二硫化鉬及他種固態潤滑劑應用之八年研究

An Eight-Year Study on the Application of MoS_2 and the Second Solids

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摘　　要

水泥、鋼鐵、紙漿、製紙；肥料、橡膠輪胎、塑膠、化纖以及農業機械等工業，在臺灣成長迅速，但是由於這些工業的惡劣操作環境，像陡震、水份、重負荷、過負荷、高溫、化學及骯髒的工作場地等經常破壞潤滑油膜，引起生產設備的種種問題，傳統性的潤滑劑已不再能完全滿足臺灣業界的需要。

為了觀察把１％的二硫化鉬、和２％經過研磨的其他固態潤滑劑和傳統性潤滑油均勻混合使用的效果，一項為期八年的現場實驗在臺灣的下列工廠進行：

建臺水泥公司　南港輪胎公司　臺灣水泥公司　國泰塑膠公司　三榮橡膠公司
固特異輪胎公司　臺灣肥料公司　厚生玻璃公司　厚生橡膠公司　中興紡織公司

因為空氣壓縮機與齒輪箱是這些工業中負荷最重的設備，所以本實驗以空壓機及齒輪箱為主。

這項長期實驗的結果與在實驗室中所得相似，二硫化鉬及第二種固態潤滑劑與傳統性潤滑油混合，可降低2.24～9.8％的耗電量，降低５～24°C操作溫度，降低振動及噪音，延長零件壽命，增加生產力，並且延長潤滑周期６至８倍。由於現在國內外對能源節約、環境保護及高生產力的要求愈來愈高，選擇使用有效的機械用潤滑劑具有極高的重要性。

Abstract

The heavy industries such as cement, steel, pulp & paper, fertilizer, rubber tire, plastics, chemical fiber and agricultural machinery, are growing rapidly in Taiwan, because the adverse working conditions present in these industries, shock, vibration, water, heavy load, overload, high operating temperature, chemical fumes and dirty environment always destroy the oil film and cause the manufacturing equipments problems, the conventional mineral lubricants can no more satisfy the needs of the local industries.

To observe the effect of 1% solid lubricant plus 2% second solids (graphite, metallic sulfd, metallic oxide) in micron and submicron size dispersed into conventional mineral lubricants, an 8 year field test has been conducted at the following major Taiwan companies:

Chin-Tai Cement Corp.
Taiwan Cement Corp., Kaohsiung Plant
San Jung Rubber Ind. Corp. Ltd.
Taiwan Fertilizer Corp.
Formosan Rubber Corp.
Nankang Rubber Tire Corp.
Cathay Plastic Industry Corp., Chu-Nan Factory
Goodyear Taiwan Limited
Formosan Glass Corp., Yang-Mei Plant
Chung-Shing Textile Corp., Yang-Mei Chemical Fiber Plant

Because air compressor and gear box are the heaviest duty machineries in these plants, we concentrated our tests on these machines.

The field tests show the same results as in the laboratory, MoS_2 solid lubricant plus 2nd solids mixed with conventional mineral oils can reduce energy consumption by 2.24-19%, reduce operating temperature by 5–24°C, reduce vibration and noise, extend parts life, increase productivity and extend lubrication cycle by 6 to 8 time.

The requirement of energy conservation, environment protection and high productivity is increasing, to select and employ the effective industrial lubricants is of utmost importance.

INTRODUCTION

The most important function of any lubricant is to reduce friction between elements of a mechanism moving relative to each other. It has been established in the past by Reynolds[1] and others that moving parts can be separated by a fluid film under pressure generated by hydrodynamic action. O.D. Hersey[2] later applied dimensional analysis to experimental work with hydrodynamic lubrication on bearings. Lubrication can be subdivided into four basic types.

1. Hydrodynamic (commonly called fluid film or thick film lubrication).
2. Incomplete (commonly called quasihydrodynamic or semi-fluid lubrication).
3. Boundary (sometimes called thin film lubrication and usually involving a long-chain, adsorptive type, boundary lubricant).
4. Contact[3] (where solid lubricants from part of the metal surface itself as the results of mechanical occlusion or plastic deformation during running-in).

The object of solid film lubrication is to insert between two relatively moving surfaces a solid or solids which have low shear strength. The most widely used solid lubricants with low shear strengths are the layer-lattice or laminar solids, such as graphite and molybdenum disulfide[4,5]. Both have hexagonal crystal structures, and, when rubbed under pressure, the basal planes orient on the surface so that they become parallel to the sliding interface[6,7,8]. This facilltates shear and the crystals ability to flow into themselves.

The fact that a solid has a layer-lattice crystal structure does not ensure good lubricating properties. Both boron nitride and mica have layer-lattice structures, yet neither is considered a good solid lubricant.

Some evidence suggests they are not good lubricants because they do not adequately adhere to the surfaces to be lubricated[9]. Thus, besides having low shear strength, the ability of the solid lubricant to adhere to the surface is very important[4]. Campbell and Rosenberg[10] and Barry[11] concluded that the use of solid lubricants in greases and oils offers advantages in reduction in wear and friction, lower operating temperatures, higher load carrying capacity, less power output, reduction in galling and scuffing and reduction in fretting corrosion. In their test procedure they found that less than 1% of molybdenum disulfide was more effective than common chemical anti-wear compounds or as much as 5% of graphite. The purpose of this study was to emply MoS_2 and the second solids (graphite, metallic sulfide and metallic oxide) dispersed in mineral oil to make comparison tests to see if the boundary condition and poor lubrication in heavy industries can be improved.

THEORETICAL APPROACH

Hydrodynamic lubrication is defined as a system of lubrication in which the shape and relative motion of the sliding surfaces cause the formation of a fluid film having sufficient pressure to separate the surfaces.[12]

The principles of hydrodynamic lubrication will be developed by analyzing the properties of various film shapes starting with the simple wedge (see Fig. 1).

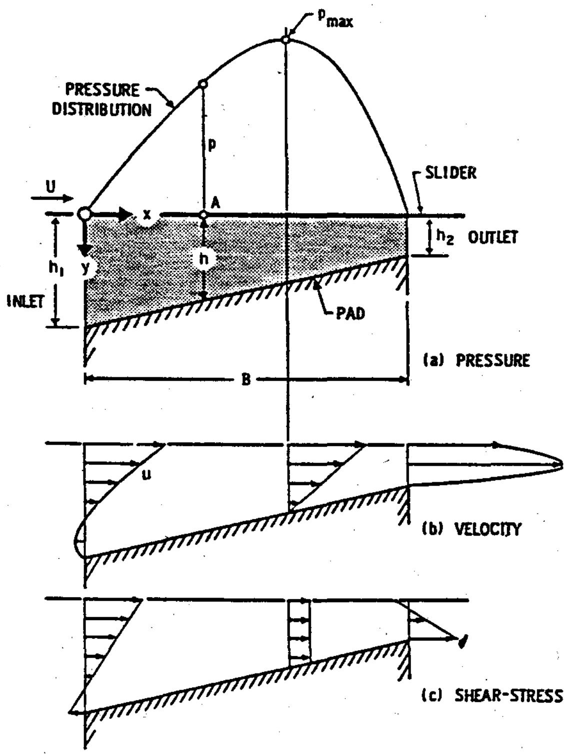


Fig. 1 Pressure, velocity, and shear stress distribution in wedge film.

Nomenclature

W = load on pad, lb
U = velocity of sliding surface, in./sec.
B = length of pad in direction of motion, in.
L = length of pad perpendicular to direction of motion, in.
P = load per unit area = W/BL, psi
μ = viscosity in reyns, lb-sec/sq. in.
h = film thickness, in.
h_1 = inlet film thickness, in.
h_2 = outlet minimum film thickness, in.
m = inclination of pad surface = $(h_1 - h_2)/B$, dimensionless

a = film ratio = h_1/h_2, dimensionless
F = friction force, lb
f = coefficient of friction = F/W, dimensionless
Q = flow of lubricant drawn into film at inlet, cu in./sec.
t_1 = inlet film temperature, F
t_2 = outlet film temperature, F
Δt = temperature rise in film = $t_1 - t_2$, F
K = performance number = μU/PB, dimensionless
K_f = characteristic number for fixed pad = $(1/m^2)\mu$U/PB, dimensionless
J = mechanical equivalent of heat = 9,336 in.-lb/Btu
γ = weight per unit volume of lubricant, lb/cu in.
c = specific heat of lubricant, Btu/lb/F
x, y = coordinates, see Fig. 1
$\bar{x}$ = distance from inlet edge of pad to resultant load, in.
x_1 = dimensionless coordinate = x/B
C_w, C_c, C_f, C_q = film factors (functions of a only), dimensionless

The manner in which the velocity μ varies across the thickness of the film at any point is given by

$$\mu = \frac{U(h - y)}{h} - \frac{1}{2\mu}\frac{dp}{dx}\, y(h - y) \qquad (1)$$

where μ is the viscosity, dp/dx is the pressure gradient, and the other symbols are as indicated in the figure. The velocity distribution at the inlet, the outlet, and the point of maximum pressure is indicated in Fig. 1b.

The manner in which the shear stress s in the lubricant varies across the thickness of the film at any point is given by

$$s = \mu \left[\frac{U}{h} + \frac{1}{2\mu}\frac{dp}{dx}(h - 2y) \right] \qquad (2)$$

The shear stress distribution at the inlet, the outlet, and the point of maximum pressure is shown in Fig. 1c.

From Eq. (1) for fluid velocity and the principle of continuity, which states that the flow into any given portion of the film must be equal to the flow out of that portion, Eq. (3), known as Reynolds' equation may be derived:

$$\frac{d}{dx}\left(\frac{h^3}{\mu}\frac{dp}{dx} \right) = 6U\frac{dh}{dx} \qquad (3)$$

Reynolds' equation may also be written in the form

$$\frac{dp}{dx} = 6\mu U \left(\frac{1}{h^2} - \frac{C}{h^3} \right) \qquad (4)$$

where C is a constant of integration.

The pressure distribution along the length of the film (in the x direction) is given by

$$p = \frac{\mu UB}{h_2{}^2} \left[\frac{6(a - 1)(1 - x_1)x_1}{(a + 1)(a - ax_1 + x_1)^2} \right] \qquad (5)$$

where B is the length of the pad in the direction of motion, a = h_1/h_2, and x_1 = x/B.

