„On the potential value of practically relevant tribotechnical Parameters for dry, boundary and mixed friction“

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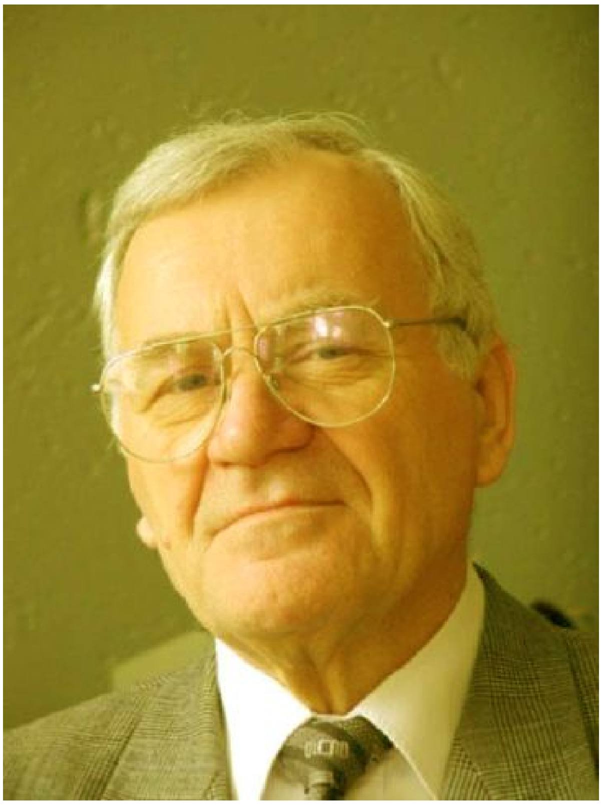

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"On the potential value of practically relevant tribotechnical Parameters for dry, boundary and mixed friction"

1. On Tribology (Tribotechnology) and Tribotechnical Systems and Parameters

The noun "tribos" comes from Greek and means "friction". Friction and frictional effects are fundamental both to our natural material world and to that world created by human technology.

Nothing can work without friction. There are many definitions of friction and tribology:
For example, DIN 50323 defines the complex field of tribology as follows,: "Tribology is the science and technology of surfaces which are affecting each other and are in relative motion," and the writer adds - "of intentional and prevented relative motion". The quote continues, "Tribology encompasses the entire field off friction and wear including lubrication, and also includes corresponding interfacial interactions, both between solid bodies and also between solid bodies and fluids or gases" (the so called R-S-V-Verhalten [Friction-Lubrication-Wear-Behaviour] of friction pairings).

With the pragmatic aim of the implementing the basic tribological principles technically, Brendel [3] "Wissensspeicher Tribotechnik" 1978 p.13, has defined tribotechnology as the,: "Sub-field of technology which strives to achieve the technical and economic mastery of friction and wear at places where friction occurs through the application of scientifically based measures to their design, manufacture assembly, operation and maintenance."' Whereby the frictional behaviour of the friction pairings largely determines the functional fulfilment, reliability, achievable quantity of work and service life of the majority of technical systems and the potential of their scientific and technological development.

Although the ancient Chinese, Arab, Greek and Roman cultures had already gained practical tribotechnical experience, with sled runners etc., and with the invention and application of the wheel, the axle and the shaft (starting about 5000 years ego), fundamental research into the friction of solid bodies over the last three hundred years can only be connected to a few European names such as: Amontous (1699), De la Hire (1732), Euler (1750), Coulomb (1785), Morin (1835) and Hirn (1854). In Europe, amongst the outstanding works during the last decades of our century, have been, for example, the fundamental work of the Englishmen Bowden and Tabor, known through the book "Friction and Wear of Solid Bodies", 1959 (cf.[1]), and the works of the Moscow School of Tribology by Kragelski, Dobycin and Kombalov with, e.g., the book published in German, "Grundlagen der Berechnung von Reibung und Verschleiß" (Principles of the Calculation of Friction and Wear) (cf.[2]). The book from Kragelski alone contains 740 relevant bibliographical references.

An almost inestimable number of international, European and especially German academic engineers and pure scientists have published and are publishing material on the friction pairings of many machine elements and automation components, from screw connections and gears tu longitudinal guides, shaft bearings, elastomer seals, as well as the working and drive units of tools, machines and automatons (e.g. [4], [5], [6], [7], [8], [9], [101, [ii], [12], [13], [14], [15], (16], [17], [18]).

One repeatedly finds confirmation of the notion that the tribopairings are largely restricted to solid mechanical elements which realize relative motions in positive and non-positive contact with the kinematic degrees of freedom f = 1. .5 and, at the same time, with the most varied types of power transmission.

In addition, wanted and unwanted lubricants, and other unavoidable fluids, powders and rheological substances together with other imponderable influencing variables and effects also determine the so-called F-L-W behaviour.

The systematic representation and evaluation of the geometrical forms and the form pairings (cf. [18]) as well es their tribological effects (e.g. hydrodynamics in the case of fluid lubrication) provide systematic design possibilities for the analysis, evaluation and synthesis of tribological active pairings and active spaces, as well as the many real technical applications and the standardised or non-standardised friction pairings in model test-benches.

This means that just about every chair for machine elements, for just the German Universities and Technical Universities alone, has its own fount of tribotechnical and tribological knowledge about a multitude of machine elements and friction pairings, The same applies tu a multitude of important companies active in the machine and apparatus building sectors.

In addition, there are well-known tribological databases such as that of the Bundesanstalt für Materialforschung (BAM) (Federal Institute for Material Research) in Berlin and increasingly more comprehensive information is becoming available in the internet (cf. [20]).

