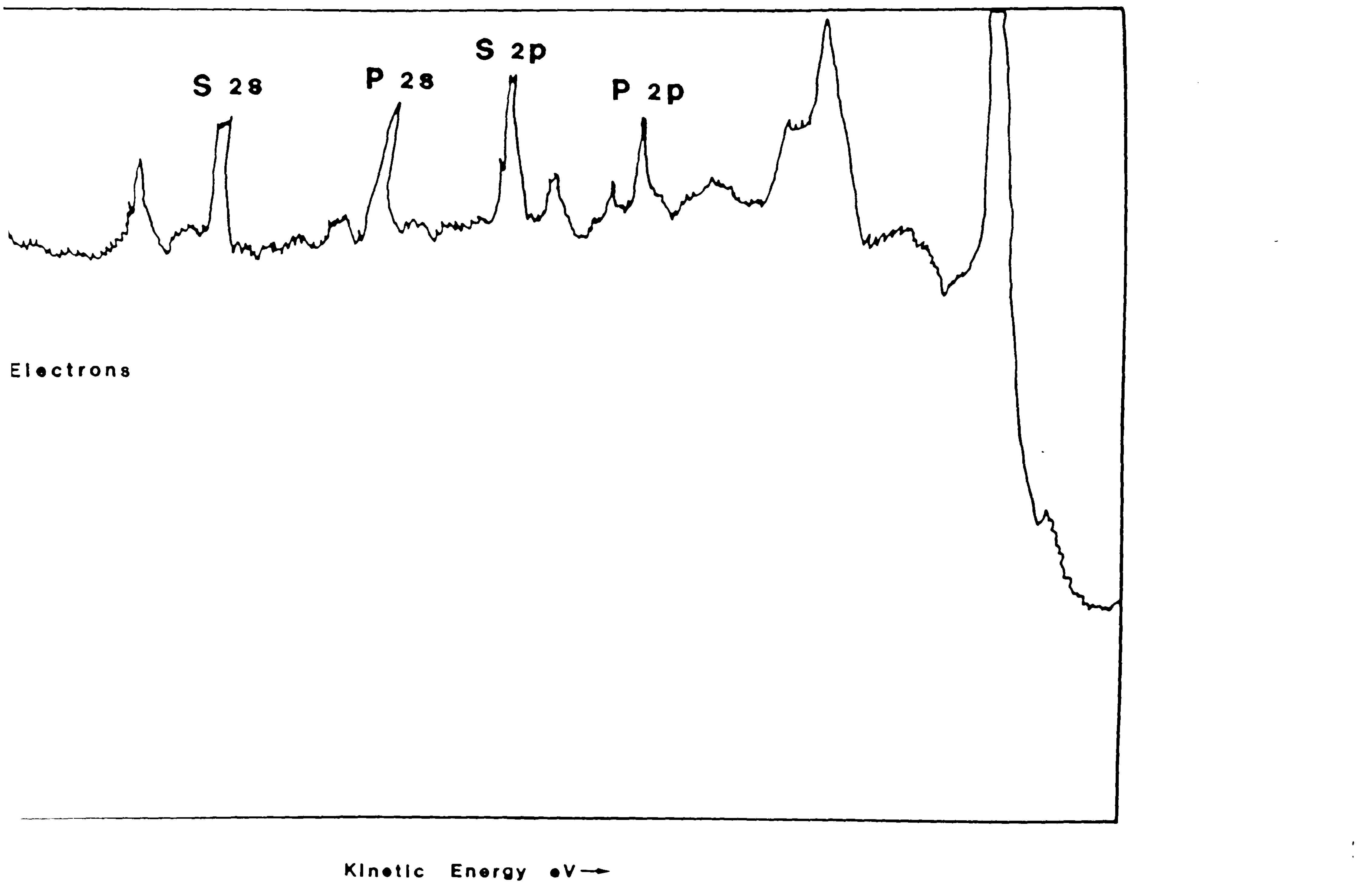
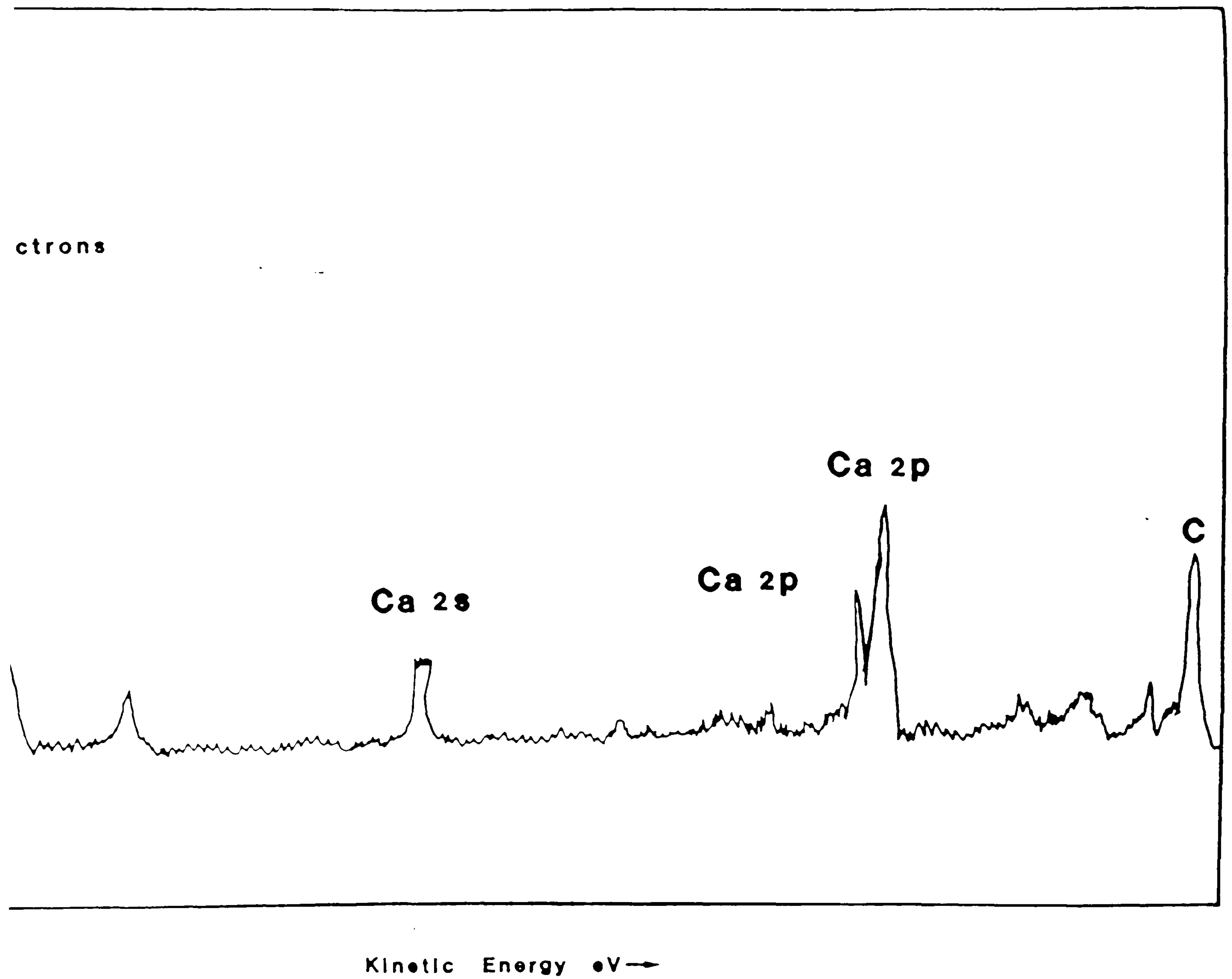


Fig.146 Energy spectrum for the polished flat using RL48 + carbon at 80°C, (load = 1 200N).


S 2s
P 2s
S 2p
P 2p
Electrons
Kinetic Energy eV →


ctrons
Ca 2s
Ca 2p
Ca 2p
C
Kinetic Energy eV→

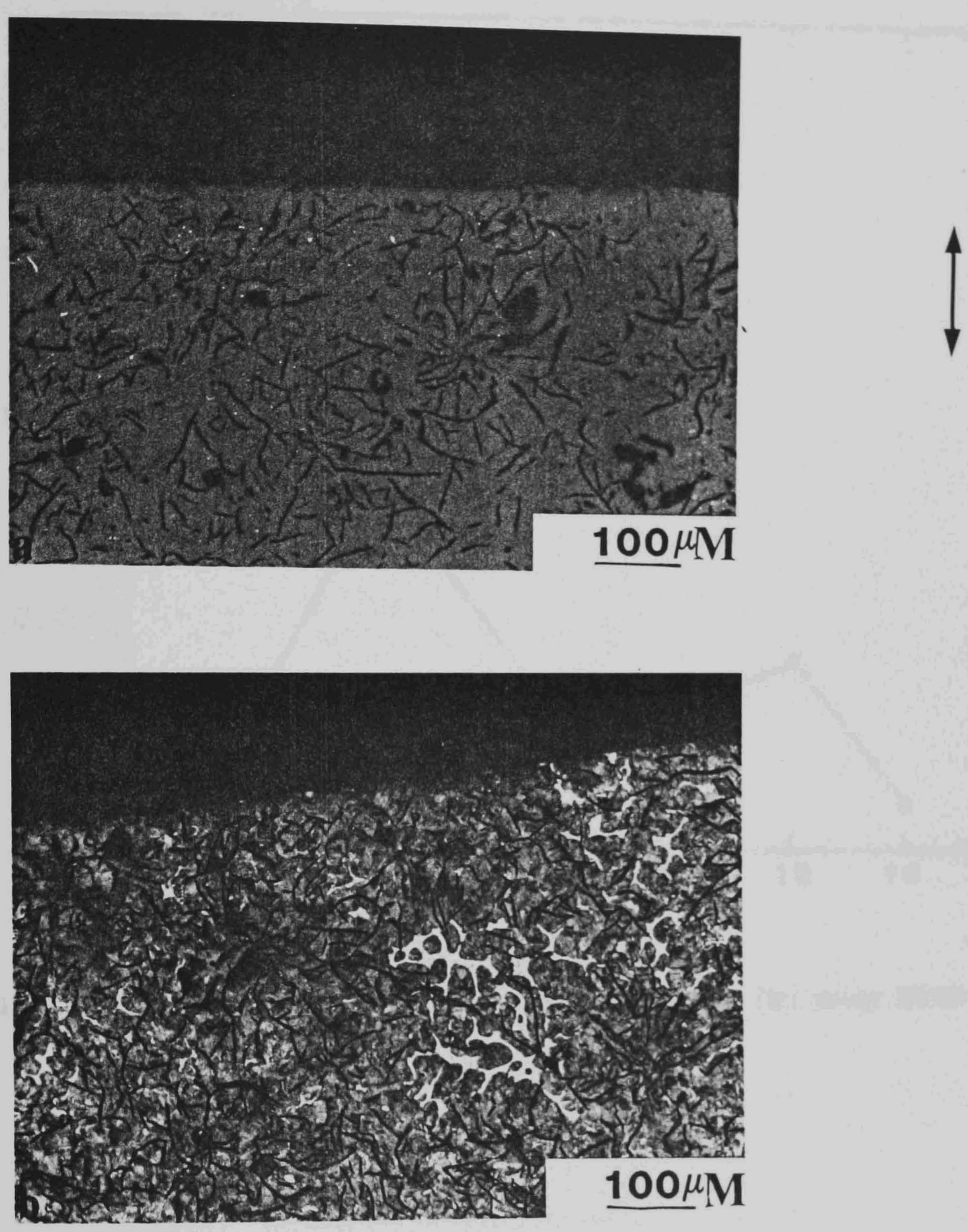


Fig.147 Taper section through polished flat using RL48 + carbon:
a) Unetched
b) Etched.