The value of the maximum pressure is given by

$$p_{max} = \frac{\mu UB}{h_2{}^2}\,\frac{1.5(a - 1)}{a(a + 1)}, \qquad (6)$$

the value of x_1 corresponding to the maximum pressure is given by

$$x_1 = \frac{a}{a + 1} \qquad (7)$$

The main characteristics of such a

bearing from a lubrication point of view include (1) minimum film thickness, (2) load capacity, (3) friction force or coefficient of friction, (4) lubricant flow, and (5) temperature rise. Assuming a pad of length L perpendicular to the direction of motion, these characteristics may be determined as follows (refer to Fig. 2):

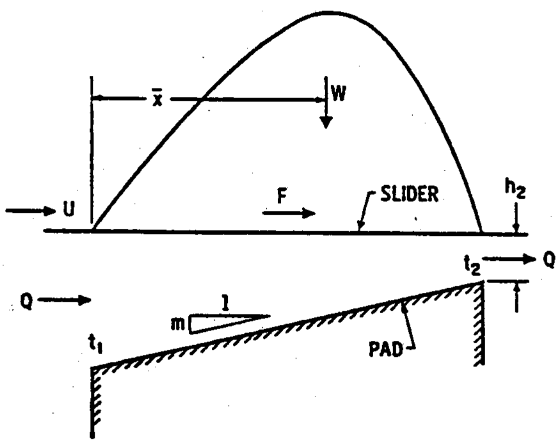


Fig. 2 Diagram for film operating characteristics.

The minimum film thickness h_2 is given by

$$h_2 = \frac{Bm}{a - 1} \qquad (8)$$

In presenting data for the solution of problems, it is helpful to have the minimum film thickness expressed dimensionlessly in terms of the minimum-film-thickness variable. Thus,

Minimum-film-thickness variable

$$= \frac{h_2}{Bm} = \frac{1}{a - 1} \qquad (9)$$

The load capacity of the wedge (or pad) is given by

$$W = \frac{\mu UL}{m^2} (a - 1)^2 C_w \qquad (10)$$

where

$$C_w = \frac{6}{(a - 1)^2} [\ell n\ a - \frac{2(a - 1)}{a + 1}]$$

the unit load P is defined as

$$P = \frac{W}{BL} \qquad (11)$$

Putting Eq. (10) in dimensionless form and substituting W = PBL give

$$\frac{m^2 PB}{\mu U} = (a - 1)^2 C_w \qquad (12)$$

For use as a chart coordinate the dimensionless form is inverted and referred to as the bearing characteristic number. It is given the symbol K_f thus:

$$K_f = \frac{1}{m^2} \frac{\mu U}{PB} = \frac{1}{(a - 1)^2 C_w} \qquad (13)$$

The distance $\bar{x}$ from the origin to the position of the resultant force W is given by

$$\bar{x} = BC_c \qquad (14)$$

where

$$C_c = \frac{1}{C_w} \frac{6}{(a - 1)^3 (a + 1)}$$

$$[a(a + 2)\ell n\ a - 2.5$$

$$(a - 1)^2 - 3(a - 1)]$$

This is expressed dimensionlessly in terms of the resultant position factor. Thus,

Resultant position factor

$$= \frac{\bar{x}}{B} = C_c \qquad (15)$$

The friction force F, which is the

force which must be applied to the slider to overcome the shear stress in the lubricant, is given by

$$F = \frac{\mu UL}{m}(a - 1)C_f \qquad (16)$$

where $C_f = \frac{1}{(a-1)}\left[4 \ln a - \frac{6(a-1)}{a+1}\right]$

The coefficient of friction f defined as F/W is given by

$$f = \frac{m}{a-1}\frac{C_f}{C_w} \qquad (17)$$

Putting Eq. (17) in dimensionless form and expressing it as the coefficient-of-friction variable give

Coefficient-of-friction variable

$$= \frac{f}{m} = \left(\frac{1}{a-1}\right)\frac{C_f}{C_w} \qquad (18)$$

The amount of lubricant Q drawn into the inlet h_1 and carried through to the outlet h_2 is given by

$$Q = \frac{BLUm}{a-1}C_q \qquad (19)$$

where $C_q = \frac{a}{a+1}$

Arranging this in dimensionless form and expressing it as the flow variable give

$$\text{Flow variable} = \frac{Q}{BLUm} = \frac{C_q}{a-1} \qquad (20)$$

If it is assumed that all the heat produced by friction goes into increasing the temperature of the lubricant as it flows from inlet to outlet, the temperature rise of the lubricant $\Delta t = t_2 - t_1$ is given by

$$\Delta t = \frac{P}{J\gamma c}\frac{C_f}{C_w C_q} \qquad (21)$$

where J is the mechanical equivalent of heat, γ is the weight of the lubricant per unit volume and c is the specific heat of the lubricant. Putting Eq. (21) in dimensionless form and expressing it as the temperature-rise variable give

$$\frac{J\gamma c\Delta t}{P} = \frac{C_f}{C_w C_q} \qquad (22)$$

Referring to Fig. 3, note that the quantity q flowing through a slot of length ℓ, width b, and thickness h is given by

$$q = \frac{\Delta p\, b h^3}{12\mu \ell} \qquad (23)$$

Δp is the pressure drop in length ℓ; hence $\Delta p/\ell$ is the pressure gradient; μ is the fluid absolute viscosity.

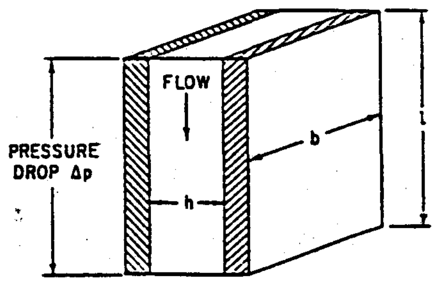


Fig. 3 Flow of visous fluid throught wide slot.

Consider first the problem of the load capacity of the film in a journal bearing. Figure 4 shows the journal displaced from its central position by a distance c. From the geometry of this figure, the following approximate formula for the film thickness is obtained:

$$h = mr(1 - \varepsilon \cos \theta) \qquad (24)$$

where $R - r = mr$

and $\varepsilon = \frac{e}{R - r} = \frac{e}{mr}$

The journal of Fig. 4 is being forced downward at some speed ν by the load W. If fluid inertia and end leakage are neglected, the pressure, load, and time relations are found as follows.

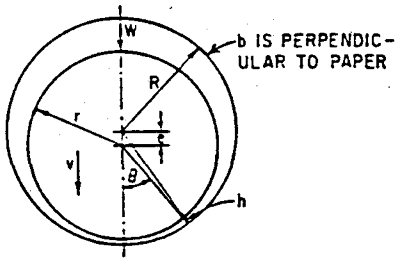


Fig. 4 Nomenclature for journal bearing.

From Eq. (23) the fluid passing a radial section at θ is given as

$$q = - \frac{bh^3}{12\mu} \frac{dp}{r\, d\theta}$$

(The negative sign arises because the pressure is decreasing as θ is increasing.)

If h from Eq. (24) is substituted,

$$q = - \frac{bm^3 r^3 (1 - \varepsilon \cos \theta)^3\, dp}{12\mu r\, d\theta} \tag{25}$$

Another relation for q can be written from consideration of displacement as follows:

$$q = \nu br \sin \theta \tag{26}$$

Combining Eqs. (25) and (26) and integrating lead to

$$p = \frac{12\mu\nu}{m^3 r} \left[\frac{1}{2\varepsilon(1 - \varepsilon \cos \theta)^2} + A \right] \tag{27}$$

The constant A is evaluated from boundary conditions. There are two cases of special interest, viz., the half and the full bearing. For a half bearing p = 0 when $\theta = \pi/2$, which makes $A = -1/2\epsilon$ and

$$p = \frac{6\mu\nu}{m^3 r\varepsilon} \left[\frac{1}{(1 - \varepsilon \cos \theta)^2} - 1 \right] \tag{28}$$

Referring to Fig. 5, the load W, which is in equilibrium with the film pressure forces, is obtained as follows:

$$W = 2br \int_0^{\pi/2} p \cos \theta \, d\theta$$

Hence

$$W = \frac{12\mu\nu b}{m^2} \left[\frac{\varepsilon}{1 - \varepsilon^2} + \frac{2}{(1 - \varepsilon^2)^{3/2}} \tan^{-1} \sqrt{\frac{1 + \varepsilon}{1 - \varepsilon}} \right] \tag{29}$$

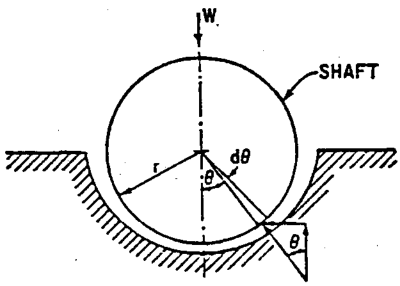


Fig. 5 Half bearing.

From Fig. 4 it is seen that $\nu = de/dt$, and putting $\epsilon = e/mr$, W can be written as

$$W = \frac{12\mu br}{m^2} \left[\frac{\varepsilon}{1 - \varepsilon^2} + \frac{2}{(1 - \varepsilon^2)^{3/2}} \tan^{-1} \sqrt{\frac{1 + \varepsilon}{1 - \varepsilon}} \right] \frac{d\varepsilon}{dt}$$

from which the time interval for the journal to move from ϵ_1 to ϵ_2 is obtained as

$$\Delta t = \frac{24\mu br}{m^2 W} \left(\frac{\varepsilon_2}{\sqrt{1-\varepsilon_2{}^2}} \tan^{-1}\sqrt{\frac{1+\varepsilon_2}{1-\varepsilon_2}} - \frac{\varepsilon_1}{1-\varepsilon_1{}^2} \tan^{-1}\sqrt{\frac{1+\varepsilon_1}{1-\varepsilon_1}} \right) \tag{30}$$

For the full journal bearing, the corresponding quantities are found as

$$p = \frac{6\mu\upsilon}{m^3 r\varepsilon} \left[\frac{1}{(1-\varepsilon\cos\theta)^2} - \frac{1}{(1+\varepsilon)^2} \right] \tag{31}$$

$$W = \frac{12\pi\mu b\upsilon}{m^3(1-\varepsilon^2)^{1/2}} \tag{32}$$

$$\Delta t = \frac{12\pi\mu br}{m^2 W} \left(\frac{\varepsilon_2}{\sqrt{1-\varepsilon_2{}^2}} - \frac{\varepsilon_1}{\sqrt{1-\varepsilon_1{}^2}} \right) \tag{33}$$

Friction coefficient for boundary-lubricated surfaces can be derived from the concept of boundary friction (Fig. 6).[13] Froce F required to shear the area in which the film has failed, plus the area in which the film exists – and must be sheared – is given by

$$F = \alpha A_r s_m + (1-\alpha) A_r s_f \tag{34}$$

where α = fraction of boundary film area which has failed and where metal-to-metal contact exists

A_r = area of real contact

s_m = shearing strength of metal-to-metal juncture

s_f = shearing strength of boundary film

Since the load $W = A_r p_m$, the friction coefficient can be written

$$f = \frac{F}{W} = \alpha \frac{s_m}{p_m} + (1-\alpha)\frac{s_f}{p_m} \tag{35}$$

or $$f = \alpha f_m + (1-\alpha) f_f \tag{36}$$

where f_m = coefficient of friction for sliding without lubricant

f_f = coefficient of friction for sliding with a complete boundary film

This equation shows that, for boundary lubrication, the total frictional force is made up of two terms: the force needed to shear metal-to-metal junctions (αf_m) and the force required to shear lubricant film $(1-\alpha)f_f$.

