# AN INVESTIGATION ON THE DEPENDENCE OF THE LUBRICATING OIL'S PERFORMANCE ON THE AMBIENT TEMPERATURE IN JOURNAL BEARINGS

A thesis
by

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First, we started in the name of almighty Allah. This thesis is accomplished under the supervision of Khandoker Mohammad Faisal Karim, Lecturer, Department of Mechanical, Sonargaon University. It is a great pleasure to acknowledge our profound gratitude and respect to our supervisor for this consistent guidance, encouragement, helpful suggestion, constructive criticism and endless patience through the progress of this work. The successful completion of this thesis would not have been possible without his persistent motivation and continuous guidance. The authors are also grateful to Md. Mostofa Hossain, Head of the Department of Mechanical Engineering and all respected teachers of the Mechanical Engineering Department for their cooperation and significant help for completing the thesis work successfully.

# ABSTRACT

Maximum temperature rise is one of the design criteria for a journal bearing. It implies that the difference between the maximum allowable journal bearing temperature and the ambient temperature is constant. On the other hand, minimum lubricating oil film thickness is another design criteria for a journal bearing. Therefore we can establish a correlation between the minimum film thickness and the ambient temperature for a given lubricating oil in a journal bearing. In this thesis, we took the minimum oil film thickness recommended by Trumpler (known as Trumpler's criteria) as the performance parameter for the journal bearing. Then we evaluated the performance of different lubricating oils used in journal bearings with respect to journal bearing diameter, length to diameter ratio, clearance to diameter ratio, rotational speed and load. We conducted this evaluation for three different kinds of journal bearings: natural fed, pressure fed and self contained journal bearings. Through this study, we investigated whether lubricating oil recommended by the machinery manufacturer after being tested in the machinery manufacturer's ambient temperature delivers acceptable performance in the user's ambient temperature.

# CHAPTER 1

# INTRODUCTION

## 1.1 Overview

Maximum temperature rise is one of the design criteria for a journal bearing. It implies that the difference between the maximum allowable journal bearing temperature and the ambient temperature is constant. On the other hand, minimum lubricating oil film thickness is another design criteria for a journal bearing. Therefore we can establish a correlation between the minimum film thickness and the ambient temperature for a given lubricating oil in a journal bearing.

In this thesis, we took the minimum oil film thickness recommended by Trumpler (known as Trumpler's criteria) as the performance parameter for the journal bearing. Then we evaluated the performance of different lubricating oils used in journal bearings with respect to journal bearing diameter, length to diameter ratio, clearance to diameter ratio, rotational speed and load. We conducted this evaluation for three different kinds of journal bearings: natural fed, pressure fed and self contained journal bearings. Both SAE grade oils and multiviscosity oils were considered for the study. Through this study, we investigated whether lubricating oil recommended by the machinery manufacturer after being tested in the machinery manufacturer's ambient temperature delivers acceptable performance in the user's ambient temperature.

## 1.2 Objectives

The objectives of the thesis are the followings:

- To study the correlation between the minimum film thickness and the ambient temperature for pressure fed, natural fed and self contained journal bearings under varying parameters (journal diameter, length to diameter ratio, clearance to diameter ratio, rotational speed and load)
- To investigate whether it is necessary to test the performance of the lubricating oils in the user's local ambient temperature

## 1.3 Thick-Film Lubrication

Let us examine the formation of a lubricant film in a journal bearing. Figure 1.1a shows a journal that is just beginning to rotate in a clockwise direction. Under starting conditions, the bearing will be dry, or at least partly dry, and hence the journal will climb or roll up the right side of the bearing as shown in Fig. 1.1a. Now suppose a lubricant is introduced into the top of the bearing as shown in Fig. 1.1b. The action of the rotating journal is to pump the lubricant around the bearing in a clockwise direction. The lubricant is pumped into a wedge-shaped space and forces the journal over to the other side. A minimum film thickness $h_0$ occurs, not at the bottom of the journal, but displaced clockwise from the bottom as in Fig. 1.1b. This is explained by the fact that a film pressure in the converging half of the film reaches a maximum somewhere to the left of the bearing center. Figure 1.1 shows how to decide whether the journal, under hydrodynamic lubrication, is eccentrically located on the right or on the left side of the bearing.Visualize the journal beginning to rotate. Find the side of the bearing upon which the journal tends to roll.Then, if the lubrication is hydrodynamic, mentally place the journal on the opposite side.The nomenclature of a journal bearing is shown in Fig. 1.2. The dimension c is the radial clearance and is the difference in the radii of the bushing and journal.

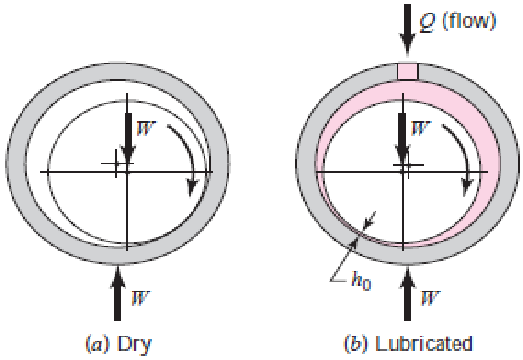


Fig 1.1: Formation of a film

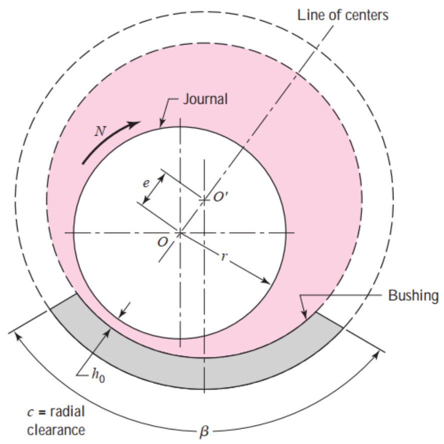


Fig 1.2: Nomenclature of a partial journal bearing

## 1.4 Hydrodynamic Theory

The present theory of hydrodynamic lubrication originated in the laboratory of Beauchamp Tower in the early 1880s in England. Tower had been employed to study the friction in railroad journal bearings and learn the best methods of lubricating them.It was an accident or error, during the course of this investigation, that prompted Tower to look at the problem in more detail and that resulted in a discovery that eventually led to the development of the theory.

Figure 1.3 is a schematic drawing of the journal bearing that Tower investigated.It is a partial bearing, having a diameter of 4 in, a length of 6 in, and a bearing arc of 157◦, and having bath-type lubrication, as shown. The coefficients of friction obtained by Tower in his investigations on this bearing were quite low, which is now not surprising. After testing this bearing, Tower later drilled a 12 -in-diameter lubricator hole through the top. But when the apparatus was set in motion, oil flowed out of this hole.

In an effort to prevent this, a cork stopper was used, but this popped out, and so it was necessary to drive a wooden plug into the hole. When the wooden plug was pushed out too, Tower, at this point,

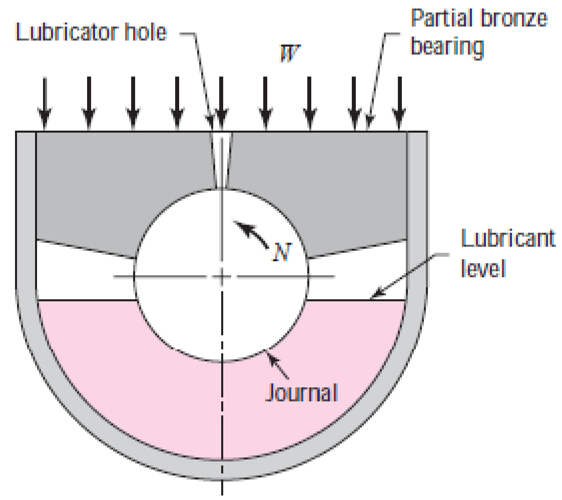


Fig 1.3: Schematic representation of the partial bearing used by Tower

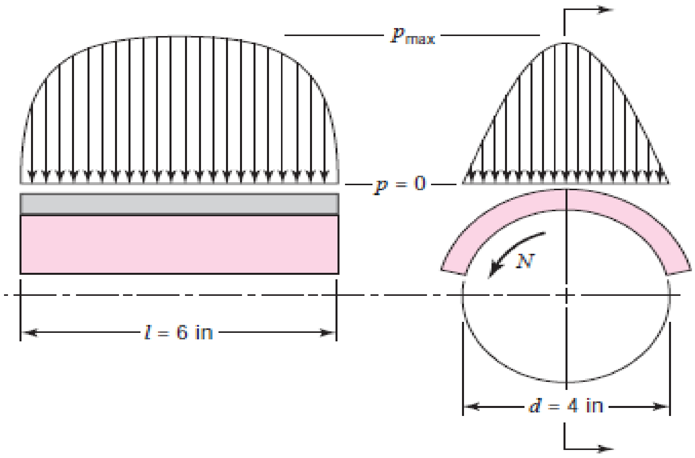


Fig 1.4: Approximate pressure distribution curves obtained by Tower

undoubtedly realized that he was on the verge of discovery. A pressure gauge connected to the hole indicated a pressure of more than twice the unit bearing load. Finally, he investigated the bearing film pressures in detail throughout the bearing width and length and reported a distribution similar to that of Fig. 1.4.

## 1.5 Trumpler's Design Criteria for Journal Bearings

Because the bearing assembly creates the lubricant pressure to carry a load, it reacts to loading by changing its eccentricity, which reduces the minimum film thickness h0 until
the load is carried. What is the limit of smallness of h0? Close examination reveals that the moving adjacent surfaces of the journal and bushing are not smooth but consist of a series of asperities that pass one another, separated by a lubricant film. In starting abearing under load from rest there is metal-to-metal contact and surface asperities are broken off, free to move and circulate with the oil. Unless a filter is provided, this debris
accumulates. Such particles have to be free to tumble at the section containing the minimum film thickness without snagging in a togglelike configuration, creating additional damage and debris. Trumpler, an accomplished bearing designer, provides a throat of at least 200 μ in to pass particles from ground surfaces.7 He also provides for the influence of size (tolerances tend to increase with size) by stipulating

$$h_0 \geq 0.0002 + 0.00004d \text{ in}$$

where d is the journal diameter in inches. A lubricant is a mixture of hydrocarbons that reacts to increasing temperature by vaporizing the lighter components, leaving behind the heavier. This process (bearings have lots of time) slowly increases the viscosity of the remaining lubricant, which increases heat generation rate and elevates lubricant temperatures. This sets the stage for future failure. For light oils, Trumpler limits the maximum film temperature Tmax to

$$T_{max} \leq 250^{o}F$$

Some oils can operate at slightly higher temperatures. Always check with the lubricant
Manufacturer. A journal bearing often consists of a ground steel journal working against a softer, usually nonferrous, bushing. In starting under load there is metal-to-metal contact, abrasion, and the

generation of wear particles, which, over time, can change the geometry of the bushing. The starting load divided by the projected area is limited to

$$\frac{Wst}{lD} \leq 300\ psi$$

If the load on a journal bearing is suddenly increased, the increase in film temperature in the annulus is immediate. Since ground vibration due to passing trucks, trains, and earth tremors is often present, Trumpler used a design factor of 2 or more on the running load, but not on the starting load.

$$n_d \geq 2$$

Many of Trumpler's designs are operating today, long after his consulting career is over; clearly they constitute good advice to the beginning designer.

## 1.6 Different Types of Journal Bearings

We considered three types of journal bearing for this study. They are:

- Pressure fed journal bearings
- Natural fed journal bearings
- Self contained journal bearings

### 1.6.1 Pressure Fed Journal Bearings

An external pump forces lubricant into the bearing under pressure through oil holes or grooves machined in the bearing shell. This ensures a continuous, abundant supply of oil regardless of shaft speed. These are reliable for film formation, handle high loads and speeds, and also carry away frictional heat. Requires a pump, reservoir, and piping — adds complexity and cost. Examples: Internal combustion engine crankshaft and camshaft bearings, large steam/gas turbines, diesel engine connecting rod bearings, industrial centrifugal pumps, aircraft engines.

### 1.6.2 Natural Fed Journal Bearings

The lubricant is supplied by gravity from a reservoir (oil cup or ring) sitting above the bearing, with no external pressure. The shaft's rotation draws oil into the clearance space through hydrodynamic wedge action. These are simple, low-cost, suitable for low to moderate speeds and loads. They require periodic oil top-up. Examples: Pedestal bearings in line shafts, ceiling fan motor bearings, small electric motor bearings, agricultural machinery, textile loom spindles.

### 1.6.3 Self Contained Journal Bearings

The bearings contain their own lubricant supply permanently sealed within them — either as grease, oil-impregnated porous metal (sintered bronze/iron), or an oil-soaked wick/pad. No external lubrication system is needed during operation. They are maintenance-free ("fit and forget"), compact and clean, ideal for inaccessible or sealed locations. They can withstand limited to moderate loads and speeds; lubricant is finite. Examples: Porous (sintered) metal bearings used in small electric motors, washing machine drums, computer fans, printer rollers; grease-packed sealed bearings used in wheel hubs, conveyor rollers, domestic appliances; wick/pad oiled bearings used in clock mechanisms, small pumps, sewing machines.

## 1.7 SAE Oils and Multiviscosity Oils

We conducted this study for both SAE oils and multiviscosity oils.

### 1.7.1 SAE Oils

The Society of Automotive Engineers (SAE) developed the SAE J300 standard to classify engine and gear oils by their viscosity — a measure of a fluid's resistance to flow. Single-grade SAE oils maintain a consistent viscosity only within a narrow temperature range, making them suitable for controlled or seasonal operating conditions. Lower SAE numbers (SAE 5, 10, 20) indicate thinner, less viscous oils that flow freely at low temperatures. Higher numbers (SAE 30, 40, 50) indicate thicker oils better suited to high operating temperatures. The "W" suffix in winter grades (SAE 5W, 10W, 20W) denotes that the oil meets cold-cranking viscosity requirements measured at sub-zero temperatures. The limitation is that a single-grade oil optimised for summer use becomes dangerously thick in winter (causing poor cold starts and bearing starvation), while a winter-grade oil thins out too much in hot conditions, reducing film strength. SAE 30 is widely used in small petrol engines, lawn mowers, and older diesel engines operating in moderate climates. SAE 40 suits heavy-duty industrial diesel engines in hot tropical conditions. SAE 10W is used in diesel engines during cold winters.

### 1.7.2 Multiviscosity Oils

Multigrade oils overcome the fundamental limitation of single-grade oils by using a base oil blended with long-chain viscosity index improver (VII) polymers. These polymers coil up at low temperatures (having little effect on viscosity) and expand at high temperatures (thickening the oil and preventing excessive thinning). The result is one oil that satisfies two viscosity grade requirements simultaneously.
How the grades are read: A designation like 10W-40 means the oil behaves like an SAE 10W oil at cold temperatures (passes cold-cranking test at −25°C) and like an SAE 40 oil at 100°C operating temperature. The "W" always refers to the cold-temperature rating. They are easier for cold starting and faster oil circulation at startup (reducing wear), stable protective film at high temperatures, year-round use without seasonal oil changes, and better fuel economy due to reduced internal friction.

## 1.8 Important Non-dimensional Parameters

### 1.8.1 Sommerfeld Number (S)

The Sommerfeld number is a dimensionless number that captures all the key operating conditions of a journal bearing into a single value. It is defined as: $S = (r/c)^2 \times (\mu N/P)$ where r is the journal radius, c is the radial clearance, μ is the dynamic viscosity of the lubricant, N is the shaft rotational speed (rev/s), and P is the bearing pressure (load per projected area). A high Sommerfeld number ($S >> 1$) indicates lightly loaded, fast-running conditions i.e. the journal rides centrally on a thick oil film. A low Sommerfeld number ($S << 1$) indicates heavy load or slow speed i.e. the journal is forced close to the bearing wall with a dangerously thin film. It is sometimes called the "bearing characteristic number" and is the master variable from which all other bearing performance parameters are determined.

### 1.8.2 Minimum Film Thickness Variable ($h_0/c$)

This dimensionless ratio relates the minimum oil film thickness $h_0$ to the radial clearance c of the bearing. It is defined as: $h_0/c = 1 - \varepsilon$ (since $h_0 = c - e$, and $\varepsilon = e/c$, where e is the eccentricity) .This variable directly determines whether the bearing is safe from metal-to-metal contact. A value of $h_0/c = 1$ means the journal is perfectly centred (no load). As load increases, $h_0/c$ decreases toward zero. At $h_0/c = 0$, the journal touches the bearing wall and the lubricant film has collapsed completely. In practice, a minimum permissible value of $h_0$ is specified based on surface roughness ($h_0$ must be greater than the combined surface asperity heights of the journal and bearing). If $h_0$ falls below this threshold, mixed or boundary lubrication occurs, causing rapid wear.

### 1.8.3 Eccentricity Ratio (ε)

The eccentricity ratio describes the positional displacement of the journal centre relative to the bearing centre, normalised by the radial clearance: $\varepsilon = e / c$ where e is the eccentricity (actual distance between the two centres) and c is the radial clearance. It ranges from 0 to 1. When $\varepsilon = 0$, the journal is perfectly centred. This occurs under no load or at very high speed (high Sommerfeld number). When $\varepsilon = 1$, the journal has shifted all the way to touch the bearing surface. This is the limit of film breakdown. Under normal operating conditions, ε typically lies between 0.4 and 0.8. A higher eccentricity ratio means a thinner minimum film, greater risk of metal contact, but also a stronger pressure wedge (higher load capacity). The eccentricity ratio and the minimum film thickness variable are directly complementary i.e. $h_0/c = 1 - \varepsilon$.

# CHAPTER 2

# LITERATURE REVIEW

## 2.1 Introduction

Journal bearings are fundamental mechanical components widely employed in rotating machinery, including turbines, compressors, internal combustion engines, and industrial gearboxes, owing to their ability to support radial loads through the hydrodynamic pressure generated within a thin film of lubricant. The operational principle relies on a sustained separation of the rotating journal from the stationary bearing surface, a condition governed by the complex interplay of rotational speed, applied load, bearing geometry, and—critically—the rheological properties of the lubricating oil [1]. Among all parameters that influence lubricant behaviour, temperature stands as arguably the most consequential, because viscosity, the primary determinant of film formation, exhibits a strong and non-linear dependence on thermal conditions [2].

The ambient temperature at which a bearing system operates determines the initial thermal state of the lubricant before any viscous dissipation occurs, thereby setting the baseline from which the steady-state operating temperature—and hence the effective viscosity—is established. In cold climates or during start-up transients, elevated viscosities may lead to excessive friction torque and parasitic power losses, whereas in high-temperature environments, dangerously thin lubricant films increase the risk of mixed or boundary lubrication, accelerated wear, and eventual bearing failure [3]. Understanding and quantifying this dependence is therefore of significant theoretical and practical importance.

The present chapter reviews the body of literature relevant to the thermo-viscous behaviour of lubricating oils, the thermohydrodynamic (THD) analysis of journal bearings, the experimental characterisation of temperature effects on bearing performance, and the numerical methods that have been developed to model these coupled phenomena. The review is structured to provide a logical progression from fundamental viscosity-temperature relationships through to recent advances in predictive modelling and advanced lubricant formulations.

## 2.2 Viscosity–Temperature Relationships of Lubricating Oils

The dependence of liquid viscosity on temperature has been the subject of systematic study since the nineteenth century. Reynolds [4] was among the first to propose an exponential relationship of the form $\eta = \eta_0 \exp(-\beta T)$, where $\eta_0$ is the viscosity at a reference temperature, T is the absolute temperature, and β is a temperature–viscosity coefficient. While adequate for limited temperature ranges, this model fails to capture the strongly curved viscosity–temperature response observed in mineral oils over wide operating windows.

Walther [5] developed an empirical double-logarithmic relationship that underpins the ASTM D341 viscosity–temperature chart, which remains the most widely adopted practical tool for estimating kinematic viscosity at arbitrary temperatures given two reference measurements. The Walther equation takes the form $\log \log(\nu + 0.7) = A - B \log T$, where ν is the kinematic viscosity in centistokes and A, B are oil-specific constants determined by regression. This formulation has proven robust for petroleum-based base stocks across the range of 0 °C to 150 °C encountered in most industrial bearing applications [6].

A theoretically grounded alternative is the Vogel–Fulcher–Tammann (VFT) equation, $\eta = K \exp[B/(T - T_0)]$, which introduces a third parameter $T_0$ interpreted as a limiting temperature below which viscous flow ceases [7]. Yasutomi et al. [8] demonstrated the superiority of the VFT equation over Reynolds' exponential model for both mineral and synthetic lubricants when pressure effects are simultaneously considered, a finding of particular relevance to elastohydrodynamic contacts. The pressure–viscosity coefficient α, which appears in the Barus equation, also exhibits temperature dependence, generally decreasing with increasing temperature, a coupling that must be accounted for in high-pressure lubrication analyses [9].

The viscosity index (VI), standardised by ASTM D2270, provides a dimensionless measure of the rate of viscosity change between 40 °C and 100 °C, with higher values indicating less sensitivity to temperature [10]. Conventional Group I mineral base oils typically exhibit VI values in the range 80–105, whereas hydrocracked Group II and Group III oils reach 100–130. Synthetic polyalphaolefin (PAO) base stocks achieve VI values of 120–160, and ester-based fluids can exceed 170, as reviewed by Mang and Dresel [11]. The practical implication for journal bearings is that higher-VI lubricants maintain more stable film thicknesses across the operating temperature range, reducing sensitivity to ambient temperature fluctuations.

Krupka and Hartl [12] measured viscosity–temperature profiles of several fully formulated engine oils using a high-pressure viscometer and found that the presence of viscosity index improvers (VII)—typically polymethacrylate or olefin copolymer additives—produced a non-Newtonian shear response at elevated temperatures that conventional Walther-type models cannot represent. This finding has motivated the application of Cross and Carreau–Yasuda constitutive models in thermohydrodynamic bearing analyses where multi-grade oils are employed [13].

## 2.3 Fundamental Theory of Hydrodynamic Lubrication in Journal Bearings

The theoretical framework for journal bearing analysis rests upon Reynolds' classical lubrication equation, first derived in 1886 from the Navier–Stokes equations under the assumptions of laminar, isoviscous, incompressible flow in a thin-film geometry [4]. In its two-dimensional form for a finite-width bearing, the Reynolds equation is written as:

$$\partial/\partial x \left[ (h^3/\eta)\, \partial p/\partial x \right] + \partial/\partial z \left[ (h^3/\eta)\, \partial p/\partial z \right] = 6U\, \partial h/\partial x + 12\, \partial h/\partial t$$

where p is the hydrodynamic pressure, h is the local film thickness, η is the dynamic viscosity, U is the surface velocity, x is the circumferential coordinate, and z is the axial coordinate [14]. Raimondi and Boyd [15] produced the first comprehensive solution of this equation for finite bearings using a finite-difference scheme, generating dimensionless design charts that remained the primary engineering tool for decades. Their results confirmed that load capacity, friction, and flow rate are all strongly dependent on the Sommerfeld number $S = (\mu NLD/W)(R/C)^2$, in which the dynamic viscosity μ appears explicitly, establishing the direct link between temperature—through viscosity—and all performance metrics.

The isoviscous assumption of the classical Reynolds equation is a significant simplification: in real bearings, viscous shearing generates heat, elevating the lubricant temperature and reducing its viscosity, which in turn alters the pressure distribution. Dowson [16] extended the theoretical framework by coupling the Reynolds equation with the energy equation to produce the first thermohydrodynamic (THD) formulation, introducing the concept of an effective viscosity computed from the temperature field across the film. This approach has since become the standard basis for advanced bearing analysis codes.

Khonsari and Beaman [17] systematically evaluated the error introduced by the isoviscous assumption across a range of operating conditions and demonstrated that, for lightly loaded bearings at moderate temperatures, isoviscous predictions are acceptable within engineering tolerances. However, at high specific loads ($p/pmax > 0.6$) and elevated ambient temperatures, the discrepancy in predicted minimum film thickness between isothermal and fully THD solutions can exceed 30%, necessitating the coupled thermal analysis. This conclusion has been confirmed by subsequent experimental studies reviewed in Section 2.5.

## 2.4 Thermohydrodynamic (THD) Analysis and Thermal Effects

Thermohydrodynamic analysis couples the Reynolds lubrication equation, the energy equation governing heat generation and transport within the lubricant film, and the heat conduction equations in the journal and bearing sleeve. The pioneering work of Cope [18] and subsequently McCallion et al. [19] established the adiabatic THD model, in which heat generated by viscous shearing is assumed to be convected axially out of the bearing with no conduction into the solid surfaces. While computationally convenient, the adiabatic assumption overestimates film temperatures by up to 40% compared with experimental measurements, as demonstrated by Mitsui et al. [20].

The fully coupled TEHD (thermoelastohydrodynamic) approach, which additionally accounts for thermally induced distortion of the journal and bearing surfaces, was developed by Boncompain et al. [21]. Their finite-element solution for a 360° bearing showed that thermal crowning of the journal could reduce the effective bearing clearance by up to 15% under high-temperature operating conditions, producing a favourable increase in film thickness relative to rigid-body predictions but also introducing risk of edge loading. The influence of ambient temperature enters this analysis not only through the initial lubricant viscosity but also through the thermal boundary conditions imposed at the outer surface of the bearing housing.

Ferron et al. [22] conducted a comprehensive parametric study of a finite-length journal bearing using a full THD model and demonstrated that the steady-state eccentricity ratio—and hence the minimum film thickness—varies substantially with supply oil temperature. Increasing the supply temperature from 40 °C to 80 °C at constant load and speed was shown to reduce the minimum film thickness by approximately 22–28%, depending on the length-to-diameter ratio, primarily due to the associated reduction in viscosity. This quantitative finding underscores the importance of controlling lubricant supply temperature as an indirect proxy for ambient temperature effects.

Gethin and El Deihi [23] investigated the combined effects of shaft misalignment and temperature on bearing performance and found that misalignment amplifies thermal gradients within the film, creating localised hot spots that can initiate oil film breakdown under conditions that would be safe in a perfectly aligned bearing. The ambient temperature was identified as a key boundary condition governing the severity of thermal gradients, with higher ambient temperatures reducing the driving force for heat conduction through the bearing housing and thereby elevating peak film temperatures.

More recently, Solghar and Gandjalikhan Nassab [24] applied a three-dimensional energy equation in generalised non-orthogonal curvilinear coordinates to a finite journal bearing and reported that axial variations in temperature, often neglected in two-dimensional THD models, become significant at L/D ratios below 0.5. At high ambient temperatures, the reduced viscosity diminishes side leakage and alters the axial temperature distribution, a finding with implications for the design of short bearings in high-temperature machinery.

## 2.5 Experimental Investigations of Temperature Effects

Experimental characterisation of journal bearing performance under controlled temperature conditions has been pursued by numerous researchers using purpose-built test rigs that permit independent control of shaft speed, applied load, lubricant supply temperature, and ambient conditions. The principal measured quantities are typically the minimum film thickness (via capacitance or inductance probes), journal eccentricity and attitude angle (via proximity transducers), bulk oil temperature rise, and friction torque.

Fitzgerald and Neal [25] performed one of the earliest systematic experimental studies of ambient temperature effects on journal bearing lubrication, testing a 100 mm diameter bearing over a range of supply temperatures from 20 °C to 90 °C and shaft speeds from 500 to 3000 rpm. They reported that the minimum film thickness correlated well with the supply-temperature-adjusted Sommerfeld number, confirming that viscosity is the dominant intermediary between ambient temperature and bearing performance. However, discrepancies of up to 18% between measured and isoviscous-theory predictions were noted at high loads, motivating the subsequent development of THD corrections.

Fillon et al. [26] tested a large tilting-pad journal bearing (TPJB) instrumented with 35 embedded thermocouples and found that the peak film temperature increased nearly linearly with ambient temperature when the lubricant supply temperature was fixed, but exhibited a non-linear response when ambient temperature influenced the oil supply through an external cooler. The authors proposed a modified Nusselt-number correlation for the housing heat transfer coefficient that accounts for the ambient-to-oil temperature differential, providing a practical tool for bearing designers.

Singhal and Khonsari [27] examined cold-start transient behaviour in a journal bearing operating at sub-zero ambient temperatures, a condition relevant to automotive and aerospace applications. Using a combination of film thickness measurements and friction torque monitoring, they demonstrated that the transient period during which the bearing operates in the mixed-lubrication regime—with attendant risk of surface damage—increased by a factor of approximately 3.5 when the ambient temperature dropped from 0 °C to −30 °C for an SAE 5W-30 oil. This result highlights the critical role of low-temperature viscometric properties (cold cranking simulator viscosity, ASTM D5293) in cold-climate applications.

Peixoto and Cavalca [28] employed optical interferometry combined with a temperature-controlled test chamber to measure the full two-dimensional pressure distribution in a journal bearing at ambient temperatures between 25 °C and 75 °C. Their measurements revealed that the peak pressure location shifts circumferentially with increasing temperature due to the change in eccentricity ratio, an effect not fully captured by classical design charts because they assume a fixed viscosity. The experimental data provided a comprehensive validation dataset for THD numerical codes used in subsequent investigations.

## 2.6 Numerical Methods and Computational Fluid Dynamics Studies

The advent of high-performance computing has enabled progressively more sophisticated numerical treatment of the coupled fluid, thermal, and structural phenomena in journal bearings. Finite difference, finite element, and finite volume methods have all been applied to the THD problem, with the choice of method influenced by the geometry, the nature of the boundary conditions, and the required fidelity of the thermal solution.

Khonsari and Wang [29] developed a finite-element THD model that iteratively solves the Reynolds, energy, and heat conduction equations to convergence, including a cavitation algorithm based on the Jakobsson–Floberg–Olsson (JFO) boundary conditions. Their simulations demonstrated that the use of a constant ambient temperature as a thermal boundary condition at the bearing outer surface—rather than a convective boundary condition with a heat transfer coefficient—can introduce errors of up to 12% in the predicted peak film temperature, with corresponding errors in computed film thickness and power loss.

Computational fluid dynamics (CFD) approaches, using commercial solvers such as ANSYS Fluent and OpenFOAM, have been applied to journal bearing analysis by several authors. Brajdic-Mitidieri et al. [30] used CFD to investigate the effect of bearing surface temperature non-uniformity—arising from localised cooling by the ambient environment—on the pressure and film thickness distributions. The fully three-dimensional simulation revealed circumferential variations in viscosity that produced asymmetric pressure profiles not predicted by conventional THD models, particularly in bearings with low L/D ratios.

Shang and Croke [31] employed a transient CFD model to simulate the thermal response of a journal bearing system following a step change in ambient temperature, simulating the scenario of a machine moved from a cold warehouse to a hot production environment. The results indicated that the bearing required approximately 15–45 minutes to reach a new thermal steady state, depending on the bearing housing material and insulation, during which the film thickness deviated significantly from both the initial and final steady-state values. This transient response has important implications for predictive maintenance strategies.

Ramos et al. [32] applied response surface methodology (RSM) combined with a THD finite-element model to generate surrogate models for journal bearing performance as a function of ambient temperature, shaft speed, and specific load. Using a central composite design of experiments, they demonstrated that ambient temperature and its interaction with shaft speed are the two most statistically significant factors affecting minimum film thickness, together accounting for over 78% of the total variance in a typical operating envelope. This result provides a rigorous statistical foundation for the commonly held engineering intuition that temperature control is paramount in bearing design.

## 2.7 Advanced Lubricant Formulations and Temperature Performance

Recognition of the limitations of conventional mineral oil lubricants under extreme temperature conditions has driven the development of advanced base stocks and additive packages designed to minimise viscosity–temperature sensitivity while maintaining adequate boundary lubrication properties.

Synthetic lubricants based on polyalphaolefin (PAO) base stocks have been extensively evaluated in journal bearing applications by Muraki and Kimura [33], who compared PAO and mineral oil performance across a temperature range of −40 °C to 150 °C. The PAO lubricant maintained adequate film thickness at temperatures 20–30 °C higher than the mineral oil baseline, a consequence of its superior viscosity index. At low ambient temperatures, the PAO also produced lower starting torque due to its more favourable cold-flow properties, as characterised by its Brookfield viscosity (ASTM D2983).

Nanofluid lubricants, in which nanoparticles of materials such as $TiO_2$, $Al_2O_3$, $MoS_2$, or graphene are dispersed in a base oil, have attracted considerable attention as a means of simultaneously improving thermal conductivity and reducing boundary friction. Reeves et al. [34] reviewed the tribological performance of nanoparticle-enhanced lubricants and noted that the temperature-dependent viscosity behaviour of nanofluids deviates from that of the base oil, typically exhibiting a higher viscosity at elevated temperatures due to nanoparticle agglomeration effects. In the context of journal bearings, this behaviour can be advantageous by partially compensating for the viscosity reduction associated with rising ambient temperature, although the associated increase in fluid density and the risk of nanoparticle sedimentation must be carefully managed [35].

Ionic liquids (ILs) represent another frontier in high-temperature lubricant research. Somers et al. [36] characterised the viscosity–temperature behaviour of phosphonium-based ionic liquids and demonstrated near-Newtonian behaviour with viscosity index values exceeding 200, substantially better than PAO. Their application to journal bearings is at an early stage, but preliminary experiments by Bermúdez et al. [37] suggest that IL-lubricated bearings maintain hydrodynamic operation at supply temperatures up to 200 °C, opening possibilities for high-temperature process machinery currently reliant on expensive solid lubricants.

## 2.8 Temperature-Induced Lubricant Degradation and Oxidation

Beyond the immediate rheological effects of temperature on viscosity, sustained high-temperature operation accelerates the chemical degradation of lubricating oil through oxidation, thermal cracking, and additive depletion, all of which progressively alter the oil's performance characteristics and ultimately lead to bearing failure if not addressed through condition monitoring.