It has become almost impossible to maintain an overview over the large quantity of existing and emerging tribological and tribotechnical date and information. This means that many things are being invented two or three times or are being discarded. As a result, scientific and technological development 5, to a certain extent, taking two steps forward and one step beck, and not just because of the fashionable ignorance of some young academics, which one sees increasingly often nowadays, who only work on, accept and quote scientific information which 3 not more than 8 tu 10 years old. As a result, many new “old mistakes” are repeatedly being made and much valuable work of the previous decades is no longer being taken into consideration. This also applies tu the tried and tested tribotechnical parameters as well es tu their potential value and limits of application.

For tribotechnical statements to be applicable tu technical tribosystems, it is particularly necessary tu define and limit the relevant existence space of the tribosystem spatially and temporally. Not until this has been done can the nature, number and effects of influencing variables and the relevant elements and relations be formulated and possibly even generalised.

The tribological credo [17] of one of the most important Nestors and the first bearer of the title “Doctor of Engineering” of German machine building in teaching, research and development and also the first principal of the TH (Technical University) Dresden after the II. World War, Prof.Dr.-Ing.habil. Enno Heidebroek still applies after over 50 years, “Whoever understands tribotechnology, understands machine elements, whoever understands machine elements, understands machines.”

In his time, he worked out over 120 variables which had an influence behaviour of F-L-W behaviour of friction pairings and, in those days, most of the currently used dopes, surface layers, plastics, compound materials, lubricants and processes were just not available.

This also applies to the multitude of modern plastic conglomerates (e.g. PTFE material mixtures) which are available nowadays as sliding and sealing materials, es well as to the high-performance elastomers such es NBR and FPM rubbers and polyurethane (AU), which are used with minimal quantities of lubricant or even in totally lubrication-free operation.

Unfortunately, apart from the technical advice on product related applications from the individual manufacturers, there are too few parameters and too few facts Which are practically applicable or even at least suitable for making rough predictions about tribotechnical pairings. That is certainly also a weakness in the state of tribological knowledge, which up to now, has not generally enabled any satisfactorily modelled, mathematical predictions to be made because of the complex frictional processes the highly-developed lubrication technology with doped and non-doped technical oils and greases with which the technical friction states of mixed, boundary to dry friction, particularly also for plastomer and elastomer plastics.
The term boundary friction is also frequently known as to as contact-layer friction, thin4ilm friction or even hunger lubrication.

Three selected tribotechnical parameters are discussed in the following text.

2. On the Coefficient of Friction μ (Sliding Friction)

The non-dimensional coefficient of friction μ, whether of motion or stasis is regarded as if it were the equivalent a natural constant in the minds of all those engineers end tribologists on the basis of their totally personal knowledge and experience, and it has established itself as a “neutral” parameter. Everyone is in some way familiar with them in connection with the so-called “non-positive” or friction contacts of most solid bodies in static and motional friction. In this way, the coefficient of friction has to serve the needs for more, less, constant, repeatable end stable friction, for friction with “increasing and decreasing” characteristics, for frictional oscillations, but also for its prevention, for friction with rapid running-in wear and low operational wear, for low starting friction (static friction) etc.

Thus one regards coefficients of friction of $\mu < 0.05$ as low (e.g. for bearings) and $\mu > 0.6$ as already high (e.g. for brakes), in which, as a rule, the coefficients of static friction μ_0 with $\mu_0 = (1.5 ... 2.5) * \mu$ are always set higher than the coefficient of motional friction.

If coefficients of sliding friction occur in which $\mu >> 1$ or $\mu << 0.1$, most users are almost even more disconcerted by this uncertainty in the usual order of things than they are by static friction μ_0 in which, unusually, $\mu_0 < \mu$.

One therefore lives, works and talks with the “normal” coefficients of friction $\mu = 0.05 ... 0,06$ one likes to believe in the universal validity and applicability of this very simple non-dimensional number. One connects one's perceptions of “smooth and sluggish running” with it, and everyone also has his very own concrete application examples, collective loads, influencing variables, geometry pairings, wear patterns etc. in his mind's eye: from the PTFE and GG glide bush through the self-inhibiting thread, the elastomer movement seals to disc brakes or multi-disc clutches. Sometimes preference is even given to using the so-called “clean” coefficients of friction μ which have been determined on all sorts of tribometers under defined tribotechnical conditions, nevertheless, their transferability, reproducibility and practical application are always having to be discussed.

It is known that the sliding friction is the limiting factor for the frictional resistance (frictional force):
“The frictional resistance and with that the frictional force F_R is the sum of all the resistive forces F_{RW} acting against the intended or already existing relative motion of two non-positive contacting surfaces in the common macrogeometrically definable tangent planes.”

For the degree of freedom of motion of the friction partners RP_1 and RP_2 (cf. figure 1) f = 1 to 3 applies, and with that $- \operatorname{sgn} v_{rel} = \operatorname{sgn} F_R$ with $v_1 - v_2 = v_{rel} = v$.

The pragmatic user is often not particularly interested in the real micro- and macrogeometrical structures or functions, processes and effects which are present and occur in the contact zones of the friction partners under dry to mixed friction. One also ignores the existence and the influence of the real and nominal (apparent) contact surfaces A_r and A_n, the contact relationship $\eta = A_r / A_n < 1$ ($\eta = 0,01 ... 0,08$ depending upon the material pairings in dry to contact layer friction) and the corresponding frictional shearing strain τ_R.

It is known that the frictional shearing strain τ_R is composed of the two main components “adhesion and cohesion tensions”. $F_R = \tau_R * A_r$ applies.