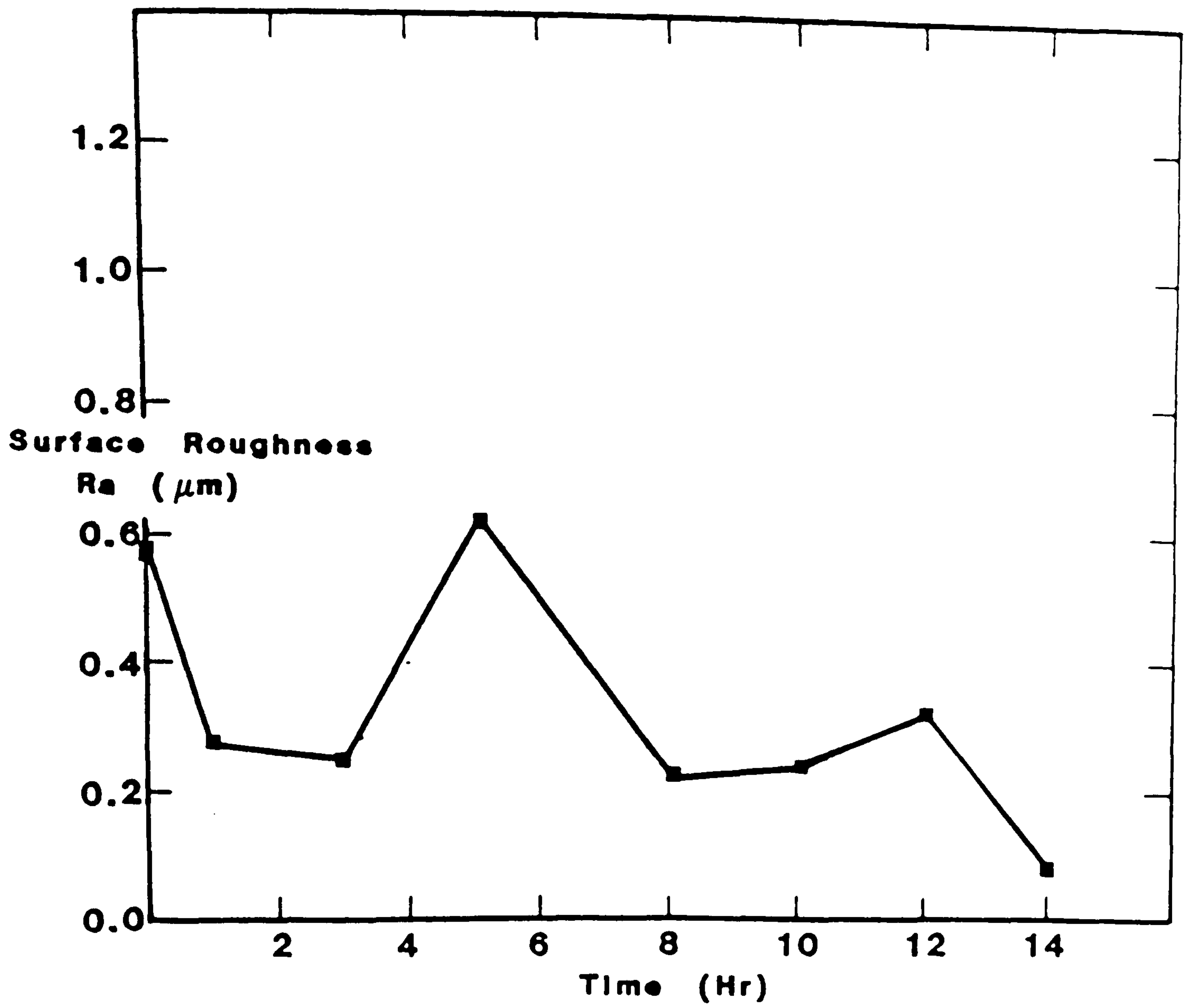


Fig.148 C.L.A. values for different test periods for the flat using RL48 + carbon at 80°C, (load = 1200N).

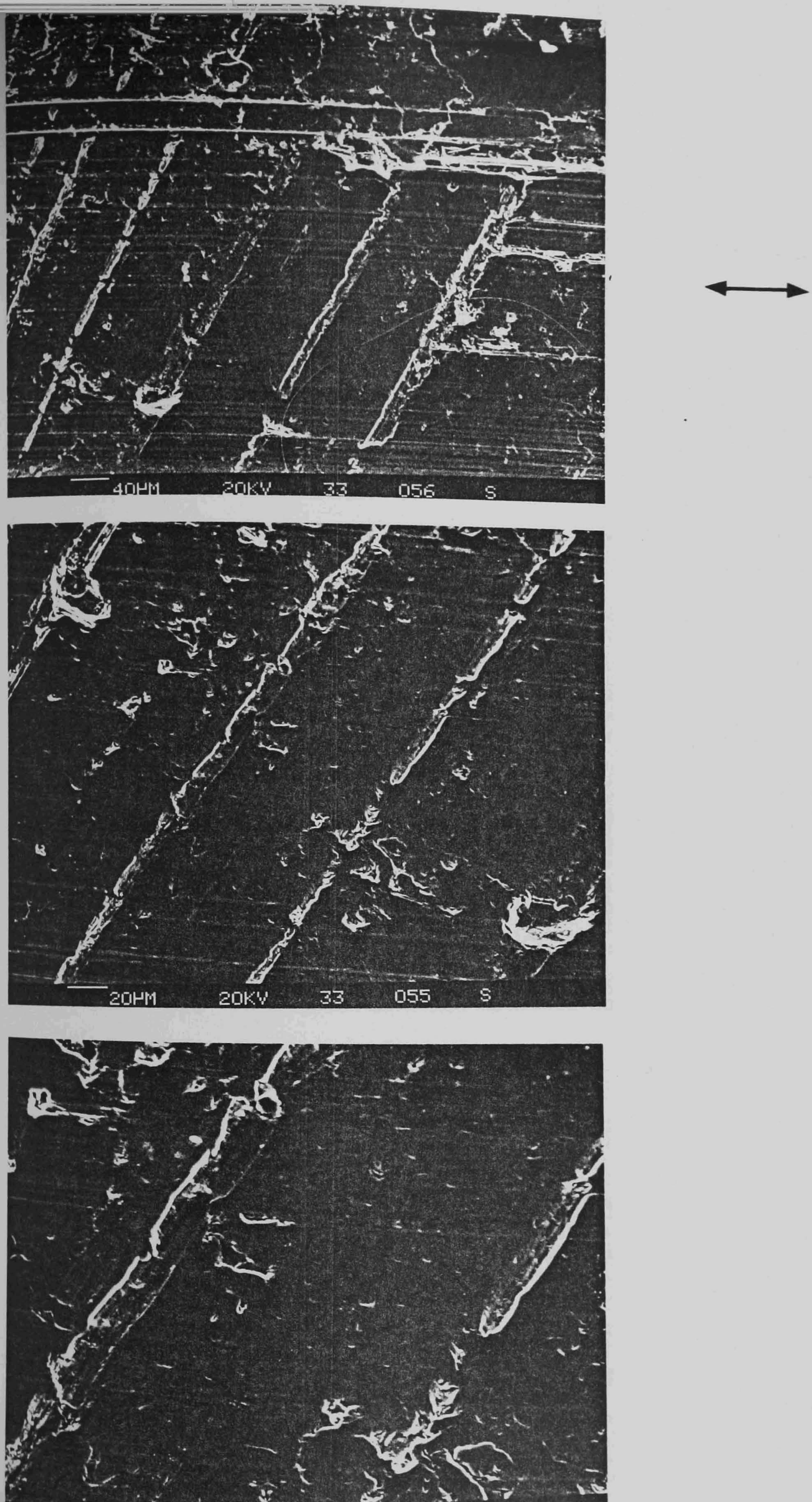


Fig.149 SEM appearance for the flat after 1 hour test using RL48 + carbon at 80^oC, (load = 1200N).

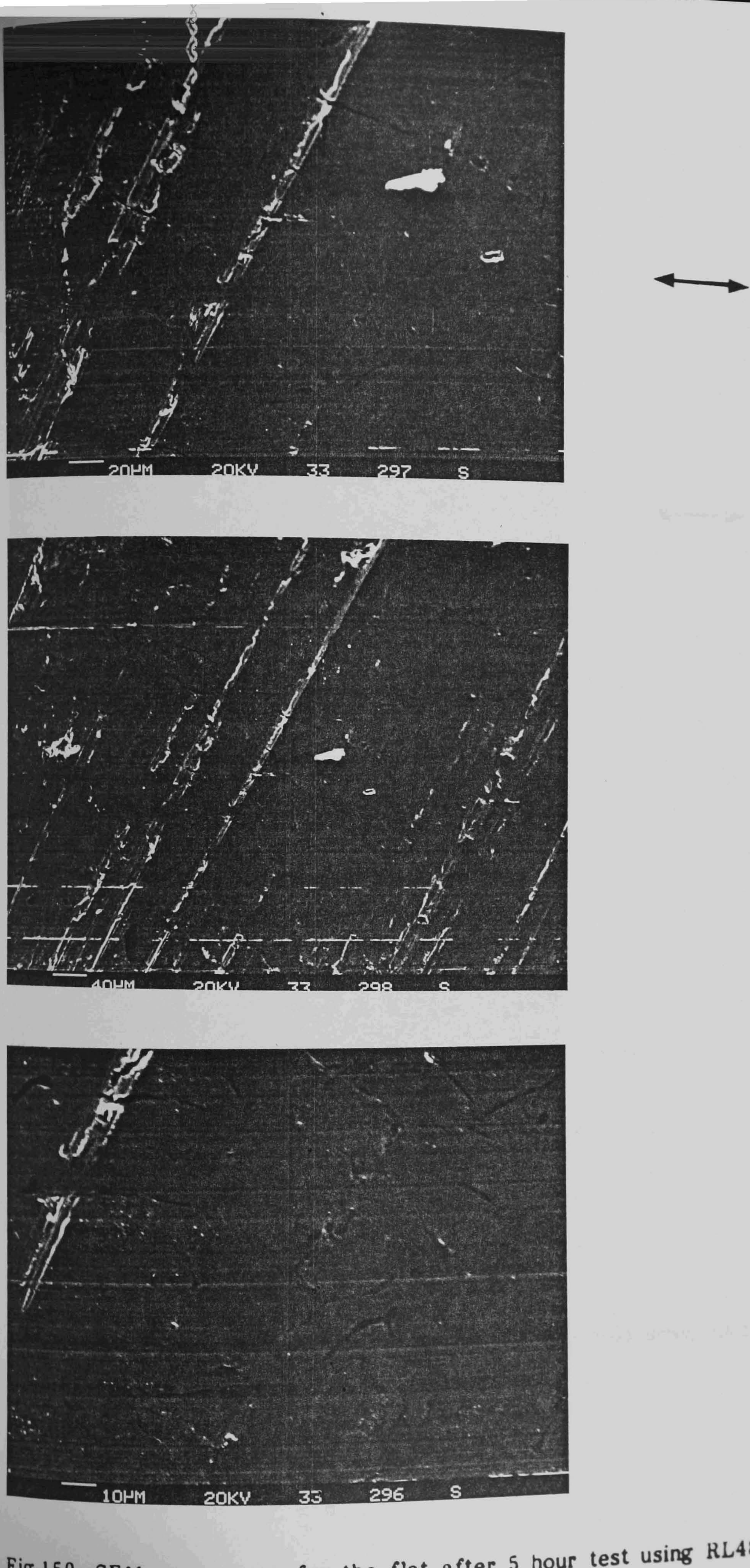


Fig.150 SEM appearance for the flat after 5 hour test using RL48 + carbon at 80^oC, (load = 1200N).

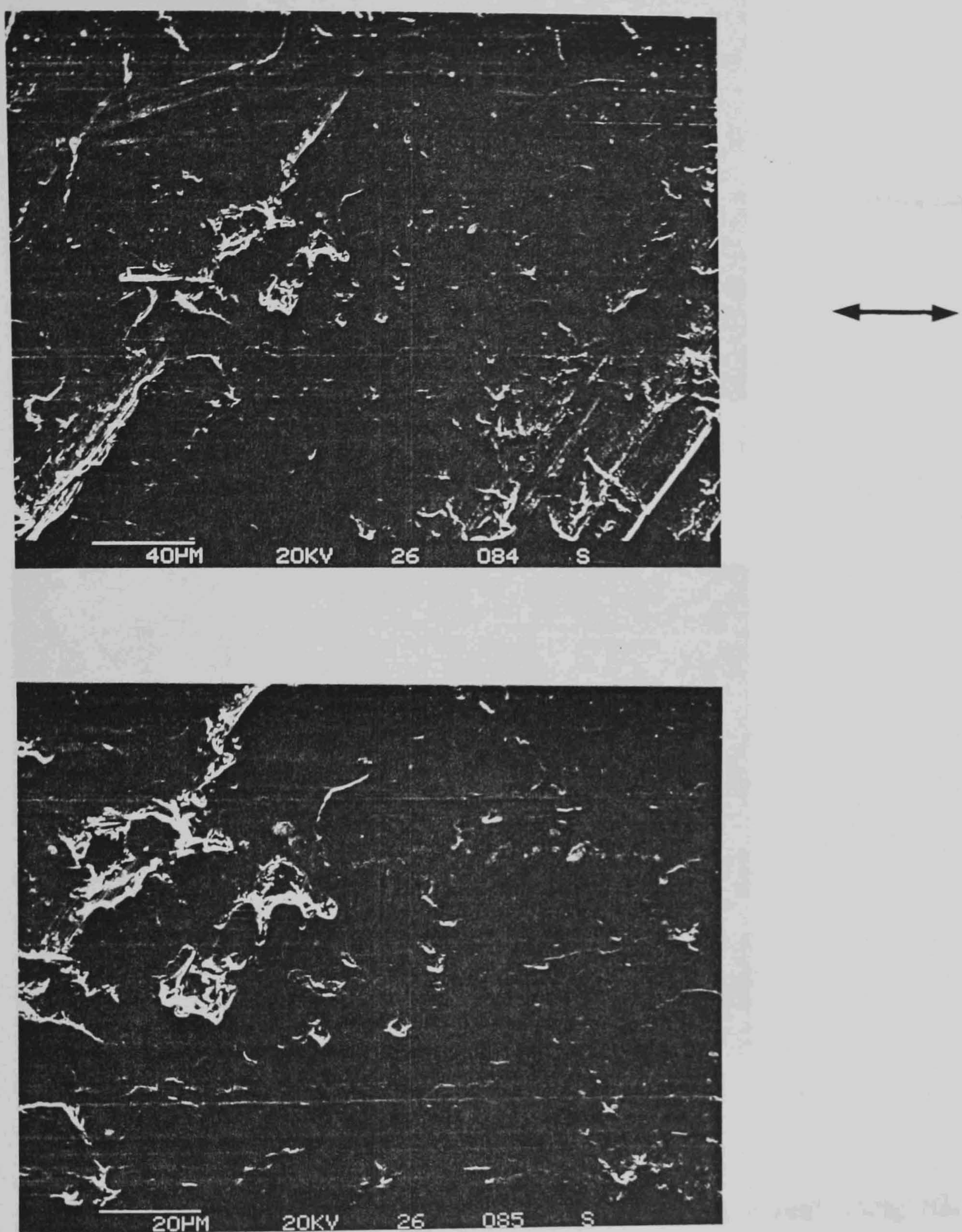


Fig.151 SEM appearance for the flat after 8 hour test using RL48 + carbon at $80^{o}C$, (load = 1200N).