Oxidation is the dominant degradation mechanism in circulating oil systems. Following the Arrhenius rate law, oxidation rate approximately doubles for every 10 °C increase in oil temperature, a relationship known as the van't Hoff rule and widely cited in lubricant engineering practice [38]. Mortier et al. [39] described the chain-reaction mechanism of hydrocarbon oxidation and the role of antioxidant additives—hindered phenols and aromatic amines—in scavenging free radicals to retard this process. Depletion of antioxidants leads to a sharp increase in oxidation rate, manifesting as an increase in oil viscosity and total acid number (TAN), both of which adversely affect bearing performance.

Toms and Toms [40] reviewed field data from industrial rotating machinery and found a strong correlation between sustained operating temperatures above 80 °C and the frequency of unscheduled bearing replacements. Their analysis identified oxidative viscosity increase—causing the Sommerfeld number to rise above design intent—as the most common mechanism linking elevated ambient temperature to premature bearing failure, emphasising the importance of oil condition monitoring alongside the direct measurement of operating temperature.

## CHAPTER 3

## METHODOLOGY

### 3.1 Governing Equations

Viscosity of lubricating oil μ in reyn is correlated with oil film temperature T in degree Fahrenheit by [41]

$$\mu = \mu_0 e^{\frac{b}{T+95}}$$

where $\mu_0$ is viscosity at 0ºC in reyn and b is a constant in degree Fahrenheit. The values of $\mu_0$ and b for different lubricating oils are given in table 3.1 and 3.2. The Sommerfeld number of the lubricating oil is correlated to the journal bearing radius r, clearance c, viscosity μ, rotational speed N and average oil film pressure P by [42]

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{P}$$

where r, c, μ, N and P are in m, m, Pas, Rev/s and $N/m^2$ respectively and S is a dimensionless number. The average oil film pressure P can be expressed in terms of load W, bearing length l and bearing diameter d for natural fed and self contained bearings as [41]

$$P = \frac{W}{ld}$$

where W, l, d are in N, m and m respectively. For a pressure fed bearing, half bearing length l′ is used in place of l. Therefore [41],

$$P = \frac{W}{l'd}$$

where [41]

$$l' = \frac{l}{2}$$

The eccentricity ratio ε can be expressed in terms of lubricating oil's Sommerfeld number, bearing length l and bearing diameter d for natural fed and self contained bearings as [43]

$$\varepsilon = \sqrt{\frac{2+2\pi S\left(\frac{l}{d}\right)^{-2} - \sqrt{\left(2+2\pi S\left(\frac{l}{d}\right)^{-2}\right)^2 - 4}}{2}}$$

where l, d both are in m and ε is dimensionless. For a pressure fed bearing, half bearing length l′ is used in place of l. Therefore [43],

$$\varepsilon = \sqrt{\frac{2+2\pi S\left(\frac{l'}{d}\right)^{-2} - \sqrt{\left(2+2\pi S\left(\frac{l'}{d}\right)^{-2}\right)^2 - 4}}{2}}$$

Minimum oil film temperature $h_0$ is correlated with eccentricity ratio ε and clearance c by [41]

$$h_0 = c(1 - \varepsilon)$$

where $h_0$ and c are in μm. The minimum acceptable oil film thickness $h_{0min}$ is taken from one of the Trumpler's criteria [44]:

$$h_{0min} = 0.0002 + 0.00004d$$

where $h_{0min}$ and d are in inches. The maximum temperature rise for a pressure fed bearing is 120ºF. Therefore ambient temperature $T_a$ can be expressed by oil film temperature T as [45]

$$T = T_a + 120$$

where T and $T_a$ both are in degrees Fahrenheit. On the other hand, the maximum temperature rise for a natural fed bearing is 70ºF [46]. Therefore,

$$T = T_a + 70$$

Again, the maximum temperature rise for a self contained bearing is 75ºF [47]. Hence,

$$T = T_a + 75$$

Table 4.1: The values of $\mu_0$ and b for SAE oils [41]

| Oil Type | Viscosity $\mu_0$ (reyn) | Constant b (ºF) |
|---|---|---|
| SAE 10 | $1.58 \times 10^{-8}$ | 1157.5 |
| SAE 20 | $1.36 \times 10^{-8}$ | 1271.6 |
| SAE 30 | $1.41 \times 10^{-8}$ | 1360.0 |
| SAE 40 | $1.21 \times 10^{-8}$ | 1474.4 |
| SAE 50 | $1.70 \times 10^{-8}$ | 1509.6 |

Table 4.2: The values of $\mu_0$ and b for multiviscosity oils [1]

| Oil Type | Viscosity $\mu_0$ (reyn) | Constant b (ºF) |
|---|---|---|
| SAE 0W-30 | $2.2473 \times 10^{-7}$ | 728.7 |
| SAE 5W-40 | $2.5031 \times 10^{-7}$ | 773.5 |
| SAE 10W-40 | $2.8703 \times 10^{-7}$ | 753.5 |
| SAE 10W-60 | $3.9583 \times 10^{-7}$ | 798.7 |
| SAE 15W-40 | $1.6673 \times 10^{-7}$ | 896.5 |

## 3.2 Python Software

Python is an interpreted, object-oriented, high-level programming language with dynamic semantics. Its high-level built in data structures, combined with dynamic typing and dynamic binding, make it very attractive for Rapid Application Development, as well as for use as a scripting or glue language to connect existing components together. Python's simple, easy to learn syntax emphasizes readability and therefore reduces the cost of program maintenance. Python supports modules and packages, which encourages program modularity and code reuse. The Python interpreter and the extensive standard library are available in source or binary form without charge for all major platforms, and can be freely distributed.

Often, programmers fall in love with Python because of the increased productivity it provides. Since there is no compilation step, the edit-test-debug cycle is incredibly fast. Debugging Python programs is easy: a bug or bad input will never cause a segmentation fault. Instead, when the interpreter discovers an error, it raises an exception. When the program doesn't catch the exception, the interpreter prints a stack trace. A source level debugger allows inspection of local and global variables, evaluation of arbitrary expressions, setting breakpoints, stepping through the code a line at a time, and so on. The debugger is written in Python itself, testifying to Python's introspective power. On the other hand, often the quickest way to debug a program is to add a few print statements to the source: the fast edit-test-debug cycle makes this simple approach very effective.

### 3.2.1 Numpy Module

Numpy is the fundamental package for scientific computing in Python. It is a Python library that provides a multidimensional array object, various derived objects (such as masked arrays and matrices), and an assortment of routines for fast operations on arrays, including mathematical, logical, shape manipulation, sorting, selecting, I/O, discrete Fourier transforms, basic linear algebra, basic statistical operations, random simulation and much more.

At the core of the Numpy package, is the **ndarray** object. This encapsulates *n*-dimensional arrays of homogeneous data types, with many operations being performed in compiled code for performance. There are several important differences between Numpy arrays and the standard Python sequences:

- Numpy arrays have a fixed size at creation, unlike python lists (which can grow dynamically). Changing the size of an **ndarray** will create a new array and delete the original.
- The elements in a Numpy array are all required to be of the same data type, and thus will be the same size in memory. The exception: one can have arrays of (Python, including Numpy) objects, thereby allowing for arrays of different sized elements.
- Numpy arrays facilitate advanced mathematical and other types of operations on large numbers of data. Typically, such operations are executed more efficiently and with less code than is possible using Python’s built-in sequences.
- A growing plethora of scientific and mathematical Python-based packages are using Numpy arrays; though these typically support Python-sequence input, they convert such input to Numpy arrays prior to processing, and they often output Numpy arrays. In other words, in order to efficiently use much (perhaps even most) of today’s scientific/mathematical Python-based software, just knowing how to use Python’s built-in sequence types is insufficient - one also needs to know how to use Numpy arrays.

### 3.2.2 Matplotlib Module

Matplotlib is a Python library for creating static, interactive and animated visualizations from data. It provides flexible and customizable plotting functions that help in understanding data patterns, trends and relationships effectively.

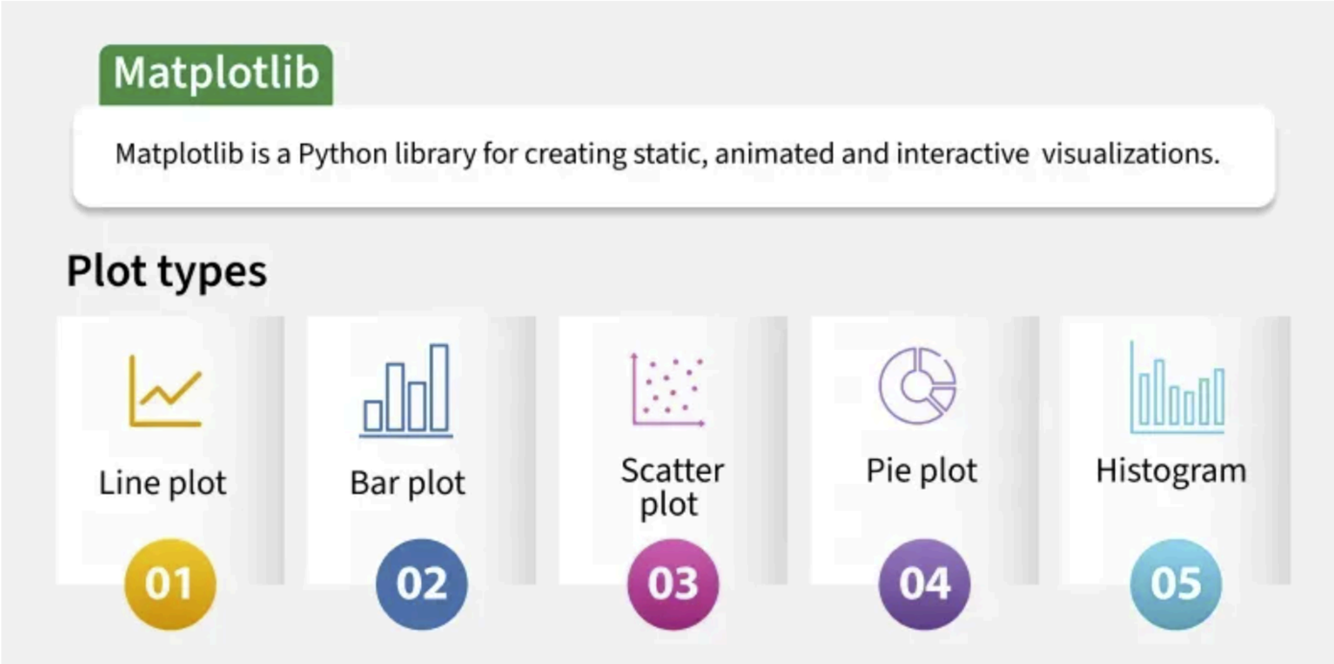


Fig 3.1: Introduction to Matplotlib

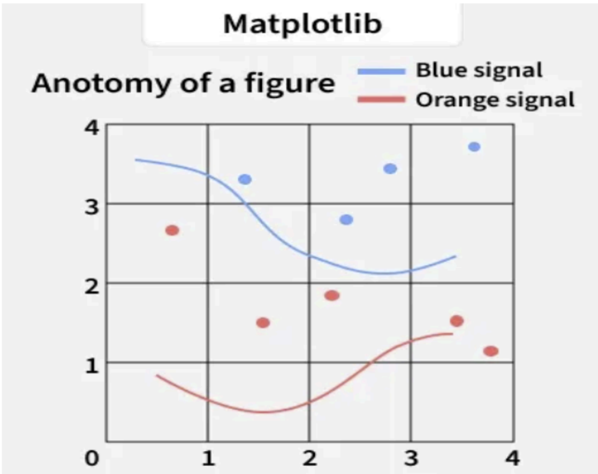


Fig 3.2 : Sample plots generated by Matplotlib

The parts of a Matplotlib figure include (as shown in the figure above):

- **Figure**: The overarching container that holds all plot elements, acting as the canvas for visualizations.
- **Axes**: The areas within the figure where data is plotted; each figure can contain multiple axes.
- **Axis**: Represents the x-axis and y-axis, defining limits, tick locations and labels for data interpretation.
- **Lines and Markers**: Lines connect data points to show trends, while markers denote individual data points in plots like scatter plots.

- **Title and Labels**: The title provides context for the plot, while axis labels describe what data is being represented on each axis.

## 3.3 Validation

The values of minimum film thickness computed by the codes are in good agreement with existing literature. Table 4.3 illustrates this agreement.

Table 4.3: Comparison of the values of $h_0$ found by the current study and that by Budynas and Nisbett

| **Condition** | **$h_0$ Found in Current Study (µm)** | **$h_0$ Found by Budynas and Nisbett [41] (µm)** |
|---|---|---|
| d = 44.45 mm, l′/d = 0.5, c/d = .0009, N = 50 rev/s, W = 4000 N, S = 0.088, pressure fed bearing | 8.3 | 8.3 |
| d = 38 mm, l/d = 1, c/d = .001, N = 30 rev/s, W = 2210 N, S = 0.135, natural fed bearing | 16 | 16 |
| d = 50 mm, l/d = 1, c/d = .0005, N = 15 rev/s, W = 450 N, S =0.67 , self contained bearing | 19.8 | 19.8 |

# CHAPTER 4

# RESULTS AND DISCUSSIONS

## 4.1 Results and Discussions

Figure 4.1 to figure 4.162 are generated by the Python code for pressure fed, natural fed and self contained bearings. The y axis represents the minimum film thickness which is taken as the performance parameter for this study and the x axis represents the ambient temperature. We can see from these figures that minimum film thickness reduces with increasing ambient temperature.

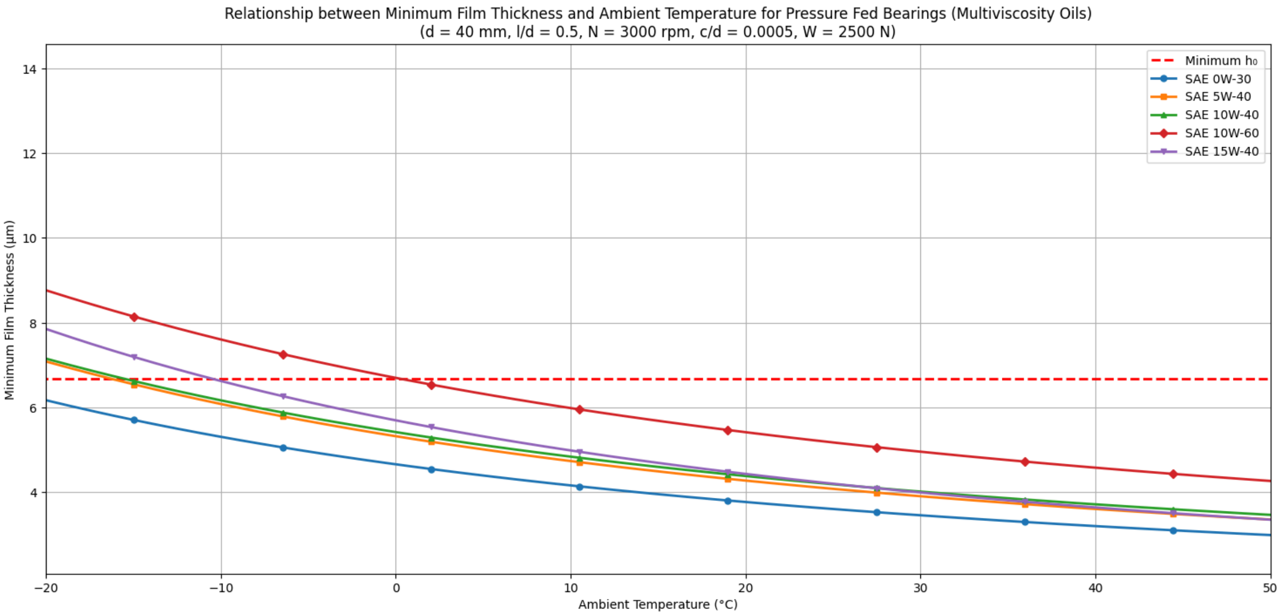


Fig 4.1: Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (d = 40 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

### 4.1.1 Effect of Varying Journal Diameter (Pressure Fed Bearing)

Figure 4.1 to figure 4.54 represents the performance of pressure fed bearings for different multiviscosity and SAE oils. The journal diameter is varied keeping length to diameter ratio, rotational speed, clearance to diameter ratio and load constant in figure 4.1 to figure 4.10. We can see that minimum film thickness increases with increasing journal diameter. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

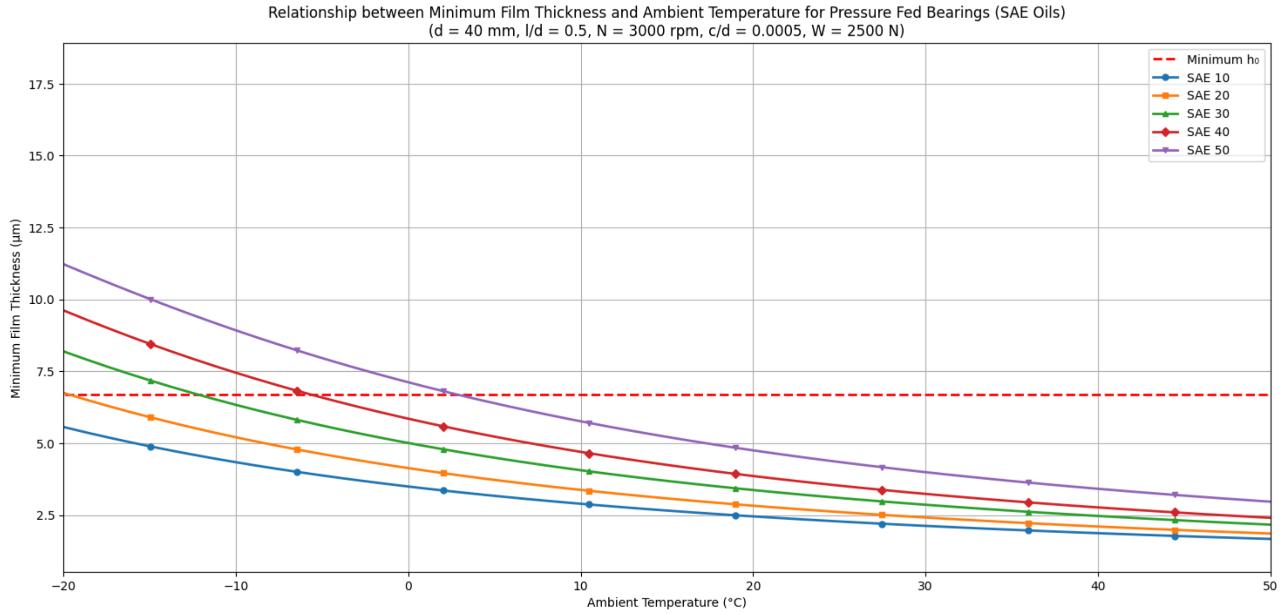


Fig 4.2: Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (d = 40 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

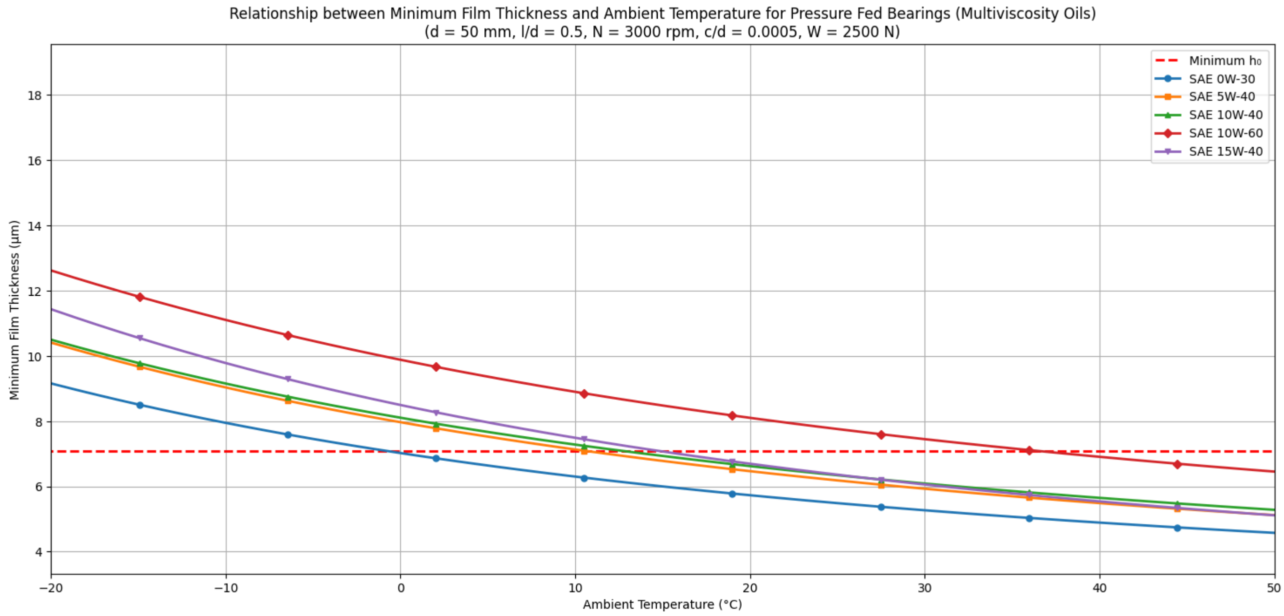


Fig 4.3:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (d = 50 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

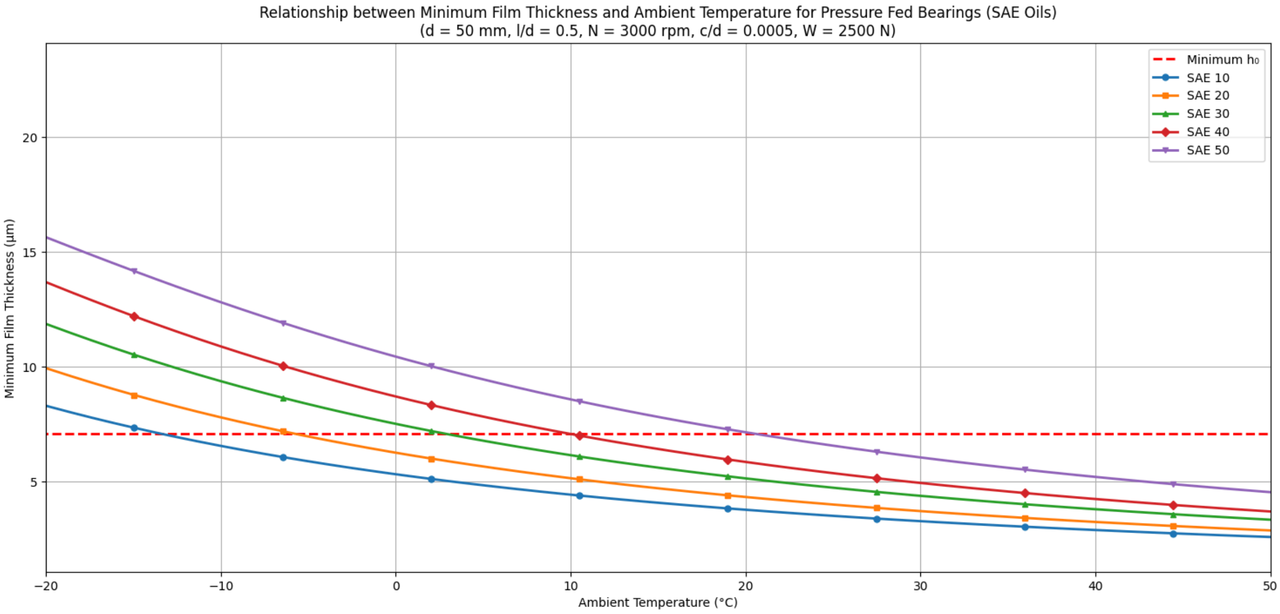


Fig 4.4:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (d = 50 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

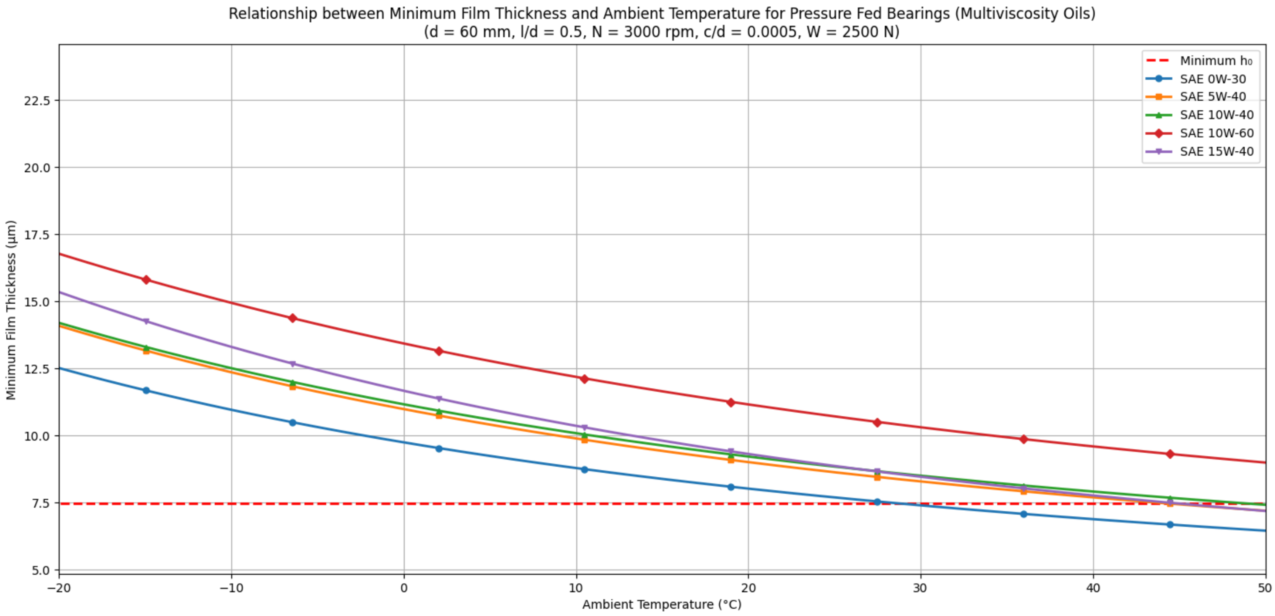


Fig 4.5:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (d = 60 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

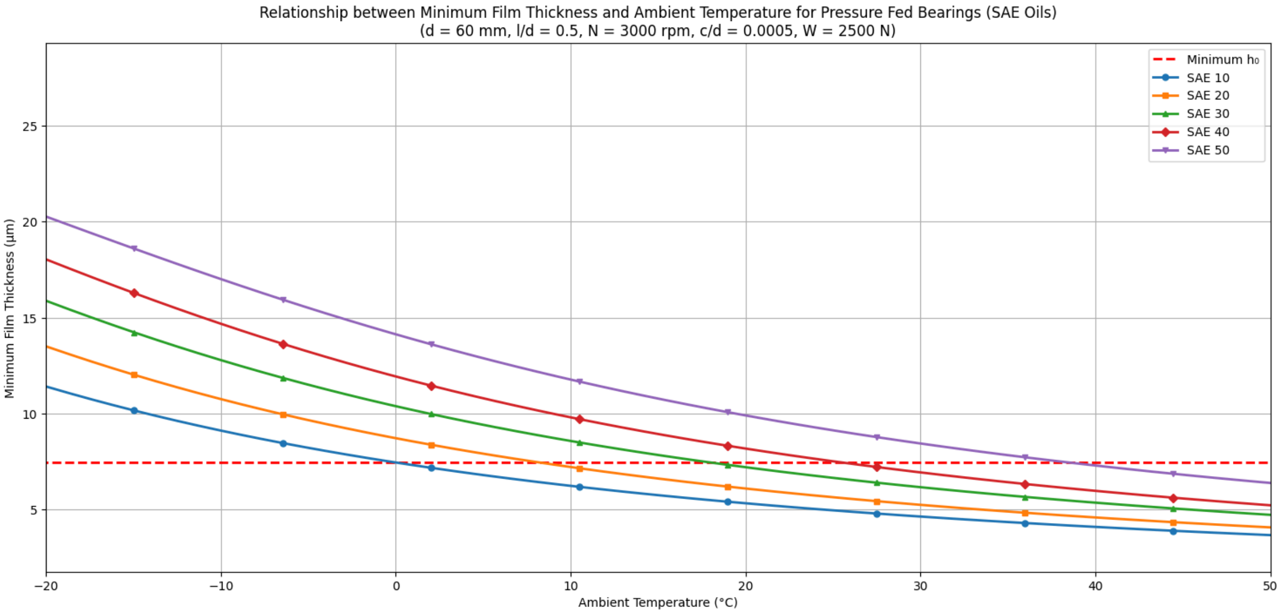


Fig 4.6:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (d = 60 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

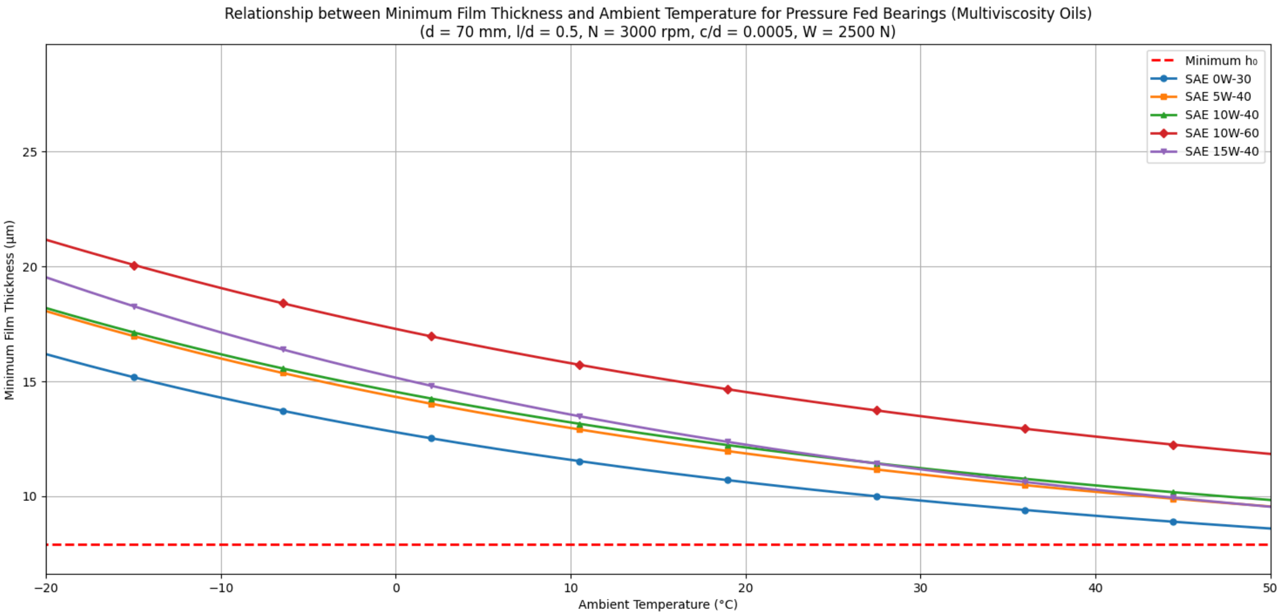


Fig 4.7:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (d = 70 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

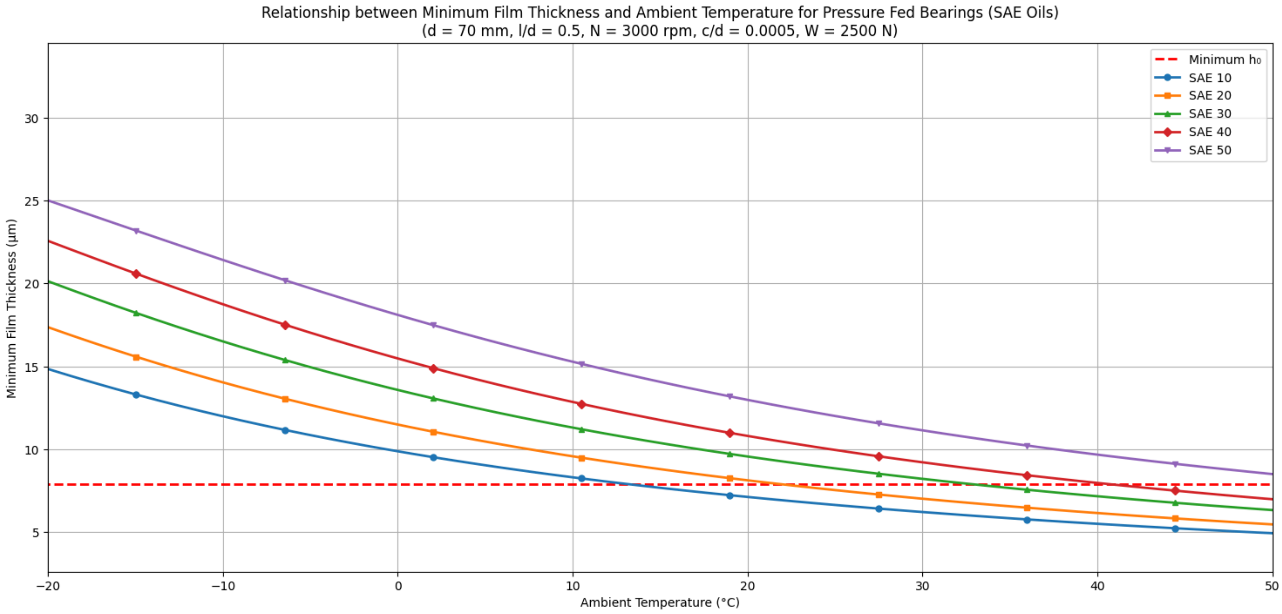


Fig 4.8:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (d = 70 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