For an effective lubricant, α is a small value, and the lubricant shearing force may be larger than the metal shearing force.

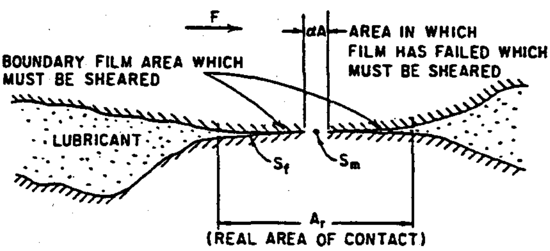


Fig. 6 Concept of boundary friction.

Effective boundary lubrication arises from the formation of a surface film of polar molecules which are absorbed out of a nonpolar or base fluid. Polar molecule content causes the so-called oiliness or lubricity property.

The polar additives condense on the surface to form a solid film. Because of their polarity, they all orient themselves with the one end at the metal surface (preferred orientation).[14,15,16] Many molecules pack in as closely as possible and strengthen the film with lateral cohesive forces.

This solid film, adhering to the surface and with the molecules cohering to each other, then has the ability to resist penetration of asperities and thus inhibit metal-to-metal contact. In addition, a zone of low shear strength is formed between the outermost surfaces of two monolayers adsorbed on opposing metal surfaces. This concept is represented schematically in Fig. 7. In the figure, effective lubrication is occurring at A and B and lubricant failure is shown at C.

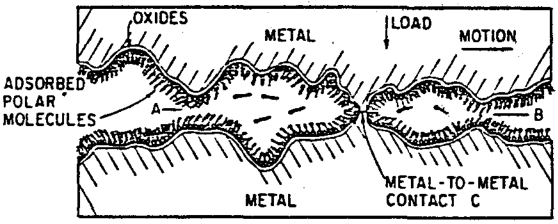


Fig. 7 Schematic of boundary – lubricated surfaces showing effectively (A and B) and ineffectively (C) lubricated contacts.

A solid lubricant is a thin film of a solid interposed between two rubbing surfaces to reduce friction and wear. The need for solid lubricants has grown rapidly with advances in technology requiring operation under such severe conditions of pressure, temperature, and environmental reactivity that no existing organic fluid possesses sufficient stability.[17,18]

This equation gives a good representation of the frictional behavior of an effective solid-film lubricant when it is thin enough that p_m determines the real area of contact. When the film is continuous, $\alpha = 0$ and $f = s_f/p_m$. Thus, for minimum f, s_f must be low and p_m high. Note. however, that as the film thickens, the real area of contact will be determined in part by p_f, the yield pressure of the solid film material, and f will increase to s_f/p_f, p_f being much lower than p_m. If the film is penetrated, f will increase as α increases reaching s_m/p_m when $\alpha = 1$, when the film is completely removed.

For moderately loaded, unlubricated metals of similar hardness it has been shown that

$$V = K \frac{W}{P_m} \tag{37}$$

Where V is the volume worn off in unit sliding distance, W is the load, and K is a constant. For a tightly adherent solid-lubricant film of optimum thickness, which wears by lubricant-lubricant encounters, so that p_m is replaced by p_f, Eq. (37) can be used as a general guide to solid-lubricant life.[19]

It is possible to deduce many of the desirable friction and wear reducing properties of a solid-lubricant film from the relations and discussed above. They are:

1. Low shear strength
2. Low hardness.
3. High adhesion to the substrate material.
4. Continuity. To be effective the film must prevent all metal-to-metal contact.
5. Self-healing ability. The film should reform immediately if broken.
6. Freedom from abrasive impurities.

MECHANISM OF SOLID LUBRICANTS

Many inorganic solids have been tested and used for lubrication. A partial listing of the more common ones would be the metallic sulfides, such as molybdenum disulfide; metallic halides, such as cadmium iodide, the metallic oxides, such as lead oxide and other layer-lattice crystals such as mica and vermiculite.[20]

Other examples are tungsten disulfide, boron nitride, borax and silver sulfate.

Solid lubricants composed of two or more materials often combine their most favorable characteristics to provide synergistic lubricating properties that are superior to any single lubricating solid. A brief review of the characteristics of the most commonly used solids can give insight into the lubrication mechanism of solids alone or in combination. The most widely used inorganic and metal base materials are graphite and MoS_2.

A theoretical explanation for the lubricity of molybdenum disulfide is similarly found in its molecular structure. Each lamina of this compound is composed of a layer of molybdenum atoms with a layer of sulfur atoms on each side. The sulfur and molybdenum layers bond tightly but the adjacent laminae interface at their layers of sulfur atoms which form weak bonds between the laminae. The weak sulfur bonds between the laminae form the slippage planes of low shear resistancce as between the carbon layers in graphite. Both graphite and MoS_2 display an affinity for metal substrates. And under high loads both have been found to alloy with ferrous metal forming even greater bonds at the surface.

Under heavy forces (loads) perpendicular to bearing surfaces, these laminae are compressed and oriented parallel to the bearing surfaces and have the strength to resist rupture. The low friction reflects the low resistance of the laminae sliding on one another. Cohesive forces within graphite and MoS_2 are sufficint to allow for self-healing of interruptions in the solid lubricant film.[21]

Whilst MoS_2 and graphite resemble each other closely in many of their characteristics, they differ in two important respects which have a fundamental effect on their lubricating action. Graphite consists only of elemental carbon, whilst MoS_2 is a compound of two elements; graphite is relatively inert towards oxidizing atmospheres whilst MoS_2 is readily oxidized under certain conditions, and it is necessary to emphasize the essential difference between the oxides formed on the surface when MoS_2 and graphite are milled. In the case of MoS_2, a definite layer of oxide is formed, and this layer can cement the small pieces together, and fill in the gaps between these pieces. In the case of graphite, the oxygen atoms are attached to carbon atoms which are still a part of the graphite lattice – they can only be detached by severe heating of the graphite, and they are broken off as volatile carbon oxides. Far from cementing the particles together, they largely prevent them from recombining. There is also a difference in the thicknesses of these layers; in the case of molybdenum disulfide the oxide layer is non-volatile, and can consequently be built up to an appreciable thickness (shown by the fact that it can fill the interstices of the particles), but the oxide layer on graphite can only be one atomic layer thick. Thus no blocking of the pores is possible and milled graphite can achieve the considerable area of $600\ m^2/g$. Such a high-area powder contains 15 percent by weight of oxygen, although the oxygen layer on the surface is only one atom thick and it may take up almost

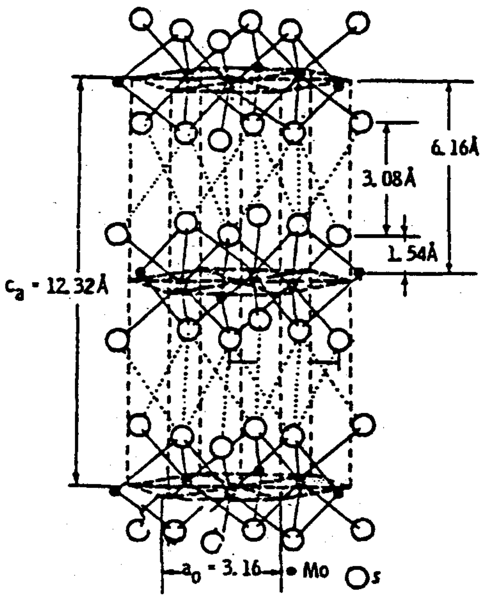


Fig. 8 Structure of MoS_2.

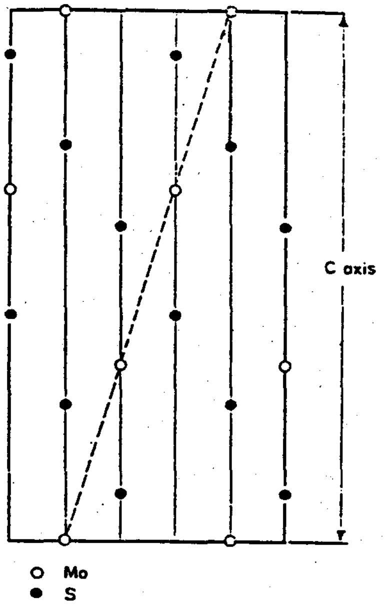


Fig. 10 Structure of single crystal of rhombohedral MoS_2.

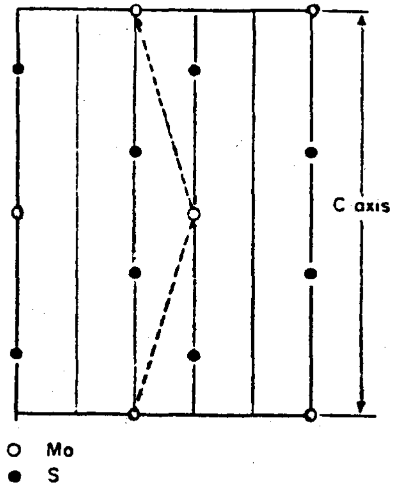


Fig. 9 Structure of single crystal of hexagonal MoS_2.

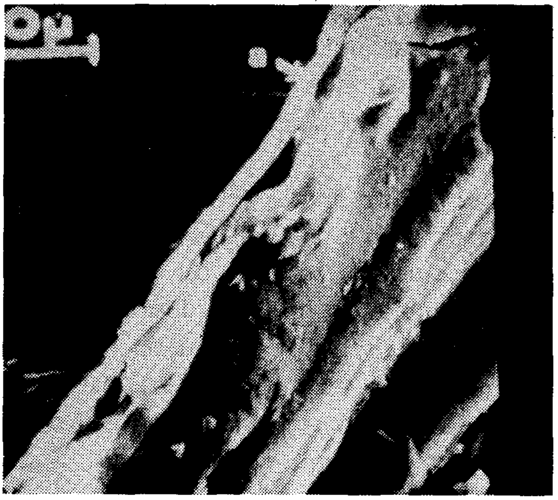


Fig. 11 Molysulfide film on a steel substrate.