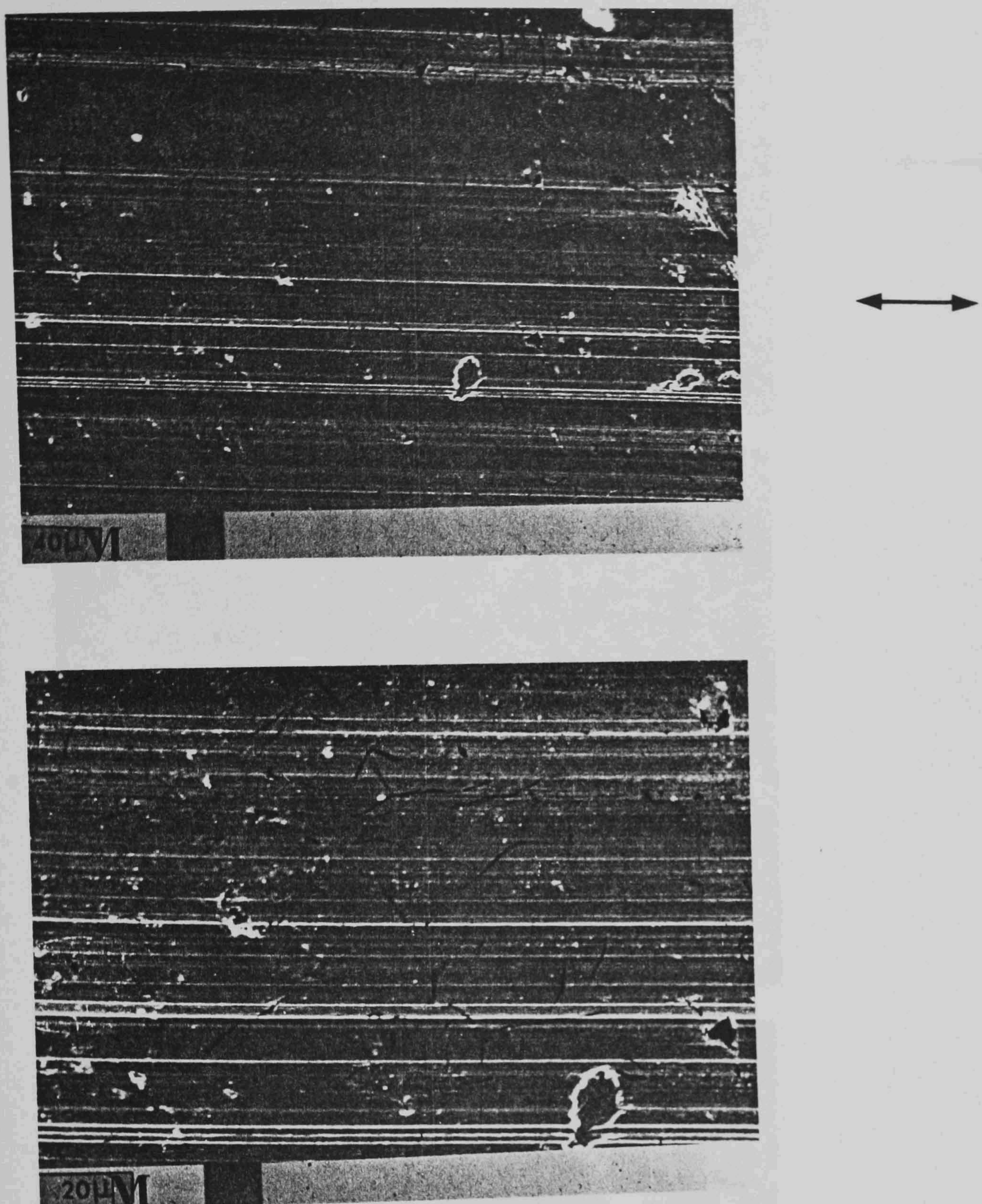


Fig.152 SEM appearance for the flat after 10 hour test using RL48 + carbon at 80°C, (load = 1200N).

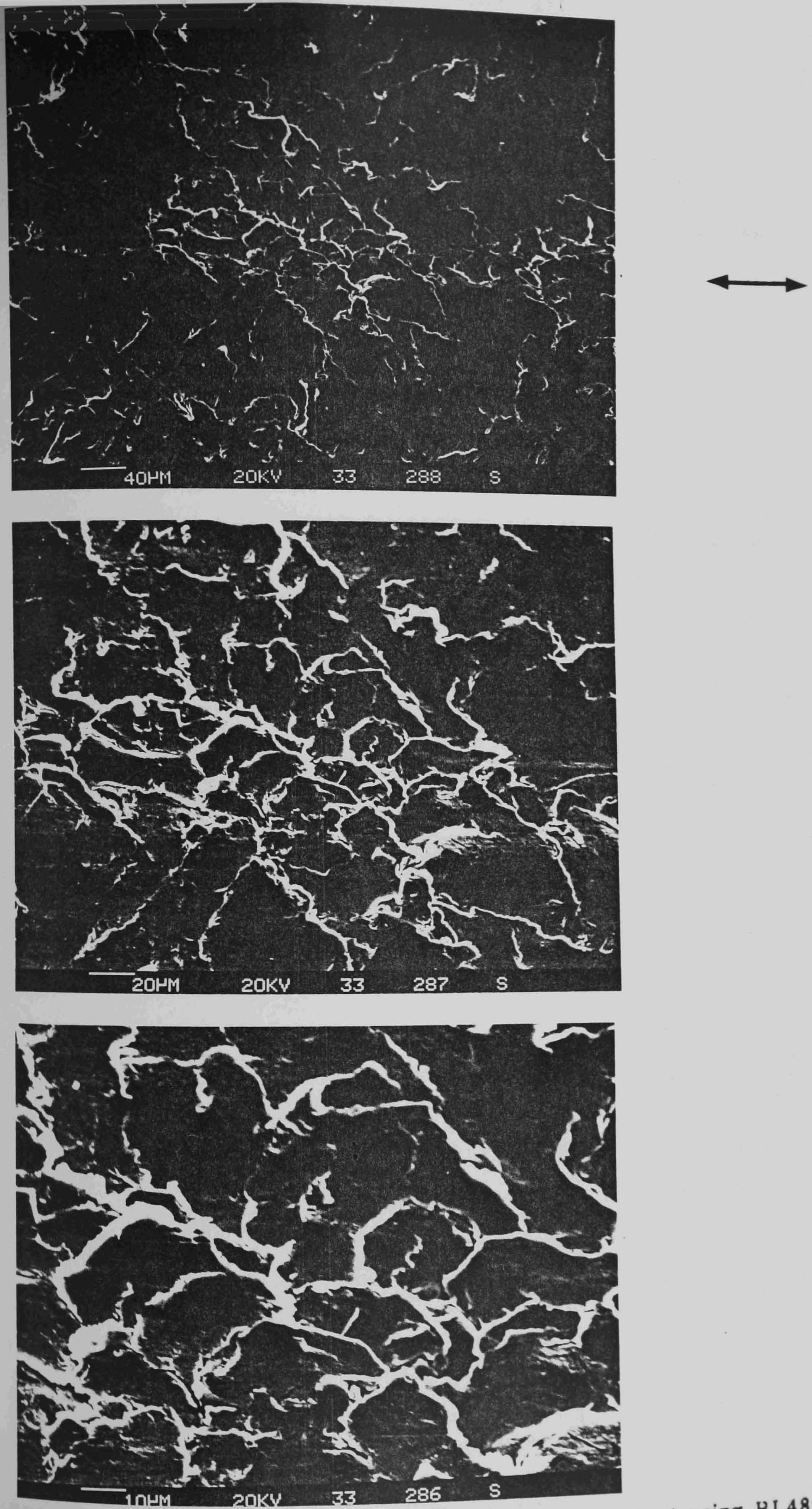


Fig.153 SEM appearance for the flat after 16 hour test using RL48 + carbon at 80°C, (load = 1200N).

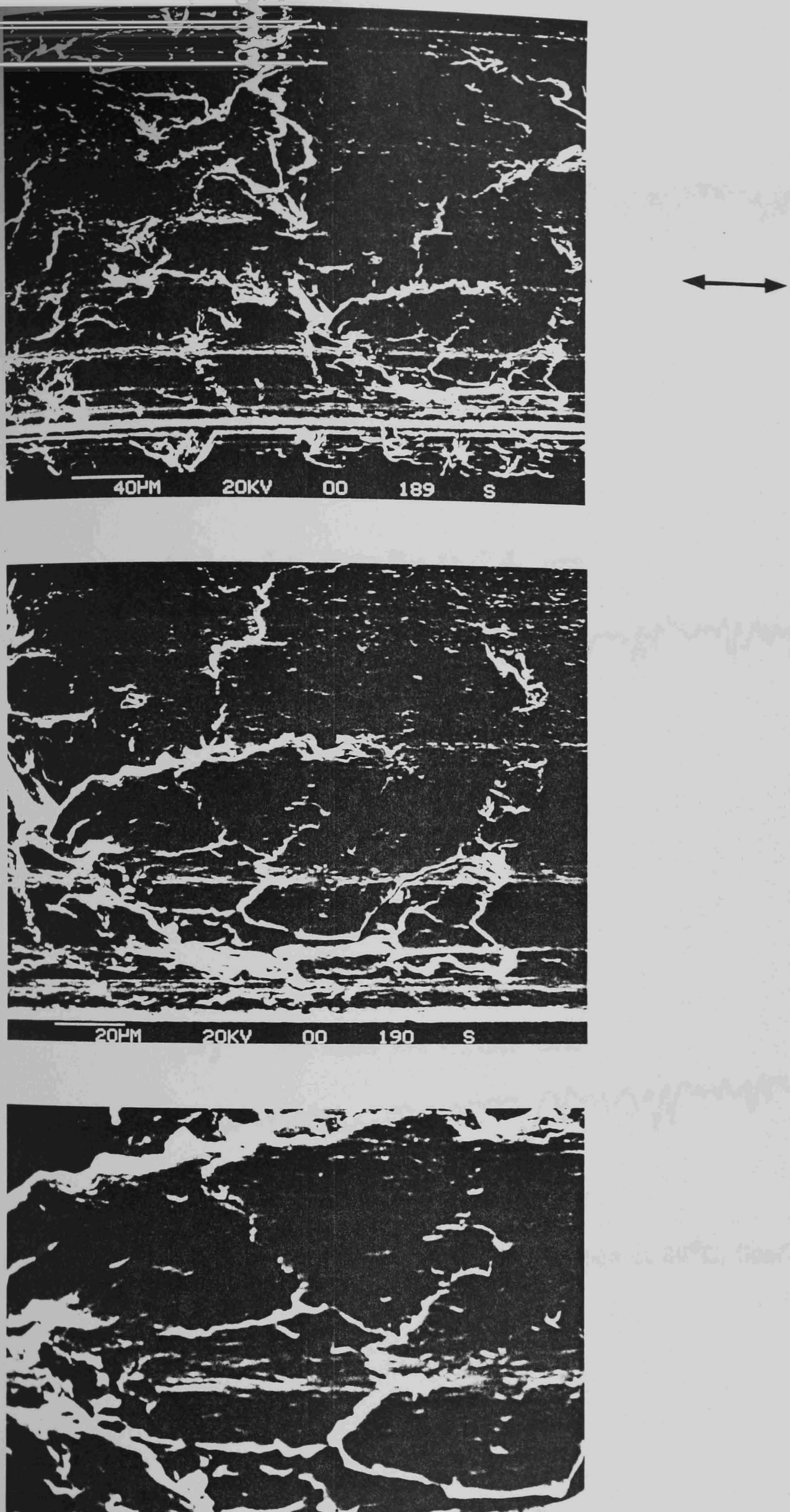


Fig.154 SEM appearance for the flat after 14 hour test using RL47 + carbon at $80^{o}C$, (load = 1200N).