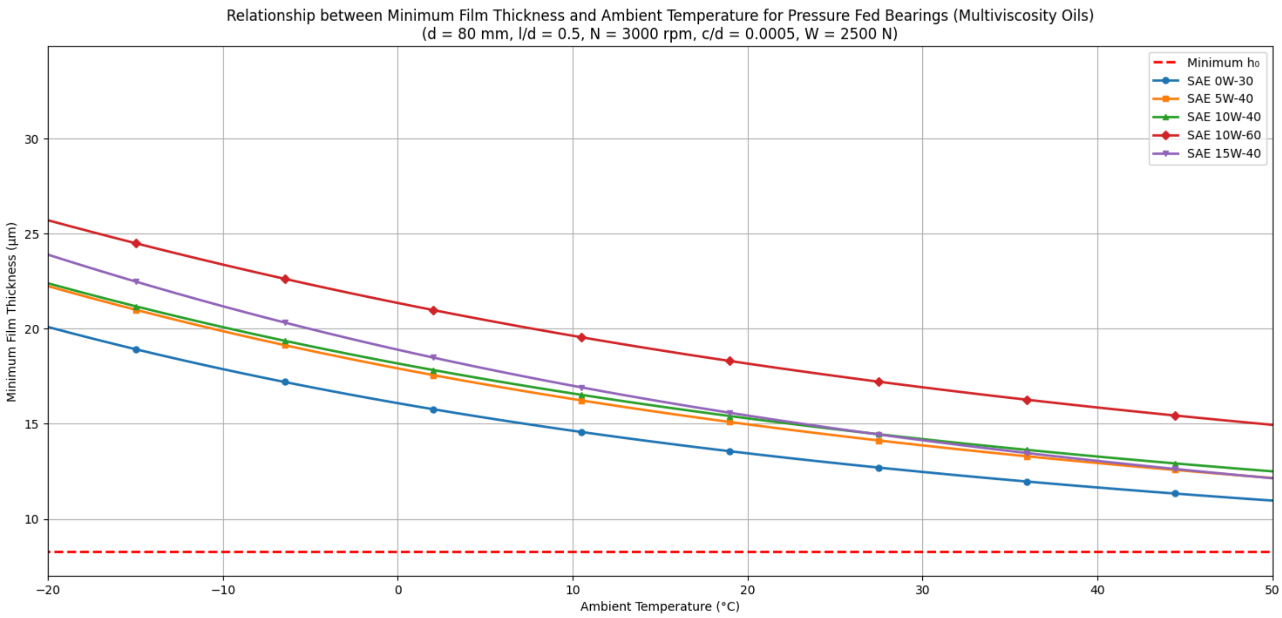


Fig 4.9:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (d = 80 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

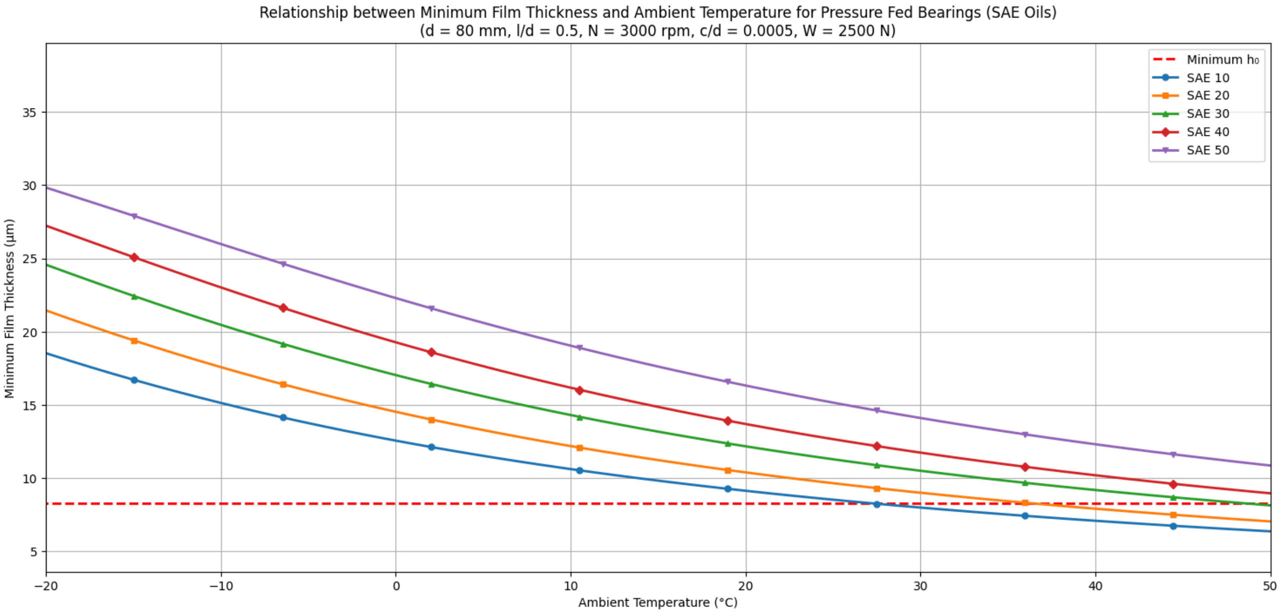


Fig 4.10:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (d = 80 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

## 4.1.2 Effect of Varying Length to Diameter Ratio (Pressure Fed Bearing)

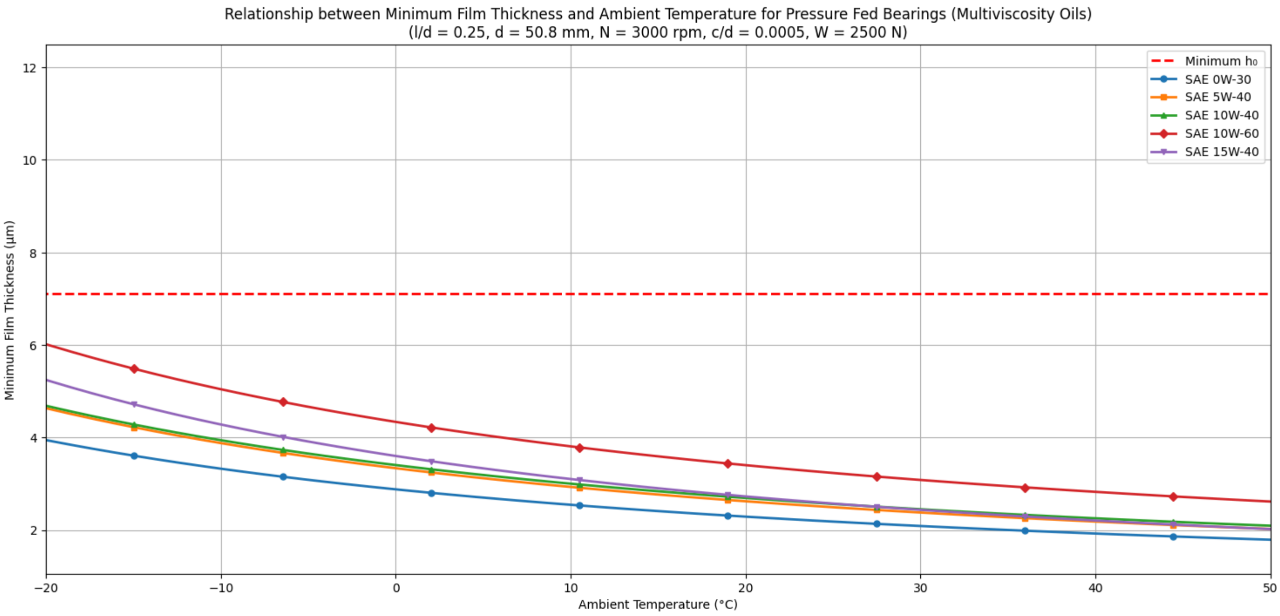


Fig 4.11:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (l/d = 0.25, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

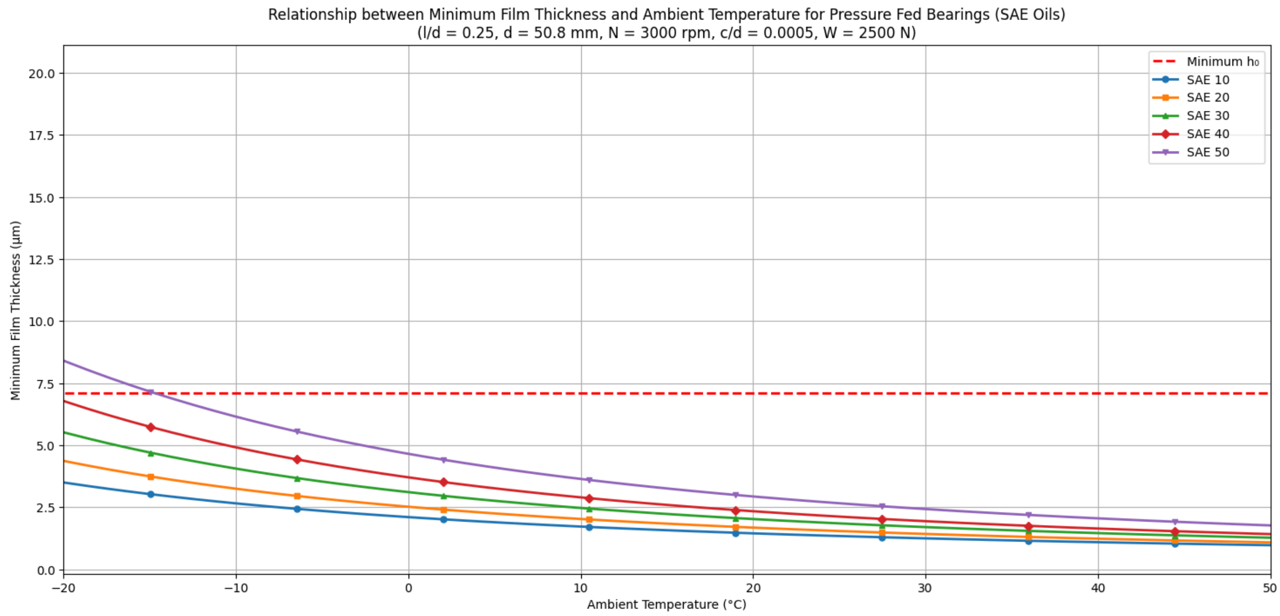


Fig 4.12:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (l/d = 0.25, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

The length to diameter ratio is varied keeping journal diameter, rotational speed, clearance to diameter ratio and load constant in figure 4.11 to figure 4.18. We can see that minimum film thickness increases with increasing length to diameter ratio. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

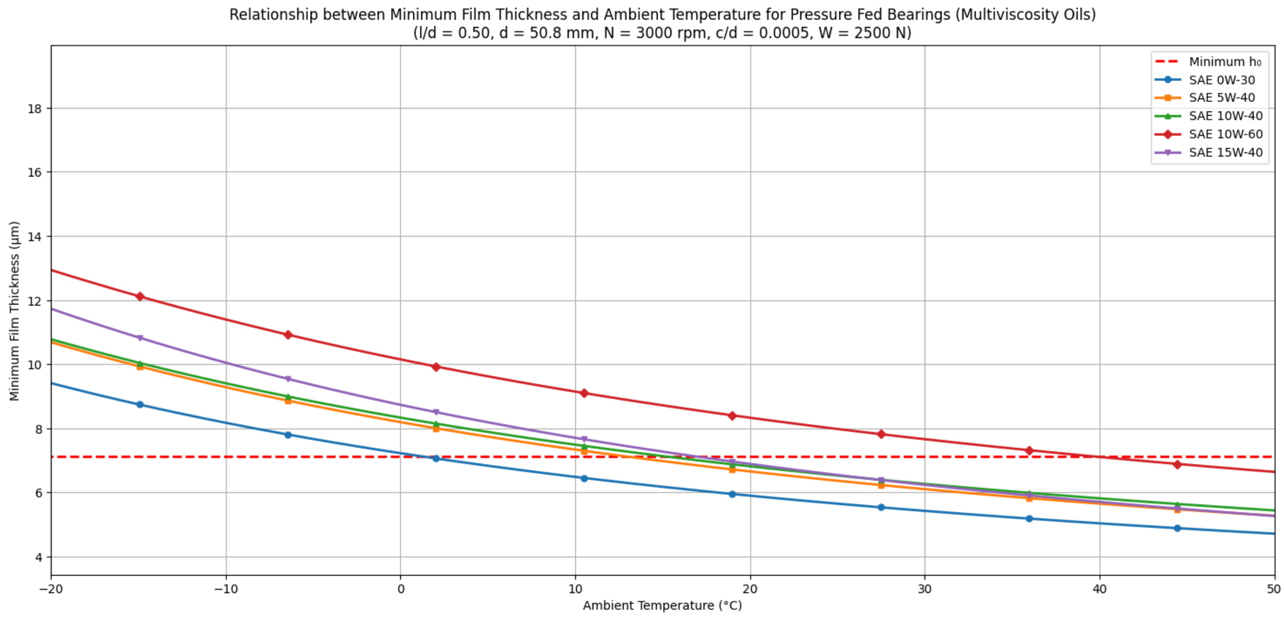


Fig 4.13:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (l/d = 0.50, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

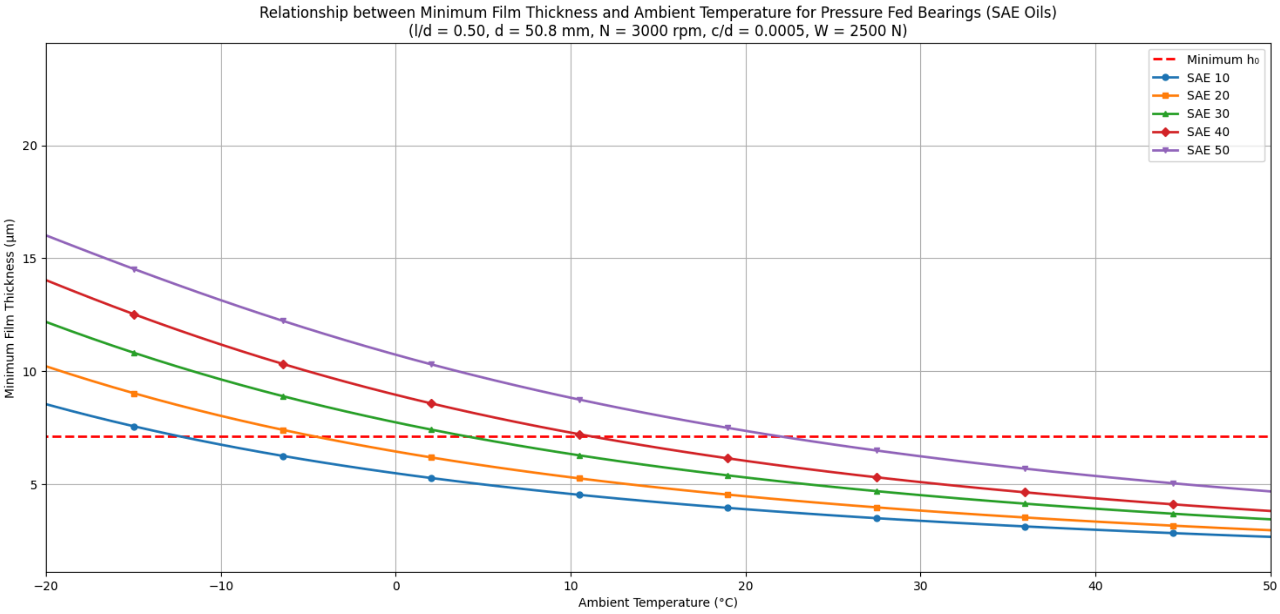


Fig 4.14:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (l/d = 0.50, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

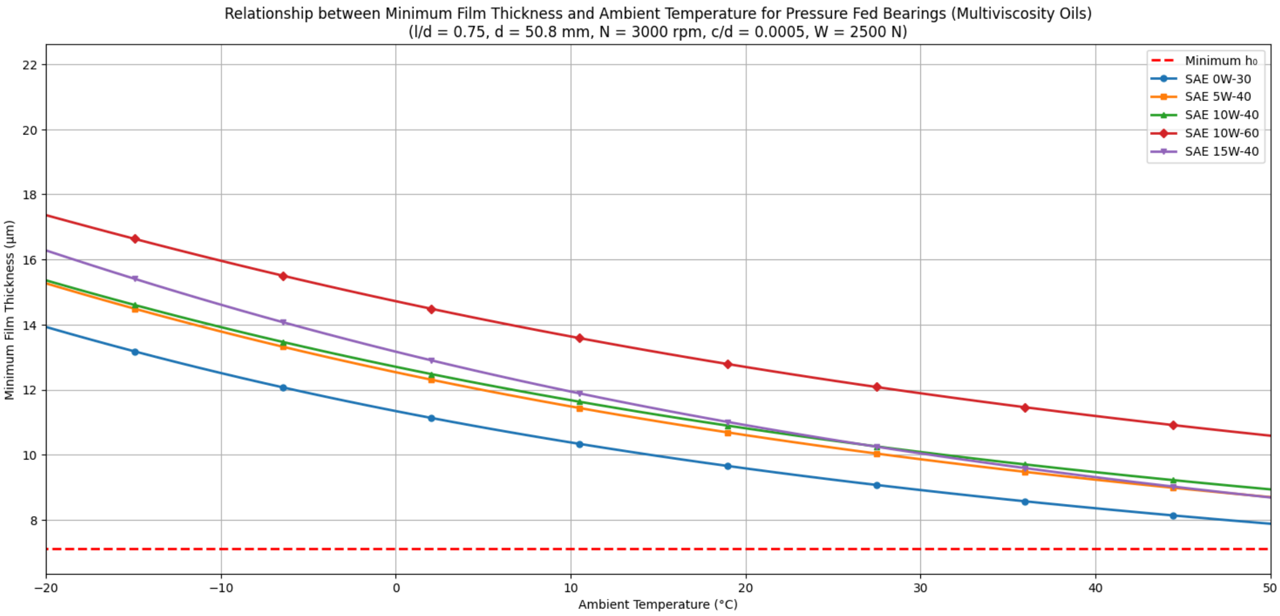


Fig 4.15:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (l/d = 0.75, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

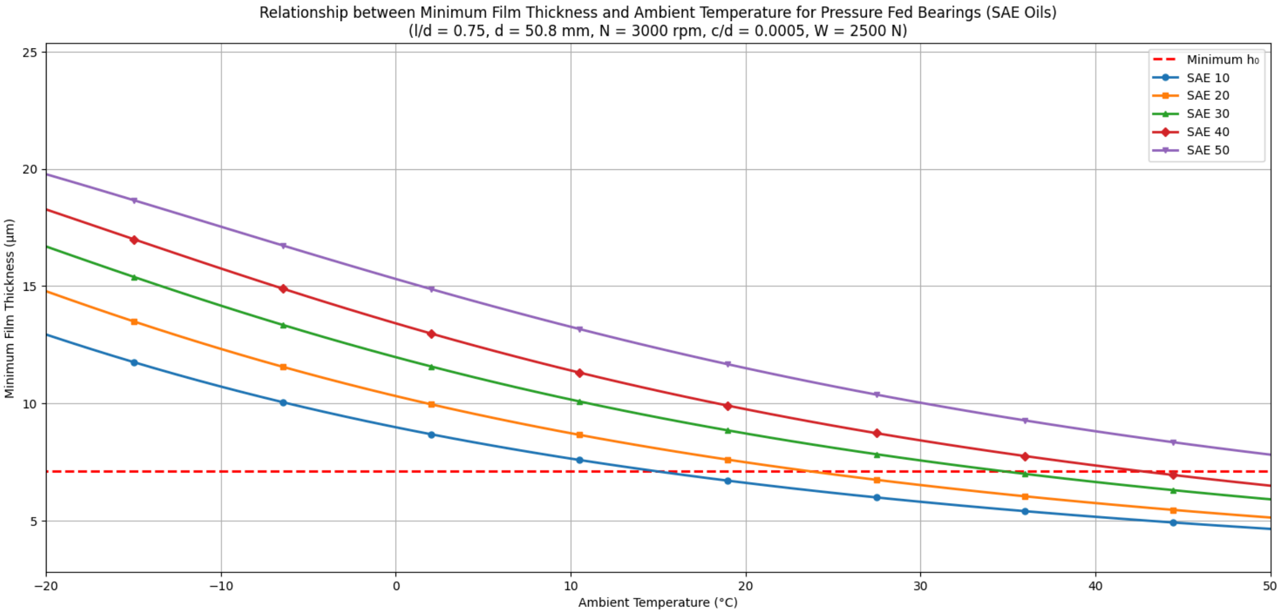


Fig 4.16:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (l/d = 0.75, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

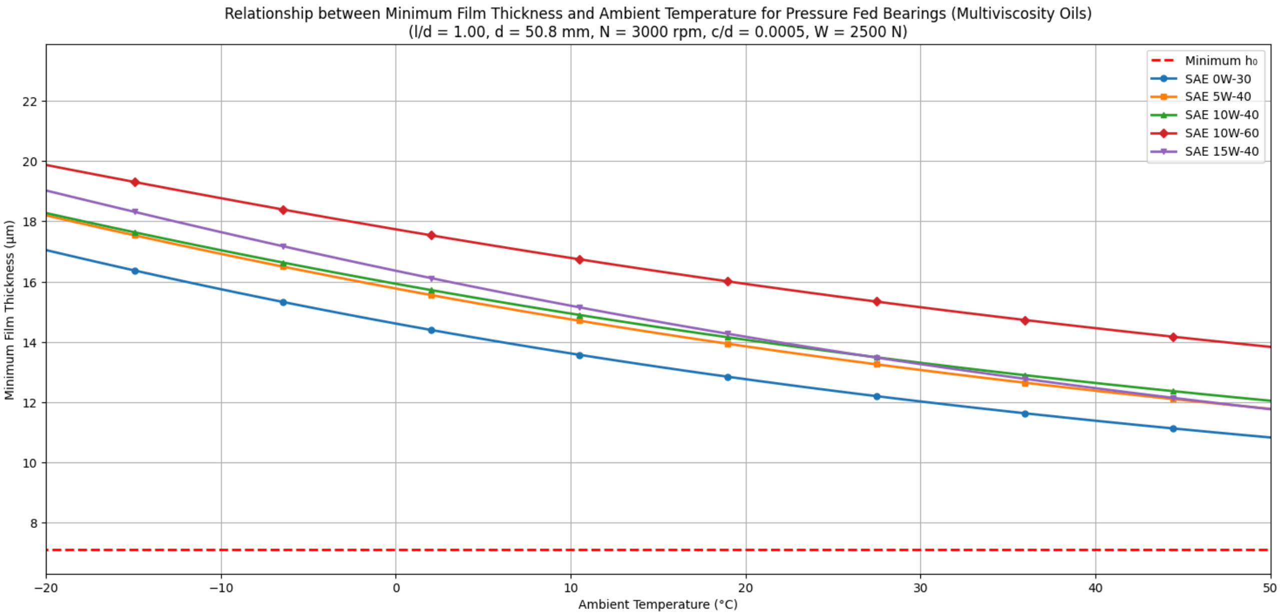


Fig 4.17:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (l/d = 1.00, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

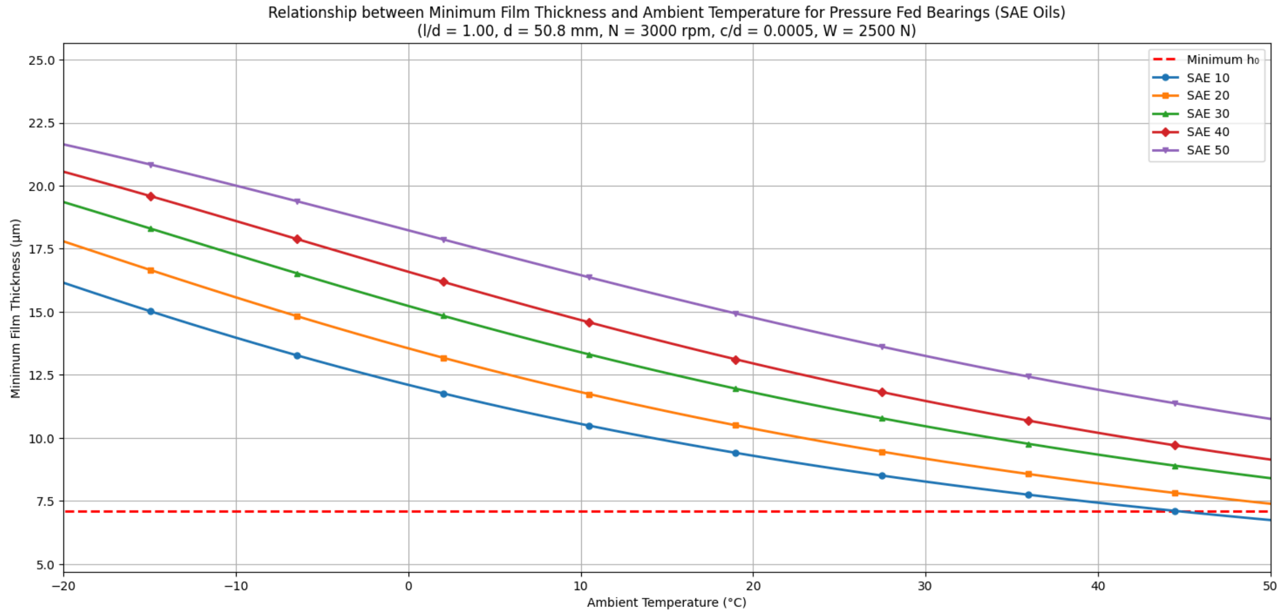


Fig 4.18:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (l/d = 1.00, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

### 4.1.3 Effect of Varying Rotational Speed (Pressure Fed Bearing)

The rotational speed is varied keeping journal diameter, length to diameter ratio, clearance to diameter ratio and load constant in figure 4.19 to figure 4.32. We can see that minimum film thickness increases with increasing rotational speed. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

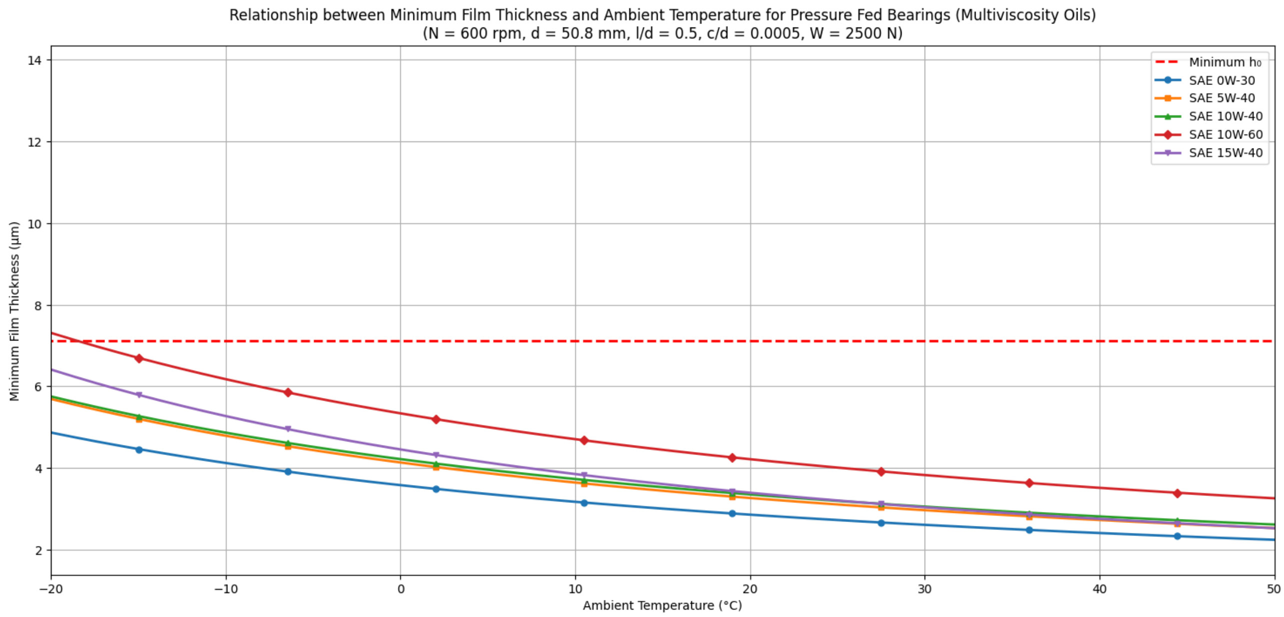


Fig 4.19:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (N = 600 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

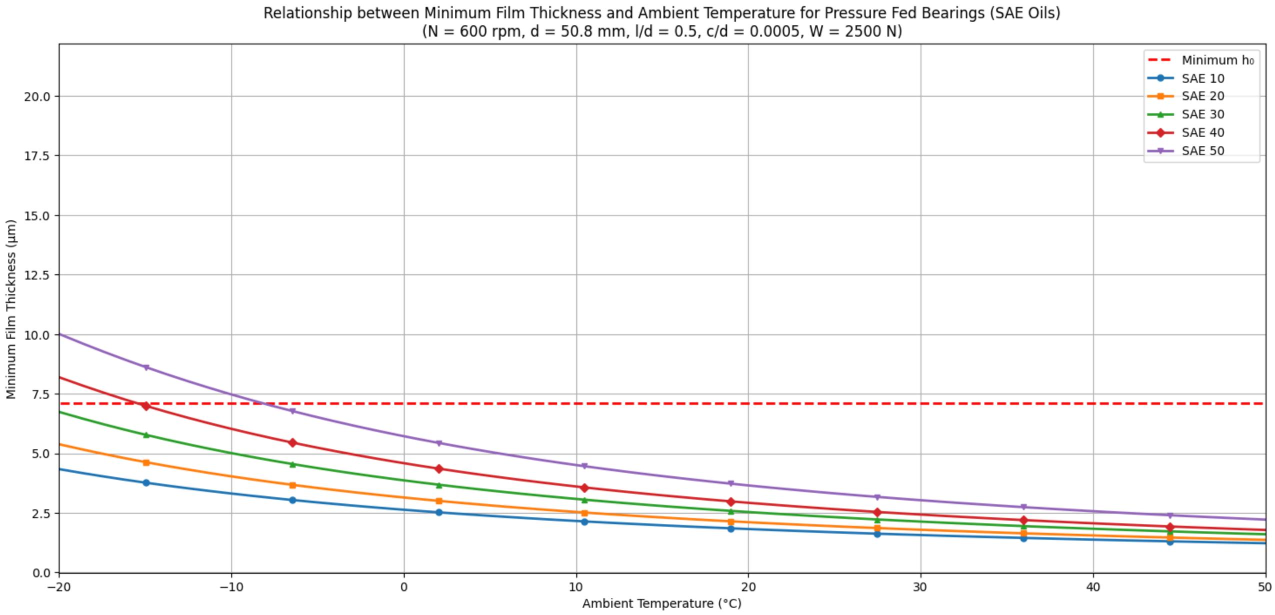


Fig 4.20:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (N = 600 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

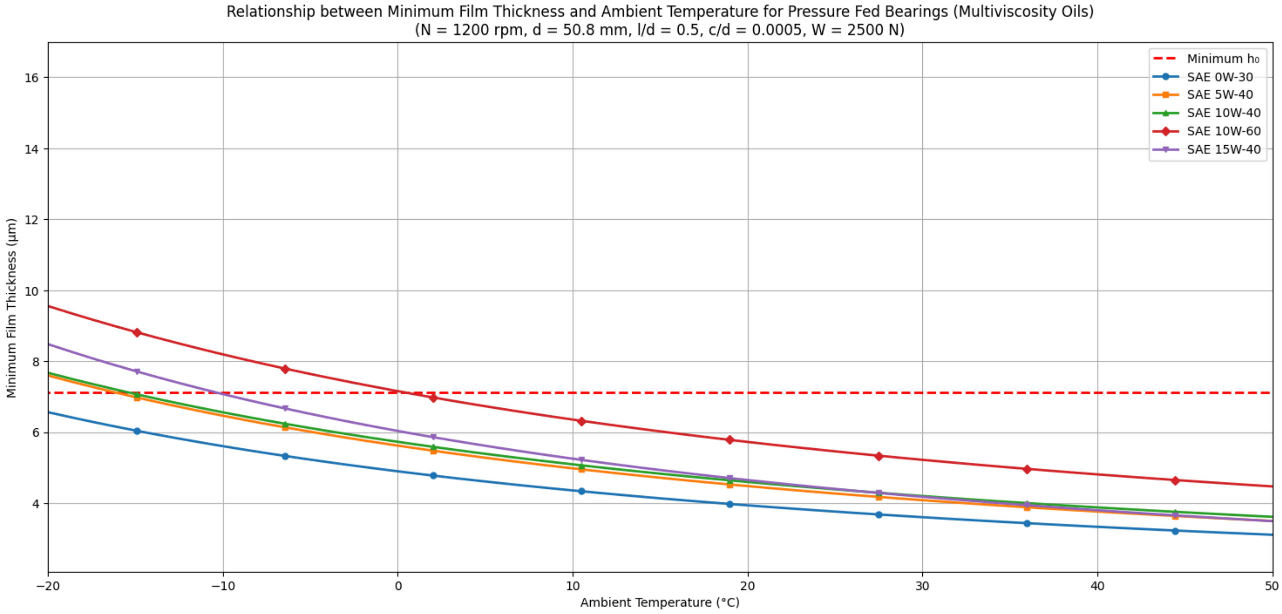


Fig 4.21:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (N = 1200 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