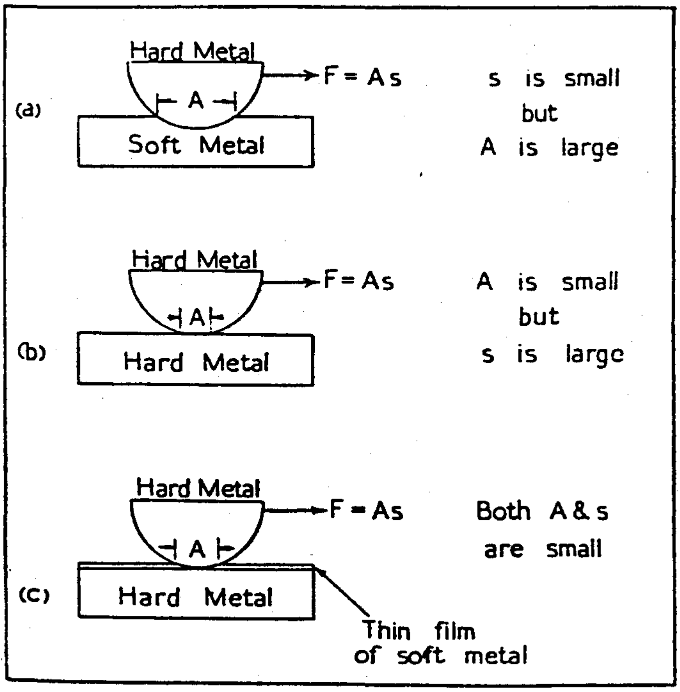


Fig. 12 The friction between metal surfaces is not greatly dependent on their hardness. A low friction may be obtained by depositing a thin film of a soft metal on a hard metal substrate.

its own weight of water in moist air.[3]

The oxygen content of MoS_2 rises rapidly with the rapid increase in surface area, and also continues to rise when the surface area is actually diminishing. This occurs because in this stage of the milling, edge atoms, with their ability to absorb oxygen, are still being exposed by breaking of the crystallites across the basal planes, but the area of the exposed basal planes is being reduced, due to the fact that the crystals have become thin enough to adhere to one anothe strongly. The lubricating action of molybdenum disulfide can be regarded as an extension of the mechanism put forward for graphite, the extent to which it is operative being dependent solely on the surface state of the molybdenum disulfid.

There are four main facts concerning the mechanism of lubrication by MoS_2 and these are now briefly stated.

1. Peterson and Johnson[22] showed that the lubricating properties of MoS_2 are dependent on temperature and humidity.
2. Johnson, Godfrey and Bisson[23] showed that the coefficint of friction decreases with increasing sliding velocity.
3. Barwell and milne[24] suggested that the coefficient of friction decreases with increasing load.
4. Johnson and Vaughan[25] attributed the low frictional coefficient of MoS_2 when outgassed at high sliding speeds to the presence of an amorphous layer of sulphur.

COMPARISON TESTS IN CONTROLLED LABORATORY

I. TEST A – on friction

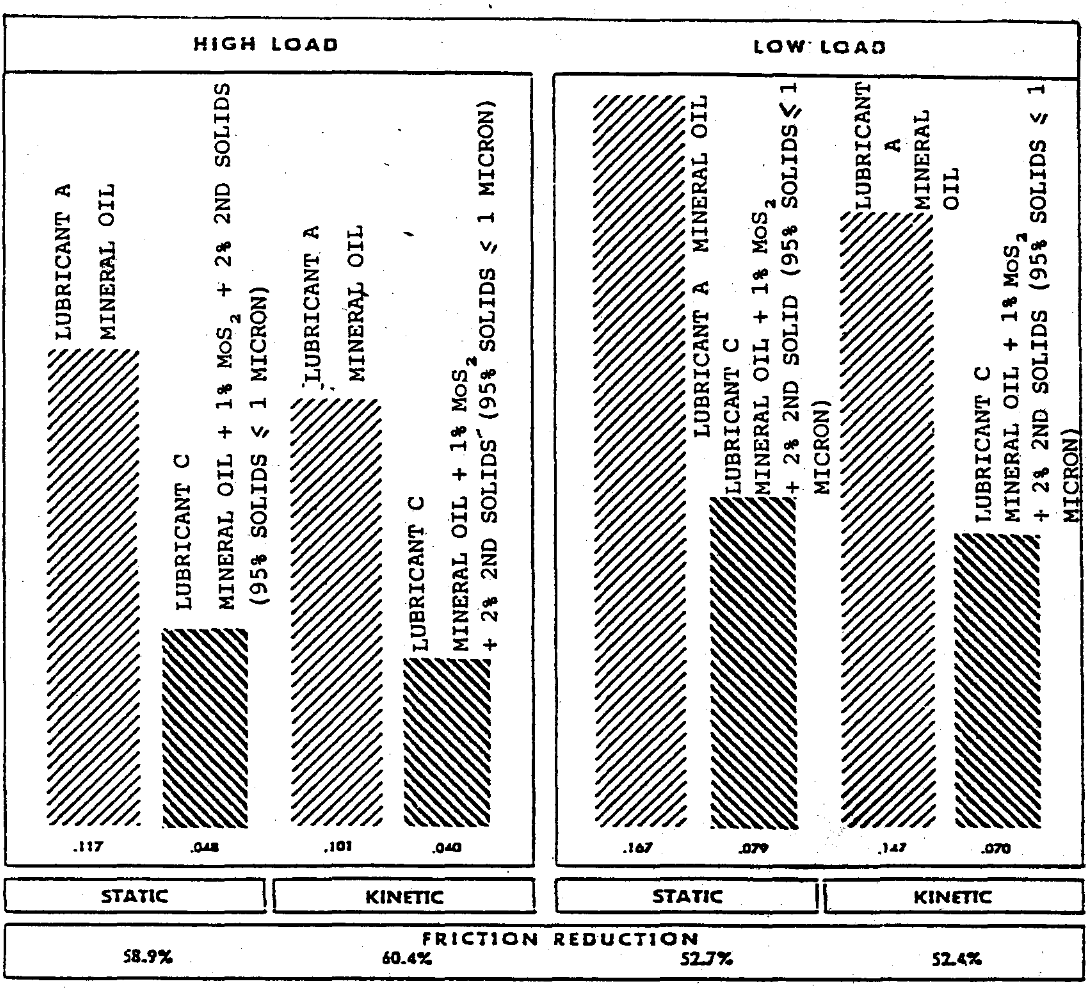


Fig. 13. lubricant c vs. lubricant a – coefficient of friction this test dramatically demonstrates the reduction in coefficient of friction with lubricant C compared to lubricant A.

The test procedure consisted of drawing a tool steel block between two Meehanite G.A. blocks. The force required to start the tool steel block moving (static friction) and th force required to maintain movement (kinetic friction) was measured accurately.

Type of Material:

Sample I – Tool Steel: Crucible Airkool – Moving Member

Typical Analysis from Crucible Handbook.

Carbon	1.00%
Manganese	0.70%
Sulphur	0.15%
Silicon	0.30%
Chromium	5.25%
Vanadium	0.30%
Molybdenum	1.15%
Actual Hardness: Rockwell "C"	61.0

Sample II – Meehanite G. A. – Stationery Members

Typical Analysis from Lincoln Foundry, Los Angeles

Total Carbon	2.90–3.10
Manganese	0.80–1.00
Silicon	1.30–1.50
Sulphur	0.10 max.
Phosphorous	0.080 max.

Actual Hardness: Brinell 3000 kg load 197

Type of Lubricants:

Lubricant A: mineral lubricating oil
Lubricant B: mineral lubricating oil + 1% MoS_2
Lubricant C: mineral lubricating oil + 1% MoS_2 + 2% 2nd solids

The test blocks were ground to obtain a similar surface finish to be comparable with the job application conditions.

Surface Finish Obtained:

Sample I (Airkool Tool Steel)	9 Micro inches
Sample II (Meehanite G. A.)	11 Micro inches

Before the application of any lubricants the test blocks were thoroughly cleaned by rinsing in Carbon Tetrachloride, blowing dry and then a rinse in Acetone. The lubrication was then applied for testing.

TEST NO. 1 – 2050 P.S.I. NORMAL PRESSURE, LUBRICANT A

SAMPLE	NORMAL PRESSURE POUNDS	LATERAL LOAD IN POUNDS STATIC	LATERAL LOAD IN POUNDS SLIDING	COEFFICIENT OF FRICTION STATIC	COEFFICIENT OF FRICTION SLIDING
I	16,400	5,500	5,000	0.167	0.152
II	16,400	5,500	4,650	0.167	0.142
			Average:	0.167	0.147

TEST NO. 2 – 2050 P.S.I. NORMAL PRESSURE, LUBRICANT B

SAMPLE	NORMAL PRESSURE POUNDS	LATERAL LOAD IN POUNDS STATIC	LATERAL LOAD IN POUNDS SLIDING	COEFFICIENT OF FRICTION STATIC	COEFFICIENT OF FRICTION SLIDING
I	16,400	5,300	4 200	0.162	0.128
II	16,400	5,300	4,500	0.162	0.137
			Average:	0.162	0.133

TEST NO. 3 – 2050 P.S.I. NORMAL PRESSURE, LUBRICANT C

SAMPLE	NORMAL PRESSURE POUNDS	LATERAL LOAD IN POUNDS STATIC	LATERAL LOAD IN POUNDS SLIDING	COEFFICIENT OF FRICTION STATIC	COEFFICIENT OF FRICTION SLIDING
I	16,400	2,400	2,100	0.073	0.064
II	16,400	2,800	2,510	0.085	0.076
			Average:	0.079	0.070

TEST NO. 1A – 4150 P.S.I. NORMAL PRESSURE, LUBRICANT A

SAMPLE	NORMAL PRESSURE POUNDS	LATERAL LOAD IN POUNDS STATIC	LATERAL LOAD IN POUNDS SLIDING	COEFFICIENT OF FRICTION STATIC	COEFFICIENT OF FRICTION SLIDING
I	33,200	7,150	6,300	0.108	0.095
II	33,200	8,350	7,150	0.126	0.108
			Average:	0.117	0.101

TEST NO. 2A – 4150 P.S.I. NORMAL PRESSURE, LUBRICANT B

SAMPLE	NORMAL PRESSURE POUNDS	LATERAL LOAD IN POUNDS STATIC	LATERAL LOAD IN POUNDS SLIDING	COEFFICIENT OF FRICTION STATIC	COEFFICIENT OF FRICTION SLIDING
I	33,200	6,950	6,350	0.105	0.096
II	33,200	5,900	5,300	0.089	0.079
			Average:	0.097	0.088

TEST NO. 3A – 4150 P.S.I. NORMAL PRESSURE, LUBRICANT C

SAMPLE	NORMAL PRESSURE POUNDS	LATERAL LOAD IN POUNDS STATIC	LATERAL LOAD IN POUNDS SLIDING	COEFFICIENT OF FRICTION STATIC	COEFFICIENT OF FRICTION SLIDING
I	33,200	3,250	2,650	0.049	0.040
II	33,200	3,100	2,250	0.047	0.041
			Average:	0.048	0.040

II. TEST B – on gear oils

LUBRICANTS TESTED

Lubricant A – Mineral Gear Oil

Lubricant C – Mineral Gear Oil + 1% MoS_2 + 2% 2nd Solids (95% solids $\leqslant$ 1 micron)

