Lubricated machine components can be divided into those whose elements slide together and those having elements that roll against one another. In the former, such as piston ring/liners and plain bearings, the elements are generally designed to conform closely, so as to spread the applied load over a large fluid-film area and thus reduce the contact pressure. By contrast, in rolling-sliding components such as rolling bearings, involute gears, cam/follower systems and constant velocity joints, the elements must have curved surfaces and be non-conforming so as to be able to roll. These elements have a very small contacting area and in consequence experience very high "Hertzian" pressures, often of the order of 200,000 psi.

The lubrication of conforming lubricated contacts is well described by the hydrodynamic lubrication theory first developed by Reynolds in 1886. However, for many years after this it was not understood how non-conforming contacts could operate successfully, as Reynolds hydrodynamic theory predicts only very thin lubricating films in such high pressure contacts. In the early 1940s Ertel demonstrated that two beneficial effects of the very high contact pressure, the local elastic deformation of the contacting surfaces and an increase in lubricant viscosity with pressure in the contact, coupled with hydrodynamic theory would promote lubricant film formation in high pressure contacts.

This type of hydrodynamic lubrication is now known as elastohydrodynamic lubrication (EHL) and the lubricating films formed in such contacts are called elastohydrodynamic (EHD) films. The primary purpose of any lubricant is to separate moving elements via a full EHD film, wherein the film thickness is generally 1.5 to 4 times the combined surface roughness of the elements.[PULSAR, pg.5] The rolling contact load is thus carried exclusively by the lubricant film sans asperity-to-asperity contact, greatly reducing friction, wear, and surface fatigue.[SKF, pg.105] Inadequate consideration of EHD film formation may lead to the selection of an unsuitable lubricant, leading to premature part failure and reduced performance.
>>> SIDEBAR
The ratio of EHD film thickness to surface roughness is known as the 'lambda ratio'
<<<

EHL theory and EDH film formation factors

[STLE, pg.5 figure 8]

The Hertzian condition of contact is a dominating feature of EHD lubrication. It establishes the overall shape of the contacting surfaces. The hydrodynamic pressure generated in this region has the task of separating the surfaces, which are being forced together by the enormous pressure in the Hertzian region. The hydrodynamic pressure is goverend by the viscous properties of the fluid, under the pressure and temperature conditions within the inlet region. A change in any parameter, which causes a greater hydrodynamic pressure to be generated in the inlet region, will result in an increase in film thickness. [STLE, pg.4,5,6]

A key feature of almost all practical EHD contacts is that the contacting surfaces roll together. In many cases they also slide and experience mixed sliding-rolling. Film thickness is more sensitive to the *entrainment speed*; This is the mean of the rolling speeds of the two surfaces with respect to the contact, the speed with which lubricant is drawn or "entrained" into the contact. The sliding speed, which is the speed of the two surfaces relative to each other, adds to friction and heat generation but has a very low impact on film thickness. [SPIRAL, pg.5]

The influences of load and elastic modulus are also quite low. An increase in load merely increases the maximum Hertzian pressure and increases the size of the

Hertzian region, it has little effect on the shape of the inlet region where the hydrodynamic pressure is generated. It's somewhat ironic that "elastohydrodynamic" lubrication shows very little dependence on the elasticity of the materials. As long as the elastic deformation is similar to the Hertzian deformation, it doesn't matter if the elastic modulus is high as with tungsten carbide or low as with aluminum or glass.[STLE, pg.6]
>>> SIDEBAR
"An increase in the radius of curvature of the contact points would also result in a larger film thickness."[Ellipticity, pg.10]
<<<

In a typical installation the primary method for achieving ideal EHD film formation would be via lubricant selection; Since operational temperatures, loads, and speeds are known constants. An increase in either the dynamic viscosity of the lubricant (at atmospheric pressure), or the composite pressure-viscosity coefficient ([A]), would lead to an increase in film thickness. Both lubricant properties decrease with temperature, thinning the film, though the effect on the [A]-value can be considered neglegible when selecting a lubricant. The Viscosity Index is an arbitrary value used to approximate the relation between lubricant viscosity and temperature. Changes in temperature have a reduced effect on lubricants with a high VI, leading to more consistent performance within operational temperatures. [SPIRAL, pg.7]
>>> SIDEBAR
The [A]-value describes the change in the viscosity of a lubricant at different pressures. The pressure/viscosity behaviour of a lubricant is influenced significantly by the type of base oil, its molecular structure and its additive package. In many cases, precise values are not available for individual lubricants. In practice, however, the [A]-values in this chart can be generally used for rolling bearings.[SCHAEFFLER pg.13. A-VALUE CHART]
<<<