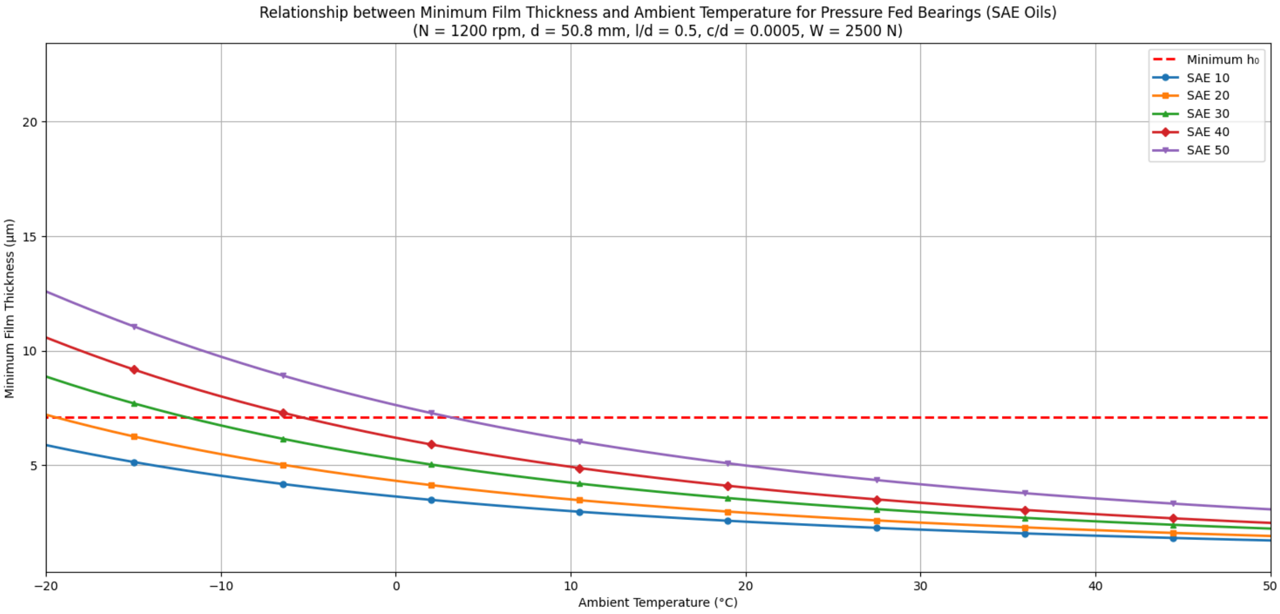


Fig 4.22:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (N = 1200 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

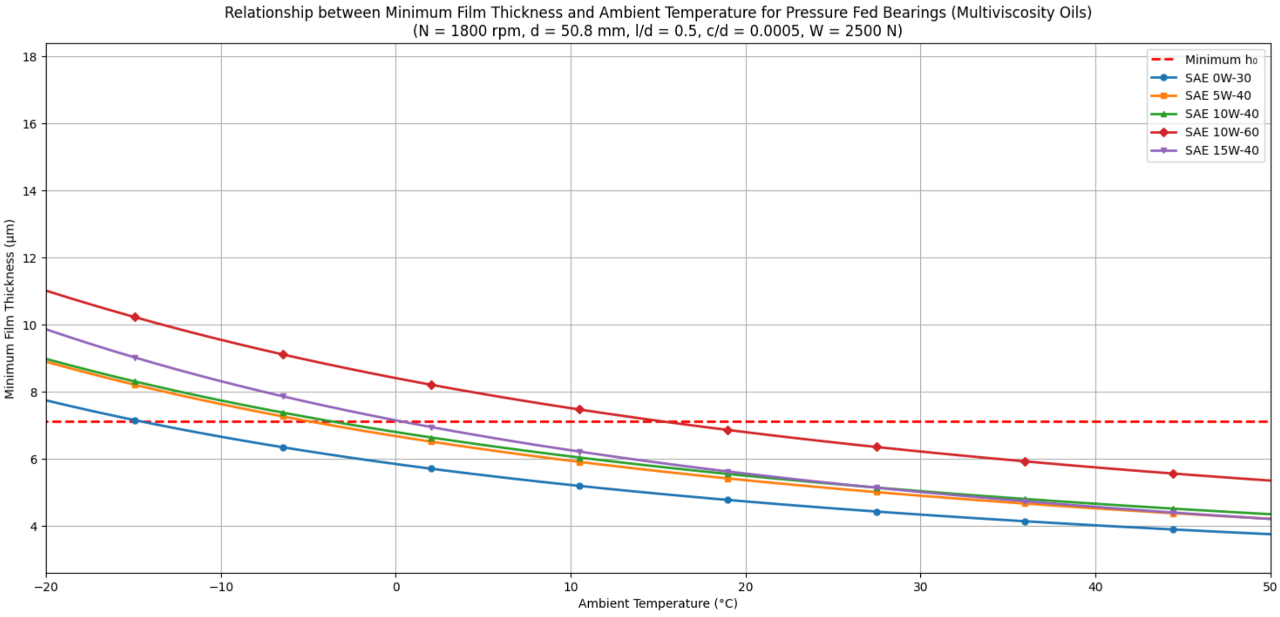


Fig 4.23:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (N = 1800 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

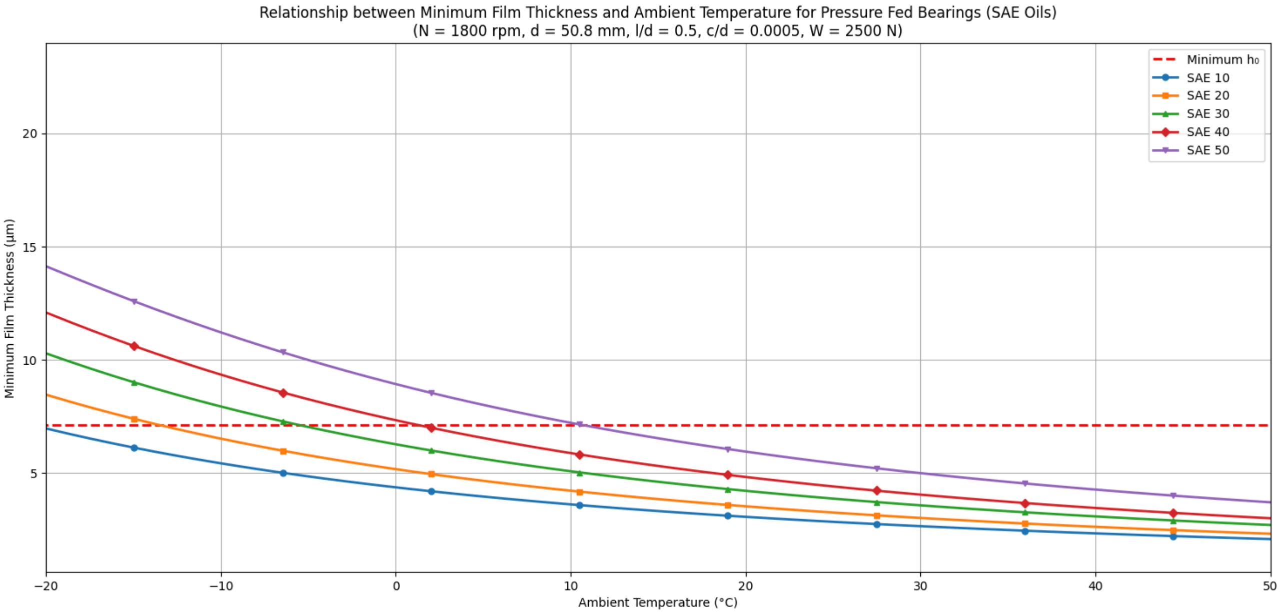


Fig 4.24:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (N = 1800 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

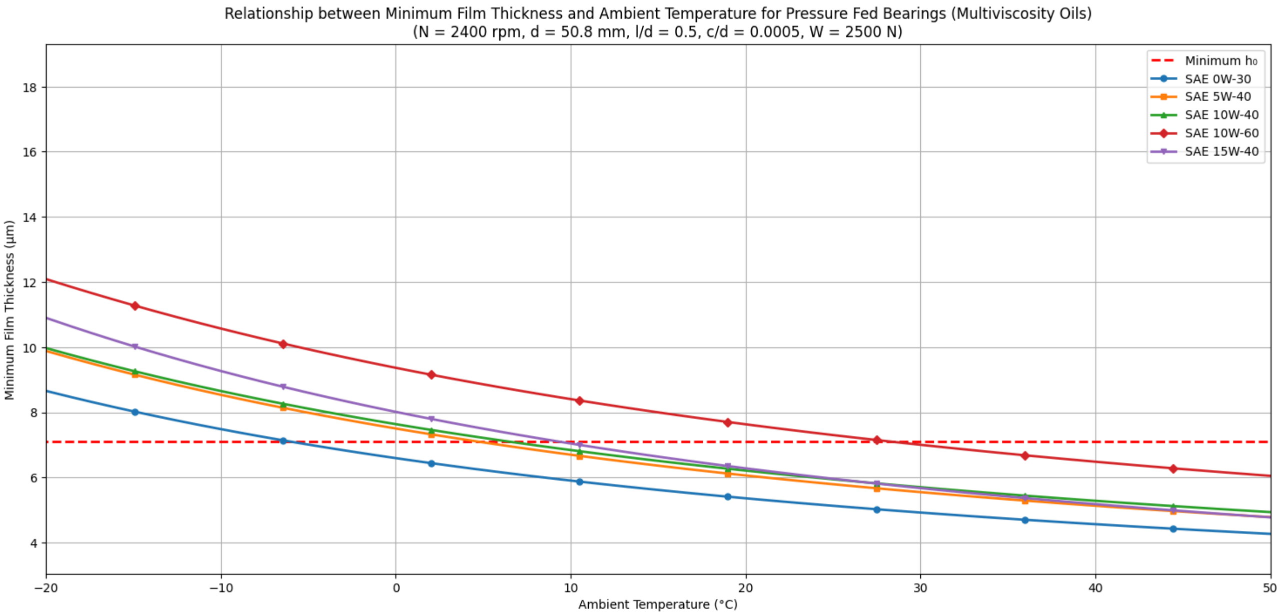


Fig 4.25:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (N = 2400 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

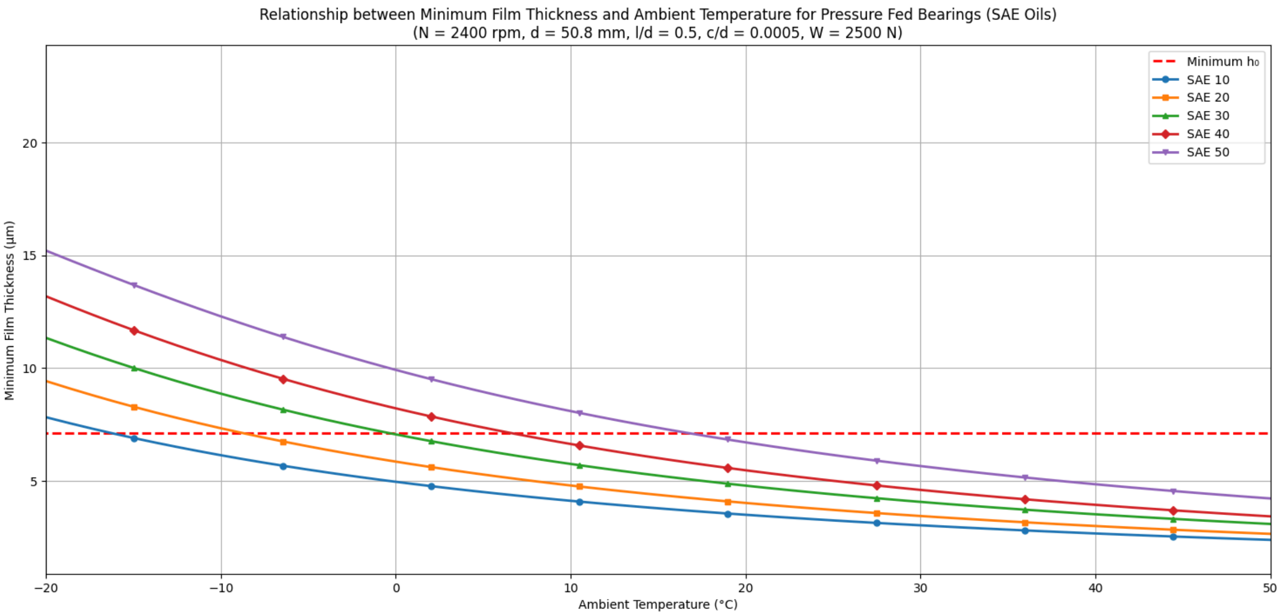


Fig 4.26:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (N = 2400 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

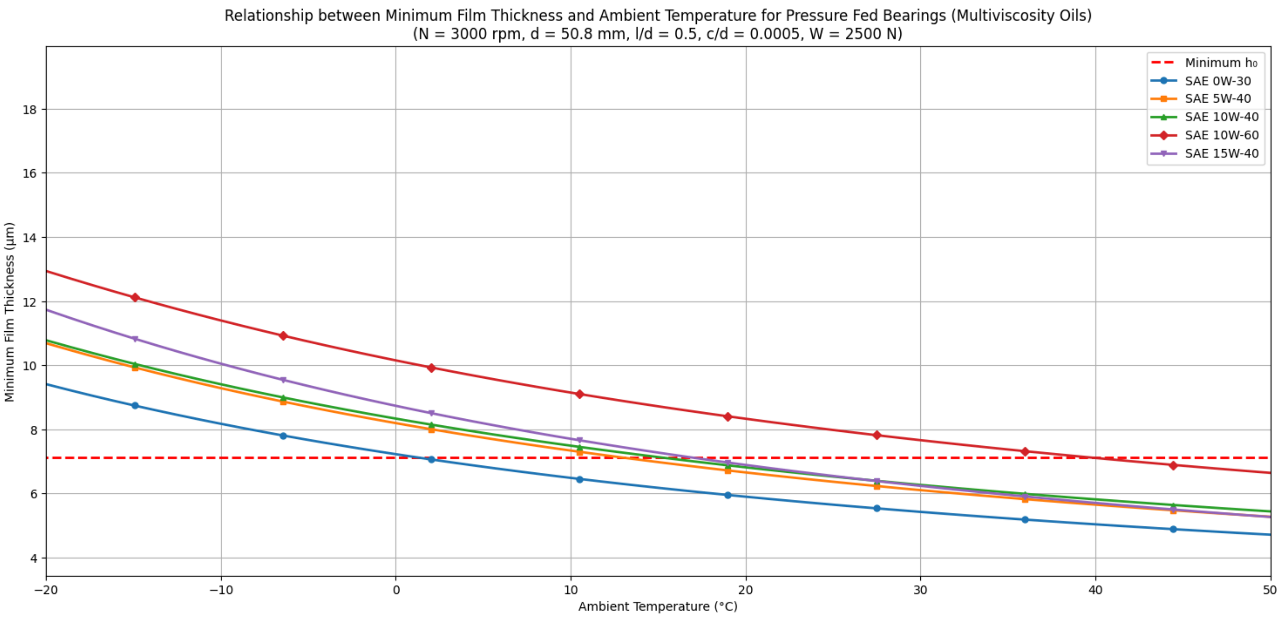


Fig 4.27:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (N = 3000 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

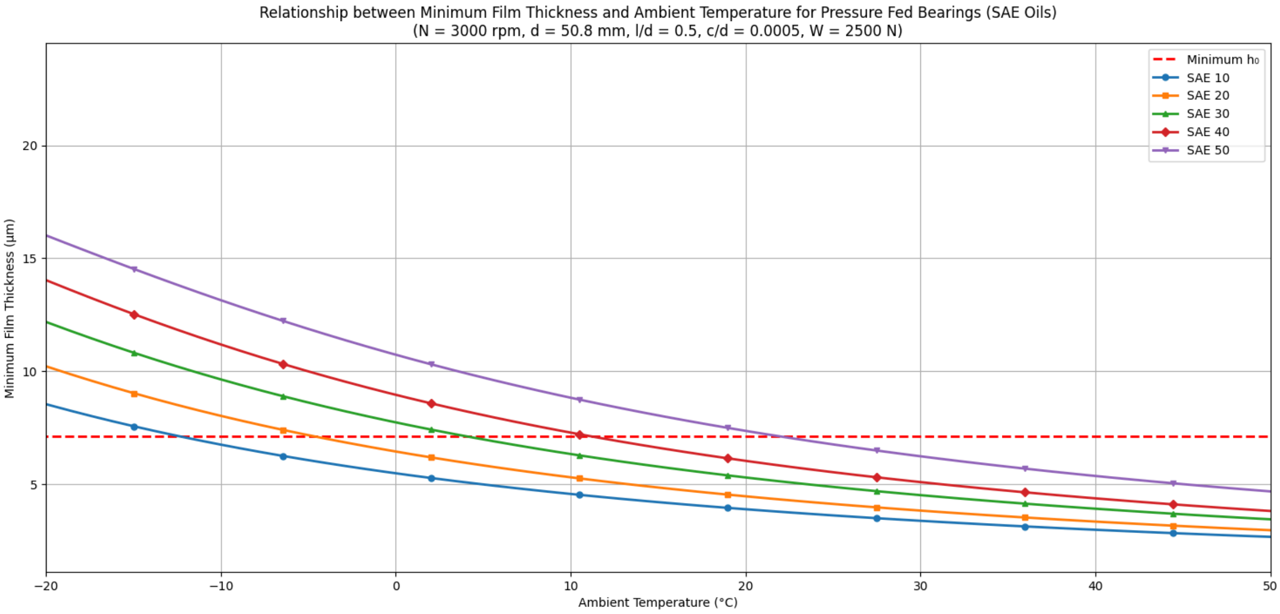


Fig 4.28:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (N = 3000 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

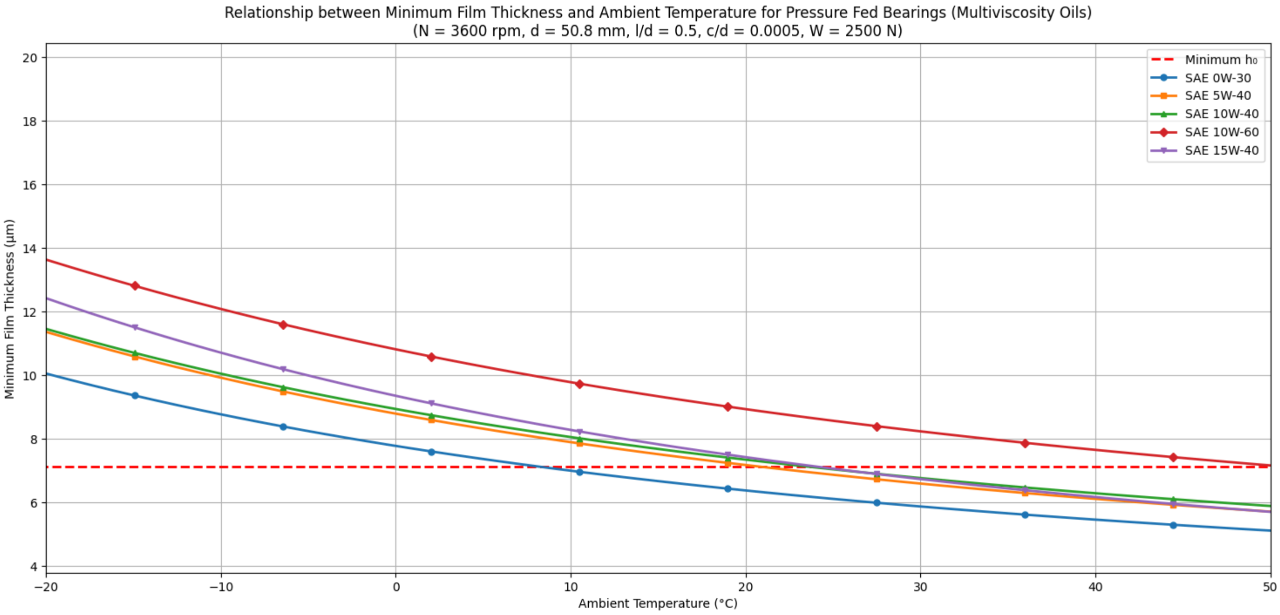


Fig 4.29:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (N = 3600 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

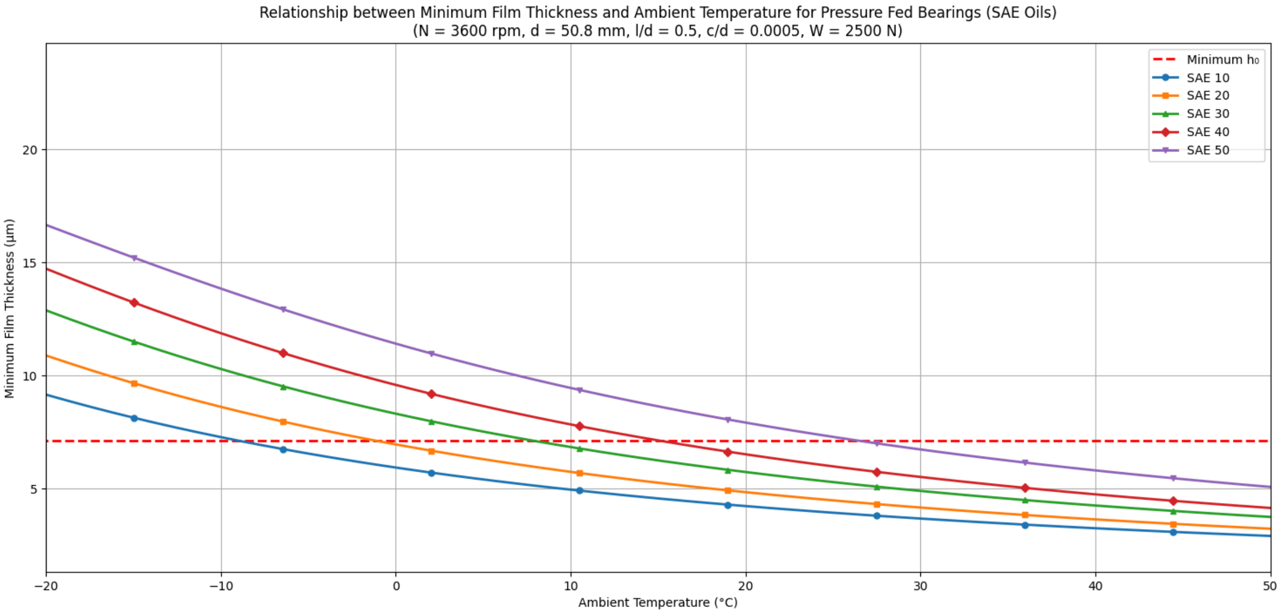


Fig 4.30:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (N = 3600 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

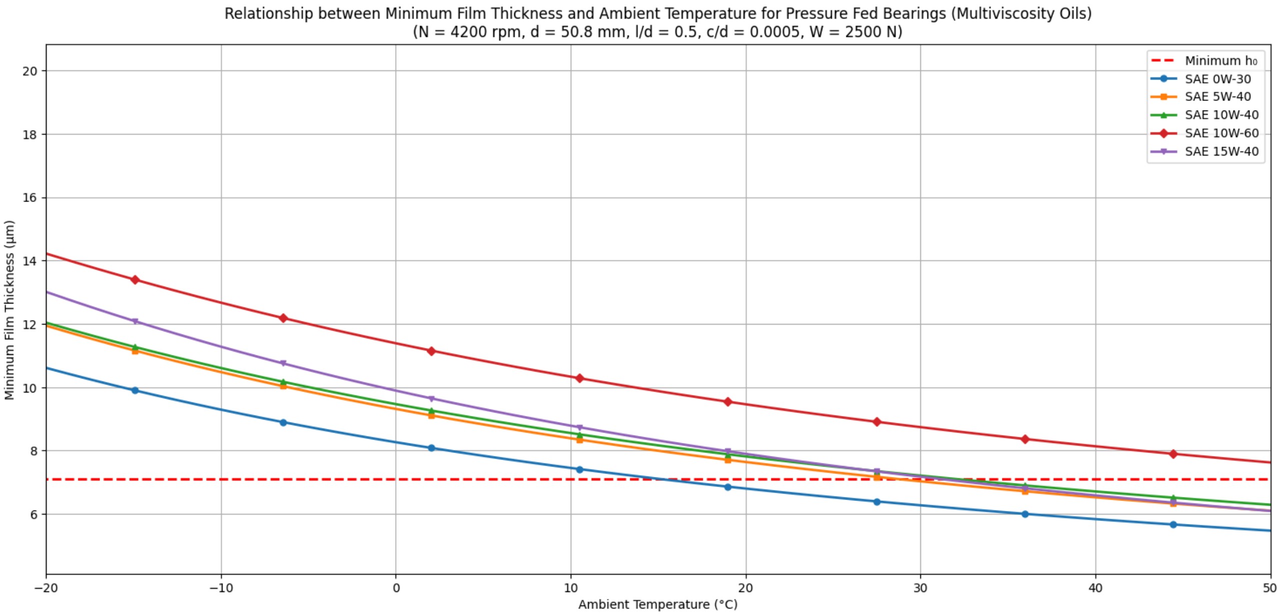


Fig 4.31:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (N = 4200 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

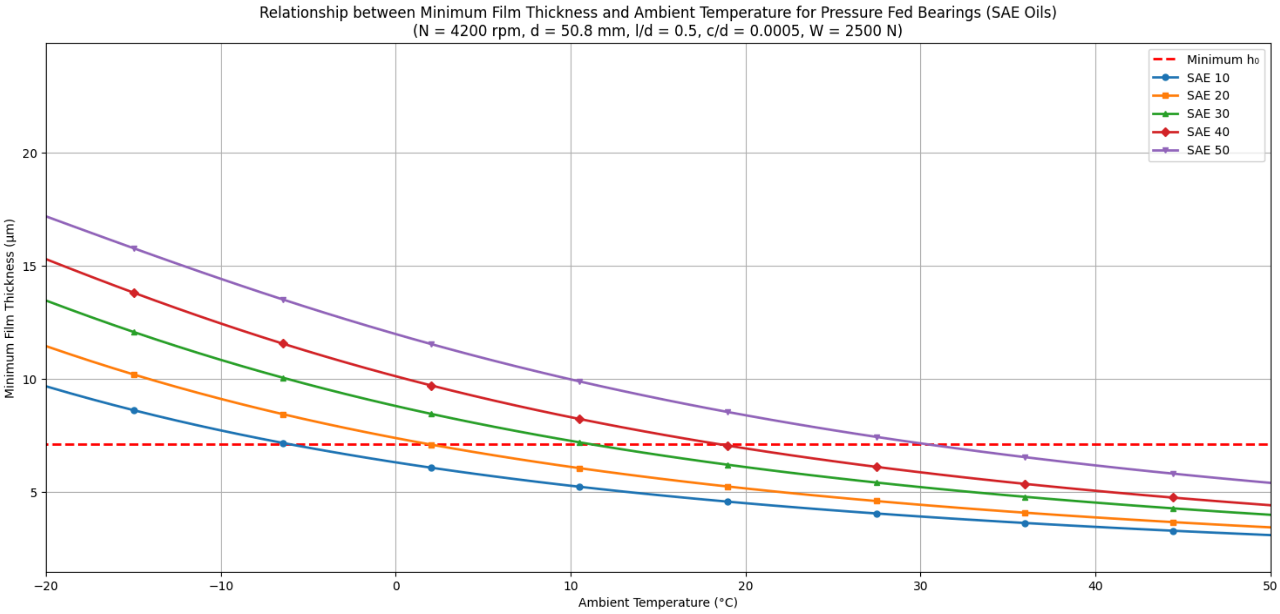


Fig 4.32:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (N = 4200 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

### 4.1.4 Effect of varying Clearance Ratio (Pressure Fed Bearing)

The clearance to diameter ratio is varied keeping journal diameter, length to diameter ratio, rotational speed and load constant in figure 4.33 to figure 4.42. We can see that minimum film thickness increases with increasing clearance to diameter ratio. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

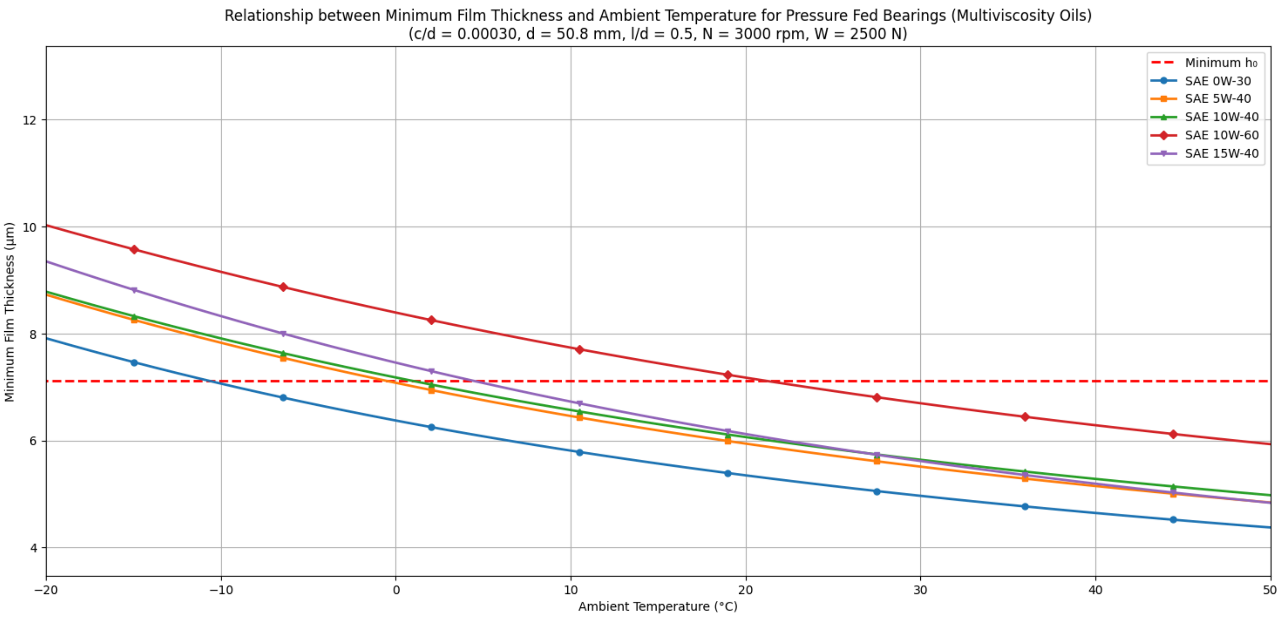


Fig 4.33:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (c/d = 0.00030, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

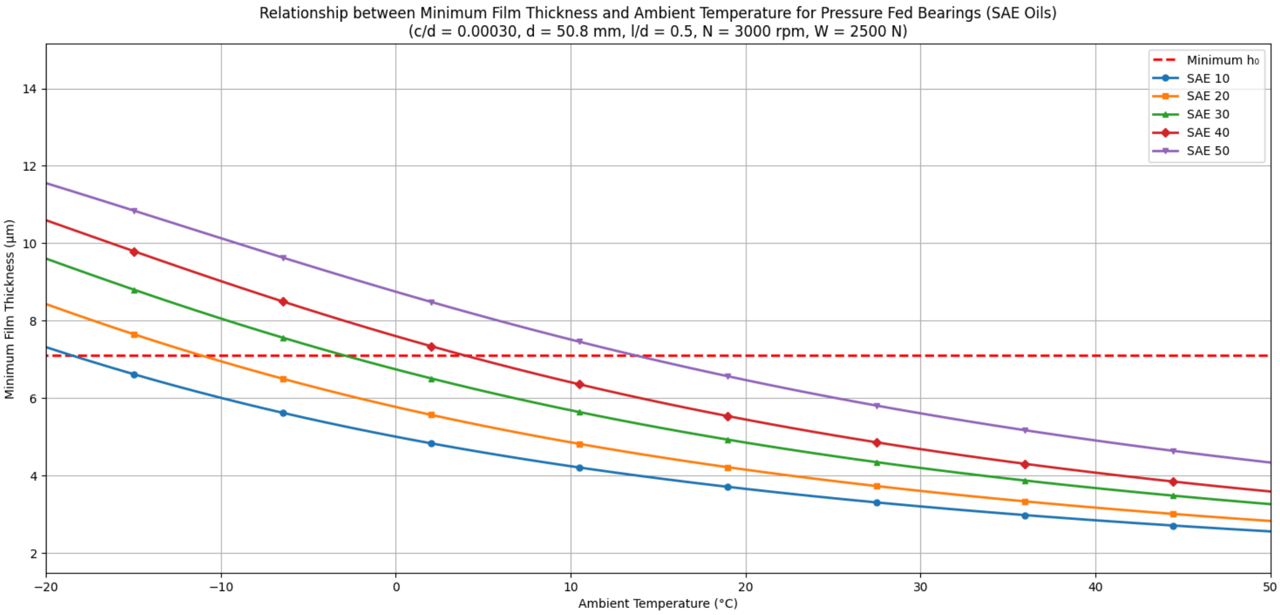


Fig 4.34:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (c/d = 0.00030, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