ITEM TESTED

FZG Test

Timken Extreme Pressure Test

Falex Extreme Pressure Test

Four Ball Extreme Pressure Test
Four Ball Wear Test

Table 1. Comparison Test Result On Gear Oils

	Lubricant A	Lubricant C
ISO Viscosity Grade, ASTM D 2422	320	320
SAE Viscosity Classification	90	90
AGMA Lubricant Number	6EP	6EP
Specific Gravity, ASTM D 1298, @60°F/15.6°C	0.8976	0.8984
API Gravity, ASTM D 1298, @60°F/15.6°C	26.1	23
Viscosity, ASTM D 445, D 2161:		
@ 100°F, SUS	1500	1582
@ 210°F, SUS	110	124
Viscosity Index, ASTM D 2270	94	105
Flash Point, ASTM D 92, COC, °F/°C	450	460
Fire Point. ASTM D 92, COC, °F/°C	480	510
Pour Point, ASTM D 97, °F/°C	-5	10
Rust Test, ASTM D 665		
Procedure A (Distilled Water)	Pass	Pass
Procedure B (Synthetic Sea Water)	Pass	Pagg
Oxidation Test (ASTM D 2893), 312 Hours @ 203°F/95°C, % Viscosity Increase @ 210°F, max	6	6
Foam Test, ASTM D 892, Seq. 1, 2, 3	Pass	Pass
FZG Test, (L/16.6/90) Load Stages Passed	11	12+
Timken Extreme Pressure Test, ASTM D 2782, OK Value, lbs/kg	60	70
Falex Extreme Pressure Test, psi (3000 psi gage)	1150	1350
Four Ball Extreme Pressure Test (ASTM D 2783)		
Load Wear Index, kg.	68.4	71.5
Weld Load, kg.	350	400
Four Ball Wear Test (40 kg., 167°F/75°C, 1800 rpm, 1 hr.)		
Scar Diameter, mm	0.52	0.50
Falex Wear Test, ASTM D 2670, wear teeth	2	3
Lubricant solids, Grade Classification	None	1% MoS_2 + 2% 2nd Solids (95% solids $\leqslant$ 1 micron)

III. TEST C – on compressor oil

LUBRICANT TESTED

Lubricant A – Mineral Oil

Lubricant C – Mineral Oil + 1% MoS_2 + 2% 2nd Solids (99% 2nd Solids $\leqslant$ 1 micron)

Table 2. Comparison Test Result on Compressor Oils

	Lubricant A	Lubrican
ISO Viscosity Grade, ASTM D 2422	220	220
Specific Gravity, ASTM D 1298, @ 15.6°C/60°F	.919	.9094
API Gravity, ASTM D 1298, @ 15.6°C/60°F	22.3	24.1
Viscosity, ASTM D 445, D 2161:		
@ 100°F, SUS	1400	1125
@ 210°F, SUS	84	76.3
Flash Point, ASTM D 92, COC, °C/°F		238/460
Fire Point, ASTM D 92, COC, °C/°F	224/435	263/505
Pour Point, ASTM D 97, °C/°F	-9.4/15	−21/-5
Rust Test, ASTM D 665		
Procedure A (Distilled Water)	pass	pass
Procedure B (Synthetic Sea Water)	pass	pass
Conradson Carbon Residue, ASTM D 189 Base Oil, wt%	0.45	.05
Four Ball Wear Test (40 kg, 75°C/167°F, 1800 rpm, 1 hr)		
Scar Diameter, mm	0.52	0.5
Falex Wear Test, ASTM D 2670, Wear Teeth	9	10
Lubricant Solids, Grade Classification	None	1% MoS_2 + 2% 2nd Solids (99% solids ≤1 micron)

COMPARISON IN FIELD TESTS

FIELD TEST NO. 1

COMPANY: Taiwan Cement Corp., Kaohsiung Plant

MACHINERY: Mitsubishi, 2000 KW Helical and Herringbone Gear Reducer of No. 9 Raw Mill

ITEMS TESTED: (a) Oil Temperature Difference (temperature difference between oil and room temperature)
(b) Sound Pressure Level
(c) Vibration
(d) Teeth Surface of Gear Reducer

LUBRICANT TESTED: Lubricant A – Mineral Oil
Lubricant C – Mineral Oil + 1% MoS_2 + 2% 2nd Solids

Tables 3. Test Data with Lubricant in Taiwan Cement

DATE	SURFACE TEMP. OF GEAR REDUCER	ROOM TEMP.	DIFF. BETWEEN GEAR REDUCER SURFACE & ROOM TEMP.	VIBRATION MILLS B(X Axis)	VIBRATION MILLS B'(Y Axis)	NOISE (dB) C(X Axis)	NOISE (dB) C'(Y Axis)
4/10/80	40	26	14	3	3.5	105.5	103
4/17/80	40	26	14	3	3.5	106.5	103
4/28/80	40	27	13	3.1	3.2	104.5	103
5/3/80	42	30	12	3.0	2.8	102	101.5
5/16/80	42	32	10	3.0	2.6	106.5	103
5/29/80	42	32	10	2.3	2.8	106.5	103
6/10/80	44	34	10	3.0	2.8	106.5	103
6/28/80	44	35	9	3.0	2.8	107	102.5
7/4/80	42	30	12	3.0	2.8	107	103
7/11/80	42	31	11	3.0	3.0	107	103
8/9/80	42	31	11	3.0	3.0	107	103
8/27/80	42	31	11	3.0	3.1	107.5	103.5
9/16/80	43	31	12	3.1	3.1	107.5	103.5
10/14/80	41	29	12	3.0	3.1	107.5	103.5
11/4/80	40	27	13	3.0	3.1	103	101.5

Table 4. Test Data with Lubricant C

DATE	SURFACE TEMP. OF GEAR REDUCER	ROOM TEMP.	DIFF. BETWEEN GEAR REDUCER SURFACE & ROOM TEMP.	VIBRATION MILLS B(X Axis)	VIBRATION MILLS B′(Y Axis)	NOISE (dB) C(X Axis)	NOISE (dB) C′(Y Axis)
3/12/81	41	34	7	2.5	2.4	102.5	103
3/17/81	40.5	34	5.5	2.4	2.4	102.5	103
4/15/81	41	34	7	2.4	2.3	103.5	102
5/6/81	41	34	7	2.3	2.3	102.5	102
5/20/81	40.5	32	7.5	2.3	2.2	102.5	101.5
6/5/81	41	32	9	2.2	2.2	102.5	102
7/9/81	40.5	36	3.5	2.2	2.1	99	98
7/17/81	40.5	34	5.5	2.1	1.9	102.5	101
7/28/81	40	34	6	2.0	2.0	103.5	101
8/17/81	42	36	6	2.1	1.9	102	101
9/17/81	40	35	5	2.0	1.9	102.5	100.5
10/22/81	39	32	7	2.0	1.9	101.5	99.5
11/25/81	34	26	8	2.0	1.9	101	101.5
11/27/81	35.5	28	7.5	2.1	2.0	101.5	101.5
12/16/81	31	24	7	2.0	1.9	101	101.5
4/12/82	41	33	8	2.4	2.4	103	103
4/17/82	41.5	34	7.5	2.3	2.3	102.5	103
5/18/82	42	35	7	2.4	2.4	102.5	103
6/15/82	42	35	7	2.3	2.3	102.5	102
8/6/82	42	35	7	2.2	2.1	102.5	102
10/20/82	41.5	33	8.5	2.2	2.2	102.5	101.5
12/5/82	41	33	8	2.1	2.2	102.5	102
1/16/83	37	30	7	2.1	2.1	100	99
2/19/83	36	30	6	2.0	1.9	102	101
4/27/83	40	34	6	2.0	2.0	103	101
6/18/83	42	35	7	2.1	2.1	102	101.5
9/12/83	41.5	34	7.5	2.0	1.9	102.5	100.5
10/26/83	41.5	33	8.5	2.0	1.9	101.5	99.5
11/20/83	36	30	6	2.1	1.9	101.5	100.5
11/29/83	35.5	29	6.5	2.0	1.9	101.5	100.5
12/20/83	35	28	7	2.0	1.9	101	100.5
2/11/84	34	28	6	2.4	2.4	102.5	103
4/20/84	37	30	7	2.5	2.4	102	103
5/25/84	41.5	35	6.5	2.4	2.4	102	102.5
7/17/84	42	36	6	2.3	2.3	102	102.5
9/12/84	42	36	6	2.3	2.3	101.5	102
11/26/84	40.5	32	7.5	2.3	2.2	102	101.5
12/5/84	37	29	8	2.2	2.2	101.5	101.5
1/10/85	36	30	6	2.2	2.2	99	98
3/21/85	40.5	34	6.5	2.1	2.0	101	100
5/30/85	40	34	6	2.0	2.0	102	101
7/16/85	42	36	6	2.1	2.1	102	101
9/25/85	40	35	5	2.0	2.0	102	100
12/20/85	39	32	7	2.0	2.0	102	100
1/22/86	36	29	7	2.0	2.0	101	100.5
4/26/86	39	31	8	2.1	2.0	101.5	100.5
6/11/86	40	33	7	2.1	2.1	101	100.5
7/14/86	41	34	7	2.4	2.3	102	102.5
8/12/86	40.5	34	6.5	2.3	2.3	102	102.5
9/13/86	40.5	33	7.5	2.3	2.2	103	102.5
10/18/86	41	34	7	2.4	2.3	103	101.5
11/5/86	40.5	32	7.5	2.3	2.3	102	101.5
12/20/86	39	31	8	2.3	2.2	102.5	101.5

Table 5. Test Data of Iron Wear Particles in Taiwan Cement

DATE	LUBRICANT	WEIGHT OF IRON WEAR PARTICLES
4/28/80	A	27 g
5/29/80	A	16 g
6/28/80	A	15 g
4/20/81	C	42 g
6/20/81	C	7 g
8/19/81	C	2 g
12/27/81	C	Negligible
2/8/82	C	Negligible
11/30/82	C	Negligible
3/25/83	C	Negligible
6/20/83	C	Negligible
10/23/83	C	Negligible
4/21/84	C	Negligible
7/18/84	C	Negligible
10/11/84	C	Negligible
1/20/85	C	Negligible
6/8/85	C	Negligible
11/28/85	C	Negligible
1/20/86	C	5 g
2/22/86	C	10 g
3/20/86	C	3 g
4/21/86	C	Negligible
5/20/86	C	Negligible
9/18/86	C	Negligible
12/20/86	C	Negligible

Fig. 14 Oil Temperature Difference (°C) of No. 9 Raw Mill Gear Reducer in Taiwan Cement 10.

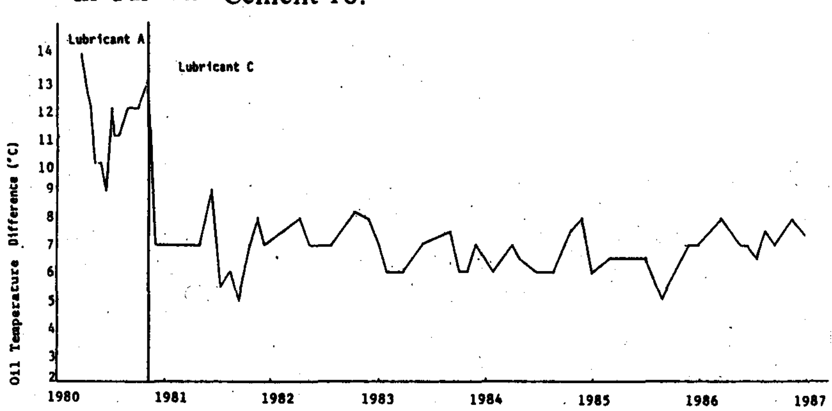


Fig. 15. Sound Pressure Level of No. 9 Raw Mill Gear Reducer In Taiwan Cement.