It might initially be inferred that the larger the [A]-value the better (the thicker the EHD film). This does not turn out to be the case for two reasons. One is that there is a broad correlation between [A]-value and viscosity index, wherein base fluids with high [A]-value tend to have low VI and vice versa. In practical terms it turns out to be more important to have a high VI rather than have a high [A]-value. Secondly, there is a tendency of fluids with high [A]-value to give high EHD friction and vice versa.[SPIRAL, pg.8] As such, selection of a lubricant is primarily derived from its *kinematic* viscosity, the lubricants dynamic viscosity relative to lubricant density.
>>> SIDEBAR
One of the most important features of EHL is that, unlike conventional hydrodynamic lubrication, there is no direct relationship between the EHD film thickness and the EHD friction. We can design for EHD film thickness and EHD friction independently, for example obtaining a combination of high film thickness and low friction.[SPIRAL, pg.6]
<<<

Lubricant selection for bearings and linear rails

Warning

The physical theory takes account only of the lubrication regime in the rolling contact. It does not cover the lubrication conditions at the other contact surfaces, for example between the rolling element and the cage pocket.

Furthermore, it does not take account of the fact that the profile geometry of the surfaces has an influence on the lubrication regime. It is therefore not sufficient to simply compare the theoretical EHD film thickness with the roughness of the surfaces.

In the selection of lubricant it is therefore necessary to take account not only of EHL theory, but also practical experience and the complete lubrication regime in the bearing as well as any possible additive reactions.[SCHAEFFLER, pg.10]

In day-to-day practice it is too cumbersome to derive the target kinematic viscosity through calculations of the EHD film thickness. Instead, the suitability of a lubricant can be estimated by using the viscosity ratio [K] as an indicator of lubricant film formation.

[K] = [V]/[V1] * (p/0.89[g/cm^3])^0.83

[V] = Kinematic viscosity of the lubricant at the operating temperature
[p] = Density of the lubricant at the operating temperature
[V1] = Reference/rated viscosity, a function of the bearing size and speed.

The reference viscosity can be determined via charts such as [SKF, pg.103] or with the formulae below:
>>> SIDEBAR
It should be noted that the curves typically presented in bearing guidebooks are only valid for a lubricant density of [p] = 0,89 g/cm3 at a temperature of +20 Â°C. For lubricants of a different density the adjusted formula should be used.

Density of oil is fairly linear relative to temperature. [SCHAEFFLER,pg.16]
<<<

[V1] = 45000 * n^(-0.83) * [dM]^-0.5   for n<1000
[V1] = 4500 * n^(-0.5) * [dM]^-0.5   for n>=1000

[n] = Operating speed
[dM] = Mean bearing diameter (OD + ID)/2
[SCHAEFFLER, pg.25]
>>>SIDEBAR
Also known as 'Pitch line/circle diameter' [KAYDON]
<<<
Experience shows that, at values of [K]>=2, a lubricant film fully capable of supporting load can be anticipated. At values of [K]>=4 a full EHD film is achieved, with all the aforementioned benefits to bearing life and performance. Values of [K] below 1 drastically reduce bearing lifespan, introducing considerable friction and wear. [SCHAEFFLER, pg.24]

However, if the lubricant is doped with suitable anti-wear (or extreme pressure) additives, separation in the contact area may also be achieved by the reaction layers formed by the additives. Through this chemical lubrication it is also possible to achieve low-wear operation even in conditions of high load(C/P>10), mixed axial/radial loads, line contact, and mixed friction([k]<1). The suitability of anti-wear additives varies and is heavily dependent on temperature. Their effectiveness can only be assessed by means of a test rig for the rolling bearing. [SCHAEFFLER, pg.25,pg.83,pg.160]

It is important to note that the operating temperature is the temperature reached by the bearing after the startup period, once an equilibrium has been achieved between heat generation and heat dissipation.It is heavily dependent on the lubrication regime as well as the temperature difference between the bearing, adjacent parts and environment.[SCHAEFFLER, pg.52] In the case of heavily loaded bearings with a high proportion of sliding motion, the temperature in the contact area of the rolling elements may be up to 20 C higher than the temperature measured on the stationary ring (without any influence from external heat). [SCHAEFFLER, pg.25]

>>> PERSONAL NOTE:
Unfortunately there is limited data on running tiny bearings at low RPM in high ambient temperatures, so the true operating temperature is not known.