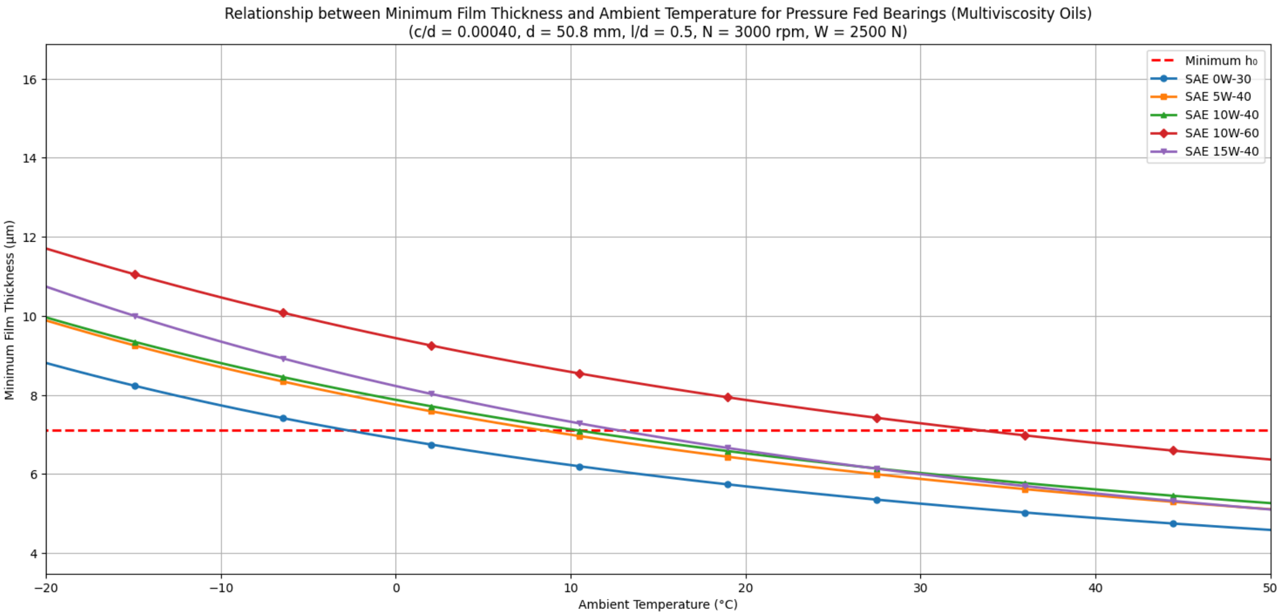


Fig 4.35:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (c/d = 0.00040, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

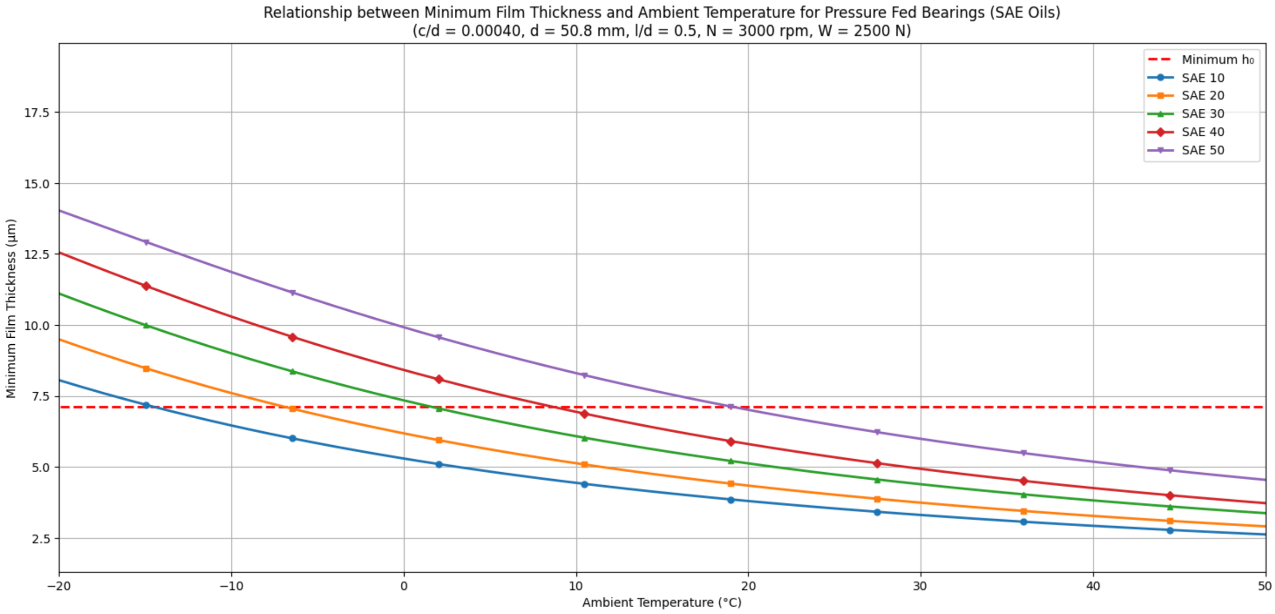


Fig 4.36:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (c/d = 0.00040, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

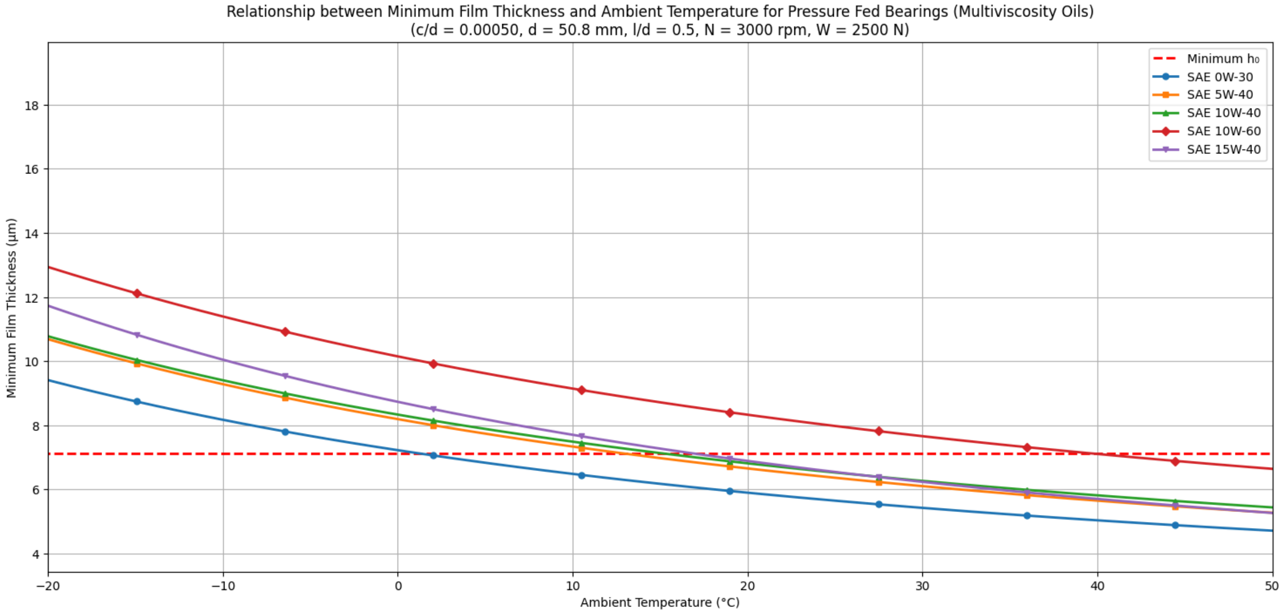


Fig 4.37:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (c/d = 0.00050, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

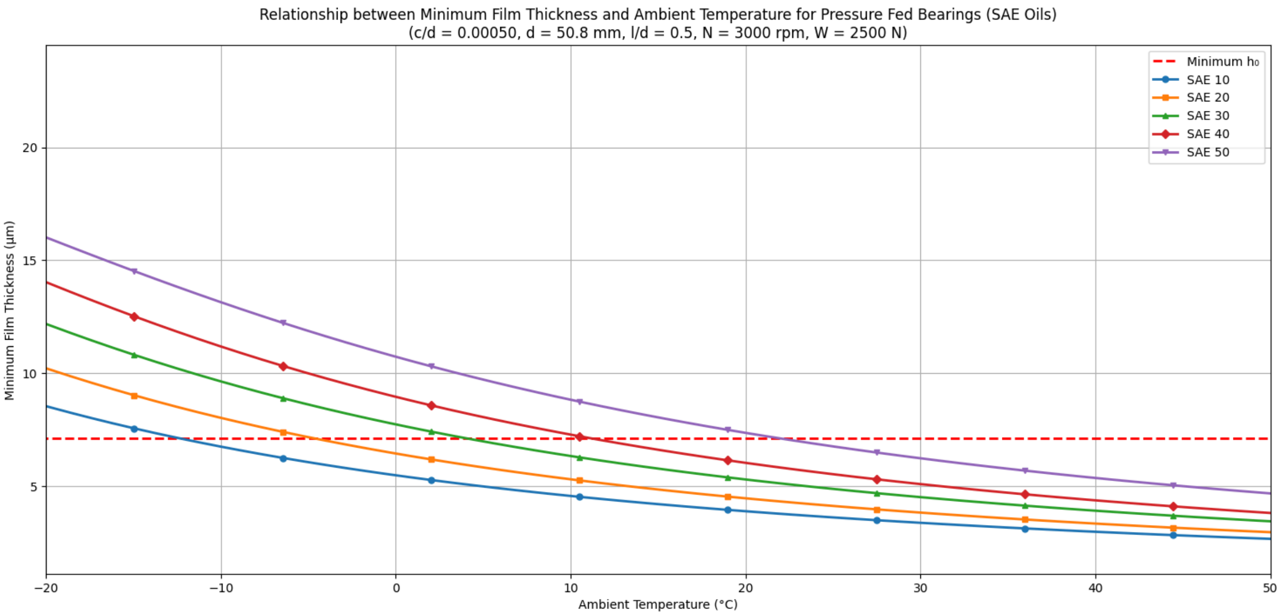


Fig 4.38:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (c/d = 0.00050, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

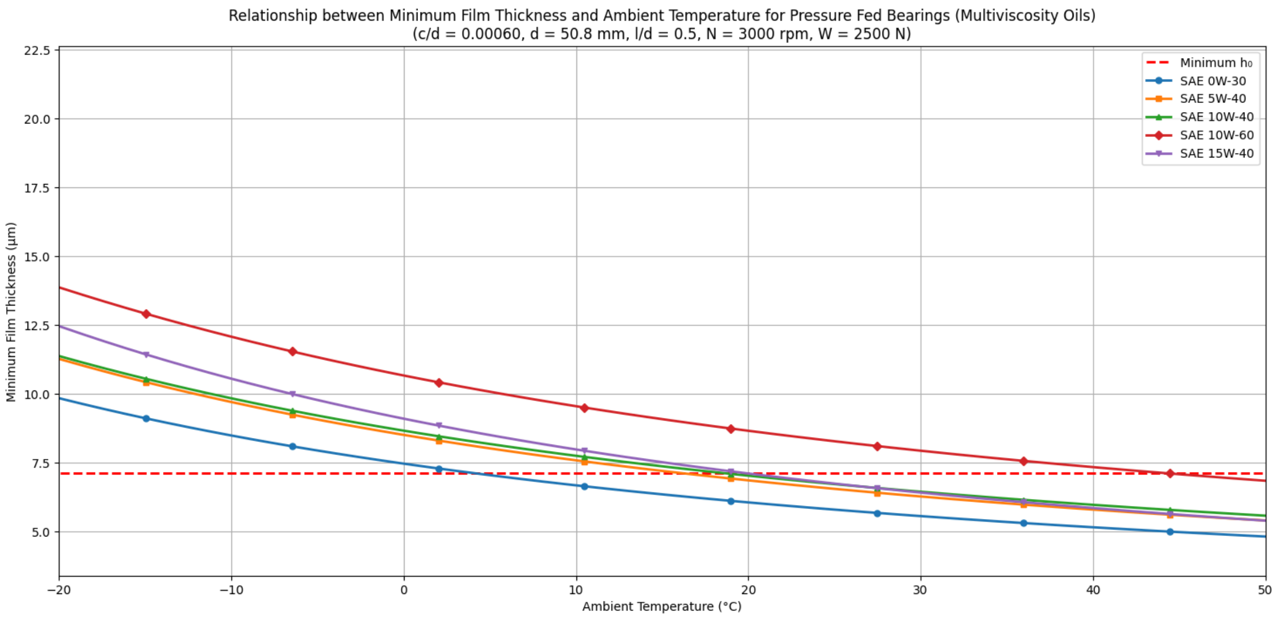


Fig 4.39:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (c/d = 0.00060, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

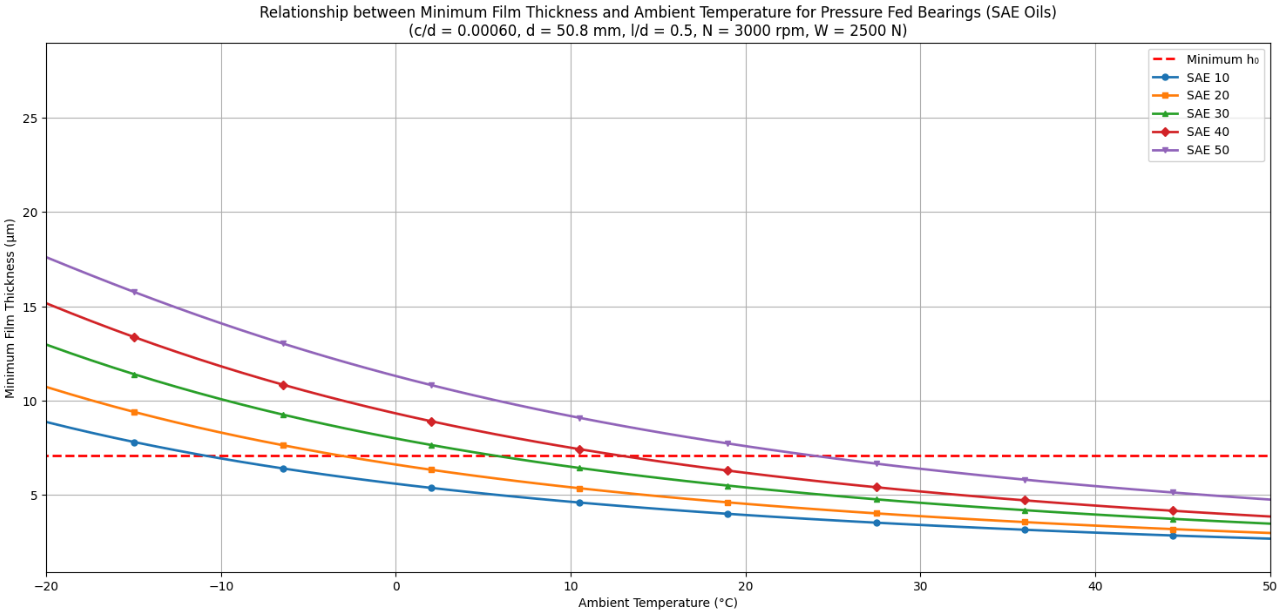


Fig 4.40:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (c/d = 0.00060, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

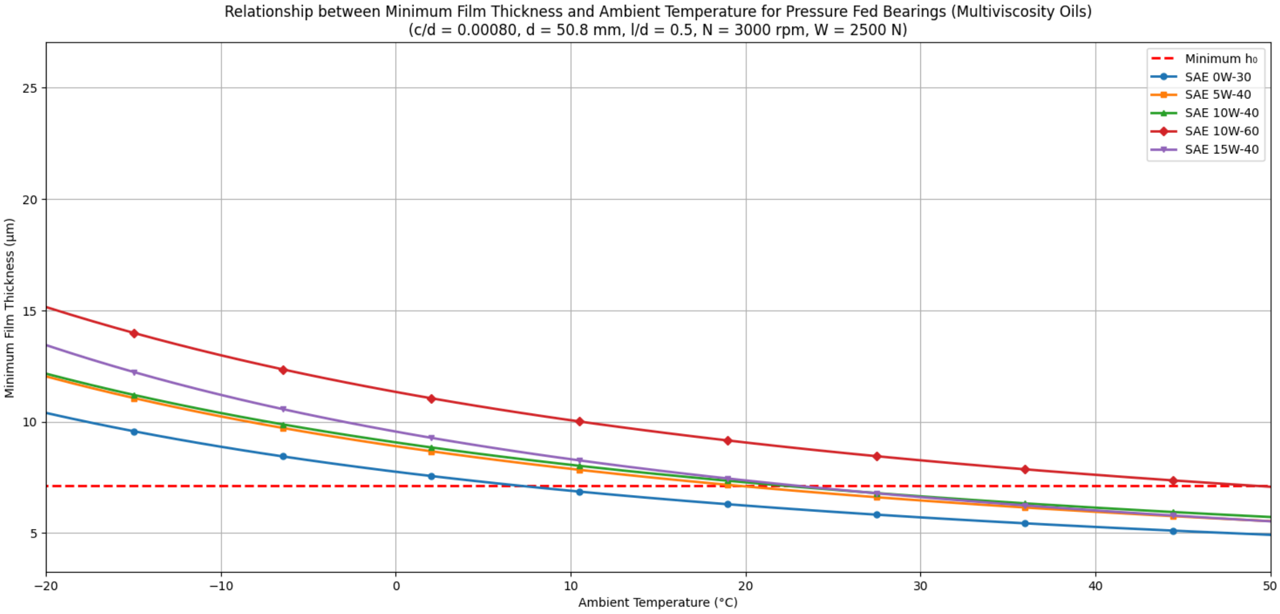


Fig 4.41:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (c/d = 0.00080, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

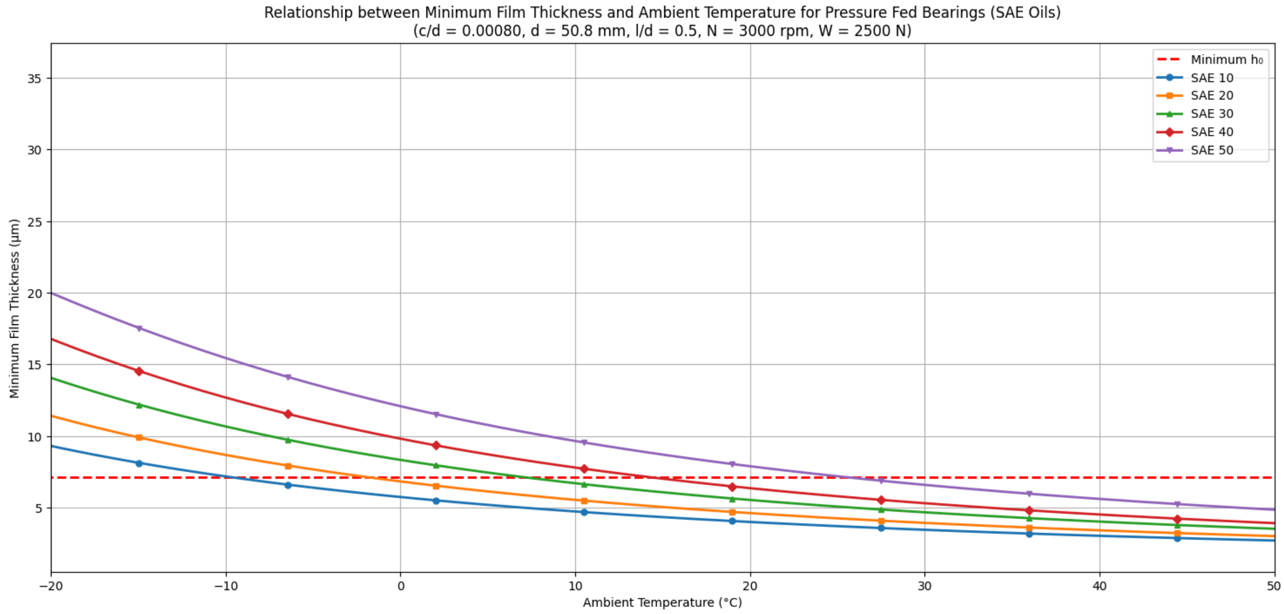


Fig 4.42:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (c/d = 0.00080, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

### 4.1.5 Effect of Bearing Load (Pressure Fed Bearing)

The load is varied keeping journal diameter, length to diameter ratio, rotational speed and clearance to diameter ratio constant in figure 4.43 to figure 4.54. We can see that minimum film thickness decreases with increasing load. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

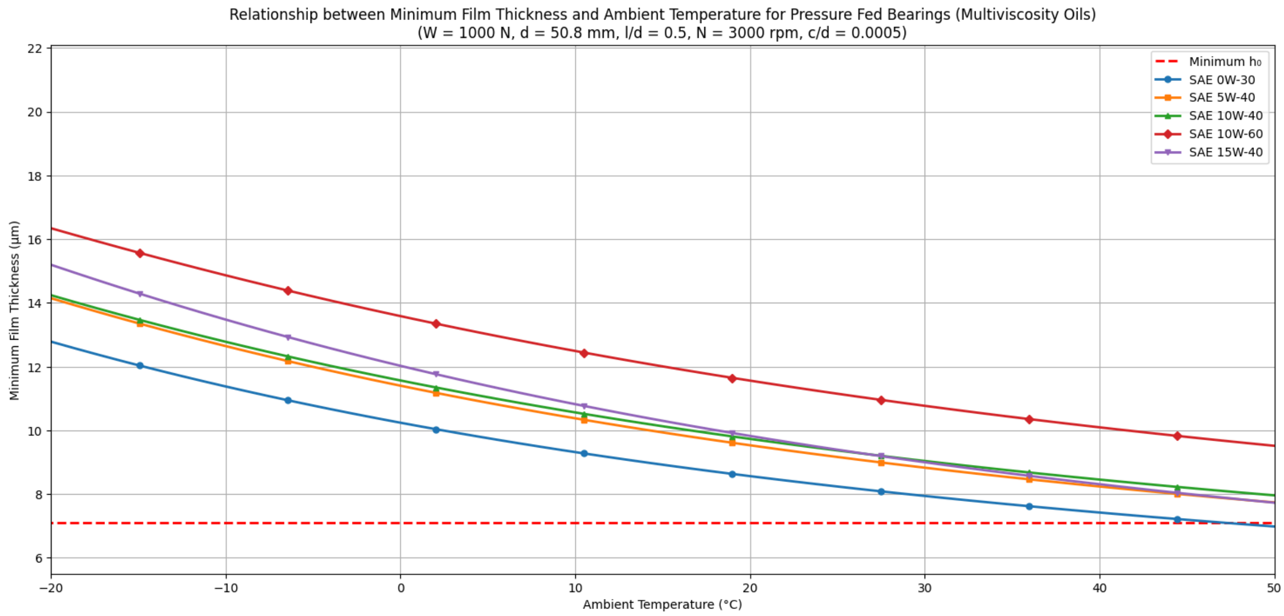


Fig 4.43:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (W = 1000 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

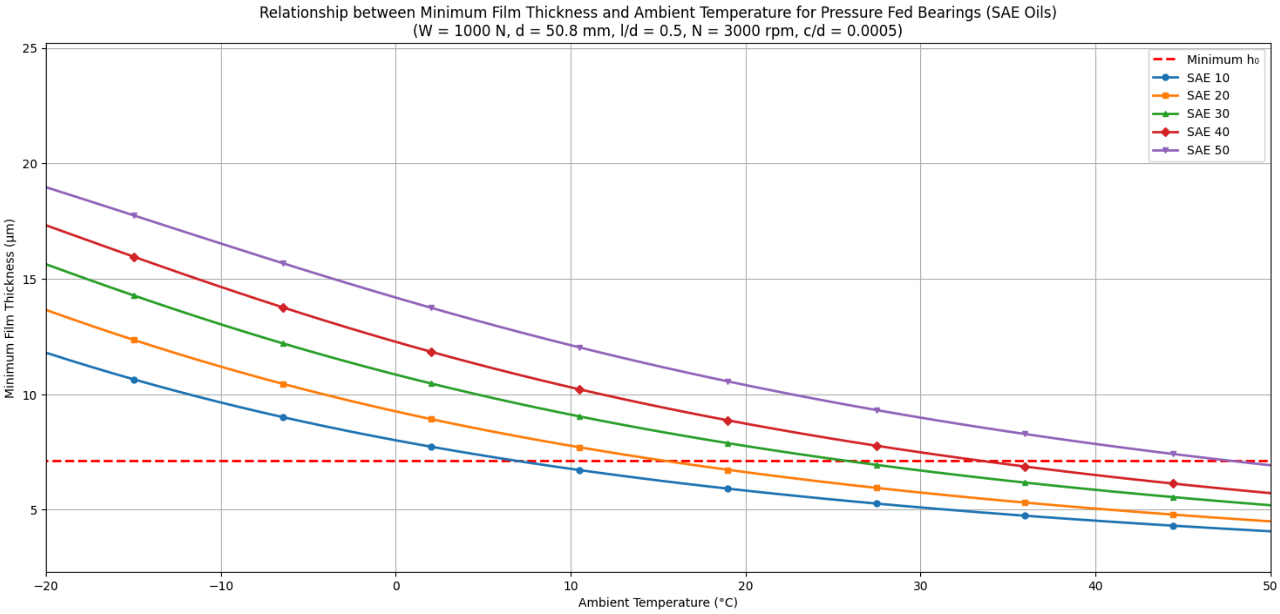


Fig 4.44:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (W = 1000 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

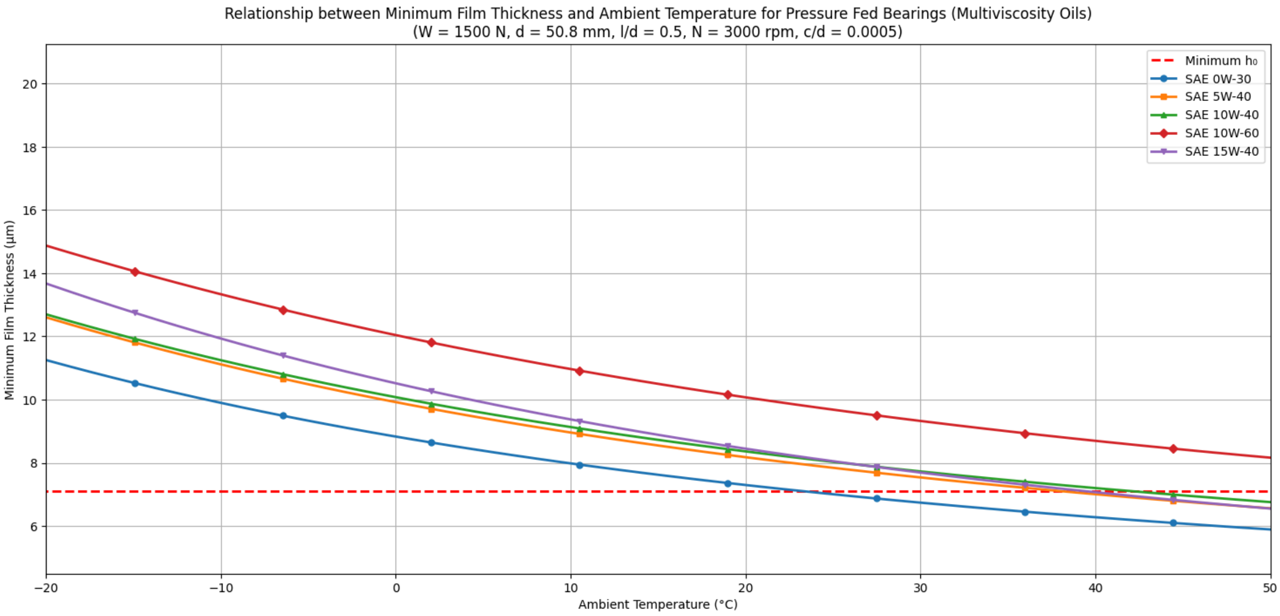


Fig 4.45:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (W = 1500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

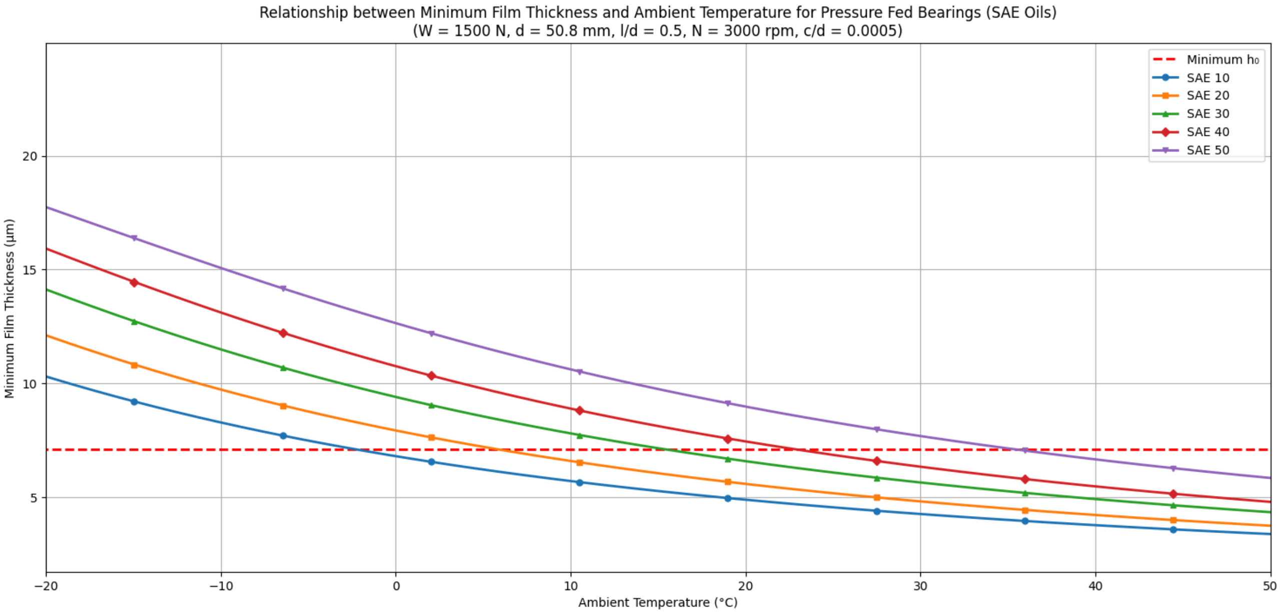


Fig 4.46:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (W = 1500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

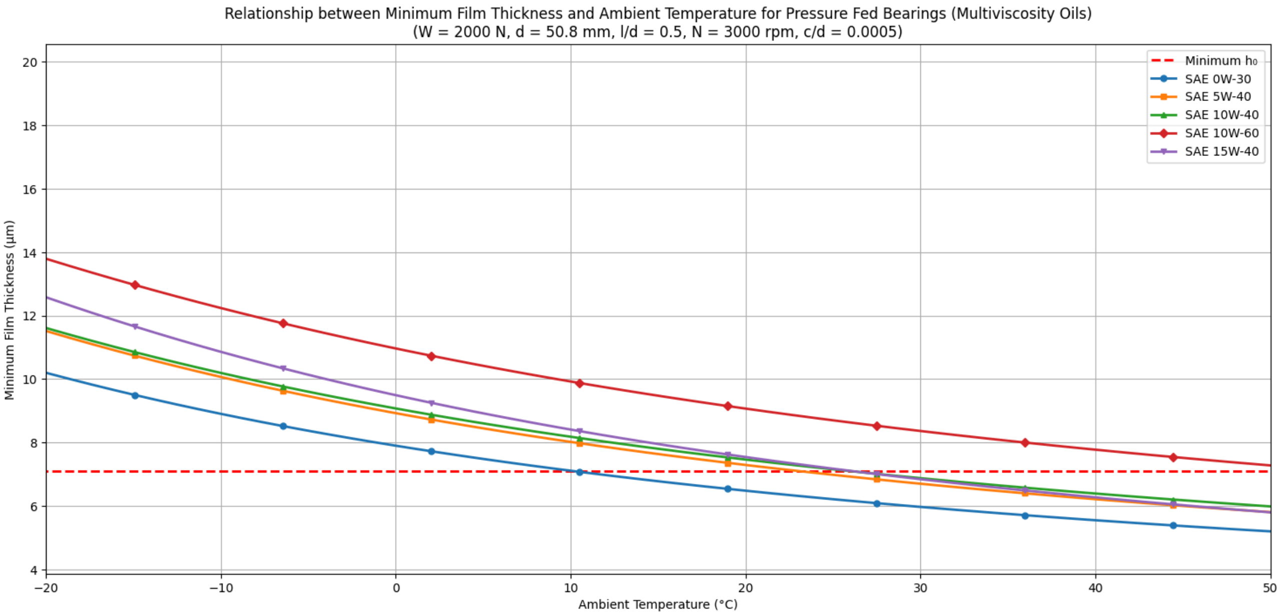


Fig 4.47:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (W = 2000 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