For the “normal” coefficient of friction, just as little or even reluctant attention is paid to the chemical and physical micro-range processes on the surfaces as is to the associated energetic and thermal processes, or to the various wear mechanisms.

One ignores the fact that surfaces can contact one another without having an external normal reaction F_N and, in this way, that the existing resistances forces in the contact plane can equally cause a frictional force F_R, e.g., through adhesion and micro-positive contacts alone. The formally calculated coefficient of friction would then be $\mu \rightarrow \infty$ because $F_N \rightarrow 0$!

Nevertheless, one refers to Coulomb, formulates that so comfortable model $F_R \sim F_N$ and thereby, as shown in figure 1, the so-simple, non-dimensional coefficient of friction μ as

$$\mu = \frac{F_R}{F_N} \qquad \text{Equation (1)}$$

with partial $\mu \sim F_R$. For $\mu < 1$ that is clearly, quasi a type of "degree of dynamic effect", that is, e.g., "How much F_N do you need to obtain a specific F_R?"

The μ-relationship Is based on the special model case, that $F_R(v)$ and $\mu(v)$ are constant and $F_R(F_N)$ is linear. Velocity exerts no influence and there is no difference between static and motional friction; all the known and imponderable influencing variables of real friction are excluded. To this day, most coefficients of friction are still being simplified in this way. We have permitted ourselves to illustrate these trivial relationships of equation clearly in figures 2.1, 2.2 and 2.3.

Figure 3 shows the known, principal velocity dependent frictional processes. Because of the extreme difficulties involved in determining and evaluating frictional forces and coefficients of friction of many technical friction pairings at velocities v < 0,001 m/s and v > 10 m/s, one generally finds velocity-dependent coefficients of friction for dry to mixed friction presented only for the range 0,01... 6 m/s.

In addition, there are further special frictional behaviours; figure 4 shows, as per [21] and [22] e.g., the frictional behaviour of mass-encumbered friction pairings under the influence of defined micro-vibration (without relevant effects of chemical and physical surface changes) witch totally changes the original, vibration-free frictional behaviour $F_R(v)$ and μ (v) because the rising $\mu_S(v)$ parameter lines prevent the creation of the self-starting, non-damped "stick-slip" friction oscillations and can reduce the static friction (starting friction) to $F_{R0S} = 0$ and $\mu_{0S} = 0$. This so-called "**vibrational or false friction**" phenomenon $F_{Rs}(v)$ and $\mu_s(v)$ in technical friction pairings can develop at remarkably low minimum frequencies, e.g. from f ≥12 Hz and amplitudes A < 1mm, and such conditions can easily occur in many real technical systems.

On account of the multiplicity and multitude of the variables Influencing the friction, one must remember some principles applying to the use of coefficients of friction:

- The statement of coefficients of friction 5 only then realistic if they are comprehensible, i.e. if the materials of the pairing, additional active partners such as lubricants, gases, water and air humidity, as well as the friction conditions (mixed to dry friction) and the type of friction (sliding, rolling, rolling-sliding friction ...) can also be stated. Furthermore the designations of the geometry of the friction partners and their form pairings and surface structures, their concrete application examples (multi-disk clutches, disk brakes, ...) as well as the influence of collective loads, e.g. forces F(t) and $F_N(t)$, velocities v(t) and $v_{rel}(t)$ are all indispensable.

- The desired problem-free transfer of so-called "neutral" coefficients of friction from one machine element to another machine element and to the coefficients of friction measured on various tribometers and tribo-testbenches is, as a rule, not possible.

Where this does function, one should not work with the non-dimensional value μ, but with concrete form and surface related frictional forces and frictional moments.

- At the moment, no one knows a representative "minimum tribological surface" in the frictional contact which can be applied to the non-dimensional coefficient of friction μ. The coefficient of friction μ and the associated normal reactions F_N are therefore reduced in practice according to experience and modelled on single points, mid-lines, average friction disk diameters, angles of belt contact, effective centres of gravity etc. of circular discs, annuluses, circles, cones, spheres, wedges, screw-threads etc. One acts as if there were not a multitude of different normal reactions related to partial surfaces, an abundance of different, velocity-related frictional forces and consequently many different coefficients of friction acting, but a single artificial value (μ, F_N) which embraces all these effects. These simplifications can be found in most calculations, from those concerning machine elements, e.g., non-positive peripheral gears (Eytelwein's equation) to those calculations for friction clutches and screw-threads. The inclined distributions of F_N, F_R,$F_R(v)$ and μ(v), e.g., in the case of multi-disk clutches, cone clutches, set screws etc., are therefore replaced by an idealised coefficient of friction μ and a correspondingly simplified model which only applies to one particular machine element or to a specific module with its existing geometrical structure and collective bad concerned. These are increasingly unacceptable approximations.

- Static friction μ_0 is even more problematical than the motional friction and its coefficients of friction μ(v).

Firstly, one should subdivide static friction into enduring "stable" stiction, coasting friction and starting or breakaway friction. Even for the same friction pairing, the coefficient of friction μ_0 can be very different, e.g., depending upon the dwell-time of the frictional condition, the rate of application of the bad F_N (t) and F(t), the enforced v(t) behaviour in ranges such as $v << 0.001$ m/s as well as in the case of displacements As of friction partners with play $\Delta s < 0.1$ mm (problem of force and distance excitations in the micro and macro ranges of the frictional contact!).
As a rule, one does not see which "bad independent" forces, depending upon the amount and direction of the friction contact, e.g., as a consequence of existing elastic potentials in the contact zone, are already loading the friction pairing with "stable" stiction or an induced "start friction". This can considerably reduce or increase the starting friction. Particularly problematical in this connection are modern friction pairings with friction partners made of rubber-elastic plastics, which, e.g., can form so-called "Schallermach-waves" and "flaking effects" during the frictional processes (cf. also [23] and [24]).