I'd guesstimate around a 10C markup over the outer ring, after a several hour 'run-in' period.
<<<

Linear rails

N/A

>>> PERSONAL NOTE:
It's 4 contact points instead of two, no formulas exist atm for this too complicated.
The surface speeds would generally be higher than expected.

I would go for GPL-225 below 80C and GPL-226 80C-120C for linear rails.

GPL-226 for <80C 625/623 bearings and XHT-AC for 80C-120C bearings.
<<<

Grease

Grease lubrication is used in approximately 90% of all rolling bearing arrangements. The advantages of grease lubrication include:
â– very little design work required
â– sealing action supported by the grease
â– long operating life with maintenance-free lubrication, eliminating the requirement for lubrication devices
â– suitability for speed parameters n<=dM 2.6*10^6 (minâ€“1 * mm)
â– longer emergency running phase in case of lubrication supply failure
â– low frictional torque
[SCHAEFFLER, pg.56]

>>> SIDEBAR
Oil lubrication presents itself as a sensible option if adjacent machine elements are already supplied with oil or if heat is to be dissipated by the lubricant. Grease is a poor heat dissipator. [SCHAEFFLER,pg.56]
<<<

The characteristics of a grease are fundamentally dependent on:
â– the base oil type
â– the base oil (kinematic) viscosity and density
â– the thickener (which is relevant to shear strength)
â– the additive package.

A distinction should be made between additives that have an effect on the oil itself (oxidation inhibitors, viscosity index improvers, detergents, dispersants) and additives that have an effect on the bearing or the metal surface (anti-wear additives, corrosion inhibitors, friction value modifiers).[SCHAEFFLER pg.67]

>>> PERSONAL NOTE
Unable to find clarification on 'relevant to shear strength'.
Maybe referencing the 'strong shearing' that grease undergoes during homogenisation?
[SCHAEFFLER, pg.204]
<<<
>>> FUTURE TODO:
Separate section for modifiers? [PULSAR, pg.8] has a good section.
Even mention some sort of modifiers that intentionally cause plastic deformation to improve surfaces permanently.
>>>

Greases are produced in various consistencies, defined as NLGI grades, ranging from #000 (nearly oil) to #6 (solid wax).If the selected NGLI rating is too high it will not pump through automatic lubrication equipment. If too low, it may run out of the bearings, particularly in vertical or inclined arrangements. The consistency rating is not representative of the lubrication ability of the grease, that is decided by the base oil.[PULSAR, pg.13][SCHAEFFLER, pg.66]

In grease lubrication, the amount of lubricant playing an active role in the lubrication process is very small. Grease is largely displaced from the rolling contact and is deposited laterally or exits the bearing arrangement through the seals. The grease remaining on the raceway surfaces, and laterally in or on the bearing, continuously releases a small quantity of oil. The effective lubricant quantity between the rolling contact surfaces is sufficient for lubrication under moderate load over an extended period.

The release of oil is dependent on:
â– the thickener (type, quantity and consistency)
â– the additives
â– the type of base oil
â– the viscosity of the base oil
â– the size of the surface releasing oil
â– the operating temperature
â– the mechanical strain on the grease.
[SCHAEFFLER, pg.18]

Typical oil bleed rates of greases for bearing lubrication are 1% to 5%, enough for a prolonged grease life while also fulfilling lubrication requirements. [PLANT_MAG]

>>> SIDEBAR
At low [K]-values in particular, the thickener and additives contribute to effective lubrication. The thickener contained in greases has an influence on formation of the lubricant film and on protection against wear. This effect has been demonstrated in practice but cannot yet be defined in theoretical terms. In order to allow an estimate of a comparable lubrication regime, current practice is based on calculation using only the base oil data.