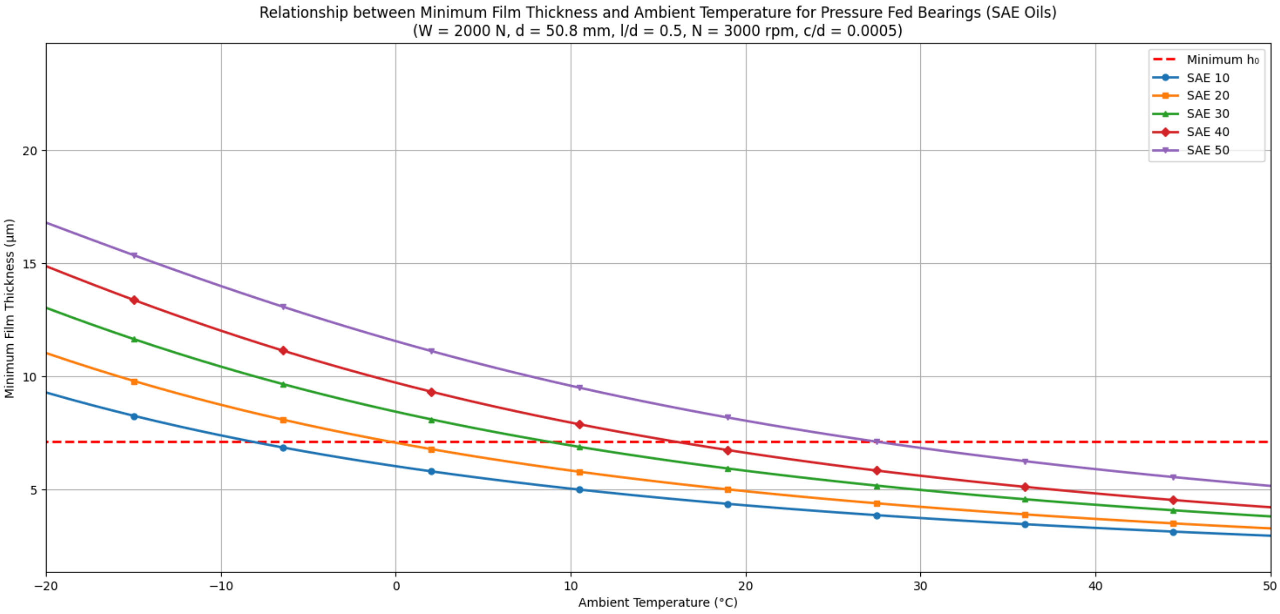


Fig 4.48:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (W = 2000 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

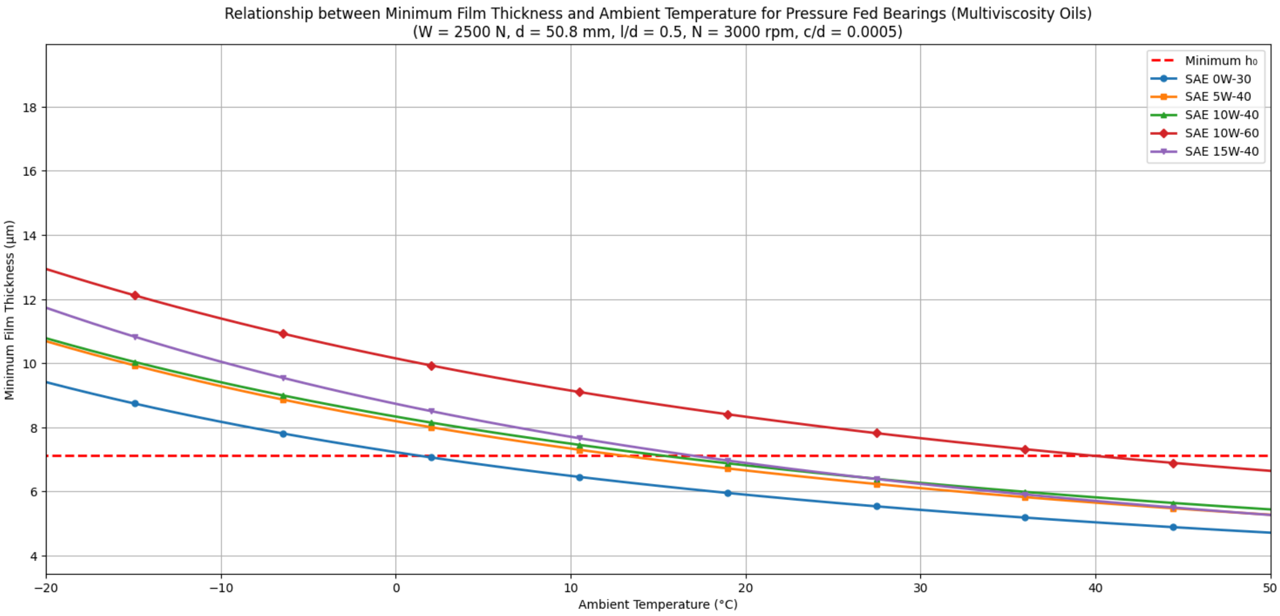


Fig 4.49:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (W = 2500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

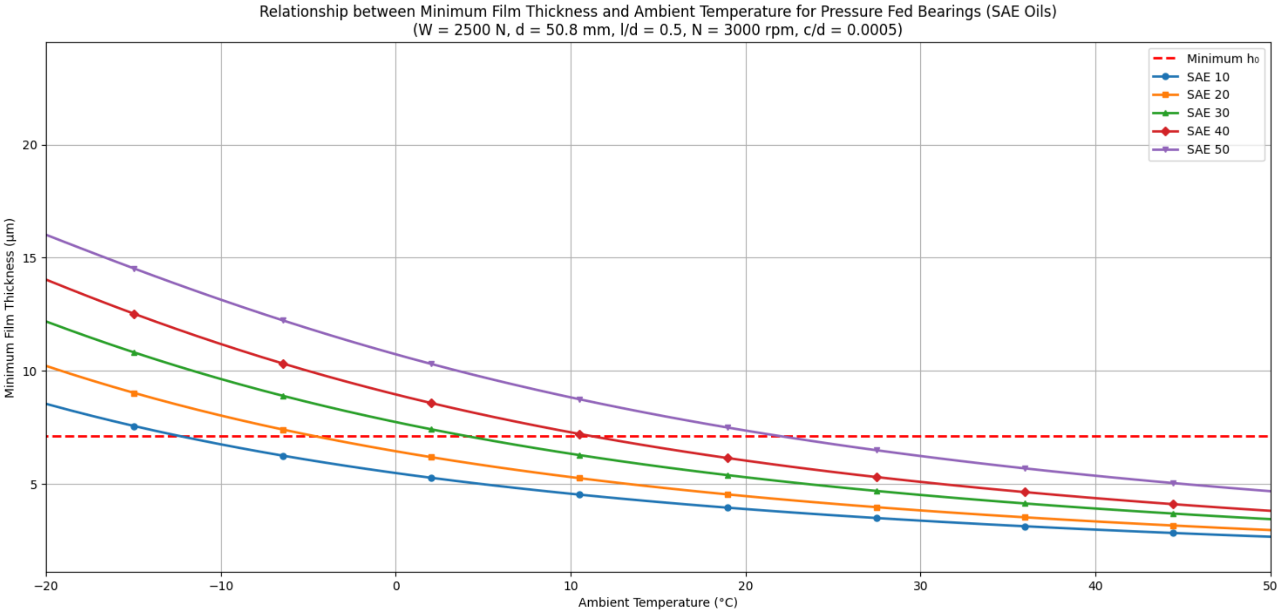


Fig 4.50:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (W = 2500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

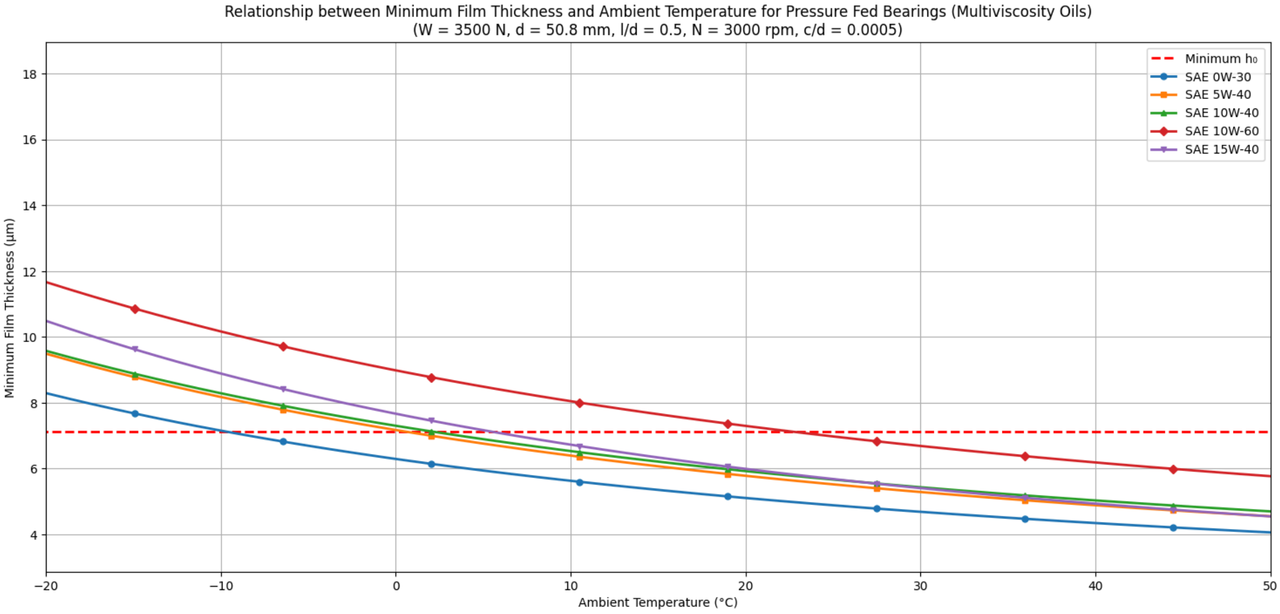


Fig 4.51:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (W = 3500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

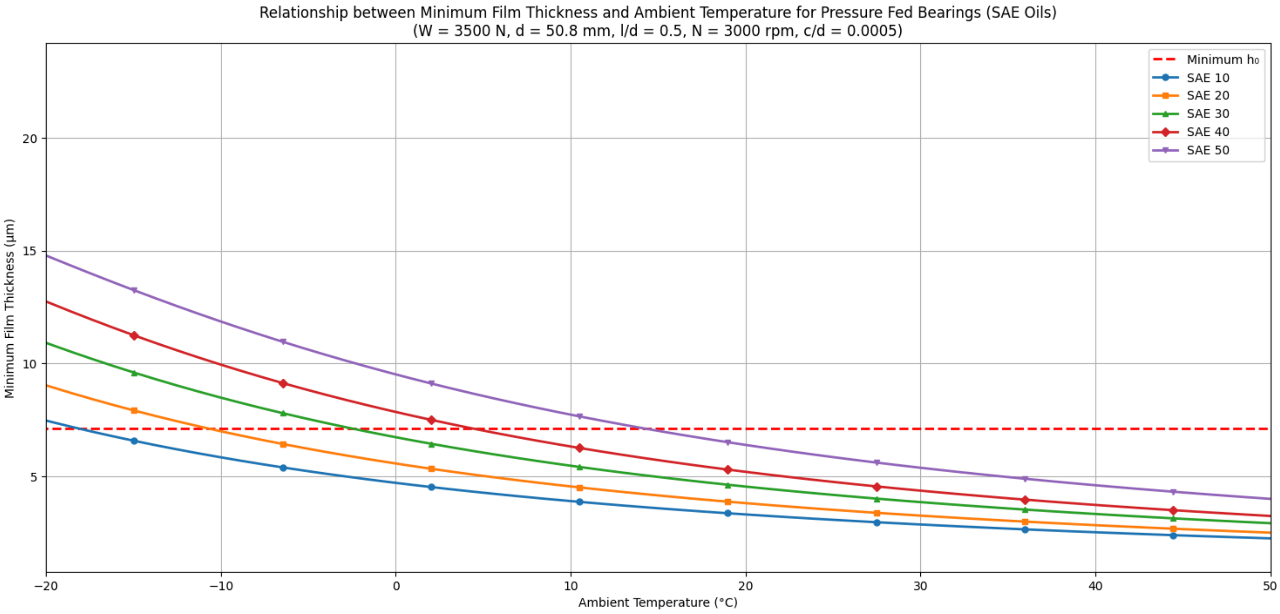


Fig 4.52:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (W = 3500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

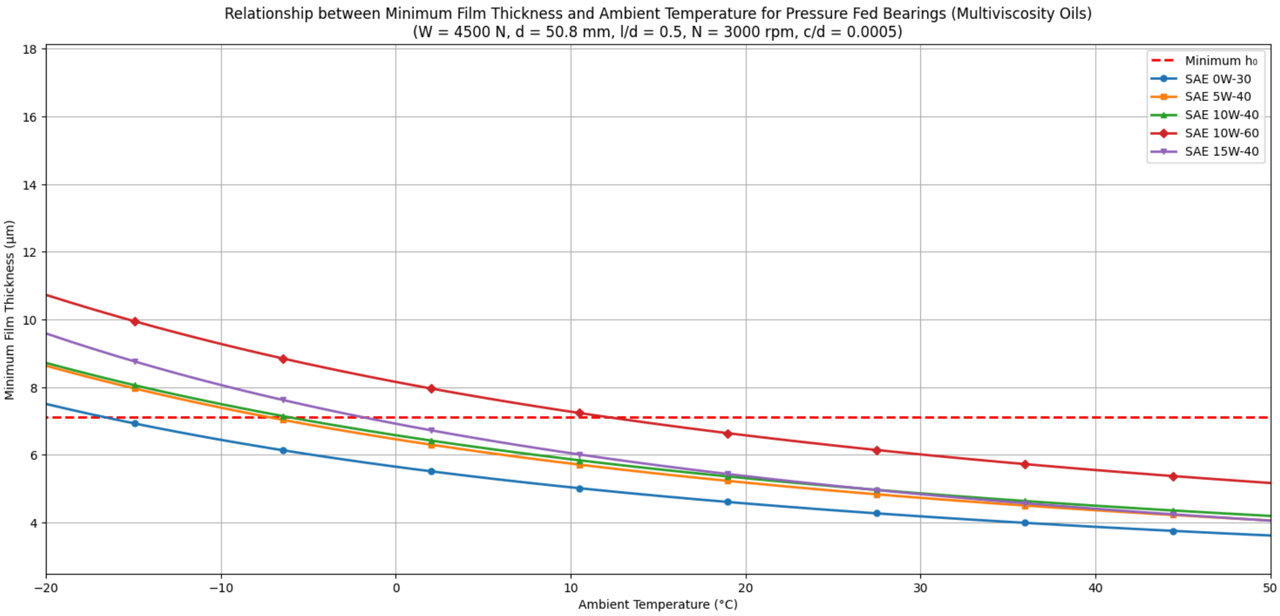


Fig 4.53:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (Multiviscosity Oils) — (W = 4500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

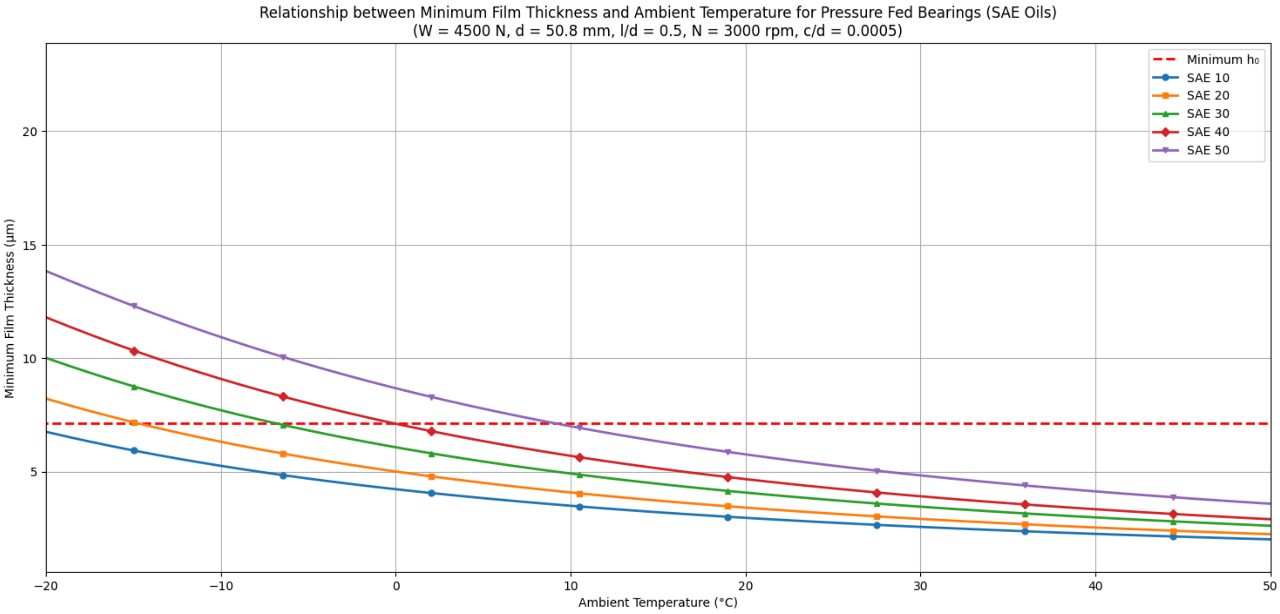


Fig 4.54:Relationship between Minimum Film Thickness and Ambient Temperature for Pressure Fed Bearings (SAE Oils) — (W = 4500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

### 4.1.6 Effect of Varying Journal Diameter (Natural Fed Bearing)

Figure 4.55 to figure 4.108 represents the performance of natural fed bearings for different multiviscosity and SAE oils. The journal diameter is varied keeping length to diameter ratio, rotational speed, clearance to diameter ratio and load constant in figure 4.55 to figure 4.64. We can see that minimum film thickness increases with increasing journal diameter. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

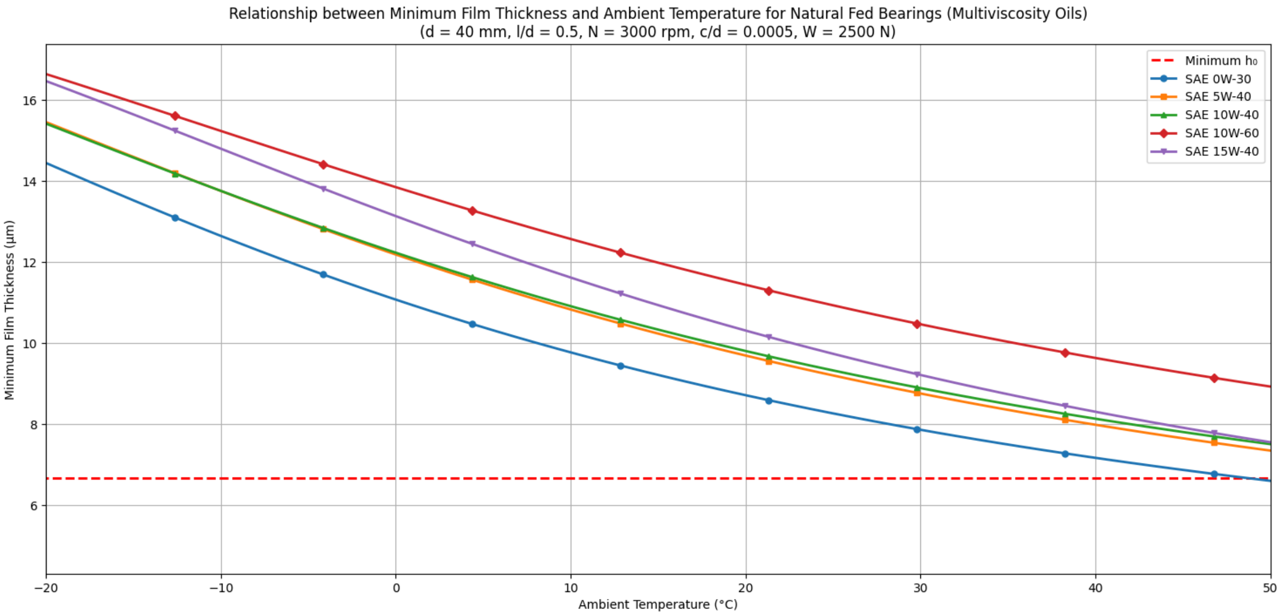

Fig 4.55:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (d = 40 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

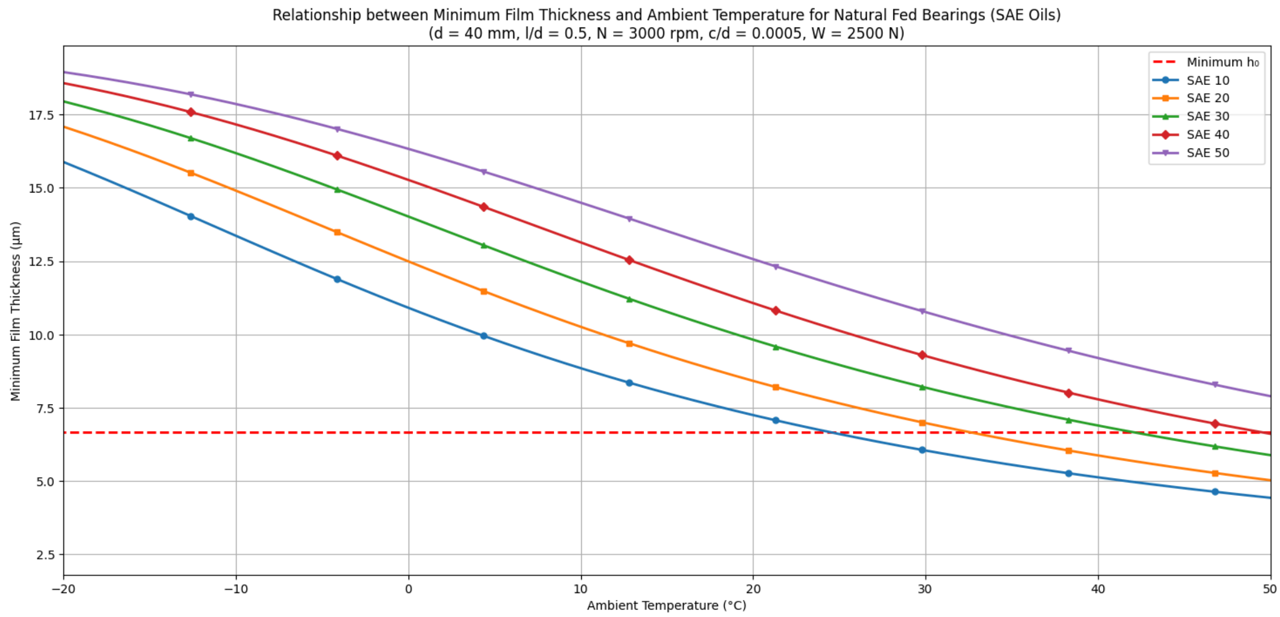


Fig 4.56:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (d = 40 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

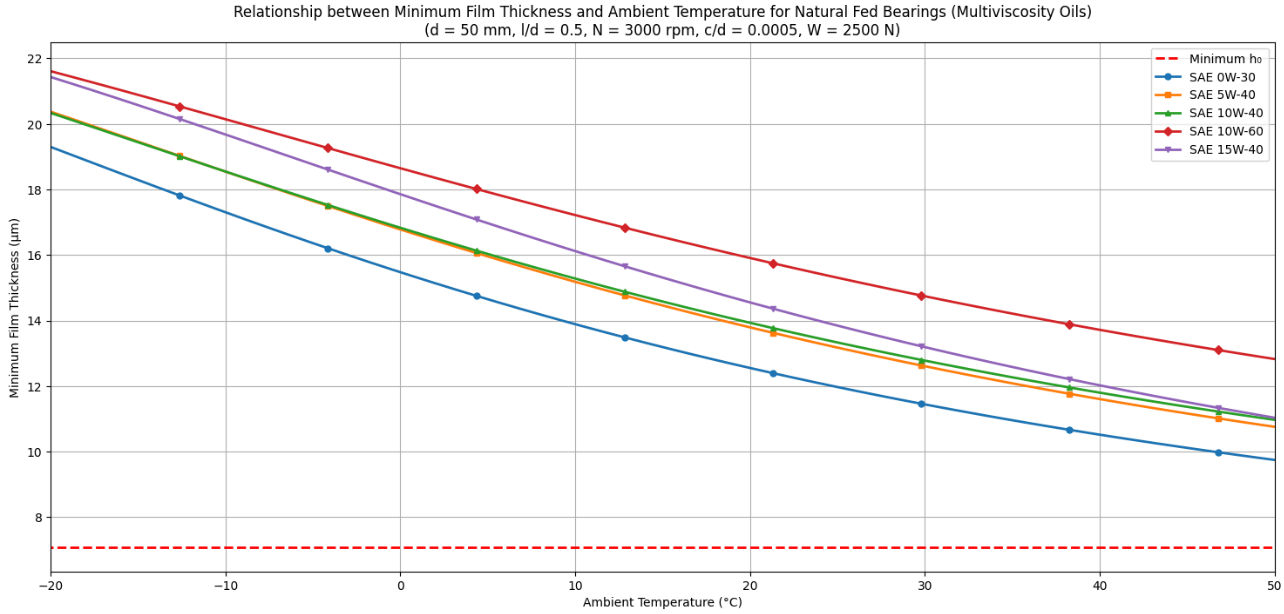


Fig 4.57:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (d = 50 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

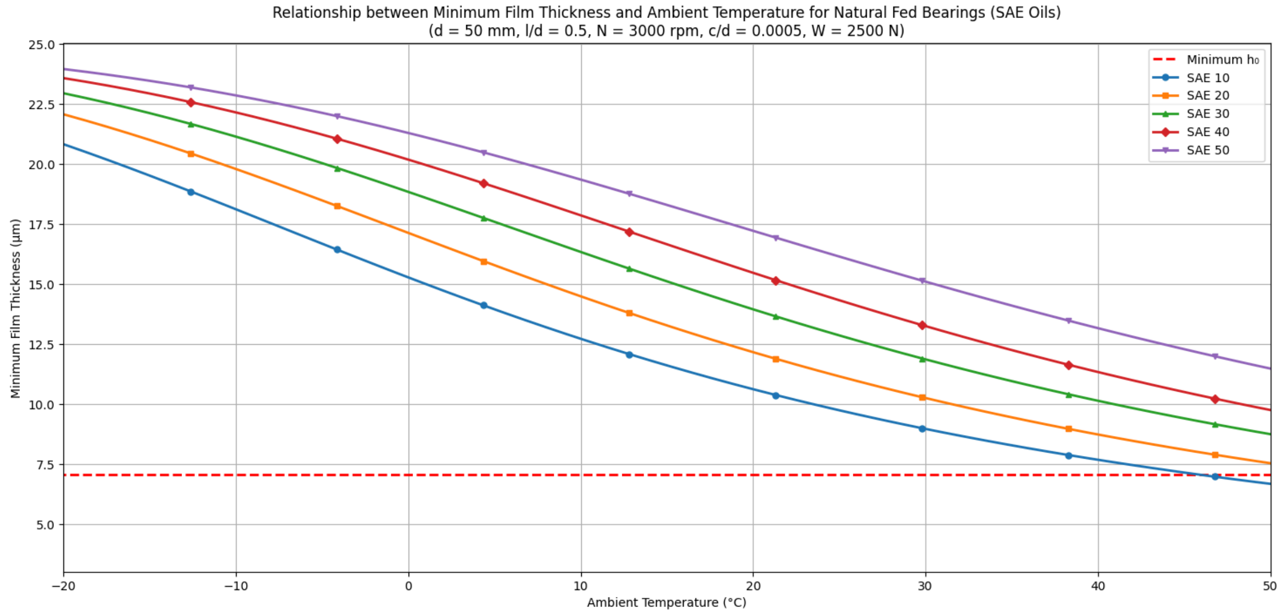


Fig 4.58:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (d = 50 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

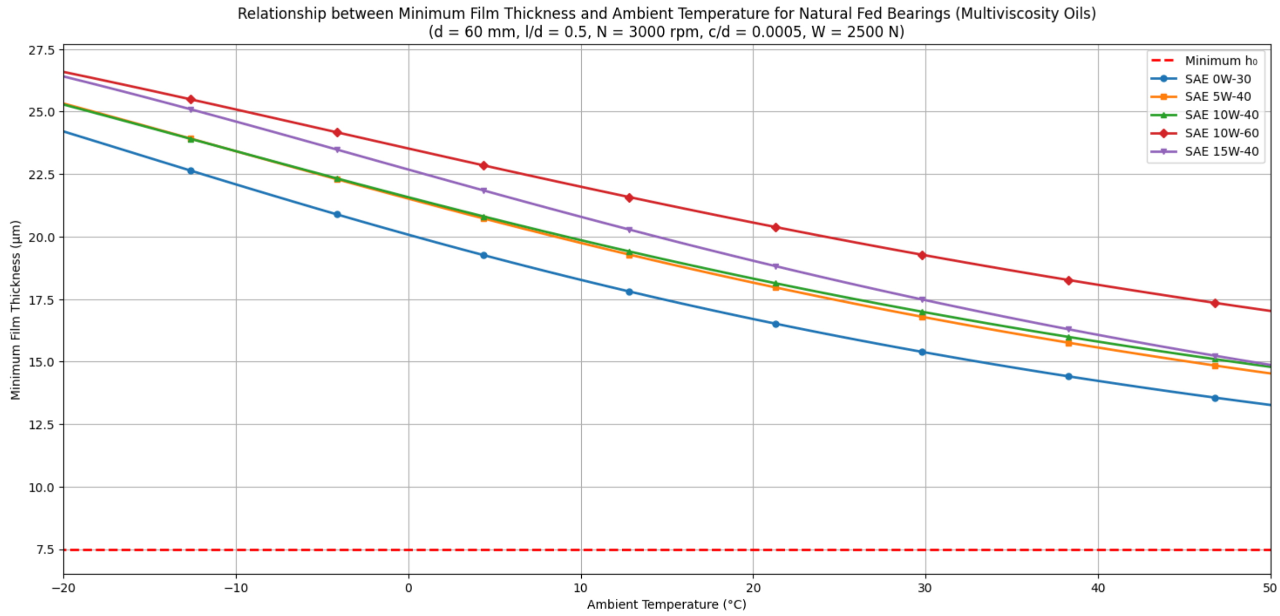


Fig 4.59:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (d = 60 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

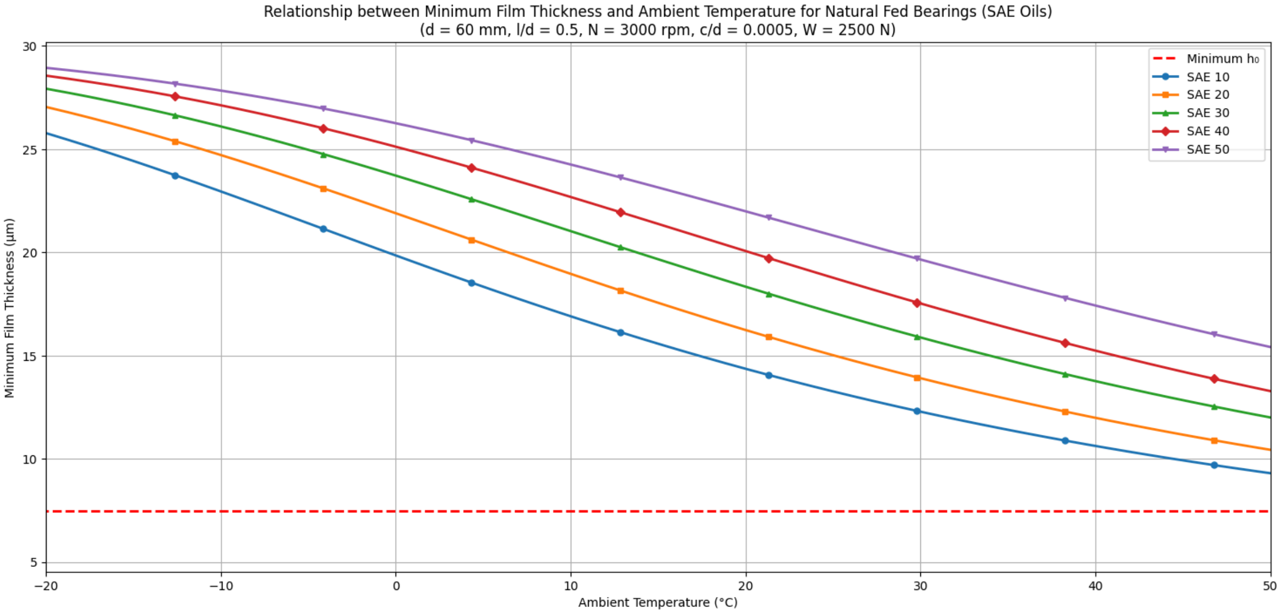


Fig 4.60:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (d = 60 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

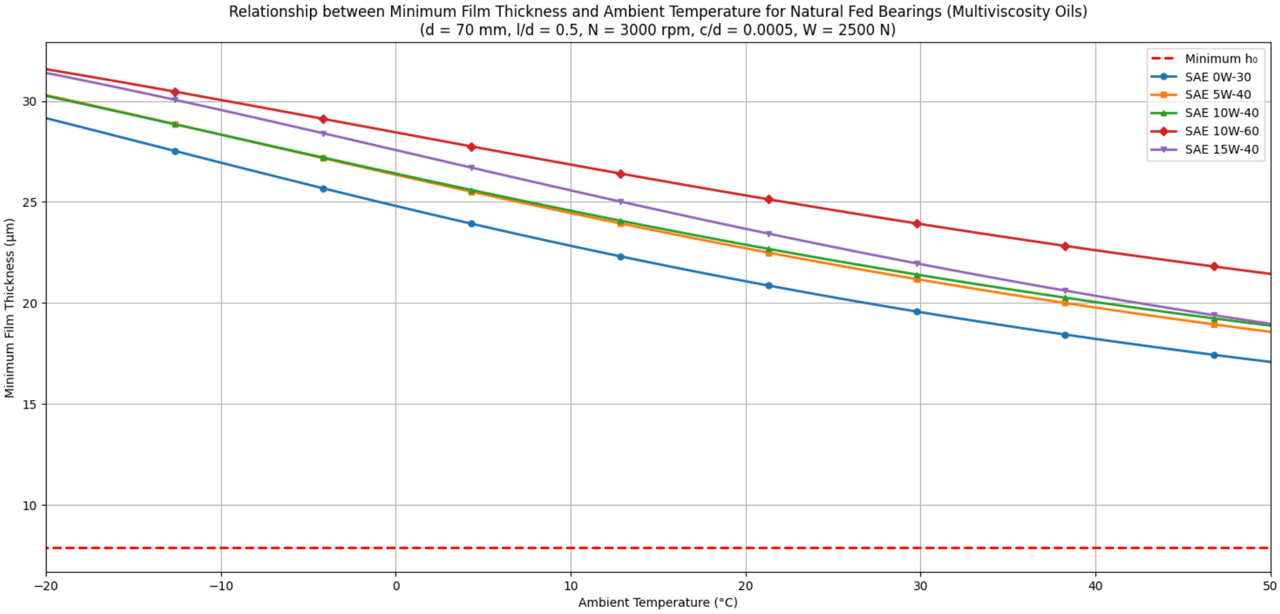


Fig 4.61:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (d = 70 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