3. Specific Surface Performance PRA / p • v Relationship

For decades, the nominal surface pressure $p = F_N / A_N$ of the frictional contact with the normal reaction F_N, the nominal or apparent contact area A_N in the presence of the velocity v or v_{rel}. as well as with an intended or prevented relative motion has been trivially regarded as the loading capacity of a friction pairing.

At the turn of the century, at the very beginning of machine tool manufacture, the experience had already been gained that, in the case of mixed friction (e.g. feeding processes of oil, grease and water-lubricated plain bearings) the temperatures achieved in bearings running at higher speeds under low pressures were roughly the same as the temperatures achieved in bearings running more slowly but under higher pressures ($p \cdot v \approx$ constant).

With the increasing use of roller bearings and hydrodynamic plain bearings together with their purpose-orientated play calculations and designed gap, for which the $p \cdot v$ value considerations are not very useful, the plain bearings with dry to mixed friction and their $p \cdot v$ value determination then became less important during the following years.

With $F_R = \mu \cdot F_N$, $P_R = F_R \cdot v$ and P_R / A_n one obtains equation (2) for the $p \cdot v$ value

$$p * v = \frac{1}{\mu} * \frac{P_R}{A_n} = \frac{1}{\mu} * P_{RA}$$ Equation (2)

and

$$P_{RA} = \mu * p * v$$ Equation (3)

for the specific surface performance.

The specific surface performance P_{RA} is therefore a surface-related frictional output P_R, which is equally represented as “heat output” in the friction pairing, which is directly proportional to the coefficient of friction μ and the bad independent external variables F_N and the $p \cdot v$ value. This heat output is correspondingly derived from the friction pairing and has to be tolerated, or limited by the choice of friction pairings at the design stage, so that the permissible temperatures and mechanical strength values of the friction partners as well es the available lubricants in the case of mixed to thin-film friction are not exceeded. The material loads on the surfaces of the friction partners (e.g. danger of scuffing, wear) are increased by the fact, that the real contact area $A_r = \eta \cdot A_n$ and $\eta < 1$ is always significantly smaller than A_n.

In practice, at an ambient temperature T_u, average operating temperatures T_m from $T_m = T_u + (15 \ldots 85)°C$ are accepted as “normal" for many lubricated friction pairings (e.g. machine elements).
The temperature T ist es all practical people know from experience, one of the most sensitive and surest indicators of tribological stability (friction, wear, lubrication status) and operating safety. Feeding processes, wear processes, lubricant dwell-time stability, lubricant film thickness as well as changes in the “external" conditions (e.g. F_N, p, V_{rel}, $T_{ambient}$) manifest themselves very rapidly through the temperature and its change over time T(t).
Plastic friction partners are particularly bad conductors of heat and therefore have lower permissible $p \cdot v$ values. Thus the permissible $p \cdot v$ values for conventional plastic bearing materials with corresponding coefficients of the friction μ under conditions of dry to mixed friction range from low $p \cdot v = 1 \ldots 10$ bar • m/s or 10 ... 100 Watt/cm² up to high $p \cdot v$ values of plastic, heavy-duty conglomerates or

composite materials in pairings with metallic or silicate surfaces, from p • v =50 ... 500 bar m/s or from 500 ... 5000 Watt/cm²! These empirical p • v values are essential for the synthesis of suitable modern friction pairings in the mixed, boundary and dry friction ranges.
One can therefore speak of a “come-back" of useful p • v limits for p(v) functions in the evaluation of modern maintenance-free friction pairings.

In this connection, figure 5 shows some examples of permissible p(v) functions for corresponding p • v values according to [17].

Thus, permissible p • v values and p(v) functions are experimentally determined in such a way that, under tolerable temperature, wear, friction and operating conditions, the corresponding p and v values vary and the usual working quantities z as sliding distances with $z_S \approx 1000 ... 10000$ km and operating time $Z_h = 3000 ... 30000$ hours can be achieved with reliable fulfilment of functions for machine elements (plain bearings, roller bearings, movement seals etc.

4. Lubrication Dwell-Time Stability / K - Factor

The achievable working quantities (sliding distances, operating hours) of friction pairings under mixed and boundary friction depend to a great extent upon the duration of the desired tribotechnical effectiveness and dwell-time stability of the lubricants or substances with a lubricating effect in the lubrication gap.

Tribotechnical effectiveness and dwell-time stability mean that the friction pairings work normally in respect of their friction, wear, oscillation, noise and temperature behaviour while they are completely fulfilling their technical I technological function, under the given collective loads and miscellaneous conditions of the entire system.

Temperature and noise (e.g. stick-slip vibrations) are always sensitive and reliable indicators of deviations from tribotechnical normality and give warning of impending danger in the active spaces of the friction pairings and to the entire technical system, and in practice they can be attributed purely pragmatically to a measurable loss of lubricant. Whereby, es a rule, only a relatively small proportion of the lubricant has been consumed by direct tribological effects (e.g. oil oxidation and reduction processes, ageing, water foaming) in the lubricating gap of the friction pairing. The greatest part of the lubricant is lost through leaks or, e.g. in the case of pneumatic aggregates, through thermodynamic and flow effects caused by the compressed air.