The effect of the grease thickener becomes clear if the film thickness is measured as a function of the running time. At the start of bearing running, the film thickness formed in the contact area as a function of the bearing type is significantly greater than the theoretically possible value of the base oil.
[SCHAEFFLER, pg.17]
<<<

While the operating temperature limits provided by lubricant manufacturers will be based on grease chemical properties, this does not mean that the grease will properly lubricate bearings within those same temperature ranges. One must also consider a base oil viscosity that supplies a satisfactory [K]-value for EHD film lubrication. Additionally, the grease should not be used for an extended period at its upper operating temperatures. At high temperatures approaching/above its operating limit, the grease will undergo excessive oil bleed and evaporation, leading to a shortened grease life and reduced performance.[SKF pg.119,pg.120]

Regardless of which lubricant you choose it will naturally lose its lubricating properties over time. A lubricating regime based on temperature, contamination factors, and predicted bearing lifespan is only an approximation. Routine lubrication frequently leads to under/over greasing, so an ultrasound assisted approach is preferred. Monitoring the bearings friction directly with an SDT device provides accurate feedback on the lubrication condition, during the initial greasing as well as routine maintenance post installation. [CBMCONNECT]

Before initial greasing, compatibility of the part preservation with the lubricant must be observed. If the bearings contain resinous oil or grease residues, precleaning by mechanical means followed by longer softening with an aqueous, strongly alkaline cleaning agent is recommended. The same applies when repacking, since in most cases lubricants are incompatible, with the resulting reaction ranging from thickening to complete lubrication failure. [SCHALEFFER pg.150, pg.78]

Deep groove ball bearings, sealed on both sides, are typically filled with a lithium soap grease of consistency grade #2 or #3 - wherein the softer grease is used for small bearings. The quantity normally introduced fills approx. 90% of the undisturbed free bearing volume. The grease is then distributed during a short running-in phase and settles to a large extent in the undisturbed part of the free bearing cavity, in other words on the inner sides of the washers. Once the running-in phase is complete, the friction is only 30% to 50% of the friction when freshly greased.
>>> SIDEBAR

In the freshly greased state, the bearing factor on friction torque can be two to five times higher. [SCHALEFFER, pg.40]

A run-in is needed to prevent dangerous temperature rise in a freshly greased bearing, but may be unecessary for low RPM bearings. [KLUBER, pg.5]
<<<
>>> SIDEBAR

Care must be taken not to overfill, this will lead to higher friction and continuous grease loss until the normal degree of filling is restored. If the egress of grease is hindered, a considerable increase in torque and temperature must be anticipated. [SCHALEFFER, pg.96]

Bearings rotating at low speeds (n*dM < 50,000 min-1*mm) and their housings must be filled completely with grease. The churning friction occurring in this case is negligible.[SCHALEFFER, pg.95]
<<<

>>>> FUTURE TODO:

Shock load, 'squeeze term' influence. Oscillation, mechanical strength of grease needed for this to work, birnelli damage, etc.

Fairly complicated and possibly out of scope for this article.

Minimum contact load and C/P ratio?

Solid film lubricants :)
<<<<

Sauces:

Ellipticity
https://www.researchgate.net/publication/
311220228_On_the_Crucial_Role_of_Ellipticity_on_EHD_Film_Thickness_and_Friction/
link/59e5ed050f7e9b0e1ab25a3e/download

STLE
https://www.stle.org/images/pdf/STLE_ORG/BOK/LS/Friction/What%20is%20EHD_tlt
%20article_Nov12.pdf

SPIRAL
https://spiral.imperial.ac.uk/bitstream/10044/1/21184/2/Basics%20of%20EHL%20for
%20practical%20application.pdf

SKF
https://www.skf.com/binaries/pub12/Images/0901d196802809de-Rolling-bearings---
17000_1-EN_tcm_12-121486.pdf

SCHAEFFLER
https://www.schaeffler.com/remotemedien/media/_shared_media/08_media_library/
01_publications/schaeffler_2/tpi/downloads_8/tpi_176_de_en.pdf

PULSAR
https://pulsarlube.com/html/_skin/seil/files/Lubrication%20Training%20%20Manual.pdf

KLUEBER
https://www.klueber.com/ecomaXL/get_blob.php?name=klu5896-1BearLubAllWP-A12b.pdf

CBMCONNECT
https://www.cbmconnect.com/ultrasound-assisted-lubrication-three-ways-to-improve-
lubrication-practices/

PLANT_MAG
https://www.efficientplantmag.com/2009/07/the-lowdown-on-lubricants-for-rolling-
bearings/

KAYDON
https://www.kaydonbearings.com/downloads/whitepapers/Kaydon-LowSpeedAppsWP.pdf