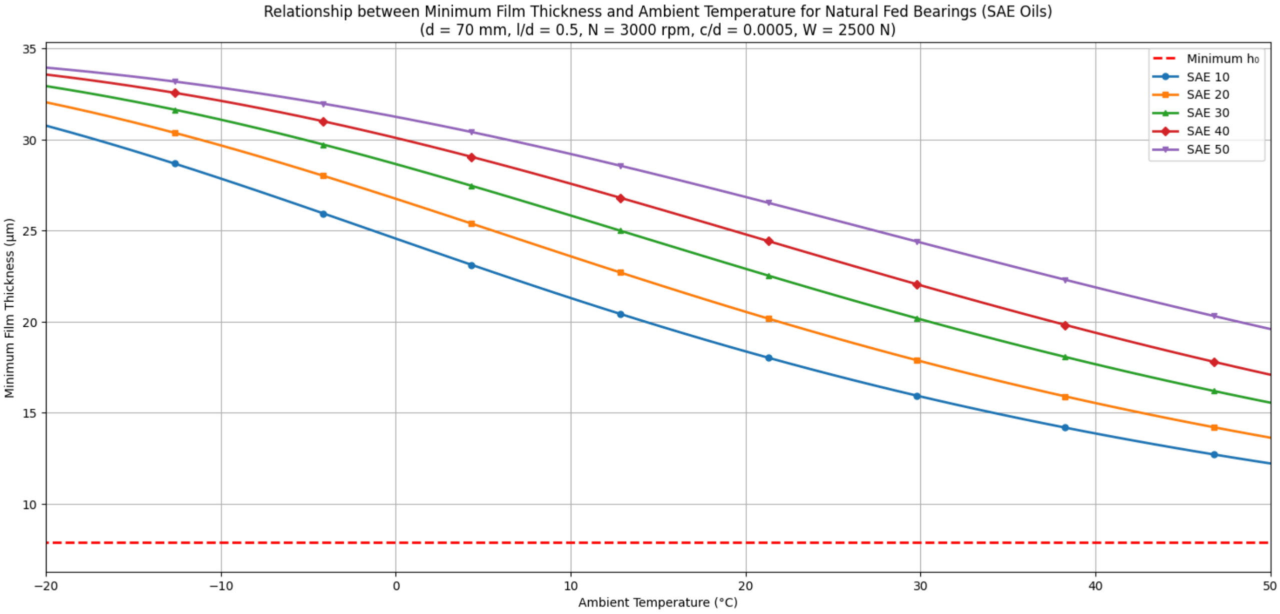


Fig 4.62:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (d = 70 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

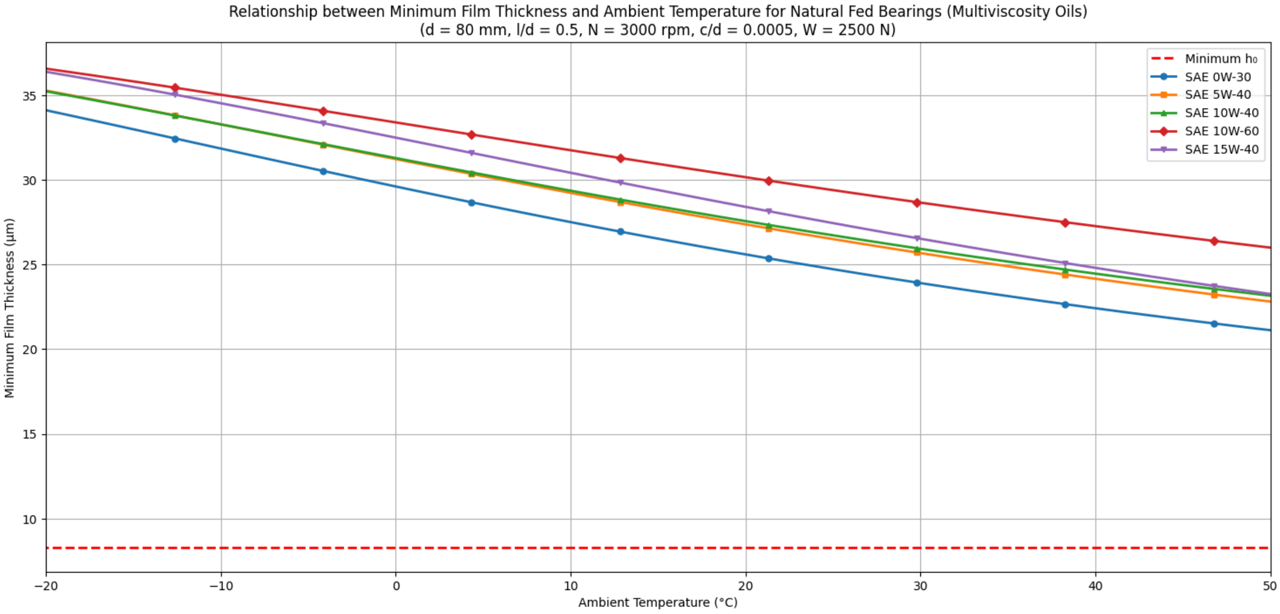


Fig 4.63:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (d = 80 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

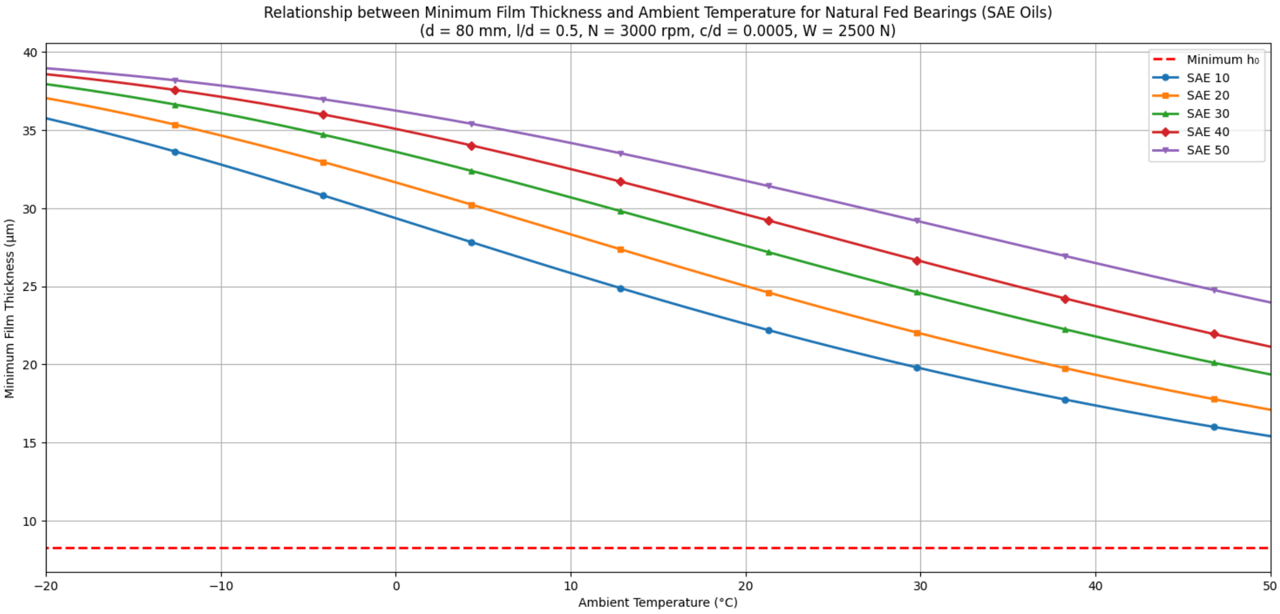


Fig 4.64:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (d = 80 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

## 4.1.7 Effect of Varying Length to Diameter Ratio (Natural Fed Bearing)

The length to diameter ratio is varied keeping journal diameter, rotational speed, clearance to diameter ratio and load constant in figure 4.65 to figure 4.72. We can see that minimum film thickness increases with increasing length to diameter ratio. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

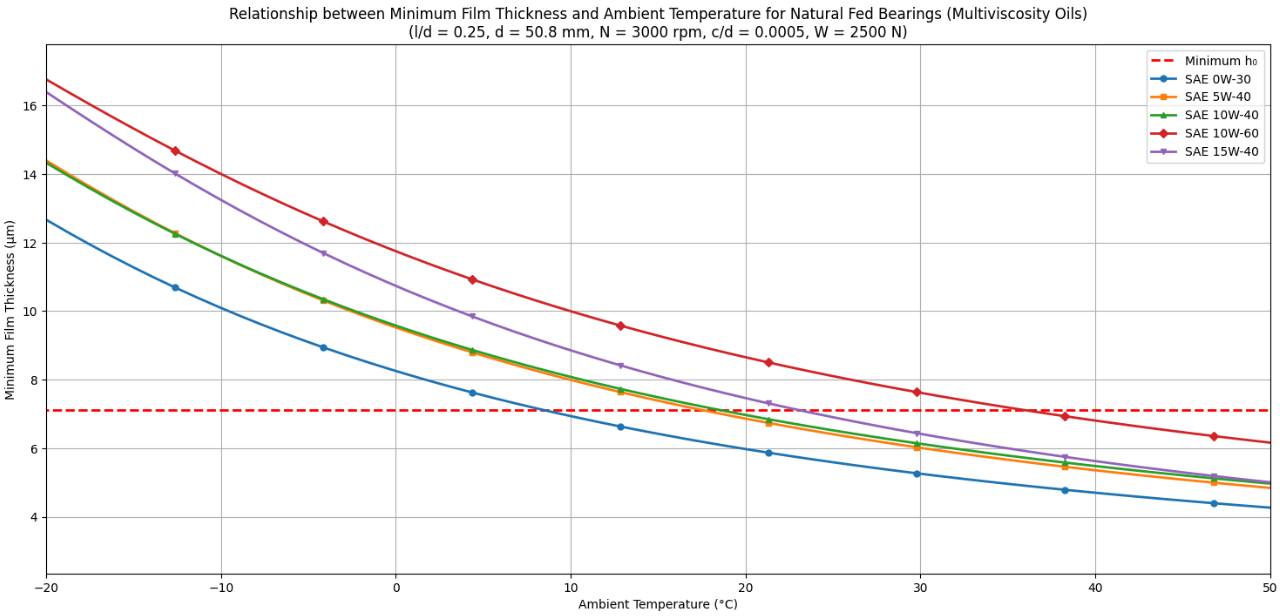


Fig 4.65:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (l/d = 0.25, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

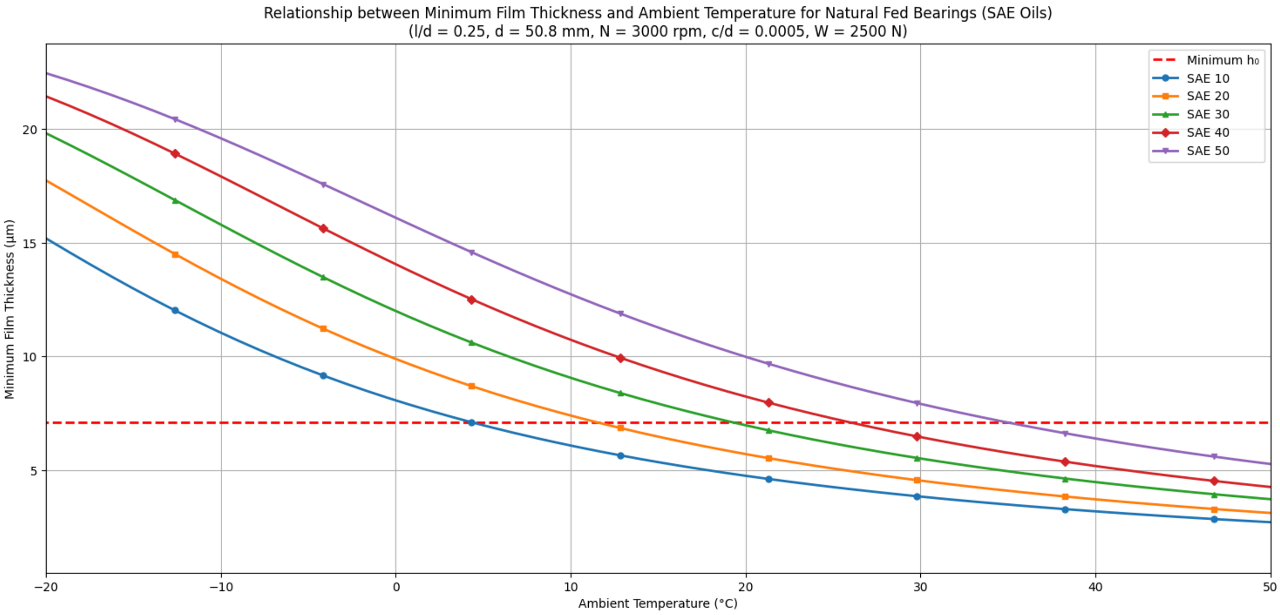


Fig 4.66:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (l/d = 0.25, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

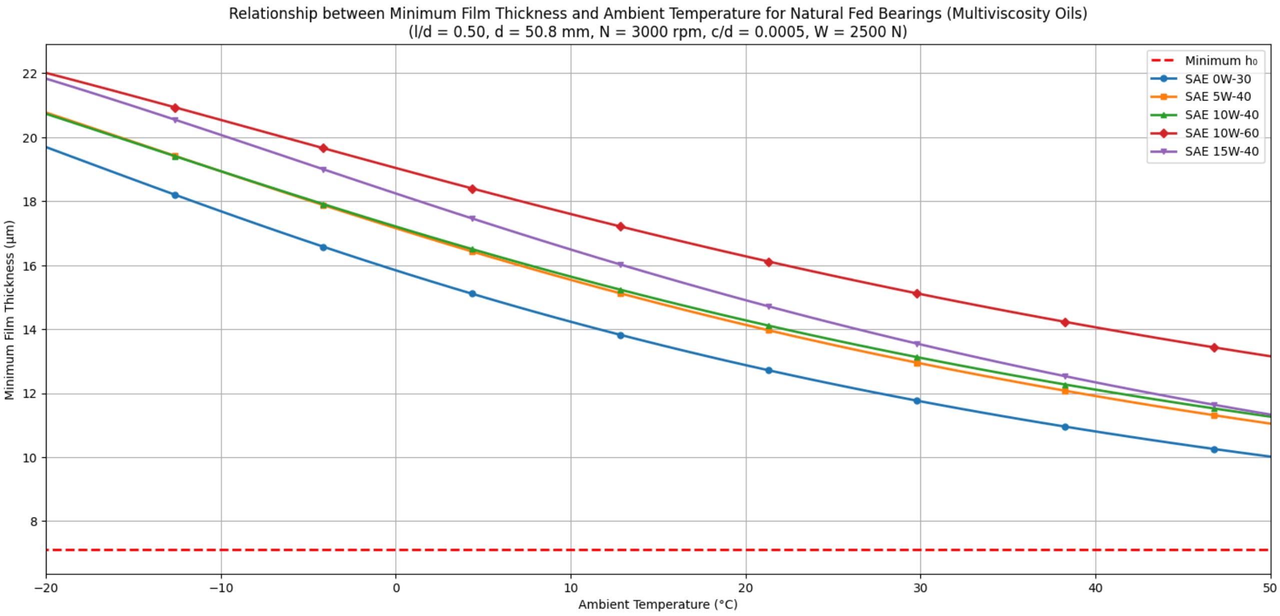


Fig 4.67:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (l/d = 0.50, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

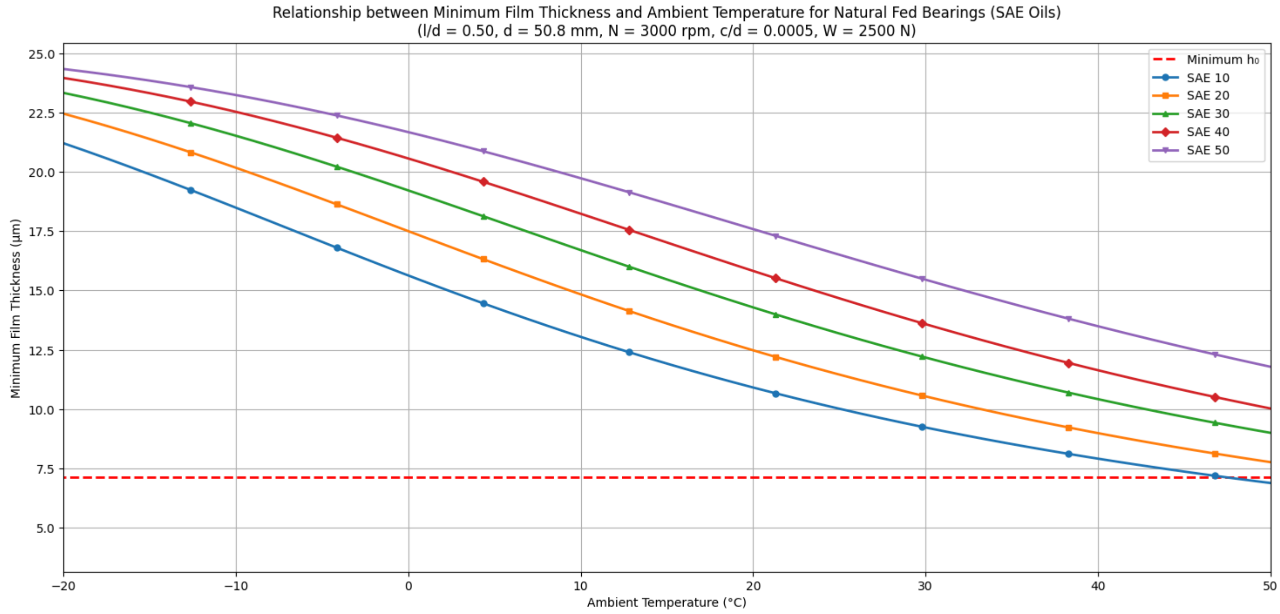


Fig 4.68:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (l/d = 0.50, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

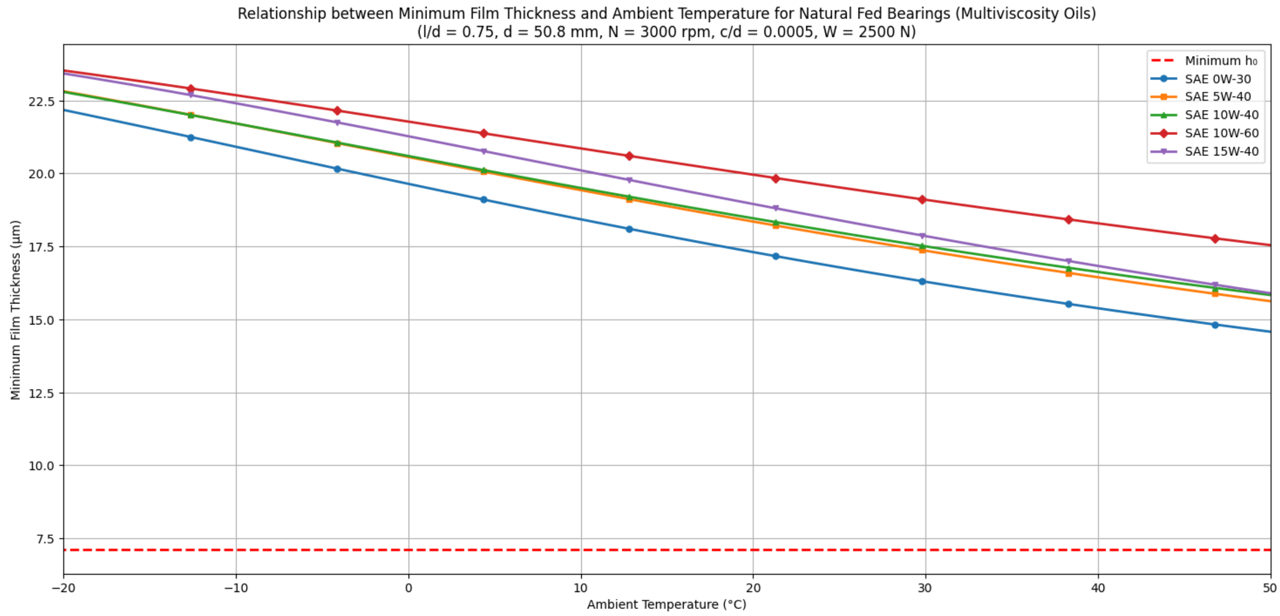


Fig 4.69:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (l/d = 0.75, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

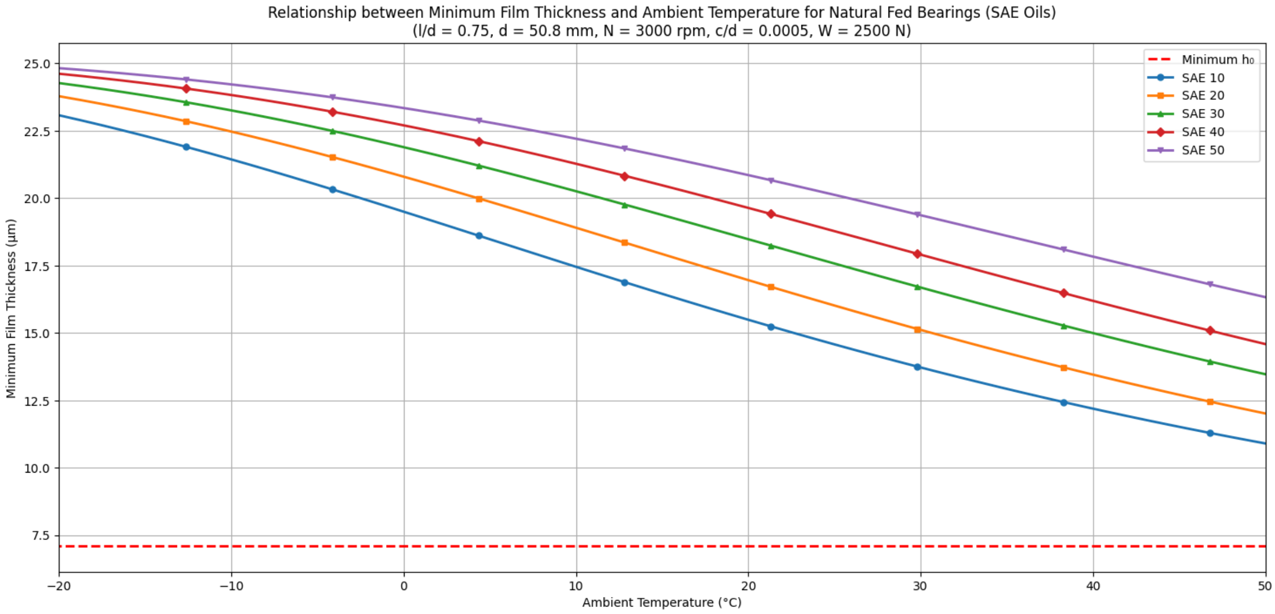


Fig 4.70:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (l/d = 0.75, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

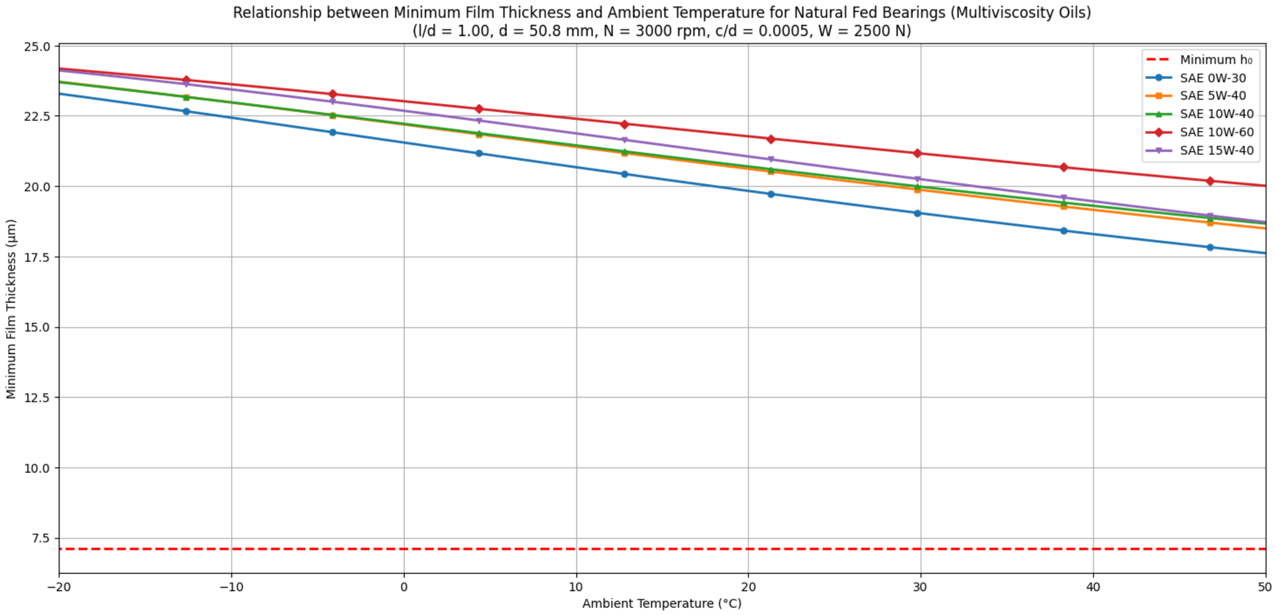


Fig 4.71:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (l/d = 1.00, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

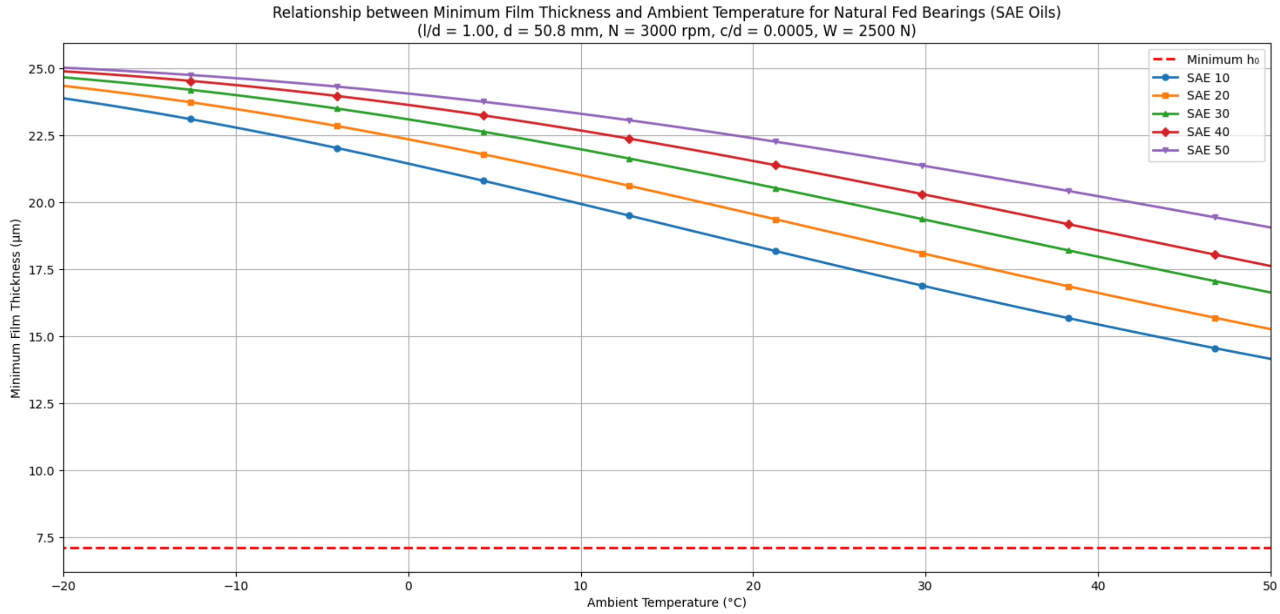


Fig 4.72:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (l/d = 1.00, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

## 4.1.8 Effect of Varying Rotational Speed (Natural Fed Bearing)

The rotational speed is varied keeping journal diameter, length to diameter ratio, clearance to diameter ratio and load constant in figure 4.73 to figure 4.86. We can see that minimum film thickness increases with increasing rotational speed. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

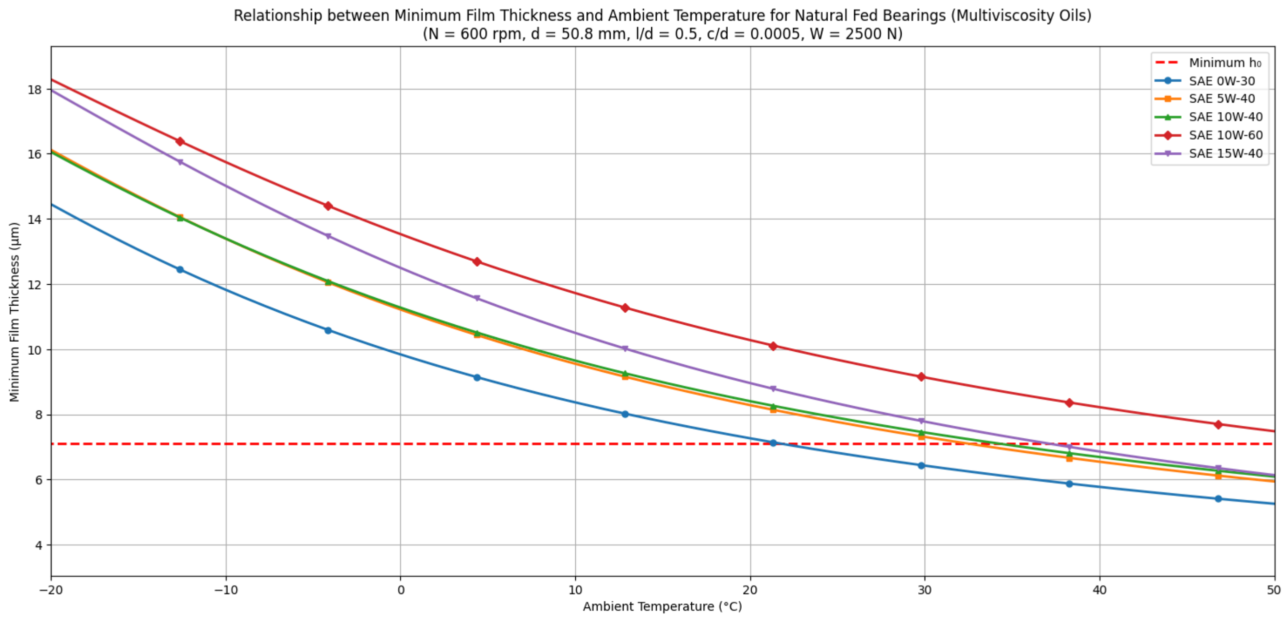


Fig 4.73:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (N = 600 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

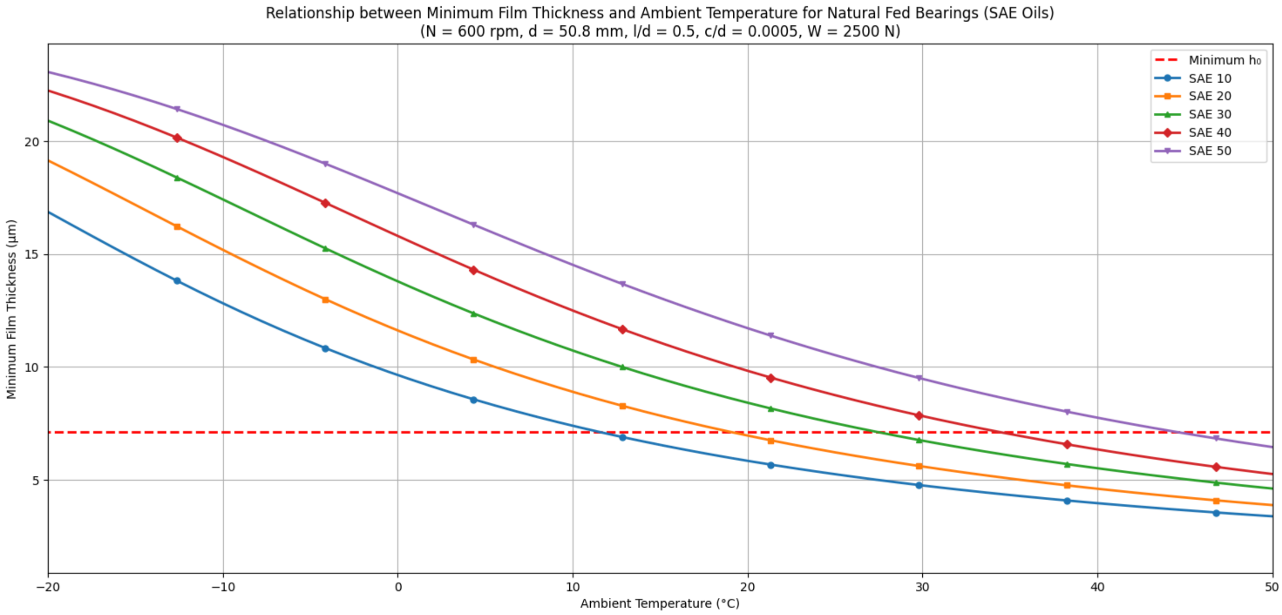


Fig 4.74:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (N = 600 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