(b) 15 mm Ra 0.34 μm

(c) 30 mm Ra 0.33 μm

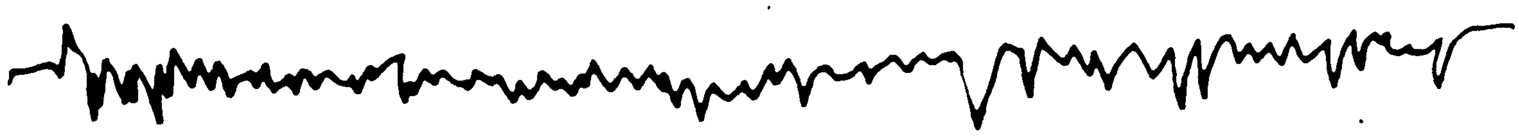


Fig.155 C.L.A. values for the flat using RL47 + carbon at 80°C, (load = 1 200N).

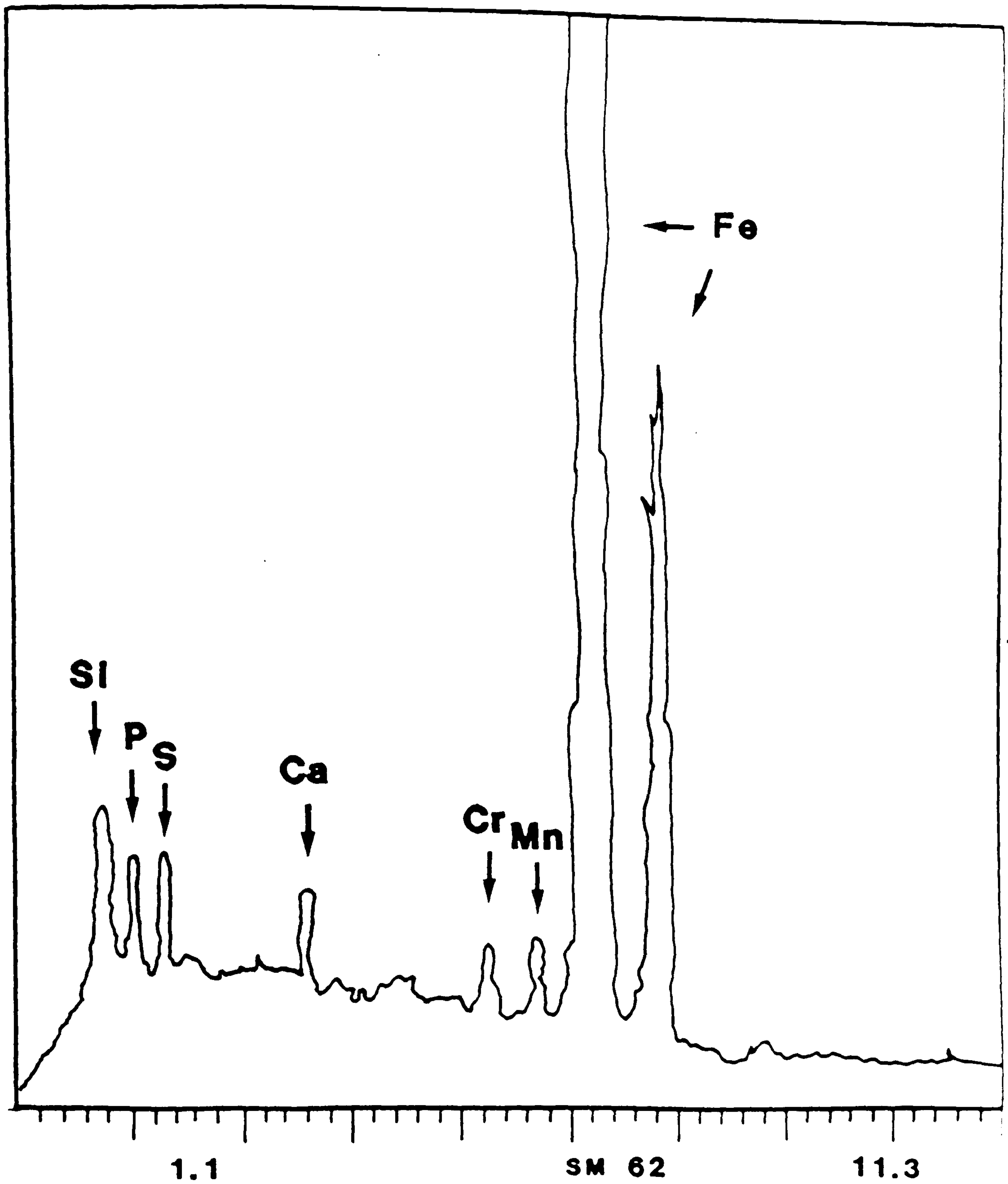
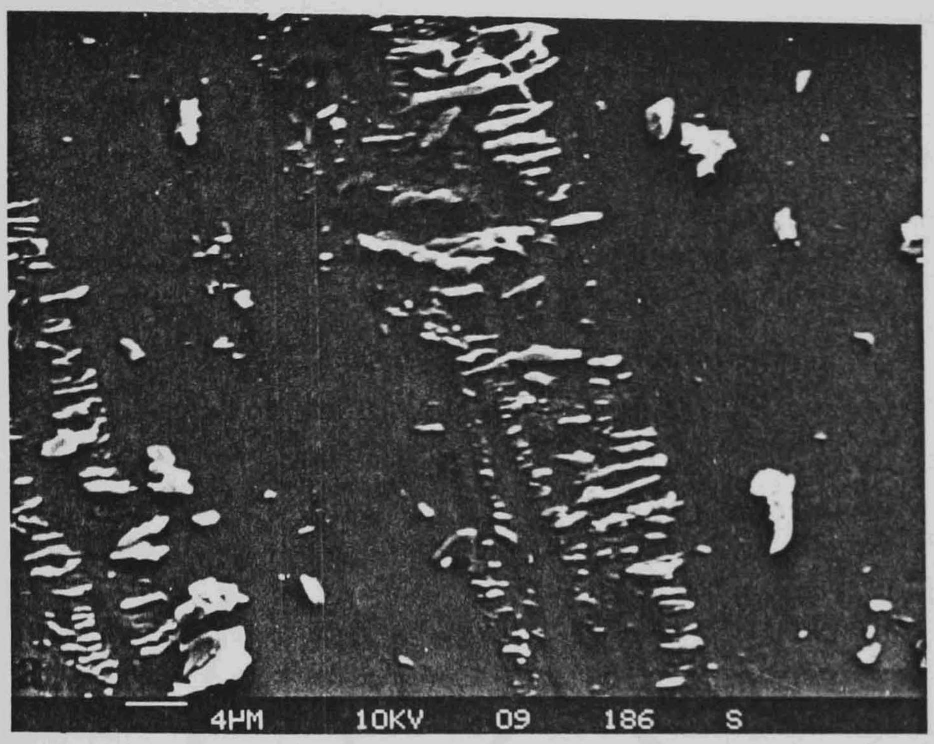


Fig.156 E.p.M.A. for the flat using RL47 + carbon at 80^{o}C, (load = 1200N



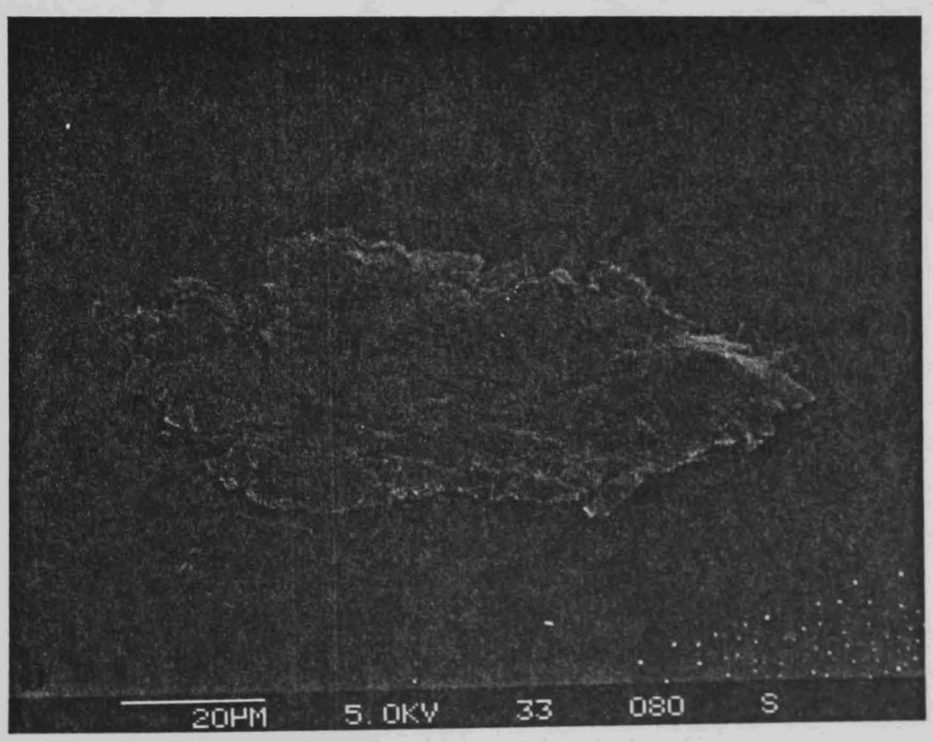


Fig.157 Debris produced using RL48 + carbon at $80^{o}C$, (load = 1200N):
a) After 1 hour test
b) After 14 hour test.

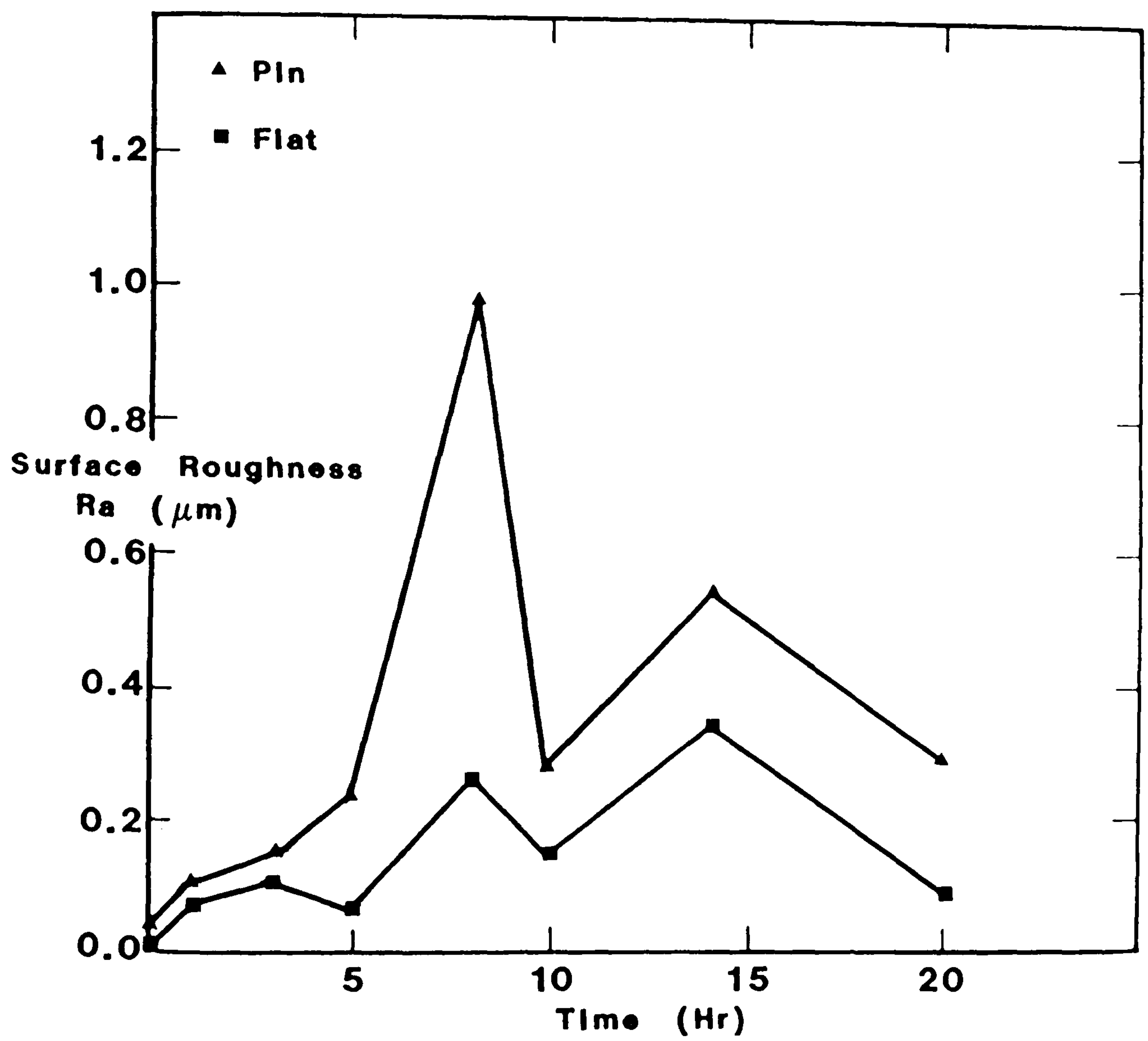


Fig.158 C.L.A. values for pin and flat for different test periods using RL48 at 80°C, (load = 1200N).