If the mass of the lubricant and possibly also its tribotechnical effectiveness are reduced, then, in order to continue normal operation, more lubricant must be supplied to the friction pairings. This is either done by feeding in lubricant either in a continuous or discontinuous flow (manual grease gun, various central lubrication systems, oil-mist lubrication etc.) from outside. The difficulty of attaining the correct dosing often results in considerable over-lubrication, which, in turn, leads to higher energy losses, lubricant losses and environmental pollution.

It is known that most technical friction pairings display a so-called “emergency running” under mixed and boundary friction (thin film friction, stiction), i.e. if the lubricant supply stops it is possible to continue with functional operation, sometimes for hours, in which case increasing temperature and the development of noises warn of the impending breakdown (increased wear, higher friction, even “scuffing”).

The emergency running properties, and also the principle relevant investigations of the dwell-time stability of tried and tested lubricants (oils, greases, solid lubricants) in the lubrication gap lead to the following statement.
“For a defined friction pairing, their exists a specific minimum volume of lubricant (minimum lubricant mass) on and in the contact surfaces of the friction partners in frictional contact and in their action space, which, as the so-called “hunger lubrication”, provides the necessary tribological stability for a certain period of operation”.

In fact, one only needs to replace this minimal mass of lubricant mass in good time and reliably in order to prevent the threatening breakdown.

This knowledge leads to the so-called “maintenance-free, life time lubrication”, which is performed during the assembly of the components (“basic or assembly lubrication") as the first or permanent provision. Whereby, one can differentiate between two types of life-time lubrication:

a) The lubricant for the friction pairings of the components is either filled in such a way that it has life-time effectiveness, in special depot spaces (hollow or filled with a lubricant carrier structure), in special reservoir or dosing units (e.g. lubricating felt, lubricating wicks) at the friction-sites or stored in the “normal” design or functional structures of the friction site which act as depots. Examples include: dead spaces, grooves, grooves, corners, lubricant pockets, edge collars, surface areas of the friction partners which do not have direct frictional contact, very porous or fibrous friction partners.

b) The contacting surfaces of the friction partners and the contact space (lubricating gap) formed by them are themselves depots for the relatively firmly adhering small quantities of lubricant (e.g. shaft I plain bearing half-lining, pneumatic cylinder / piston seal, cam plate / cam roller, slideway / slide block, roll barrel / bearing path etc.)

In both cases, the mass of the lubricants, the dwell-time stability, and the dosed resupply of the lubricant must be determined. In order to be able to determine case a) and its required reservoir volume for a desired service life or quantity of work, one needs the experimental results of case b) with minimal masses of lubricant.

Thus, experimental investigations in Erfurt at the end of the seventies, into the friction pairings of conventional pneumatic work cylinders of ØNG 63mm ($AlMg_3$ anodised, oxidised) with NBR lip seals of medium hardness had shown that after assembly lubrication with mineral oils and water-resistant lubricating greases and after a short running-in period and under normal bad conditions between the elastomer NBR seals and much larger cylinder bearing surface two edge collars and a firmly adhering coating of lubricant had formed (cf. [26]).

This lubricant coating has a fictive film-thickness of less than 1 µm and it cannot be scraped off quickly, not even by sharp-edged seals under operating pressure, e.g. at $p = 6.3$ bar and $v < 0.4$ m/s.

Despite mixed friction, one obtains the effect of a quasi “sliding lacquer”, whose “film thickness” can only be “consumed” little by little, stroke by stroke. Tribological stability exists here. The number of attainable bad alternations and with that the attainable quantity of work after removal of the edge collars and without resupplying the lubricant delivers with a linear approach the necessary, very small, mass of lubricant for a double stroke DH of the piston or, in other words, for the definable sliding distance s and operating times h.

Related to the unit of surface area A = 1 cm² and 1DH one then obtains a necessary, relative mass of lubricant m as per equation (4):

$$\bar{m} = \left(\frac{10^{-6}\, mg}{1\, cm^2 * 1\, DH}\right) * K \qquad \text{Equation (4)}$$

or, generally for any A, z, DH and with various lubricants with the lubricating factor K, a required mass of lubricant m with equation (5>.

$$m = (10^{-6}\, mg / cm^2) * z * A * K \qquad \text{Equation (5)}$$

So one obtains, e.g. for desired $z = 10^7$ DH, with A = 600 cm² cylinder bearing surface of ØNG 63 and a lubricating factor K = 2, the mass of lubricant to be deposited and dose-relased for $m = 10^{-6} \cdot 10^{-7} \cdot 600 \cdot 2$ mg = 12 g or an oil volume of von 12 cm³. These values are realistic (cf. case a), reservoir volumes).

Numerous experimental investigations of relevant work cylinders and seals with comparable collective loads and operating conditions resulted in the determination of K-factors for various lubricants and their various dwell-time stabilities.

For lubricating oils with viscosities γ = 12 ... 240 mm²/s at T = 323 K ist K = 10 ... 1 whereby the small K factors (approximately up to K = 3) are associated with the higher viscosities (γ > 50 mm²/s) and the large K-factors (approximately K = 8 ... 10) with the lower viscosities (γ < 30 mm²/s).

The grease lubrication with metal-saponified lubricating greases of middle to soft consistency lies around K = 10, fats with organic thickeners can roughly be assigned to the range K = 1.5.