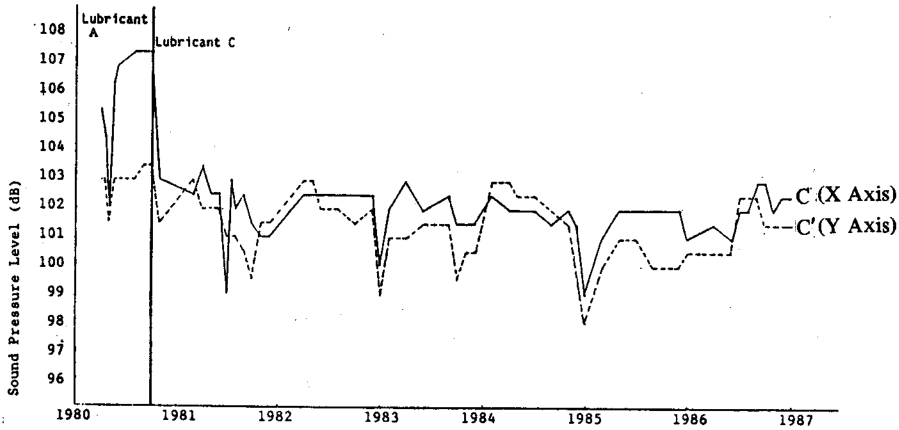
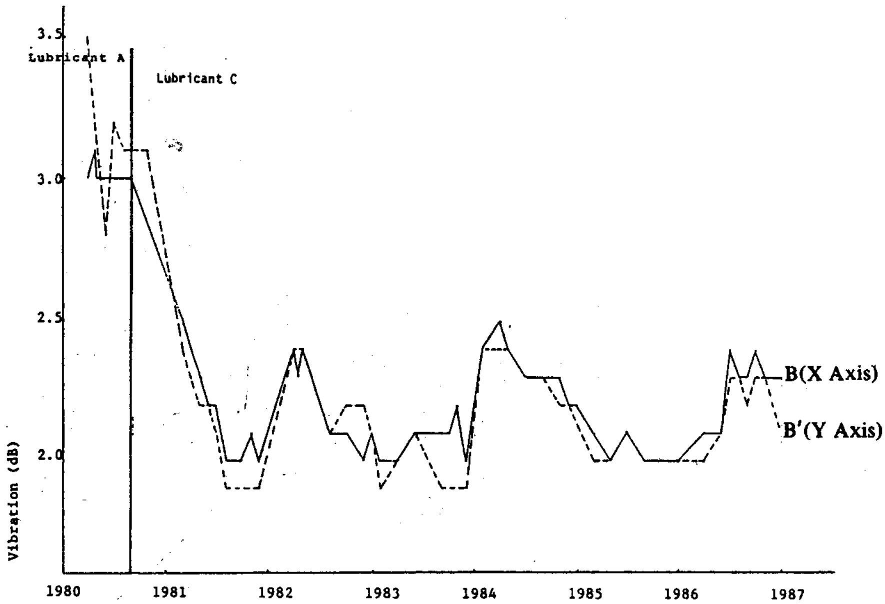


Fig. 16. Vibration of No. 9 Raw Mill Gear Reducer in Taiwan Cement



TEST RESULT

1. Oil Temperature (temperature difference between oil and room temperature):
   Using lubricant C, oil temperature is decreased by 5.5 C compared with using lubricant A.
2. Vibration:
   Using lubricant C, vibration value is reduced by 0.6 mils (Y direction), 0.6 mils (Z direction) compared with using lubricant A.
3. Sound Pressur Level:
   Using lubricant C, sound pressure level is reduced by 3% compared with using lubricant A.
4. Teeth Surface of The Gear Reducer:
   Lubricant A, serious pitting and scoring are found on the gear surface, the weight of iron wear that cleaned from filter per month is 19.3 grams. After changing to lubricant C, pitting and scoring get improved, the weight of wear iron that cleaned from filter per month is negligible.
5. Oil Service Life
   Using lubricant C the oil service life is 5 years and 2 months.
   Using lubricant A the oil service life is 1 year and 2 months.
6. The power consumption vary with process product, in this field test, power consumption only for reference.

FIELD TEST NO. 2

COMPANY: Chung-Shing Textile Corp., Yang-Mei Chemical Fiber Plant

MACHINERY: Norwalk TDR-S5T5 stages 200 HP 350 SCFM 15¾" x 14" 3000 psi air compressor

ITEMS TESTED: (a) Oil Consumption
(b) Carbon Deposit
(c) Vibration
(d) Oil Temperature Difference

LUBRICANTS TESTED: Lubricant A – Mineral Oil
Lubricant C – Mineral Oil + 1% MoS_2 + 2% 2nd Solids

Table 6. Test Date with Lubricant A in Chung-Shing Textile Corp.

	OIL CONSUMPTION							VIBRATION (dB)		TEMPERATURE (°C)															POWER
	Lub. Pt. Drops Per Stroke									1st			2nd			3rd			4th			5th			
Date	5th	4th	3rd	2nd	1st	pack	Average Drops per Storke	Crank Case (dB)	The 4th Stg. Cylinder Head (dB)	Suction	Disch	Diff.	Suction	Disch	Diff.	Suction	Disch	Diff.	Suction	Disch	Diff.	Suction	Disch	Diff.	No Load Consumption (AMP.)
8/5/79	8	8	8	8	8	8	8	87	94	33	92	59	47	123	76	47	117	70	43	130	87	43	115	72	123
9/10/79	8	8	8	8	8	8	8	88	93	30	88	58	46	120	74	46	114	68	42	128	86	41	113	72	123
10/7/79	8	8	8	8	8	8	8	87	93.5	30	87	57	46	119	73	46	115	69	42	126	84	42	112	70	123
11/8/79	8	8	8	8	8	8	8	87.5	93	26	84	58	43	118	75	43	110	67	39	123	84	39	112	73	123
12/3/79	8	8	8	8	8	8	8	87	93.5	25	84	59	40	116	76	41	107	66	36	121	85	37	110	73	123
1/11/80	8	8	8	8	8	8	8	86	94	23	83	60	39	116	77	41	106	65	36	114	78	36	108	72	123
2/15/80	8	8	8	8	8	8	8	87	93	23	82	59	39	115	76	40	106	66	35	112	77	36	106	70	123
3/3/80	8	8	8	8	8	8	8	87	93	26	84	58	41	115	74	42	109	67	38	114	76	37	106	69	123
4/21/80	8	8	8	8	8	8	8	86	93.5	27	86	59	42	115	73	42	109	67	38	114	76	38	107	69	123
5/28/80	8	8	8	8	8	8	8	87	92	35	88	53	49	121	72	49	114	65	45	120	75	44	112	68	123

Table 7. Test Data with Lubricant C in Chung-Shing Textile Corp.

	OIL CONSUMPTION							VIBRATION (dB)		TEMPERATURE (°C)															POWER
	Lub. Pt. Drops Per Stroke									1st			2nd			3rd			4th			5th			
Date	5th	4th	3rd	2nd	1st	pack	Average Drops per Storke	Crank Case (dB)	The 4th Stg. Cylinder Head (dB)	Suction	Disch	Diff.	Suction	Disch	Diff.	Suction	Disch	Diff.	Suction	Disch	Diff.	Suction	Disch	Diff.	No Load Consumption (AMP.)
5/31/80	8	8	8	8	8	8	8	84	87	32	79	47	47	110	63	46	105	59	42	109	67	41	101	60	118
6/7/80	8	8	7	7	7	7	7.3	84.5	87.5	35	83	48	49	112	63	49	108	59	45	111	66	45	105	60	119
6/26/80	8	8	5	5	5	5	6	84	87	33	81	48	47	110	63	48	107	59	44	111	67	44	103	59	119.5
7/2/80	8	8	4	4	4	4	5.3	84.5	97.5	25	73	48	42	107	65	42	102	60	36	104	68	35	94	59	118
7/11/80	6	6	4	3	3	3	4.2	84.5	97.5	22	69	47	39	102	63	38	98	60	32	99	67	32	92	60	119
8/9/80	5	5	4	3	3	3	3.8	84	87	33	81	48	48	112	64	48	110	62	44	113	69	44	100	60	118.5
9/9/80	5	5	4	3	3	3	3.8	84.5	87.5	34	81	47	48	112	64	49	110	61	45	114	69	45	105	60	120
10/4/80	5	5	4	3	3	3	3.8	84	87.5	30	78	48	46	110	64	46	107	61	41	109	68	41	100	59	118
10/11/80	3	3	3	2.5	2.5	2.5	2.75	84	87	28	77	49	44	108	64	44	105	61	39	107	68	39	99	60	118
11/7/80	3	3	3	2.5	2.5	2.5	2.75	84.5	87	26	74	48	42	106	64	43	103	60	38	106	68	37	97	60	118
12/5/80	3	3	3	2.5	2.5	2.5	2.75	84.5	87	25	73	48	42	107	65	42	102	60	36	104	68	35	94	59	118
1/20/81	3	3	3	2.5	2.5	2.5	2.75	84	87.5	22	69	47	39	102	63	38	98	60	32	99	67	32	92	60	119
2/10/81	3	3	3	2.5	2.5	2.5	2.75	84	87	20	69	49	37	101	64	37	97	60	31	98	67	31	90	59	119.5
3/3/81	3	3	3	2.5	2.5	2.5	2.75	84	87	22	70	48	38	101	63	38	98	60	33	100	67	32	92	60	118
4/1/81	3	3	3	2.5	2.5	2.5	2.75	84.5	87	25	73	48	41	104	63	40	99	59	35	102	67	34	93	59	118.5
5/15/81	3	3	3	2.5	2.5	2.5	2.75	84	87	27	75	48	44	107	63	44	104	60	40	107	67	40	100	60	120
6/7/81	3	3	3	2.5	2.5	2.5	2.75	84	87	28	75	47	44	107	63	44	104	60	40	107	67	40	100	60	118
7/30/81	3	3	3	2.5	2.5	2.5	2.75	83	86	32	79	47	47	110	63	46	105	59	42	109	67	41	101	60	118
8/11/81	3	3	3	2.5	2.5	2.5	2.75	84	87	35	83	48	49	112	63	49	108	59	45	111	66	45	105	60	119
9/9/81	3	3	3	2.5	2.5	2.5	2.75	83.5	86.5	33	81	48	47	110	63	48	107	59	44	111	67	44	103	59	119.5
10/7/81	3	3	3	2.5	2.5	2.5	2.75	84	86.5	27	74	47	44	108	64	44	104	60	39	105	66	39	98	59	118
11/1/81	3	3	3	2.5	2.5	2.5	2.75	83.5	86	24	72	48	40	104	64	39	99	60	34	101	67	34	93	59	118.5
12/5/81	3	3	3	2.5	2.5	2.5	2.75	84	86.5	21	69	48	36	101	65	37	97	60	31	97	66	31	90	59	118
1/10/82	3	3	3	2.5	2.5	2.5	2.75	84	87	20	67	47	36	101	65	37	97	60	31	97	66	31	90	59	118
2/2/82	3	3	3	2.5	2.5	2.5	2.75	83.5	86	20	67	47	36	101	65	37	98	61	31	98	67	31	91	60	120
3/19/82	3	3	3	2.5	2.5	2.5	2.75	83	86	23	71	48	39	103	64	39	99	60	34	101	67	34	94	60	119
4/15/82	3	3	3	2.5	2.5	2.5	2.75	84	86	25	73	48	40	104	64	40	99	59	35	102	67	34	94	60	119
5/3/82	3	3	3	2.5	2.5	2.5	2.75	84.5	87	28	75	47	42	106	64	42	101	59	37	103	66	36	95	59	119.5
6/17/82	3	3	3	2.5	2.5	2.5	2.75	83.5	86.5	28	76	48	43	106	63	44	103	59	40	106	66	40	99	59	118
7/15/82	3	3	3	2.5	2.5	2.5	2.75	83	86	33	79	46	47	110	63	47	106	59	43	109	66	43	102	59	118
8/7/82	3	3	3	2.5	2.5	2.5	2.75	84	87	36	82	46	50	113	63	50	109	59	46	112	66	46	105	59	119
9/9/82	3	3	3	2.5	2.5	2.5	2.75	84	87	30	78	48	46	110	64	46	105	59	41	108	67	41	101	60	120
10/1/82	3	3	3	2.5	2.5	2.5	2.75	83.5	86.5	28	76	48	42	107	65	43	103	60	38	105	67	38	98	60	119.5
11/5/82	3	3	3	2.5	2.5	2.5	2.75	84	87	24	71	47	39	104	65	39	99	60	34	101	67	33	93	60	119
12/18/82	3	3	3	2.5	2.5	2.5	2.75	84	87	23	71	48	38	103	65	38	98	60	32	99	67	32	92	60	119
1/12/83	3	3	3	2.5	2.5	2.5	2.75	83.5	86.5	21	70	49	35	100	65	35	95	60	29	96	67	29	89	60	118
2/7/83	3	3	3	2.5	2.5	2.5	2.75	83.5	86.5	20	68	48	36	100	64	36	96	60	30	96	66	30	89	59	118
3/4/83	3	3	3	2.5	2.5	2.5	2.75	84	87	23	70	47	37	101	64	37	97	60	32	98	66	32	91	59	119
4/17/83	3	3	3	2.5	2.5	2.5	2.75	84	87	25	72	47	41	104	63	42	101	59	37	103	66	36	95	59	119
5/4/83	3	3	3	2.5	2.5	2.5	2.75	83.5	86.5																119.5