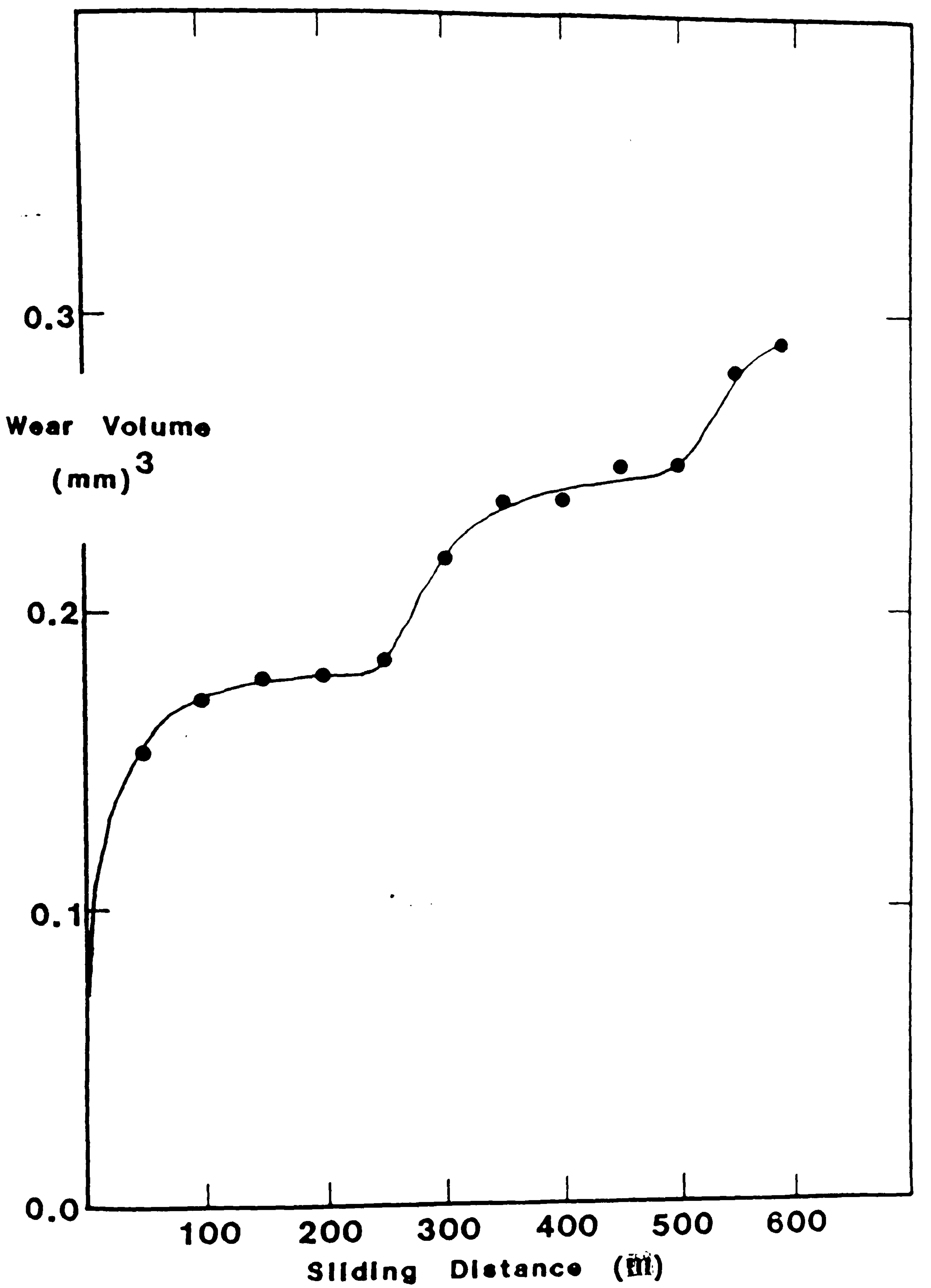


Fig.159 Wear volume versus sliding distance.

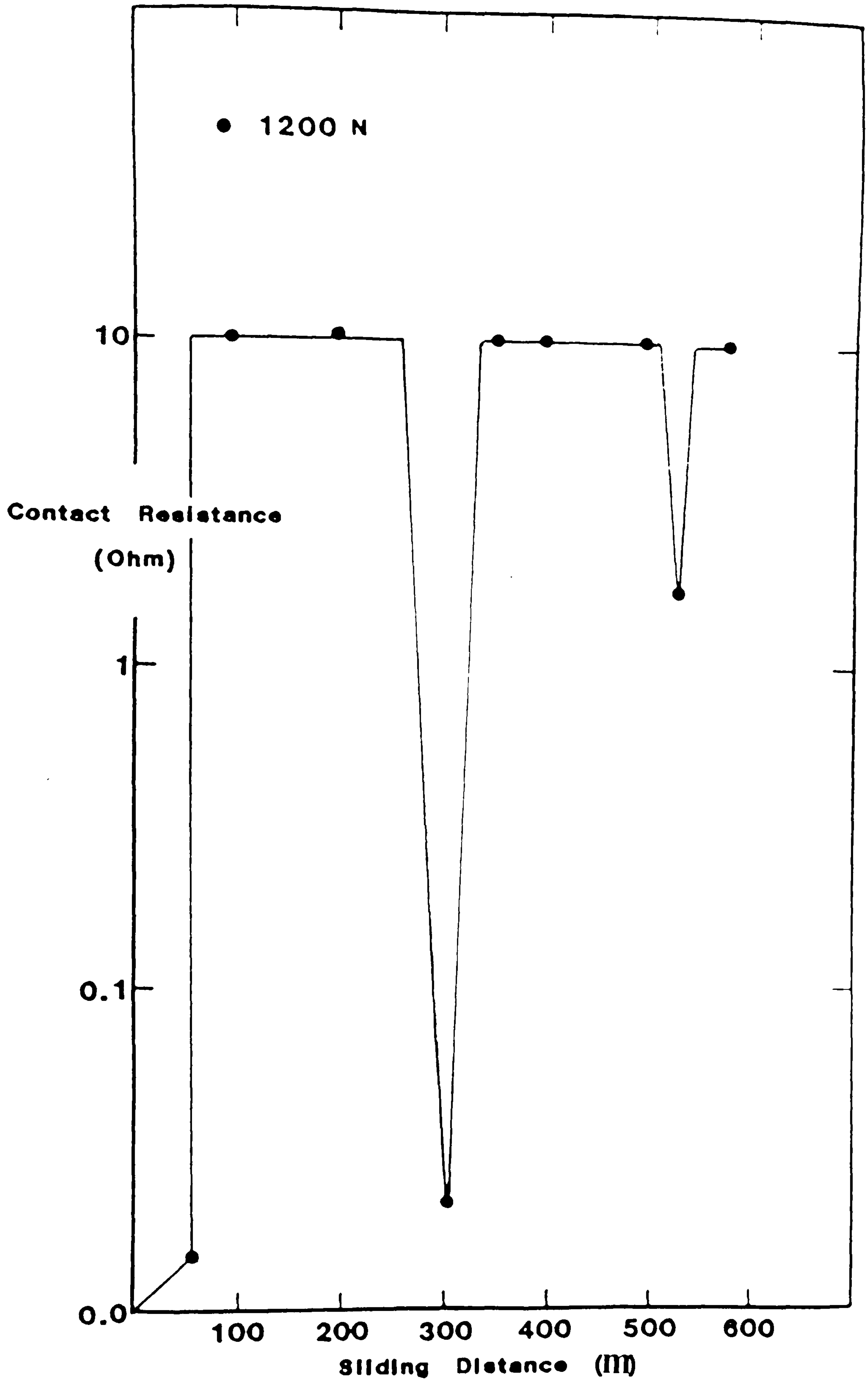


Fig.160 Contact resistance versus sliding distance.

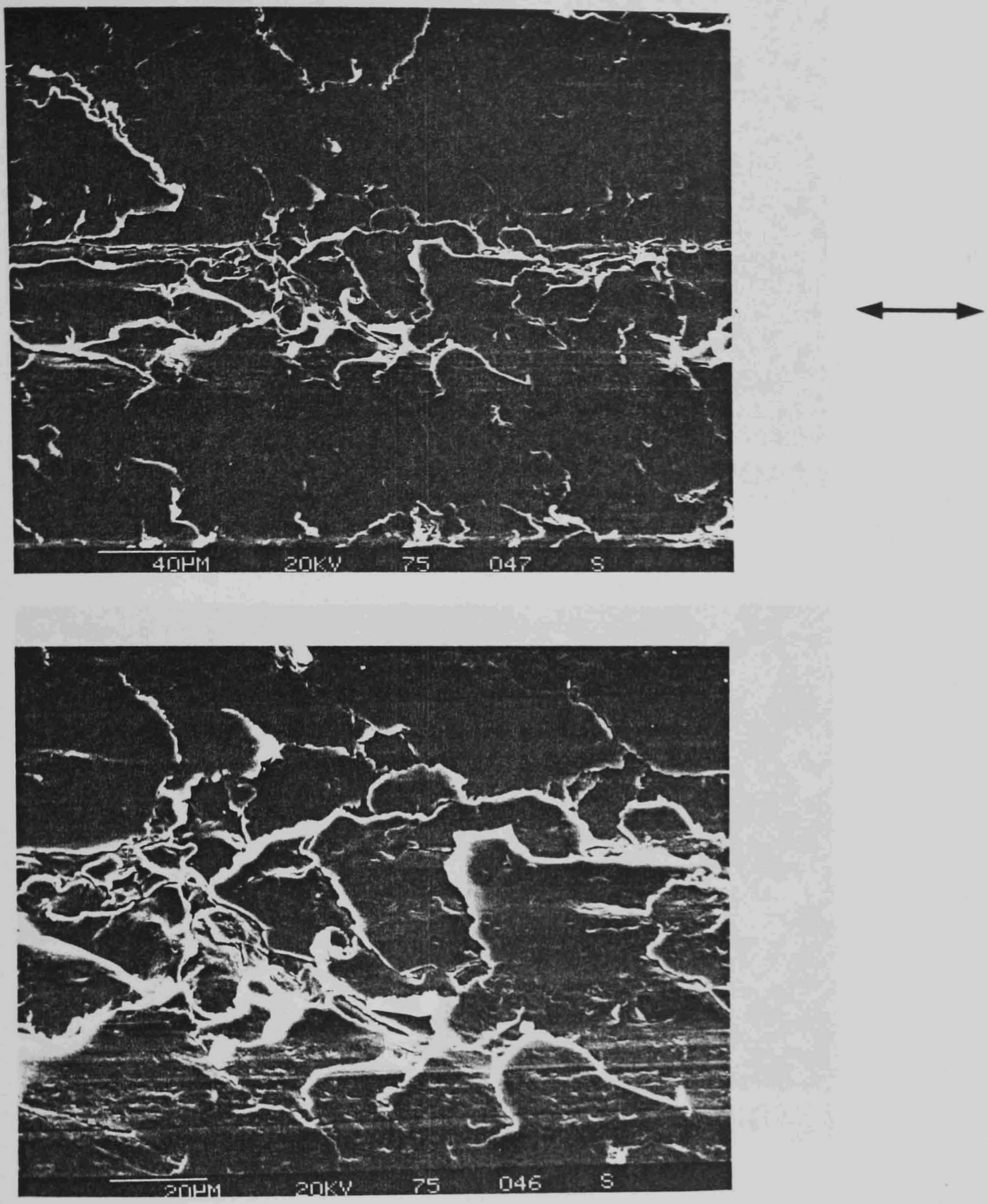


Fig.161 SEM appearance for the flat worn after 20 hour test using RL48 at 80°C, (load = 1 200N).