Despite all the reservations and simplifications needed for such an approach, the work quantities and K-factors determined provided information of practical relevance in respect of otherwise imponderable tribological processes and effects.
One actually obtained the relatively small lubricant masses and volumes of a few cm³,

which nowadays are supplied as the “basic or assembly lubrications” and mostly without special depot spaces in the “natural structure” of the cylinders, pistons, piston linear sliders and seals, which is put in as grease and also as edge collars. Furthermore, it must not be forgotten that the similarly lubricating saponification base or the thickener for the lubricating oil contained in the lubricating grease is at the same time a depot carrier. Naturally, alongside the design solution to the depot problem, and depot volume, depot location and depot stability, there is still the associated problem of the minimal lubricant dosing to be solved. Even here the design of conventional dosing processes (e.g. porous materials, wick and felt drainages, and equally with microdosing processes such as with spreading and scraping wipers need to be optimized.
Under extreme operating conditions, e.g. with heavy water condensation, heavy soiling, high rates of flow and high temperatures, corrections are needed for such a simply worked out calculation and, under certain circumstances, these are not so easy to achieve. Nevertheless, they provide a practical and manageable possibility for making tribological predictions and appropriate design forms.

This procedure, which has been illustrated by examples, is recommended by the author generally and for any comparable, lubricated, technical tribopairings under mixed to boundary friction.
One places the tribopairings under “hunger lubrication”, determines the attainable quantity of work z, e.g. es tribologically stable sliding distances, operating hours, work cycles etc. and determines the relative lubricant mass $\bar{m}$ and the corresponding K factors for the lubricants.

5. Bibliography:

1.	Bowden, F.P. Tabor, D.	Reibung und Schmierung fester Körper Springer Verlag Berlin/Göttingen/Heidelberg ,1959
2.	Kragelski, J.V. Dobycin Kombalov	Grundlagen der Berechnung von Reibung und Verschleiß Verlag Technik Berlin, 1982 Masinostroenie Moskau, 1977
3.	Brendel, H.	Wissensspeicher Tribotechnik Fachbuchverlag Leipzig, 1978
4.	Fleischer, G. Gröger, H. Thum, H.	Verschleiß und Zuverlässigkeit Verlag Technik Berlin, 1980
5.	Niemann, G.	Maschinenelemente Springer-Verlag Berlin/Göttingen/Heidelberg, 1981...1983
6.	Vogelpohl, G.	Betriebssichere Gleitlager Springer-Verlag Berlin/Göttingen/Heidelberg, 1967
7.	Fronius, St.	Maschinenelemente-Antriebselemente Verlag Technik Berlin, 1971
8.	Decker, K.H.	Maschinenelemente Carl Hanser Verlag München/Wien, 1989
9.	Winkelmann, S. Hartmuth, H.	Schaltbare Reibkupplungen Konstruktionsbücher 34 Springer-Verlag Berlin/Göttingen/Heidelberg, 1985
10.	Bartz, W.J.	Energieeinsparung durch tribologische Maßnahmen Expert-Verlag, Elmingen, 1988
11.	Drescher, H	Die Mechanik der Reibung zwischen festen Körpern VDI-Zeitschrift 101,1959
12.	Heinicke, G	Tribochemle Akademieverlag Berlin, 1984
13.	Czichos, H. Habig, K.H.	Tribologiehandbuch Reibung und Verschleiß Vieweg-Verlag Braunschweig/Wiesbaden, 1992
14.	Holland, J.	Die Ermittlung der Kenngrößen für zylindrische Gleitlager Zeitschrift Konstruktion 13,1961

15.	Freitag, E.	Tribotechnische Untersuchungen von NBR-Dichtungskautschuk auf ausgewählten, mit Haftschichten versehenen Gleitpartnern der Hochdruckpneumatik Dissertation PH Erfurt/Fakultät Math./Nat., Erfurt, 1987
16.	Deters, L. Hadler, J.	Berechnung von stationär belasteten Radialgleitlagern unter Mischreibungsbedingungen Gif-Fachtagung Tribologie Göttingen, 1995
17.	Schneider, G.	Zur Bestimmung und Beherrschbarkeit Tribotechnischer Kenngrößen bei der Konstruktion Technischer Systeme unter besonderer Berücksichtigung elastomerer Kunststoffe bei Misch- bis Trockenreibung 41. Internationales wissenschaftliches Kolloquium TU Ilmenau,Bd.2, Ilmenau, 09.1996
18.	Lutherdt, M.	Systematisierung und Strukturierung ausgewählter Wirkungsweisen mit festkörpermechanischen Wirkungspartnern, ein Beitrag zur Lehre maschinentechnischer Sachverhalte Dissertation PH Erfurt/Maschinentechnik, Erfurt, 1976
19.	Heidebroek, E.	Richtlinien für den Austausch von Wälziagern gegen Gleitlager KdT Dresden/Berlin A; Lager/Schmierung (5,1949/01) Berlin 1949
20.	Müller, B.	Kampf gegen die Reibungsverluste Zeitschrift „Bild der Wissenschaft“ 6,1996
21.	Schneider, G.	Zum Einfluß der Vibration auf das Reibungsverhalten von Reibpaarungen Dissertation, TU Dresden, Fakultät Maschinenwesen, Dresden, 1968
22.	Schneider, G.	Theoretische Betrachtungen und experimentielle Untersuchungen an Reibpaarungen unter Vibrationseinfluß 8. Internationales Symposium ,>Schmierstofte und Schmierungstechnik", Leipzig, 1967
23.	Jendro, H.	Zum Startreibungsverhalten von Arbeitszylindern der Hochdruckpneumatik Dissertation, PH Erfurt, Fakultät Math.-Nat., Erfurt, 1987

24. Schneider, G, Zum Phänomen und zur Verringerung der Losbrechreibung von NBR-Dichtungen in pneumatischen Arbeitszylindern und Ventilen
9.Internationale Dichtungstagung Dresden 1990, Tagungsband 2 S.85-90.