Test Data with Lubricant C in Chung-Shing Textile Corp.

Date	OIL CONSUMPTION: Lub. Pt. Drops Per Stroke 5th	4th	3rd	2nd	1st	pack	Average Drops per Storke	VIBRATION (dB): Crank Case (dB)	The 4th Stg. Cylinder Head (dB)	TEMPERATURE (°C): 1st Suction	1st Disch	1st Diff.	2nd Suction	2nd Disch	2nd Diff.	3rd Suction	3rd Disch	3rd Diff.	4th Suction	4th Disch	4th Diff.	5th Suction	5th Disch	5th Diff.	POWER: No Load Consumption (AMP.)
6/9/83	3	3	3	2.5	2.5	2.5	2.75	83	86	26	74	48	41	104	63	42	101	59	37	104	67	36	96	60	118
7/3/83	3	3	3	2.5	2.5	2.5	2.75	84	86.5	26	74	48	41	104	63	41	101	60	37	103	66	36	95	59	118.5
8/15/83	3	3	3	2.5	2.5	2.5	2.75	83.5	86	29	76	47	44	107	63	44	104	60	40	106	66	39	99	60	118.5
9/17/83	3	3	3	2.5	2.5	2.5	2.75	84	86	34	81	47	49	113	64	49	109	60	44	110	66	43	102	59	118
10/15/83	3	3	3	2.5	2.5	2.5	2.75	83.5	86	29	77	48	45	109	64	45	105	60	41	107	66	41	101	60	118
11/2/83	3	3	3	2.5	2.5	2.5	2.75	84	86	25	73	48	42	106	64	42	102	60	38	104	66	38	97	59	118
12/11/83	3	3	3	2.5	2.5	2.5	2.75	83.5	85.5	23	71	48	40	103	63	40	101	61	35	101	66	35	94	59	119
1/7/84	3	3	3	2.5	2.5	2.5	2.75	84	86	19	68	49	39	103	64	39	99	60	34	100	66	34	93	59	118
2/3/84	3	3	3	2.5	2.5	2.5	2.75	84	86	18	67	49	39	103	64	39	99	60	35	102	67	34	94	60	119
3/7/84	3	3	3	2.5	2.5	2.5	2.75	84	86	23	72	49	40	104	64	41	101	60	36	103	67	35	95	60	118.5
4/19/84	3	3	3	2.5	2.5	2.5	2.75	84.5	86.5	25	73	48	41	105	64	41	101	60	36	103	67	35	95	60	119
5/12/84	3	3	3	2.5	2.5	2.5	2.75	84	87	26	74	48	41	104	63	41	101	60	36	103	67	35	95	60	118
6/11/84	3	3	3	2.5	2.5	2.5	2.75	84	86.5	28	76	48	44	107	63	44	103	59	40	107	67	39	99	60	118
7/10/84	3	3	3	2.5	2.5	2.5	2.75	84	86	32	80	48	47	110	63	47	106	59	41	107	66	40	100	60	118.5
8/7/84	3	3	3	2.5	2.5	2.5	2.75	84	86	36	83	47	50	113	63	50	108	58	45	111	66	44	103	59	119
9/3/84	3	3	3	2.5	2.5	2.5	2.75	84.5	86	30	77	47	45	109	64	45	105	60	39	105	66	39	98	59	118
10/16/84	3	3	3	2.5	2.5	2.5	2.75	84	87	28	75	47	43	107	64	43	103	60	37	103	66	37	96	59	118.5
11/10/84	3	3	3	2.5	2.5	2.5	2.75	84.5	87	25	72	47	42	106	64	42	102	60	36	102	66	36	95	59	118
12/1/84	3	3	3	2.5	2.5	2.5	2.75	83.5	86.5	23	71	48	41	106	65	41	101	60	36	101	66	36	95	59	119
1/8/85	3	3	3	2.5	2.5	2.5	2.75	84	86.5	20	69	49	39	104	65	39	99	60	33	99	66	33	92	59	119
2/3/85	3	3	3	2.5	2.5	2.5	2.75	84	86.5	20	69	49	39	103	64	39	99	60	33	100	67	33	93	60	119.5
3/14/85	3	3	3	2.5	2.5	2.5	2.75	84	86	22	71	49	40	104	64	40	100	60	35	102	67	36	96	60	118
4/3/85	3	3	3	2.5	2.5	2.5	2.75	84	86.5	24	72	48	42	105	63	42	101	59	37	104	67	37	97	60	118
5/9/85	3	3	3	2.5	2.5	2.5	2.75	84.5	87	26	74	48	42	106	64	42	101	59	38	105	67	37	97	60	118
6/7/85	3	3	3	2.5	2.5	2.5	2.75	84	87	28	76	48	44	108	64	43	102	59	39	105	66	39	98	59	119
7/11/85	3	3	3	2.5	2.5	2.5	2.75	84	87	31	79	48	45	108	63	44	103	59	40	106	66	40	99	59	118
8/13/85	3	3	3	2.5	2.5	2.5	2.75	84.5	86.5	33	80	47	46	109	63	46	105	59	42	108	66	41	100	59	118
9/2/85	3	3	3	2.5	2.5	2.5	2.75	84	87	29	77	48	44	108	64	44	103	59	40	106	66	40	100	60	119
10/5/85	3	3	3	2.5	2.5	2.5	2.75	84	86.5	27	75	48	43	107	64	43	103	60	38	104	66	38	98	60	120
11/3/85	3	3	3	2.5	2.5	2.5	2.75	84.5	87	26	73	47	43	107	64	43	103	60	37	104	67	38	98	60	119
12/1/85	3	3	3	2.5	2.5	2.5	2.75	84	86.5	24	72	48	40	105	65	40	100	60	34	100	66	35	94	59	119
1/5/86	3	3	3	2.5	2.5	2.5	2.75	84	86.5	23	71	48	39	103	64	39	99	60	33	99	66	33	92	59	118.5
2/15/86	3	3	3	2.5	2.5	2.5	2.75	84	87	21	70	49	39	103	64	39	99	60	33	100	67	33	92	59	118
3/7/86	3	3	3	2.5	2.5	2.5	2.75	84.5	86	24	72	48	41	105	64	41	101	60	36	102	66	36	96	60	119
4/11/86	3	3	3	2.5	2.5	2.5	2.75	84	86	26	74	48	42	105	63	42	102	60	38	104	66	38	98	60	119
5/10/86	3	3	3	2.5	2.5	2.5	2.75	84.5	86.5	27	75	48	44	107	63	44	103	57	40	107	67	40	99	59	119
6/7/86	3	3	3	2.5	2.5	2.5	2.75	84	86.5	29	76	47	46	109	63	46	105	59	42	108	66	43	102	59	118
7/3/86	3	3	3	2.5	2.5	2.5	2.75	84	86.5	30	78	48	45	109	64	46	104	58	42	109	67	43	102	59	119
8/4/86	3	3	3	2.5	2.5	2.5	2.75	84	86	35	82	47	49	112	63	49	107	58	45	111	66	45	105	60	118.5
9/16/86	3	3	3	2.5	2.5	2.5	2.75	84.5	86.5	28	75	47	44	108	64	44	104	66	40	106	66	40	99	59	118.5
10/23/86	3	3	3	2.5	2.5	2.5	2.75	84	86	27	75	48	42	106	64	43	103	60	38	104	66	38	97	59	118
11/20/86	3	3	3	2.5	2.5	2.5	2.75	84	86.5	25	72	47	39	103	64	39	99	60	33	100	67	33	92	59	118
12/17/86	3	3	3	2.5	2.5	2.5	2.75	84	86.5	25	72	47	39	103	64	39	99	60	33	100	67	33	92	59	118
1/14/86	3	3	3	2.5	2.5	2.5	2.75	84	86	18	66	48	35	98	63	35	95	60	29	95	66	30	89	59	118

Fig. 17. Oil Consumption of Norwalk Compressor in Chung-Shing Textile Corp.