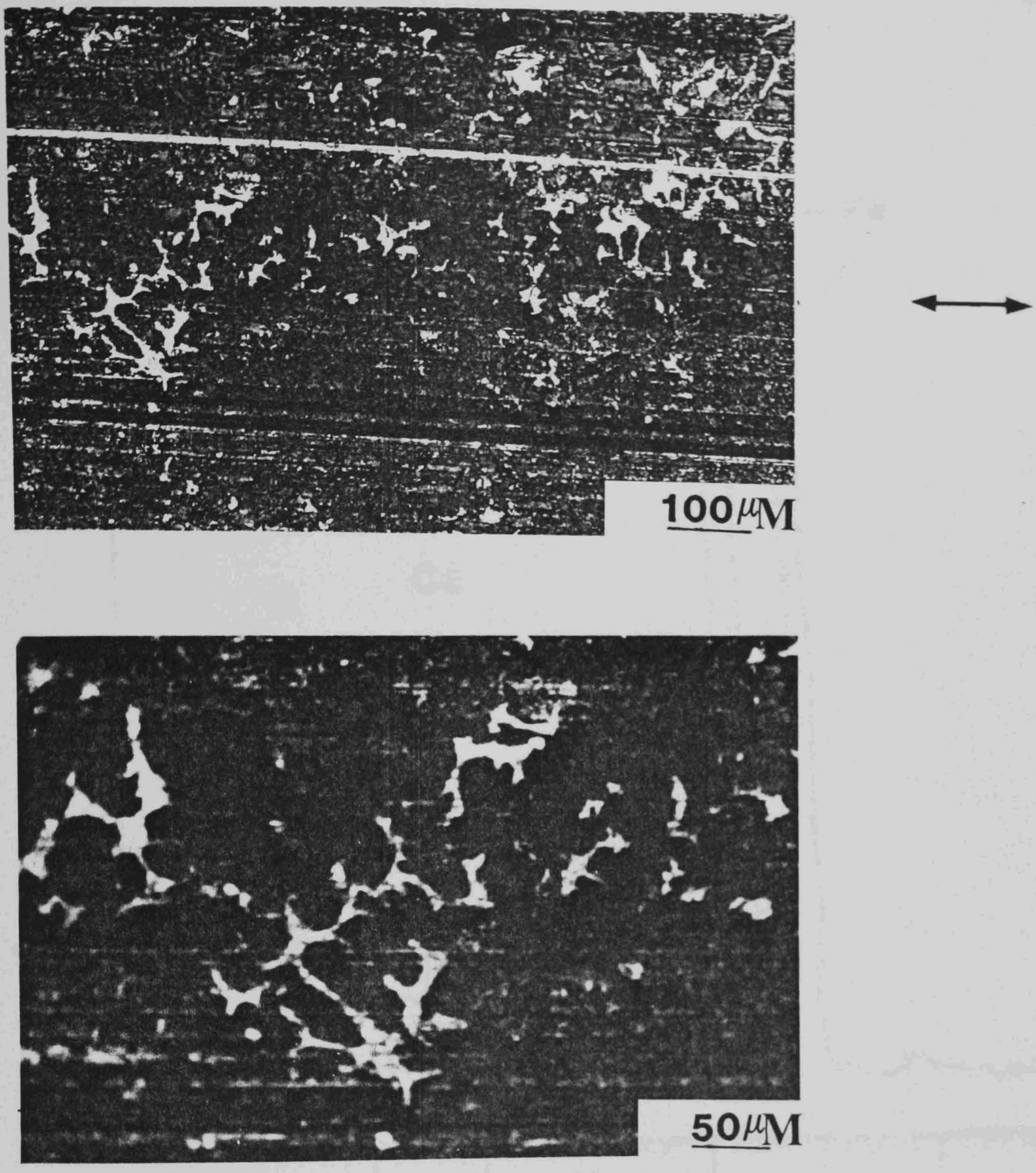


Fig.162 The flat worn after 20 hour test using RL48 at 80^{o}C, (load = 1200N).

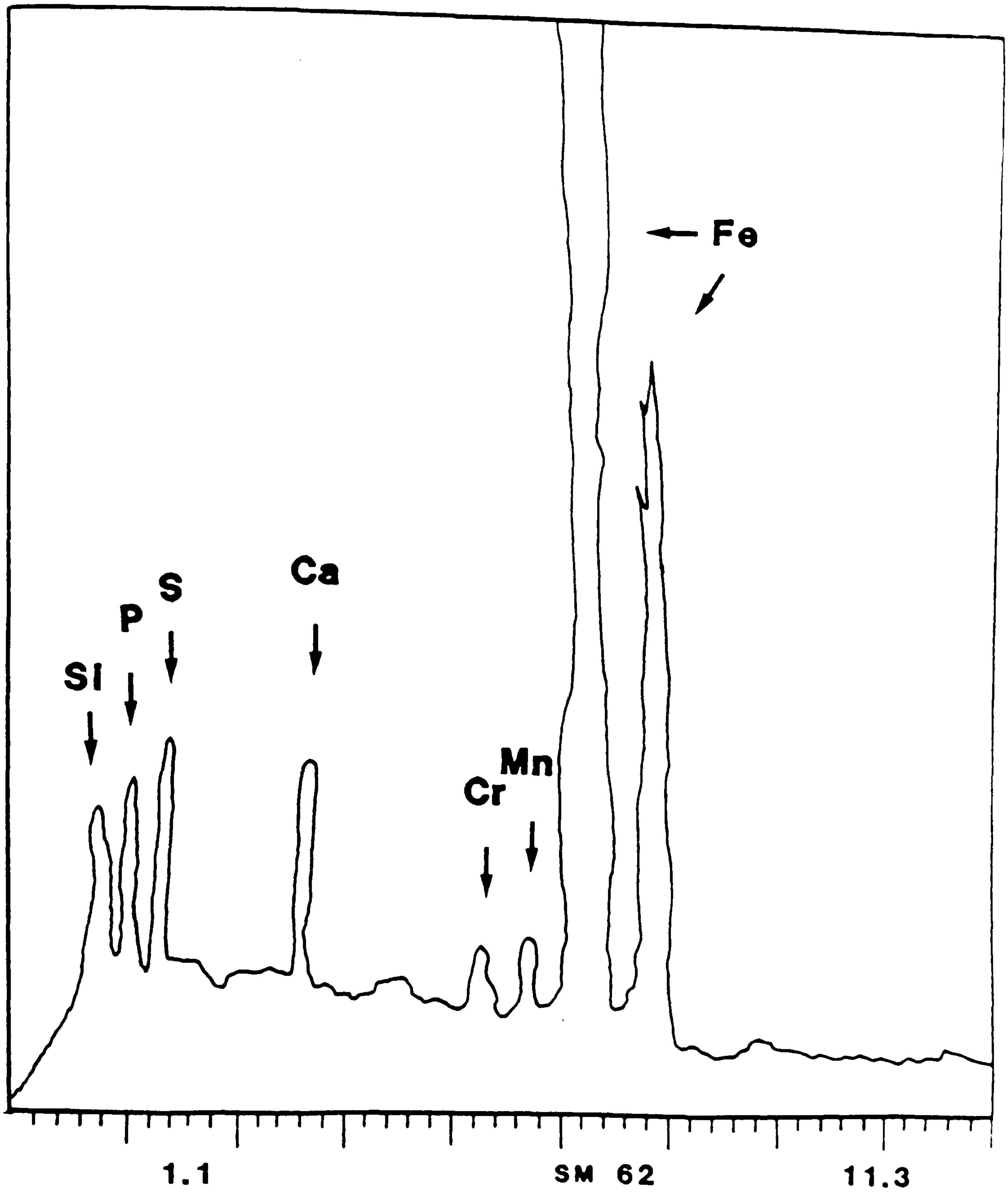
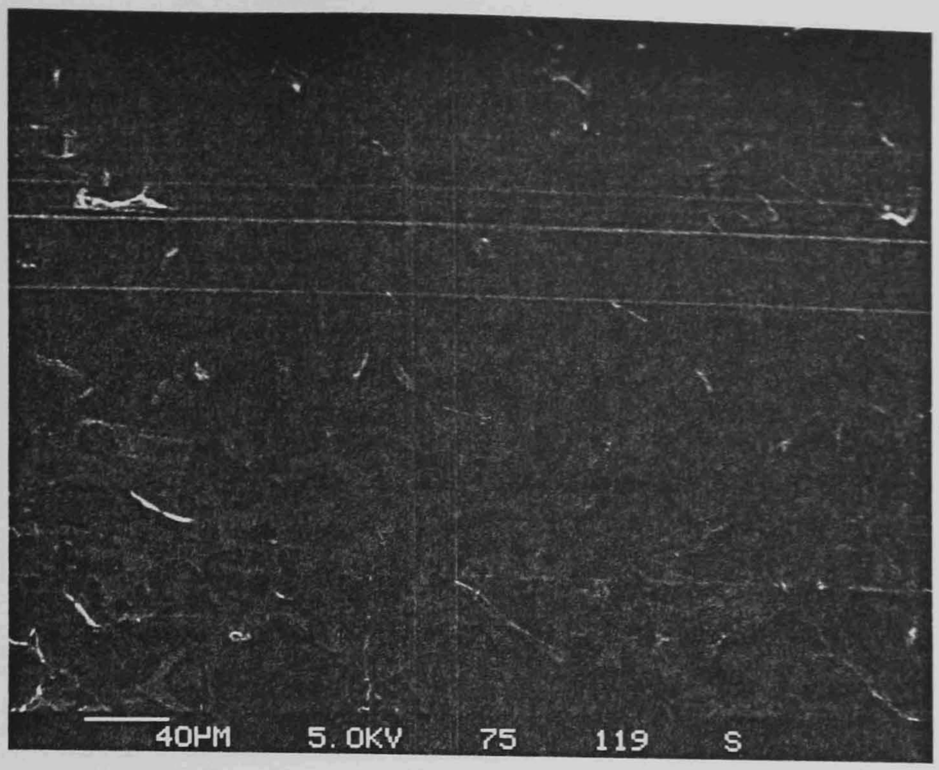
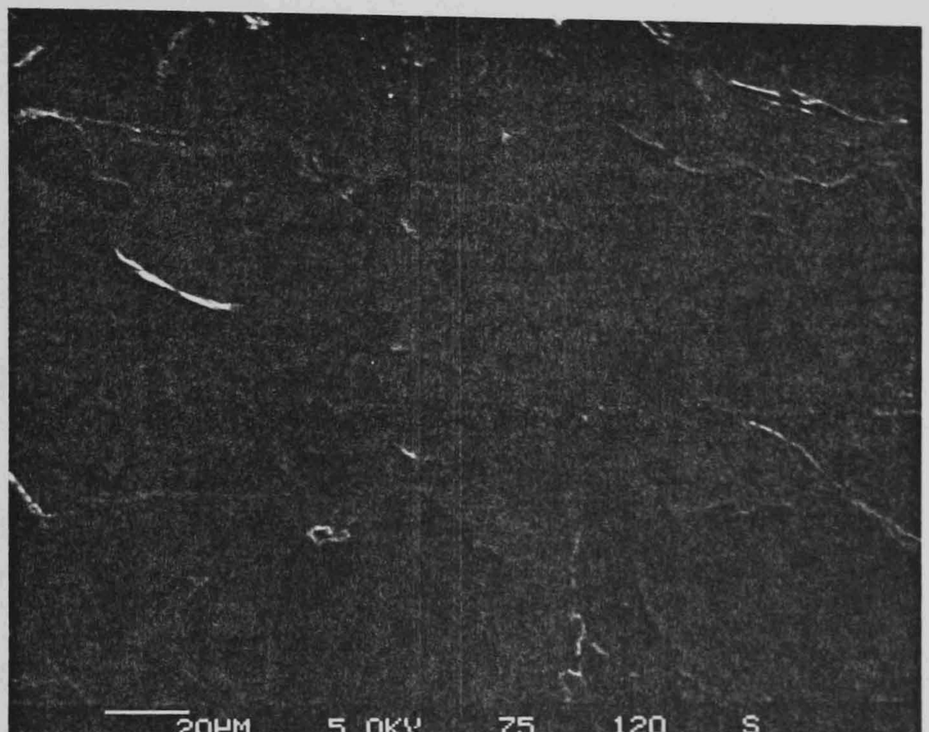


Fig.163 E.p.M.A. for the flat worn after 20 hour test using RL48 at 80°C, (load = 1200N).





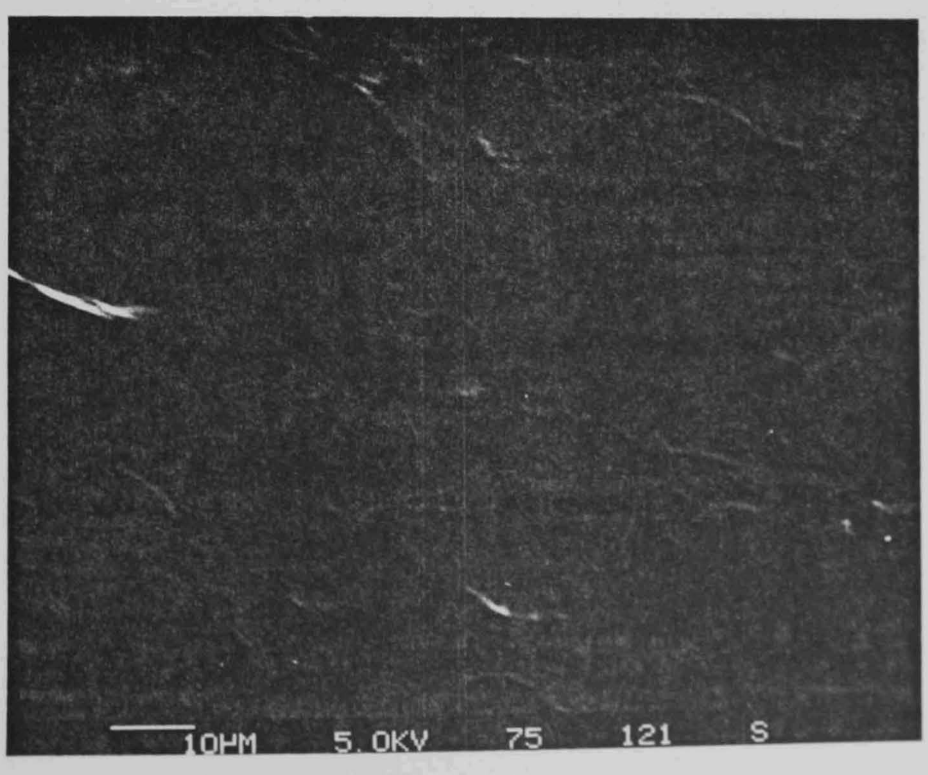


Fig.164 The flat worn after 20 hour test using RL48 at 80°C showing the internal corrosive wear and some grooves of abrasive wear.