25. Schneider, G. Zur Erfassung von Schmierstoffbedarf und tribologischer Zuverlässigkeit der Dichtungselemente lebensdauer geschmierter Bauelemente bei ölnebelfreiem Betrieb
Vortrag Internationale Fachtagung 1993 in Fulda / „Die Kautschukindustrie: High Tech in elastomeren Dichtsystemen“
Zeitschrift Kautschuk-Gummi-Kunststoffe, 11,1993

26. Schneider, G. Experimentelle Erfahrungen mit ausgewählten wartungs- und ölnebelfrei betriebenen praxisnahen Bauelementen bei minimalen Schmierstofteinsatz
Internationale Hydraulik und Pneumatik-Tagung, TU Magdeburg, 1981, Magdeburg

27. Schneider, G. Siebert, R. Gestaltung und Anordnung von Schmiermitteldepots für Arbeitszylinder und Wegeventile der Leistungspneumatik
WP F 16N1208 148(22) vom 11.12.1978 und 01.02.1979, Berlin DPA, Nr.140076 A1

6. List of Figures

Bild 1
Reibpaarungsmodell

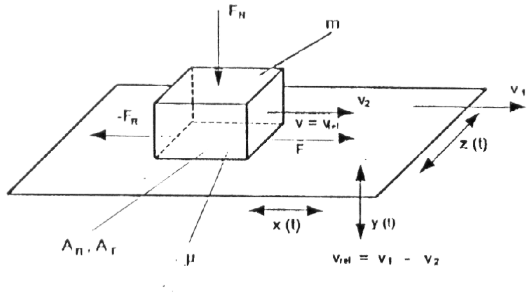


Bild 2 (2.1 ... 2.3)
Reibungskraft / Reibungszahl (Coulomb); μ(v) = konstant

Bild 2.1

μ : Reibungszahl
v : Relativgeschwindigkeit

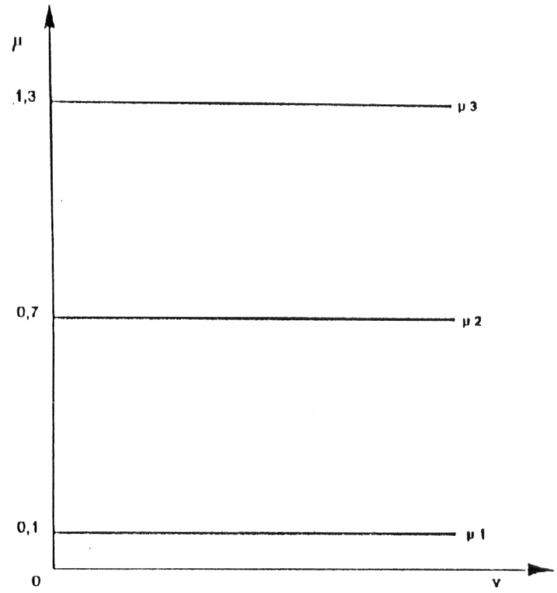


Bild 2 (2.1 ... 2.3)
Reibungskraft / Reibungszahl (Coulomb); $F_R = \mu \cdot F_N$

Bild 2.2 $F_R \sim F_N$

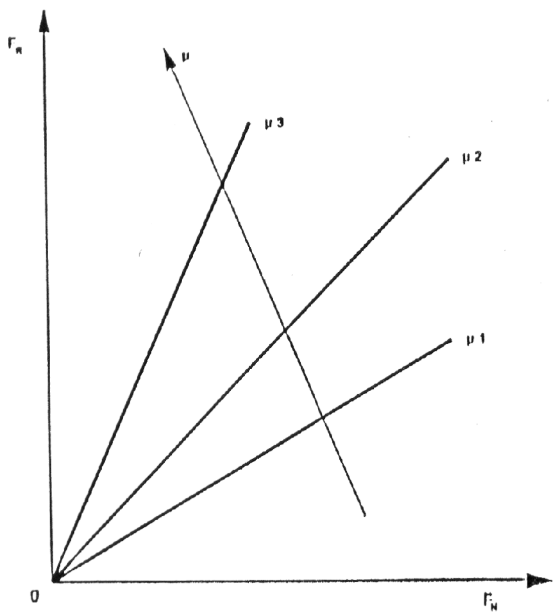


Bild 2 (2.1 ... 2.3)
Reibungskraft / Reibungszahl (Coulomb) $\mu = \frac{F_R}{F_N}$