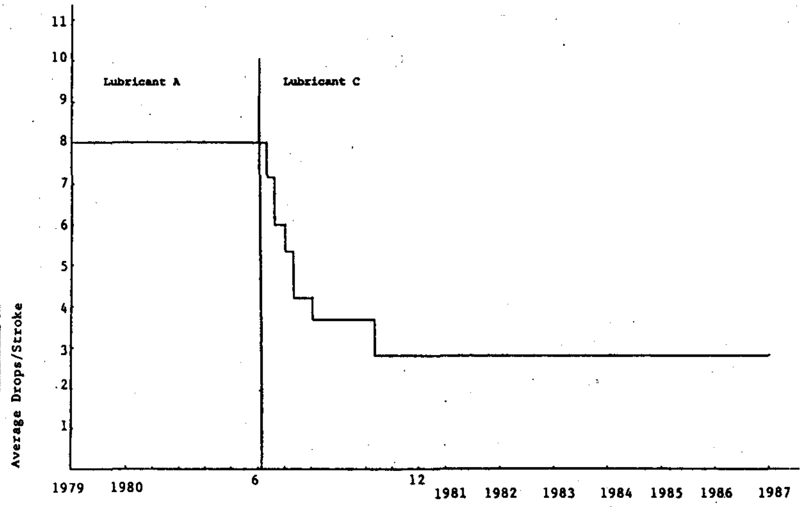


Fig. 18. Temperature Difference Between Suction Air & Discharge Air (°C) of Norwalk Compressor in Chung-Hsing Textile Corp.

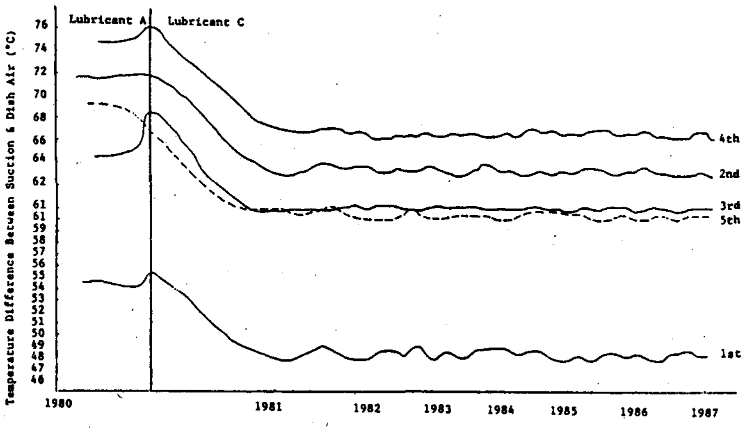


Fig. 19. No Load Power Consumption (Amp.) of Norwalk Compressor in Chung-Shing Textile Corp.

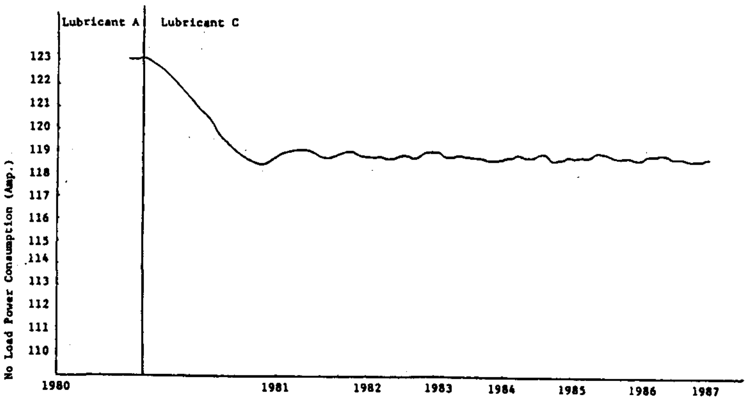


Fig. 20. Vibration of Norwalk Compressor in Chung-Shing Textile Corp.

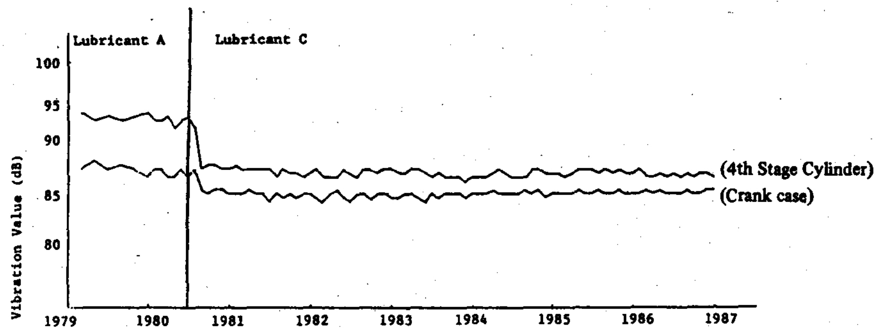


Table 8. Test Data of Carbon Deposit in Chung-Shing Textile Corp.

	Lubricant A					Lubricant C
Test Time	From Feb. 2, 1980 to May 20, 1980					From May 27 1980 to Jan. 15, 1987
	1st stg. disch. valve	2nd stg. disch. valve	3rd stg. disch. valve	4th stg. disch. valve	5th stg. disch valve	
Operating Hours	328	224	136	328	328	About 6 years & 7 Months
Carbon Deposit Thickness	Max. 4MM Average 1MM	Average 0.9 MM	Average 0.6 MM	Average 1.6 MM	Average 0.5 MM	Clean valve, Little Carbon Deposit

TEST RESULTS

1. Oil Consumption
   The oil consumption of lubricant C is about 1/3 of the oil consumption of lubricant A.
2. Carbon Deposit
   The lubricant C create little carbon deposit on discharge valves than lubricant A.
3. Vibration
   The vibration values of the compressor tested on crankcase & cylinder when using lubricant C are 1.5 dB, (4.5 dB – 6 dB) lower than using lubricant A.
4. Temperature
   The temperature difference between suction air and discharge air when using lubricant C is 5°C–8°C lower than using lubricant A.
5. No Load Power Consumption
   The no load power consumption of the compressor when using lubricant C is 2.24%–4% lower than using lubricant A.

Table 9. The Fieid Test in Taiwan Field

Company	Machinery	Power	Sound Pressure level or Vibration (dB)	Temperature Difference between Gear Reducer Surface and Room Temp.	Teeth Surface of Gear Reducer	Oil Servece Life
Chin-Tai Cement corp.	Maschinetabrik, 1500 KW Helical Gear Reducer of Raw Grinding Mill	With Lubricant C 5.3% lower than Lubricant A	With Lubricant C Sound pressure level is 4.5 dB, Vibration is 2.5 dB lower than using Lubricant A	The oil Temperature vary with process Temperature, the Temperature difference only for reference.	The weight of iron particle with Lubricant A is 72.4g/500 operating hours. With Lubricant C is 20.9g/500 operating hours.	With Lubricant A 8 months, With Lubricant C 5 year and 5 months
San-Jung Rubber corp.	100 HP Helical & Herring bone Gear Reducer of NO.6 Extruder	With Lubricant C 19.8% lower than Lubricant A	With Lubricant C Sound pressure level is 8 dB, Vibration is 6 dB, lower than using Lubricant A	With Lubricant C Temperature Difference is 24°C lower than using Lubricant A	Pitting and Scoring get improved	With Lubricant A 10 months, With Lubricant C 1 year and 4 months and still running
Taiwan Fertilizer Corp.	Maschinetabrik, 108HP Gear Reducer of Ammonia Pump	With Lubricant C 11.0% lower than Lubricant A	With Lubricant C Sound pressure level is 6 dB, Vibration is 3dB, lower than using Lubricant A	With Lubricant C Temperature Difference is 8.5°C lower than using Lubricant A	Pitting and Scoring get improved	With Lubricant A 8 months, With Lubricant C 4 year
Formosa Rubber Corp.	Nippon Roll, 400HP, Helical & Herring bone Gear Reducer of Mixing and Kneading Rubber	With Lubricant C 8.6% lower than Lubricant A	With Lubricant C Sound pressure level is 9.5dB, lower than using Lubricant A	With Lubricant C Temperature Difference is 6°C lower than using Lubricant A	Pitting and Scoring get improved	With Lubricant A 1 year, With Lubricant C 5 year and 5 months
Nanking Rubber Tire Corp.	Nippon Roll, 800HP, Helical & Herring bone Gear Reducer of Banbury	With Lubricant C 8.77% lower than Lubricant A	With Lubricant C Sound pressure level is 11dB, lower than using Lubricant A	With Lubricant C Temperature Difference is 19°C lower than using Lubricant A	Pitting and Scoring get improved	With Lubricant A 1 year, With Lubricant C 4 year and 2 months
Cathay Plastic Industry Co.	Nippon Roll, 200KW, Helical & Herring bone Gear Reducer of Calender	With Lubricant C 8.8% lower than Lubricant A	With Lubricant C Sound pressure level is 12.3dB, Vibration is 13.3dB, lower than using Lubricant A	With Lubricant C Temperature Difference is 10.2°C lower than using Lubricant A	Pitting and Scoring get improved	With Lubricant A 1 year, With Lubricant C 5 year and 1 month
Goodyear Taiwan Ltd.	Farrel 500HP, Helical & Herring bone Gear REducer of 9D Banbury	With Lubricant C 7.7% lower than Lubricant A	With Lubricant C。 Sound pressure level is 12.5dB, lower than using Lubricant A	With Lubricant C Temperature Difference is 17°C lower than using Lubricant A	Pitting and Scoring get improved	With Lubricant A 1 year, With Lubricant C 4 year and 8 months
Formosan Glass Corp.	Ingersoll-Rand XLE Compressor	With Lubricant C 3.2% lower than Lubricant A	With Lubricant C Vibration is 6dB, lower than using Lubricant A	With Lubricant C Temperature Difference Between Suction Air & Discharge Air is 10°C lower than using Lubricant A	With Lubricant C Creat little Carbon Deposit on Discharge Valves Than Using Lubricant A	The Oil Consumption of Lubricant C is about 1/3 of the Oil Consumption of Lubricant A

CONCLUSION

In heavy industries there are always difficult working conditions such as heat, load, wet, dirt, speed, etc. which make the lubrication and maintenance of machines very difficult.

Although pure hydrodynamic lubrication is always seeked for, in real life boundary or mixed condition appears more often, it causes the failure of lubrication and destroys the machine parts.

It is concluded after this 8 year study that with MoS_2 and 2nd solid lubricants mixed with mineral oils, the friction can be reduced by the solid lubricants and the metal surface can be improved and be further protected, two or more metallic sulfides in combination even produce a synergistic effect and increase the lubricity greatly.

We found from the periodic lab oil analysis that the average life of lubricant A in our tests is 8 to 12 months, and lubricant C can last for 4 to 6 years. The author also found the vibration, noise, power consumption, metal wear and operating temperature were all reduced because the MoS_2 and the 2nd solids.

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Table 3. LC-Periodic Model Estimates

Series	d_f	ρ	ω_0
Gota River Flow	0.57	0.0274	0.0016
Gunpowder River Flow	0.50	0.1060	0.0016
St. Lawrence River Flow	0.79	0.0040	0.0016
Blacksmith River Flow	0.66	0.0983	0.0069
Sunspot Number	0.85	293.45	0.0100
Central England Annual Temp.	0.42	0.3826	0.0016
Eastern American Annual Temp.	0.60	0.2322	0.0016

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