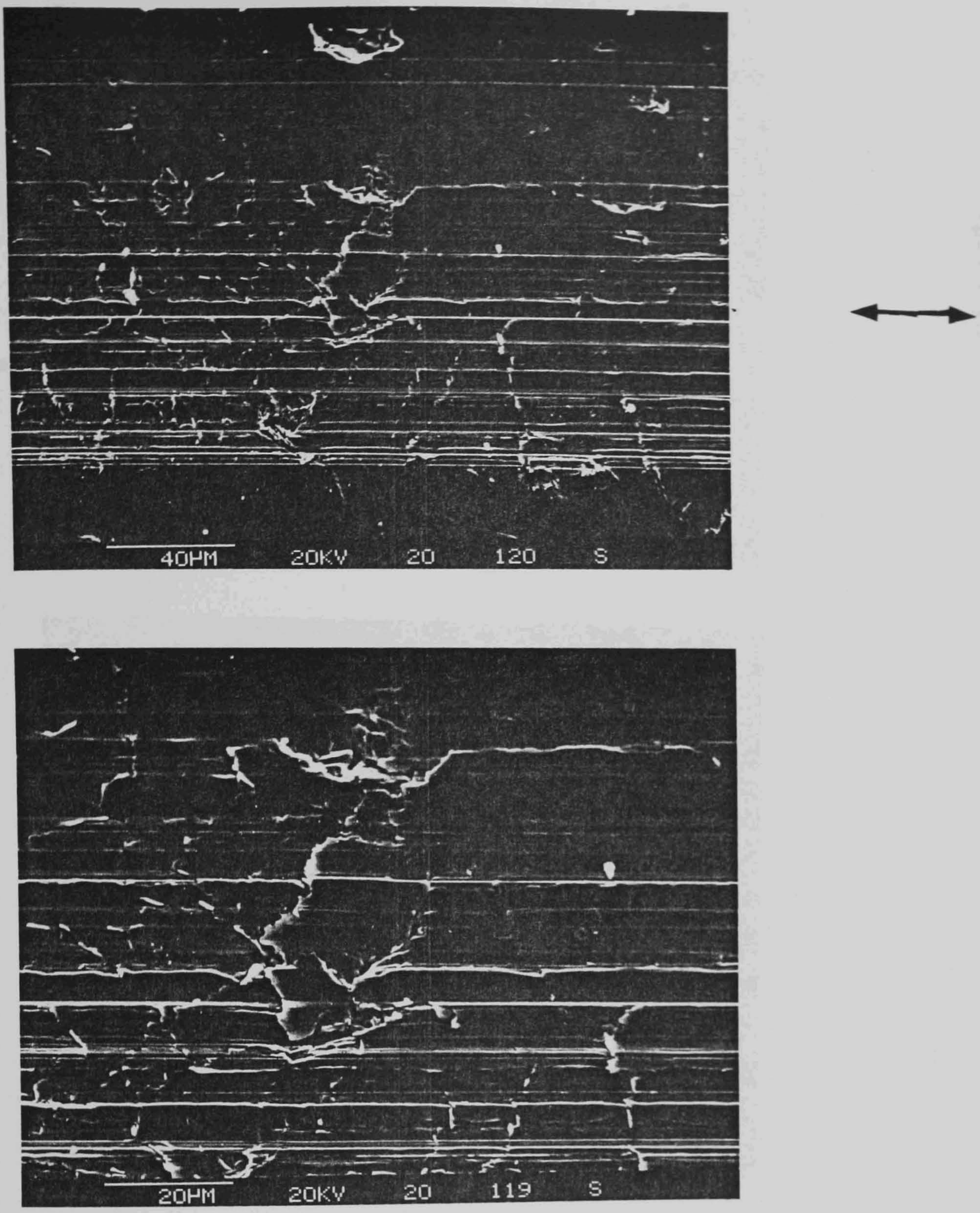


Fig.165 Equilibrium wear for the flat worn using RL48 at 80^{o}C, (load = 1 200N).

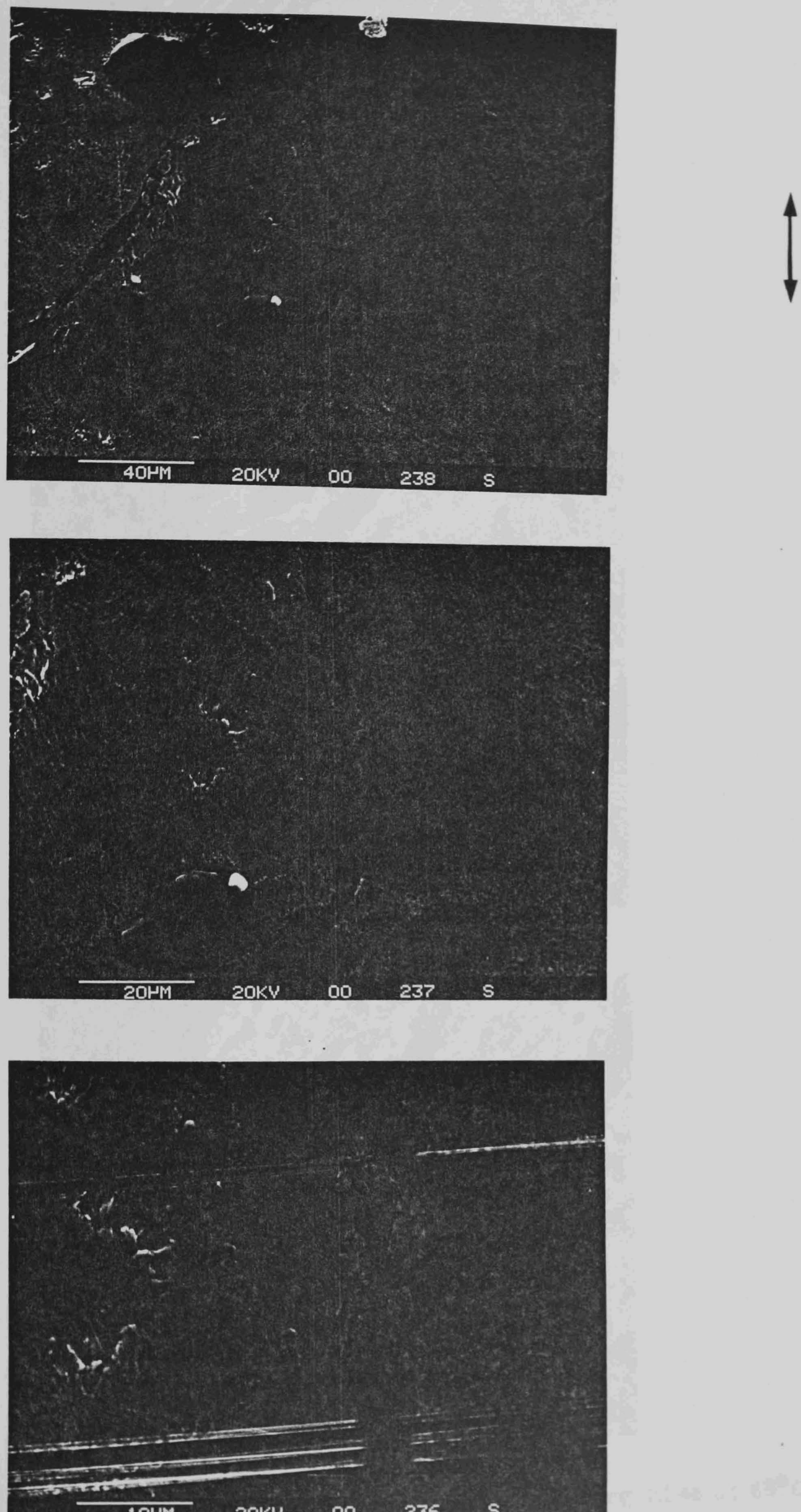


Fig.166 SEM appearance for the pin worn after 20 hour test using RL48 at 80°C, (load = 1200N).

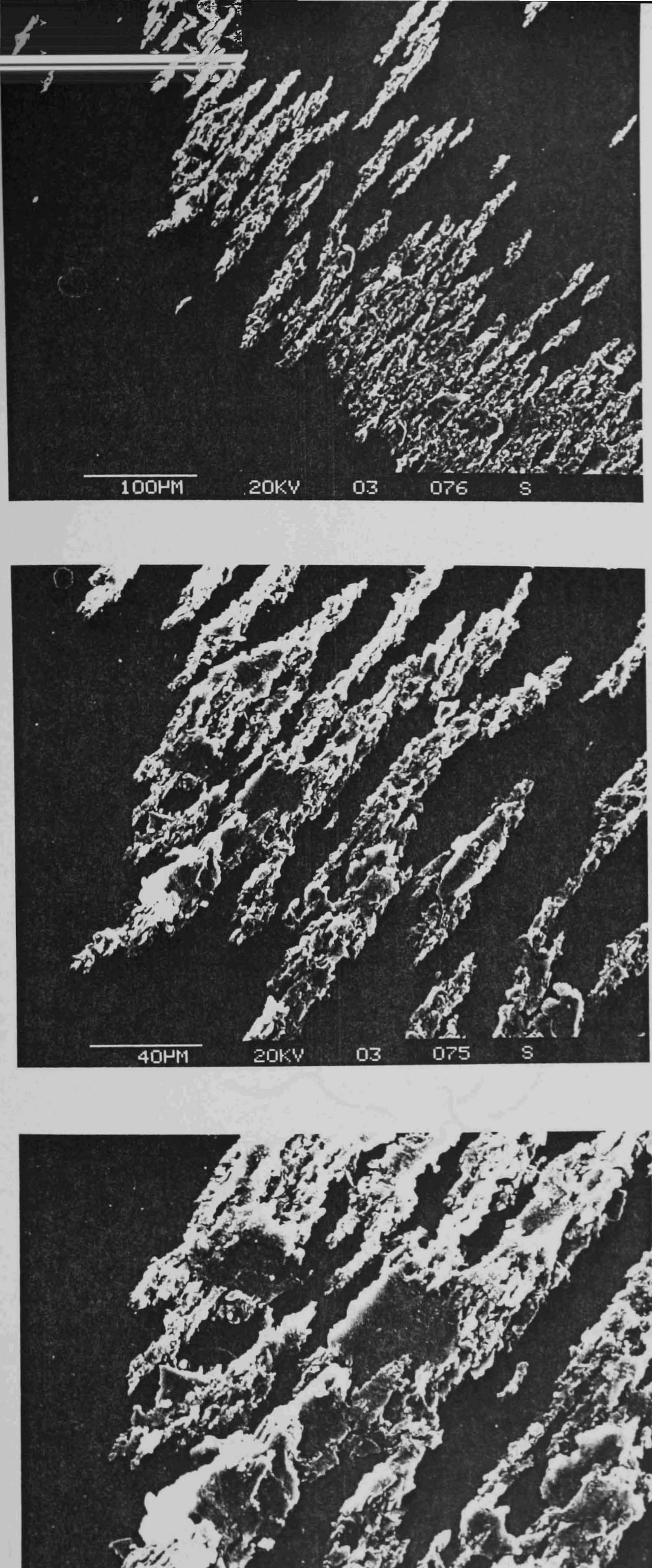


Fig.167 Debris produced after 20 hour test using RL48 at 80°C, (load = 1200N).