Bild 2.3 $\mu \sim \frac{1}{F_N}$

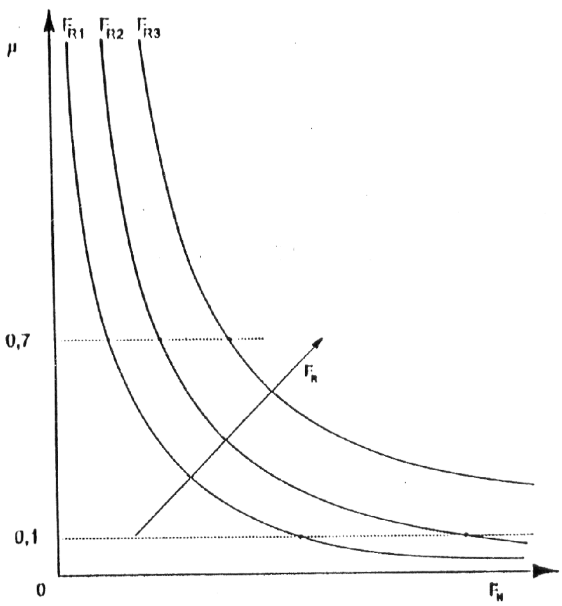
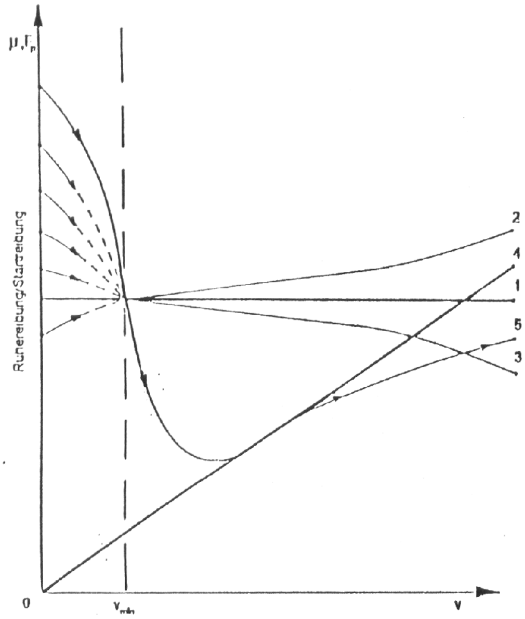
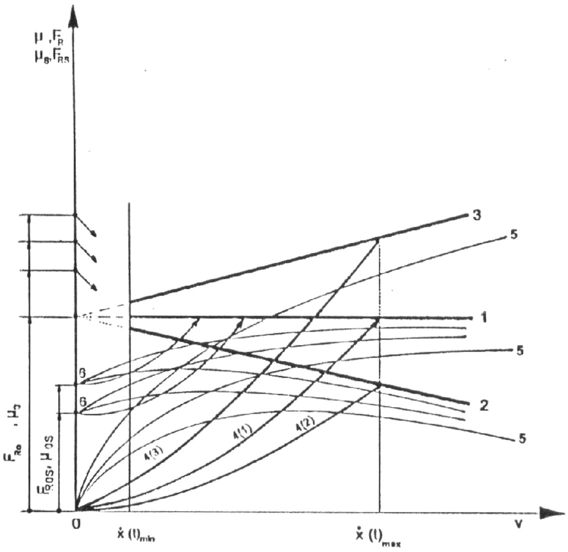


Bild 3
Geschwindigkeitsabhängige Reibung; $\mu(v)$ bzw. $\mu(v_{rel})$
$F_R(v)$ bzw. $F_R(v_{rel})$



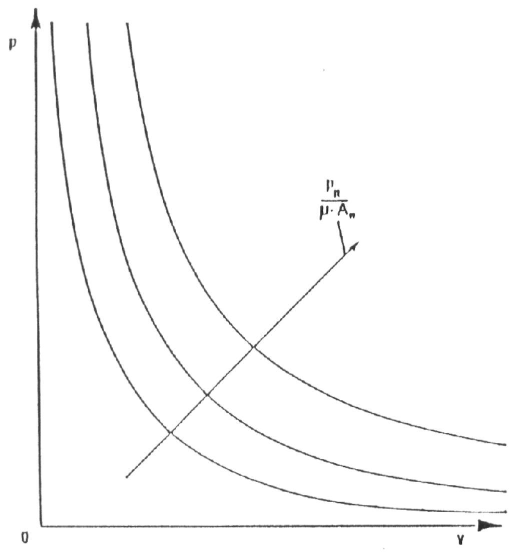
1 Coulombsche Reibung
2 "Steigende" Reibung
3 "Fallende" Reibung
4 Newtonsche Reibung
5 Stribecksche Reibung

Bild 4
Vibrationsreibung (Scheinreibung) $\mu(v)$, $F_R(v)$
$\mu_S(v)$, $F_{RS}(v)$



1 Coulombsche Reibung
2 "Fallende" Reibung
3 "Steigende" Reibung
4 Vibrationsreibung $F_{RS}, \mu_S \longleftrightarrow x(t)$
5 Vibrationsreibung $F_{RS}, \mu_S \longleftrightarrow z(t)$
6 Partielle Vibrationsreibung $x(t), z(t)$

Bild 5
p (v) - Charakteristiken



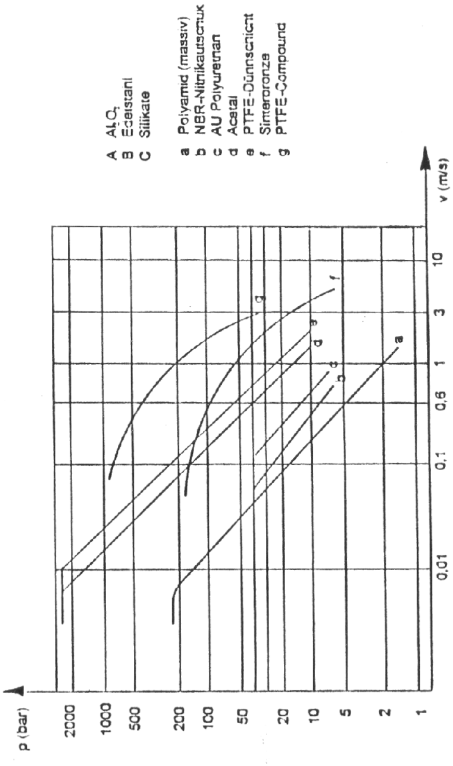


Bild 6
p (v) - Diagramm für Misch- bis Trockenreibung
(Paarungen A ... C mit a ... g)