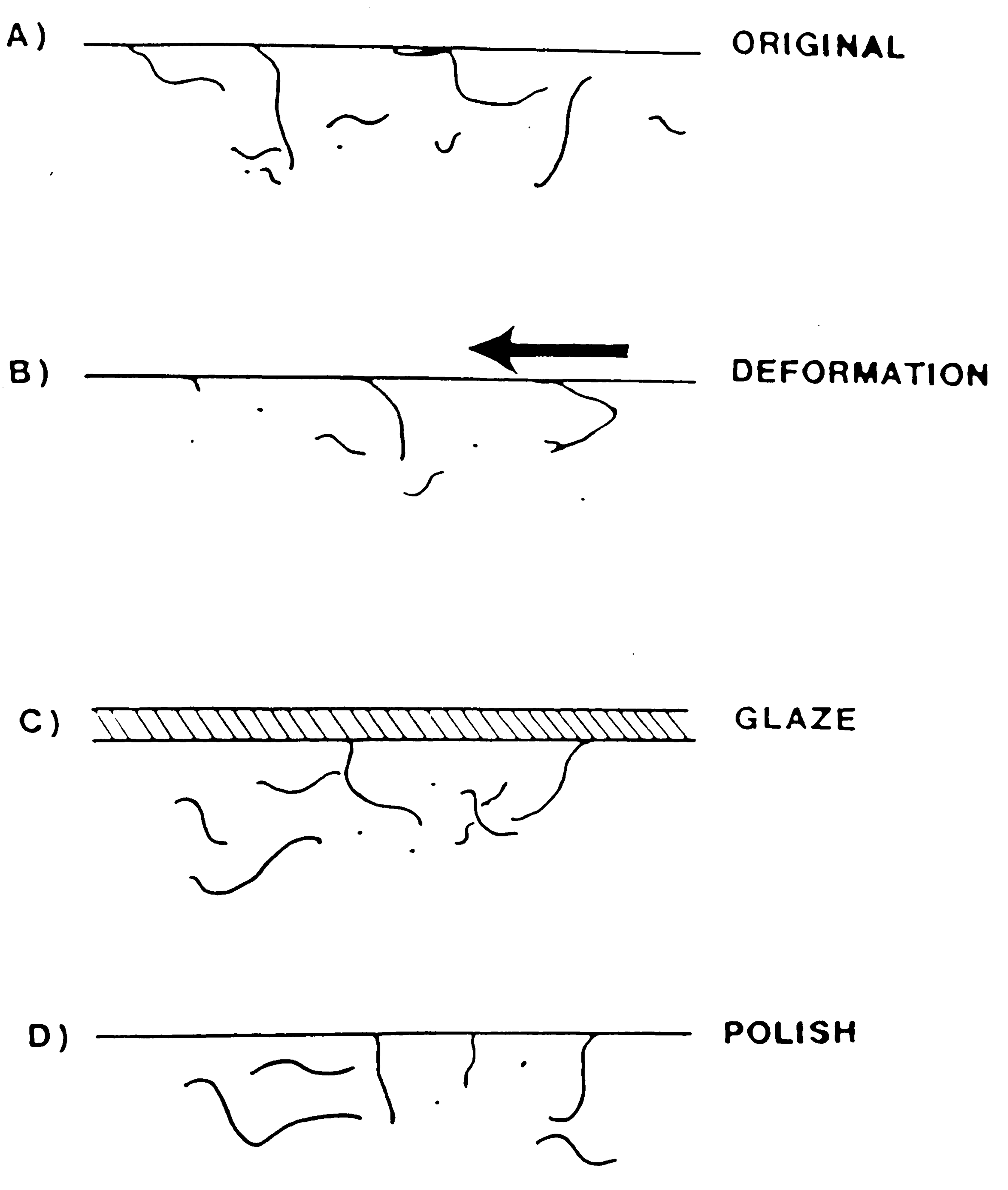


Fig.168 Schematic view of the surface prouduced by different processes.

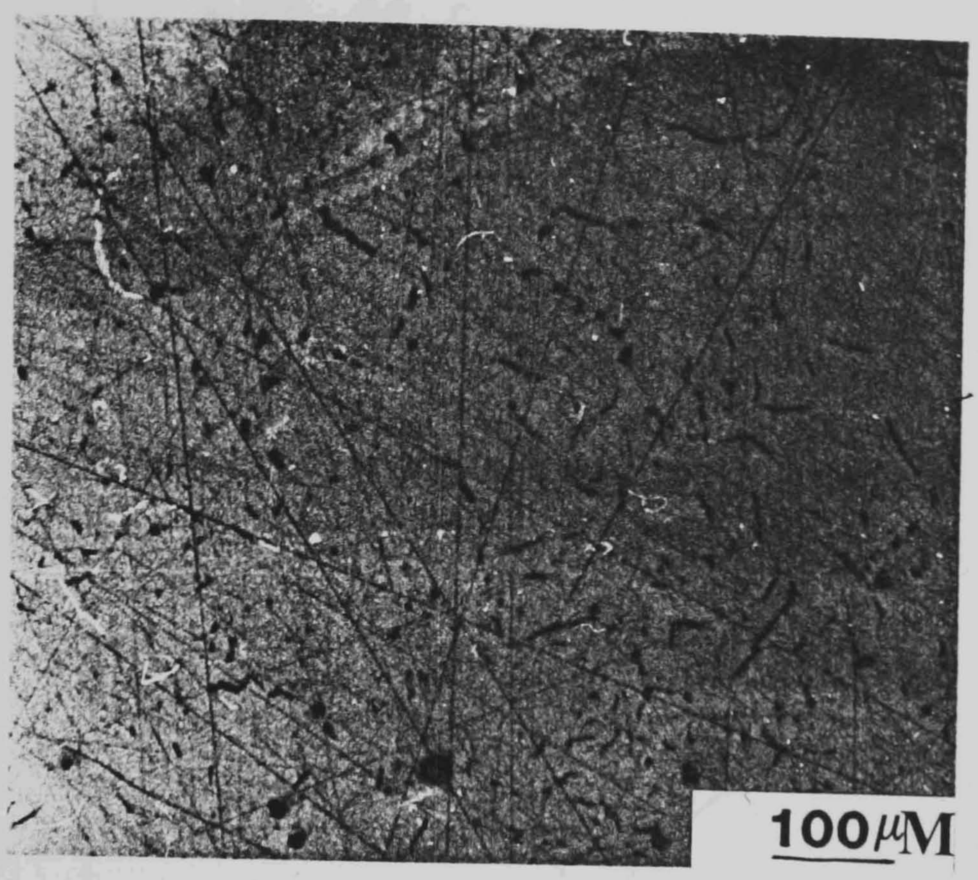


Fig.169 Microstructure of grey cast iron polished with 6 μ diamond paste.

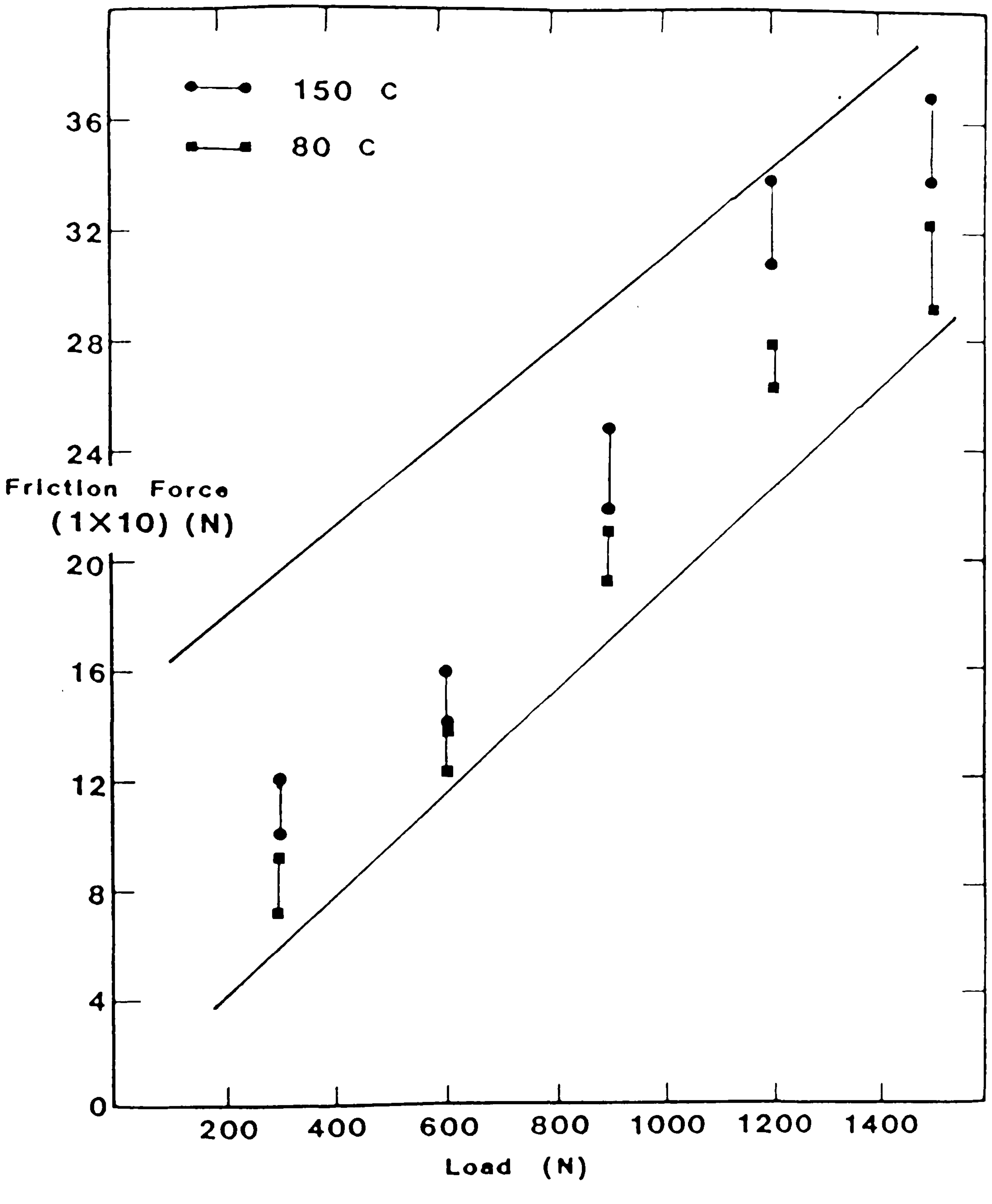


Fig.170 Friction force versus load for RL48 at two different temperatures

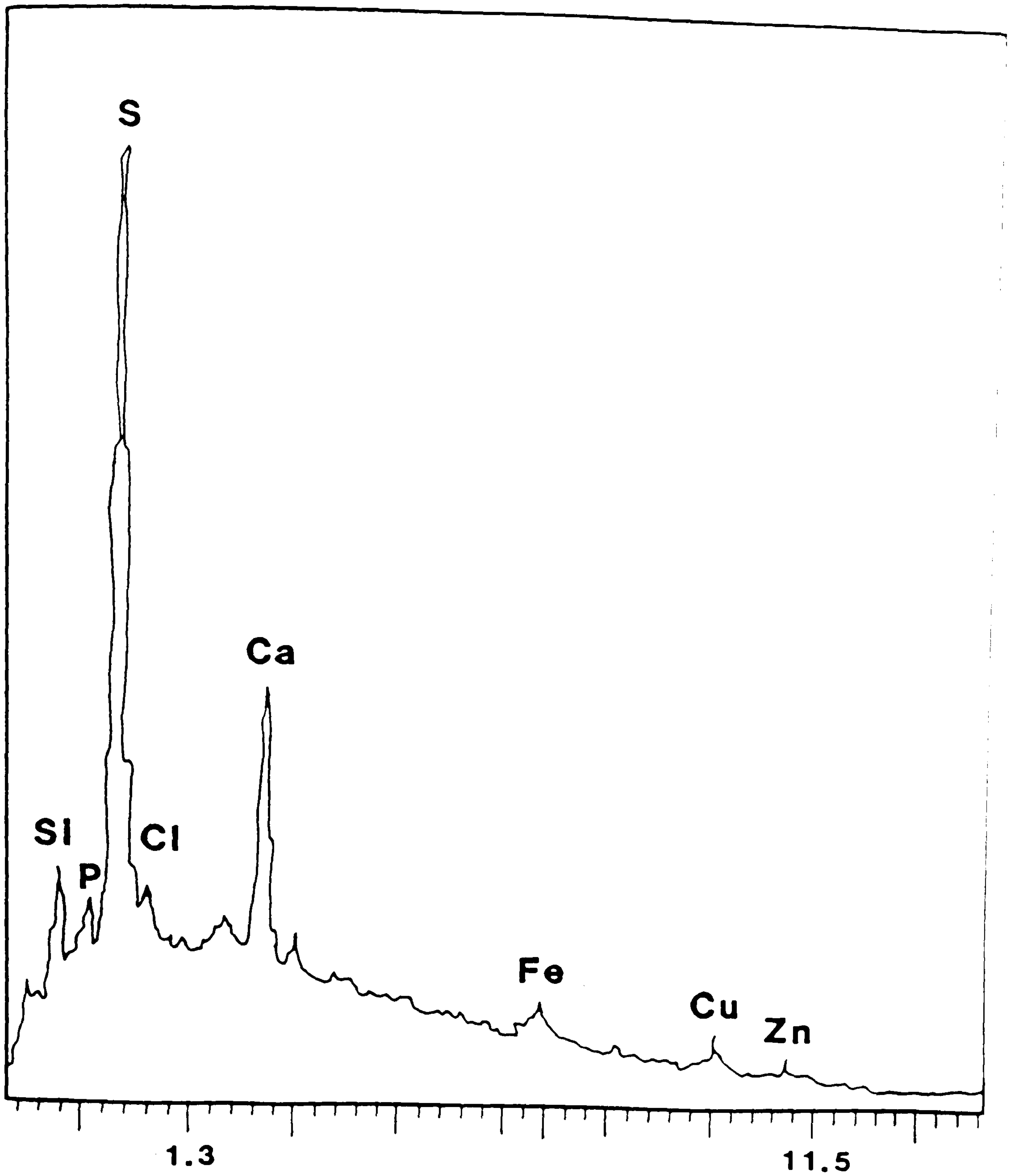


Fig.171 E.p.M.A. for the carbon removed fromthe piston top land.

Appendix 1 CEC Cylinder Bore Polishing Reference Oils

Reference	RL47	RL48
SAE grade	20W/40	20W/20
Viscosity @ 210°F., cSt	14.77	9.46
Viscosity @ 100°F., cSt	122.6	77.93
V.I.	133	103
TBN mgKOH/g	16.8	8.3