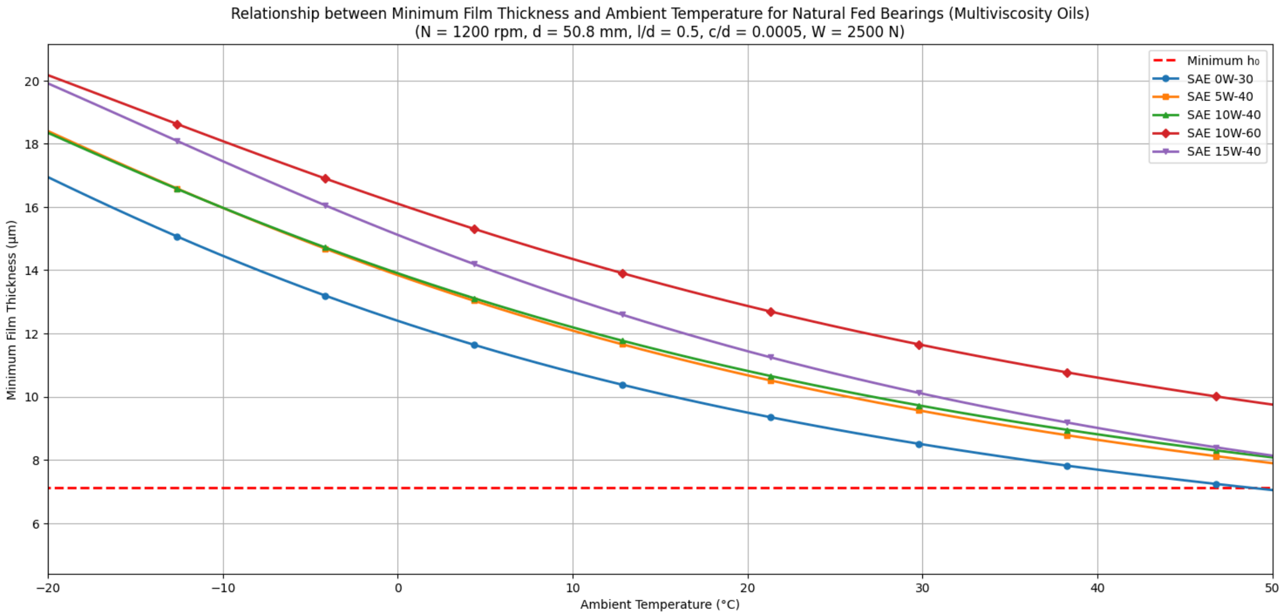


Fig 4.75:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (N = 1200 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

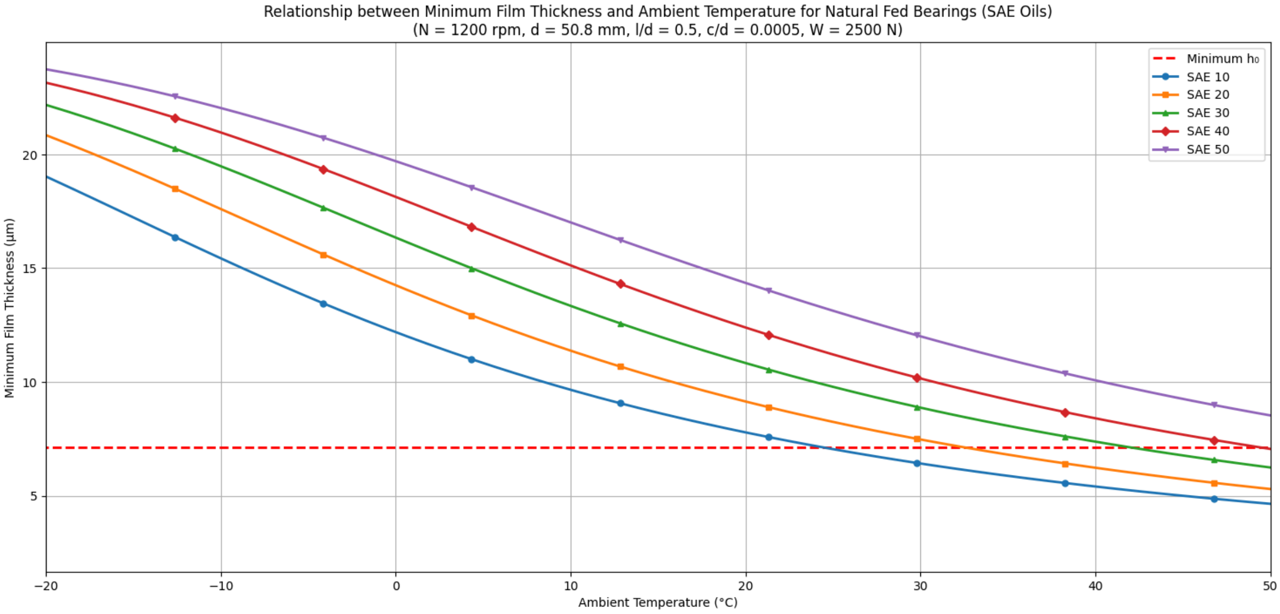


Fig 4.76:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (N = 1200 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

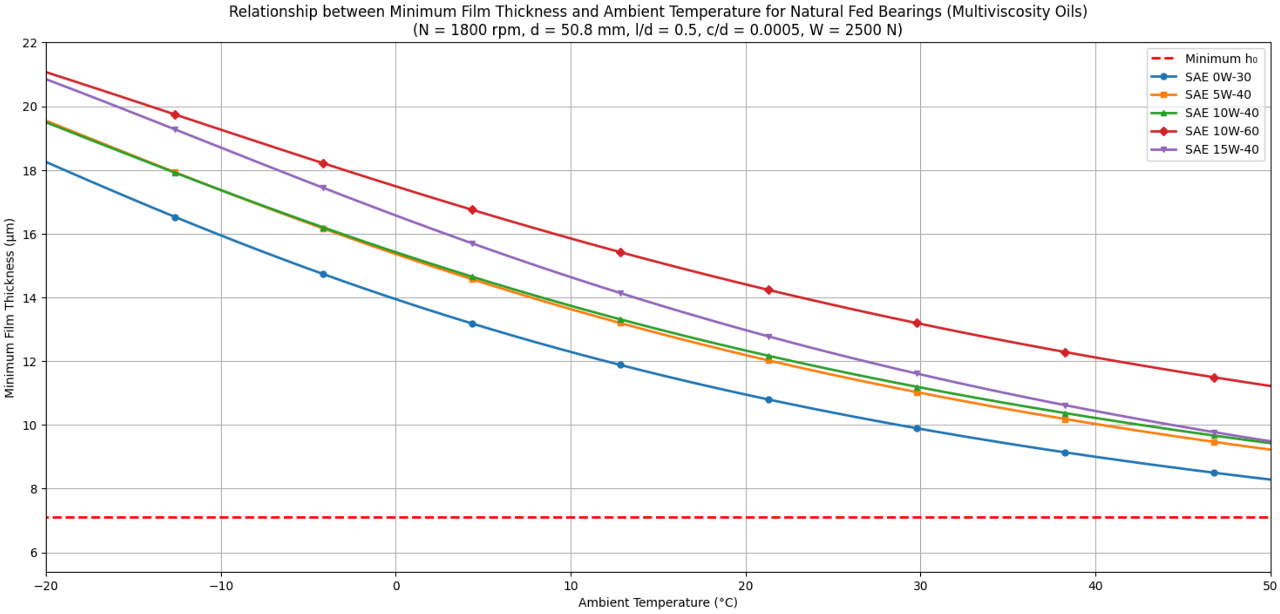


Fig 4.77:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (N = 1800 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

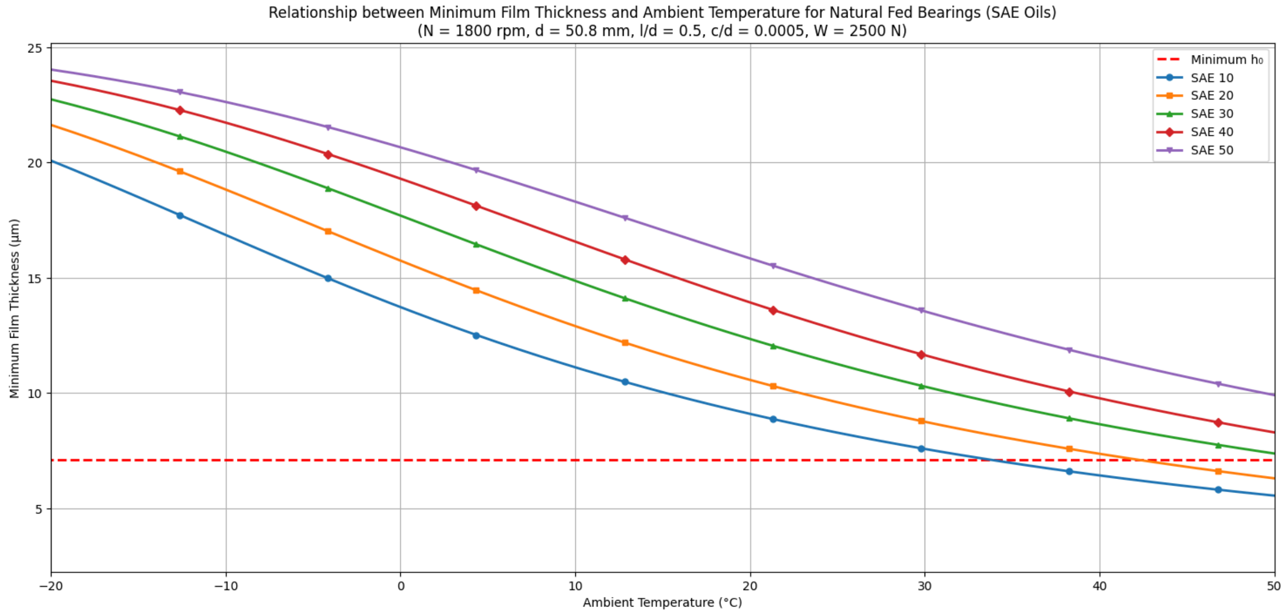


Fig 4.78:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (N = 1800 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

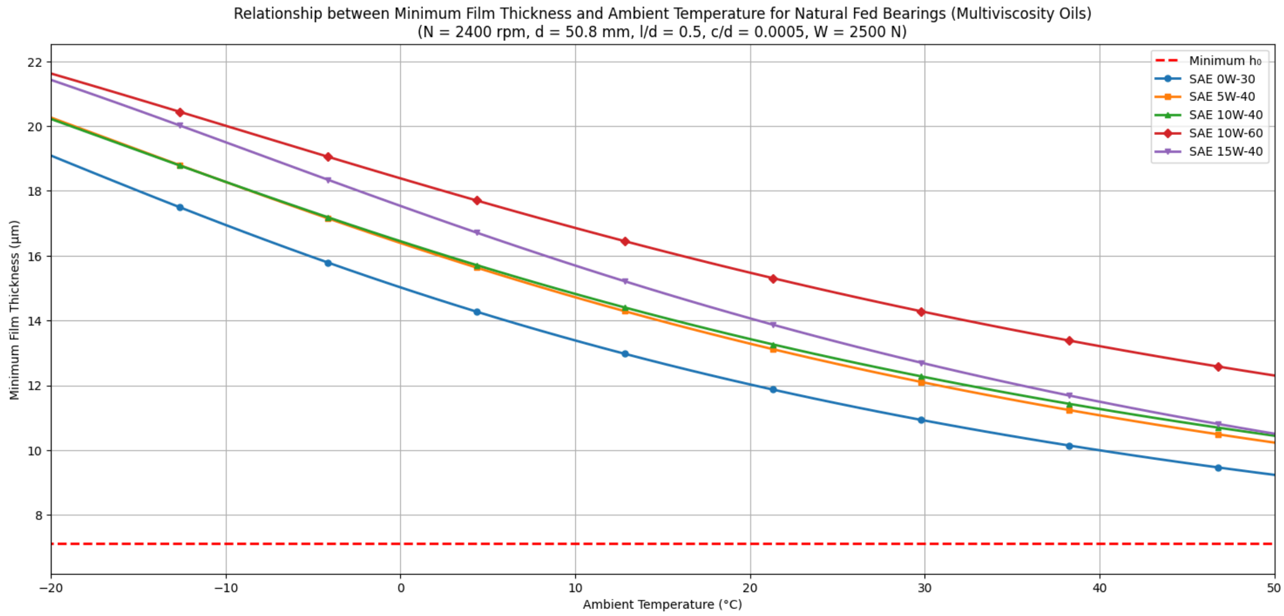


Fig 4.79:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (N = 2400 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

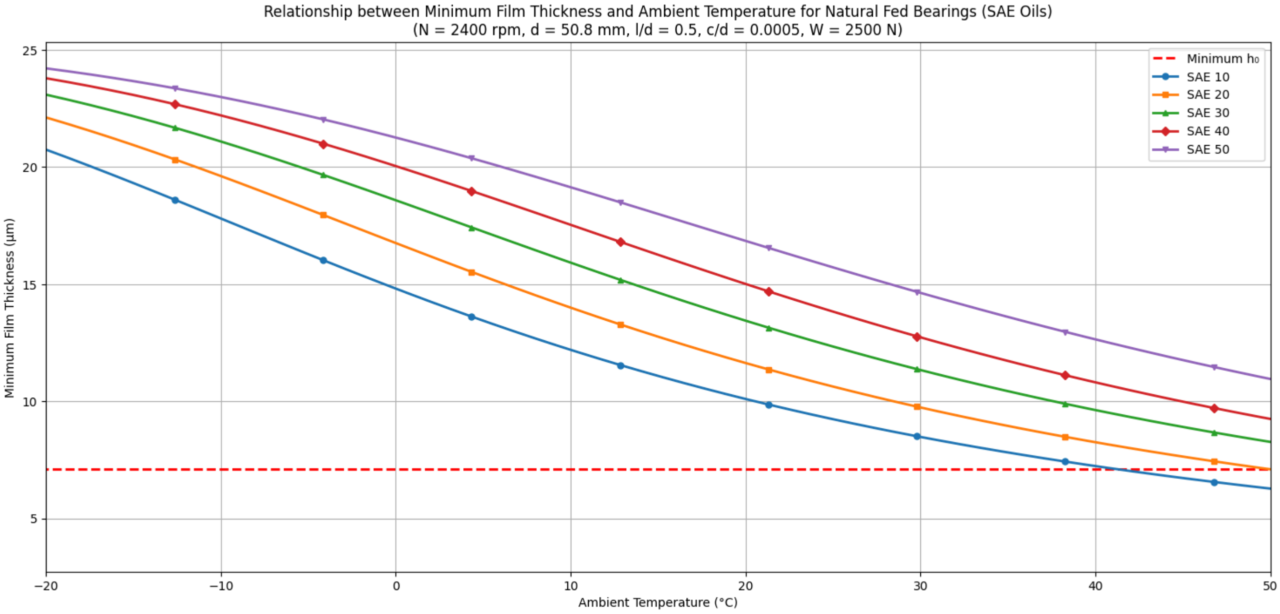


Fig 4.80:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (N = 2400 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

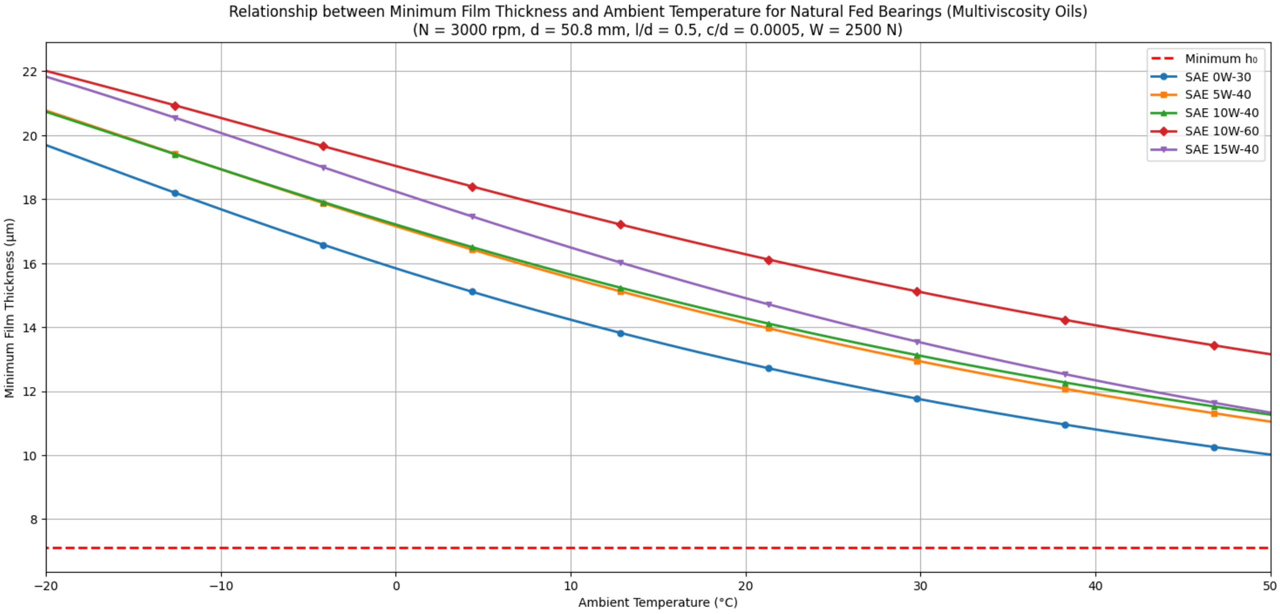


Fig 4.81:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (N = 3000 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

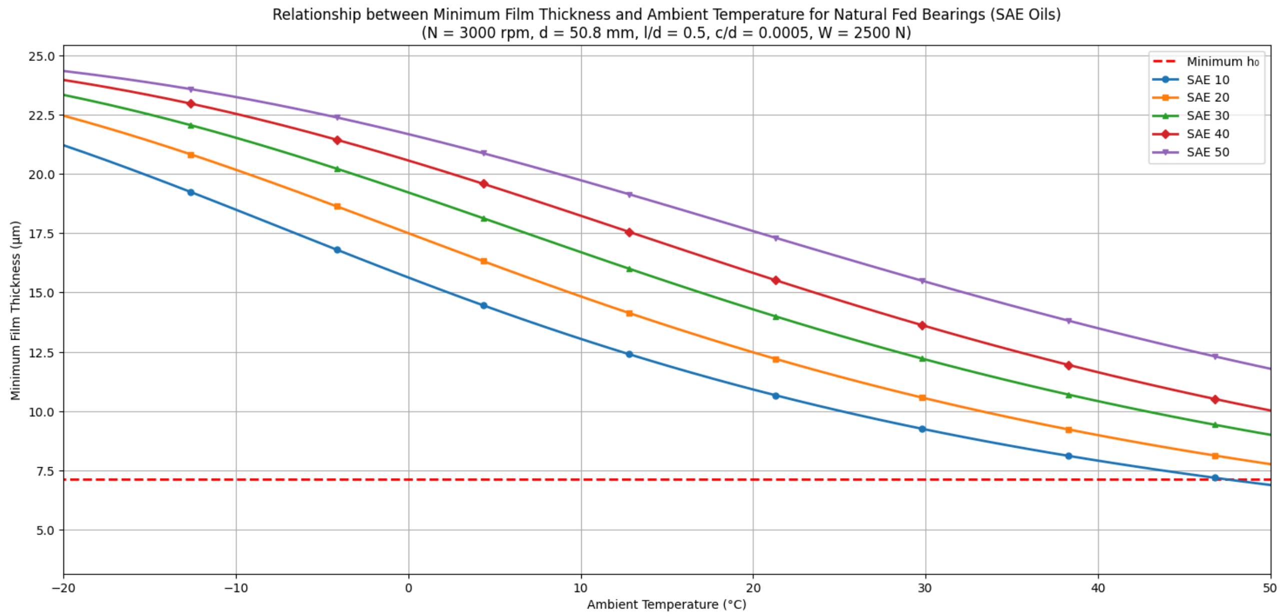


Fig 4.82:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (N = 3000 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

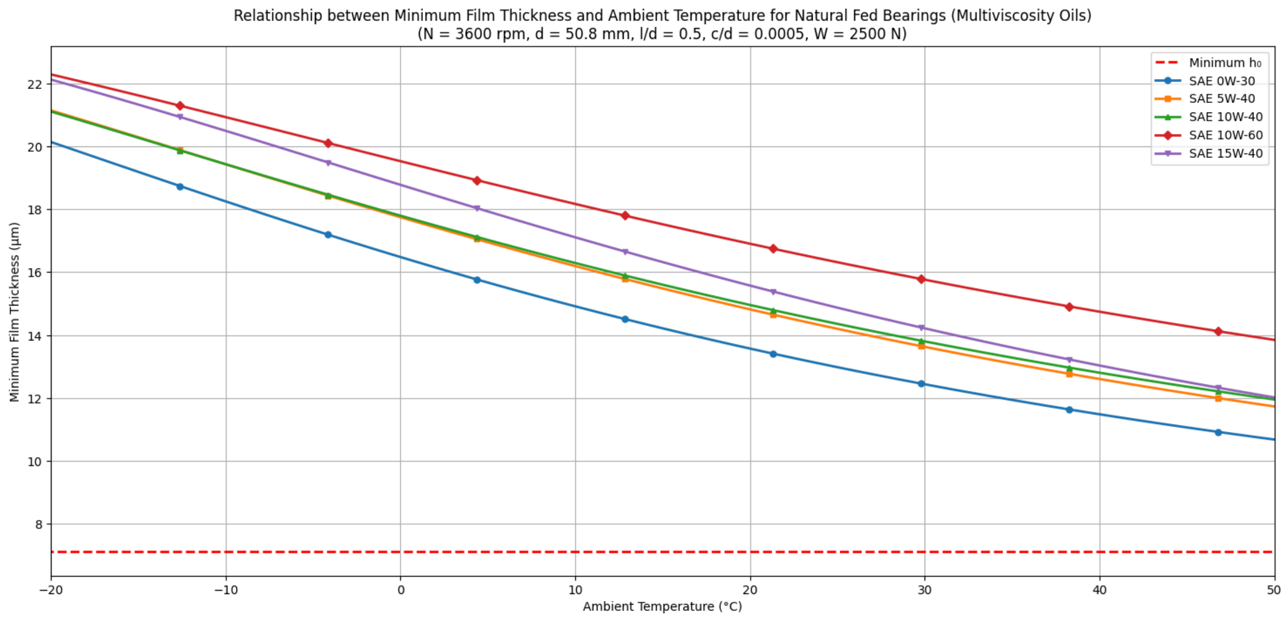


Fig 4.83:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (N = 3600 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

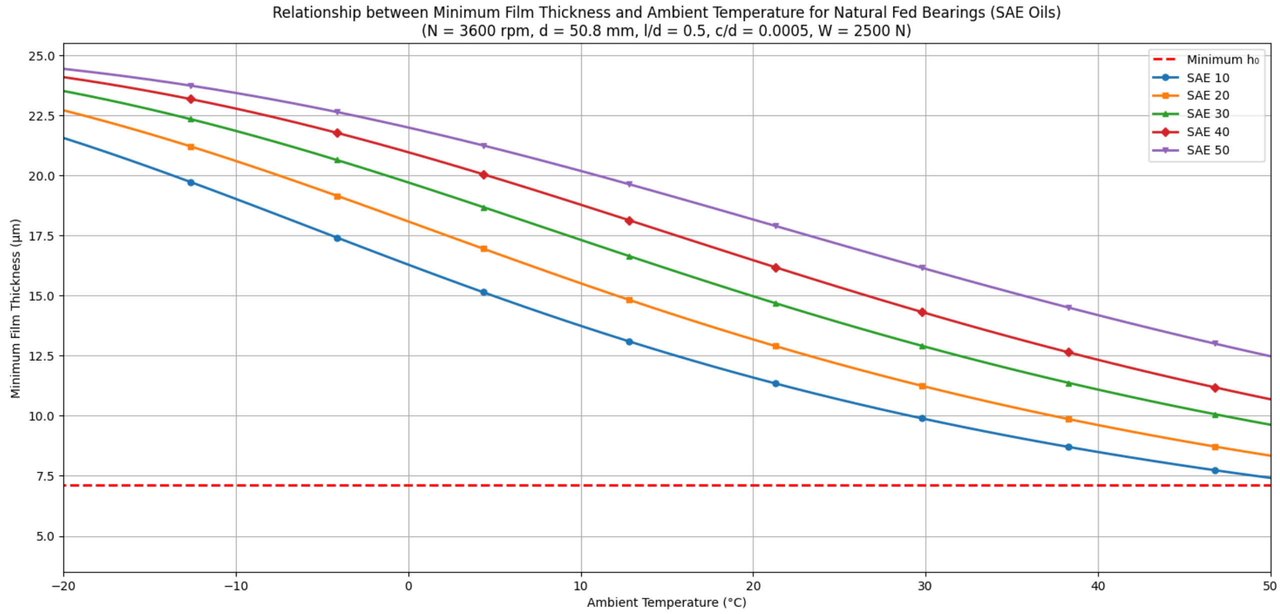


Fig 4.84:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (N = 3600 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

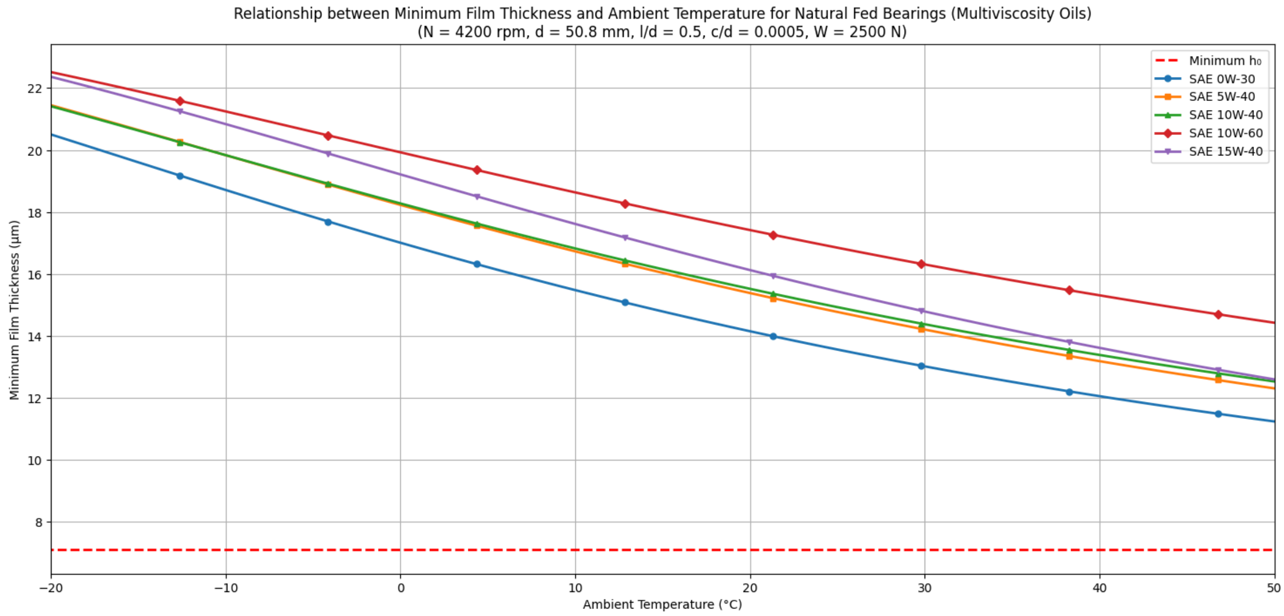


Fig 4.85:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (N = 4200 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

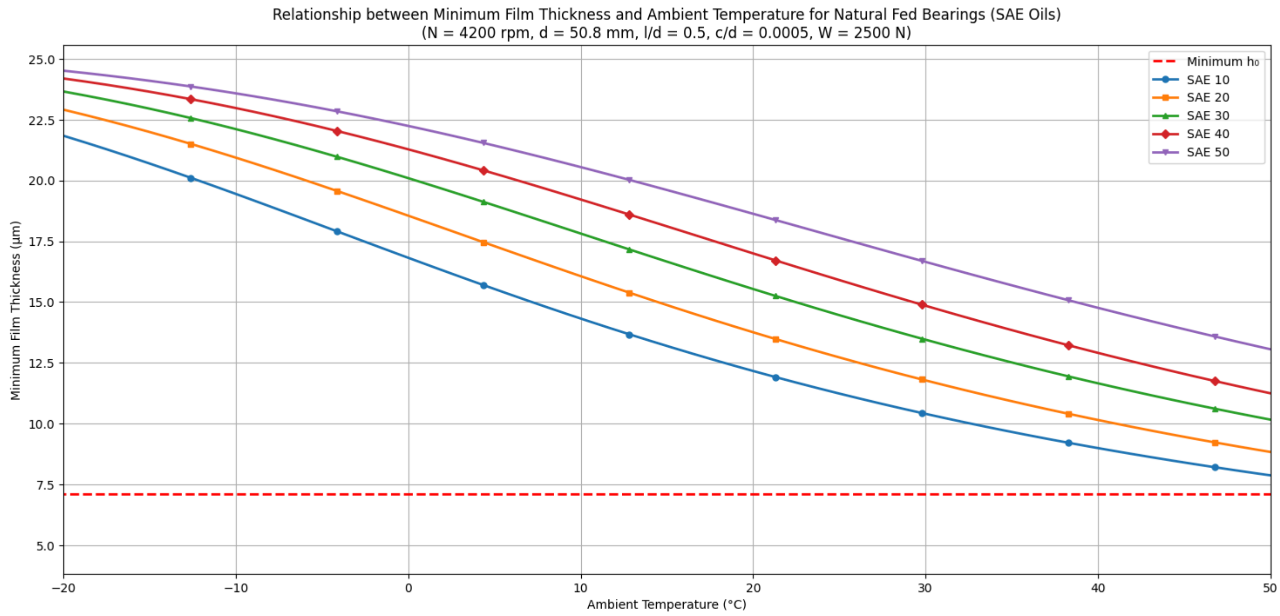


Fig 4.86:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (N = 4200 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

### 4.1.9 Effect of Varying Clearance Ratio (Natural Fed Bearing)

The clearance to diameter ratio is varied keeping journal diameter, length to diameter ratio, rotational speed and load constant in figure 4.87 to figure 4.96. We can see that minimum film thickness increases with increasing clearance to diameter ratio. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

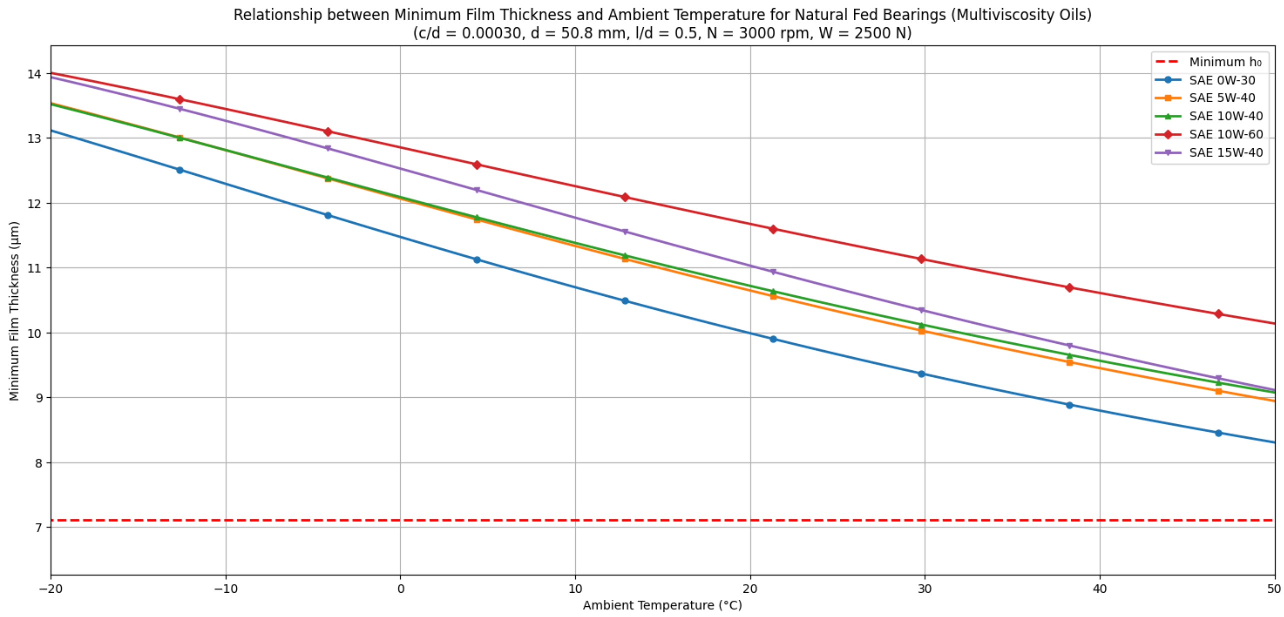


Fig 4.87:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (c/d = 0.00030, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

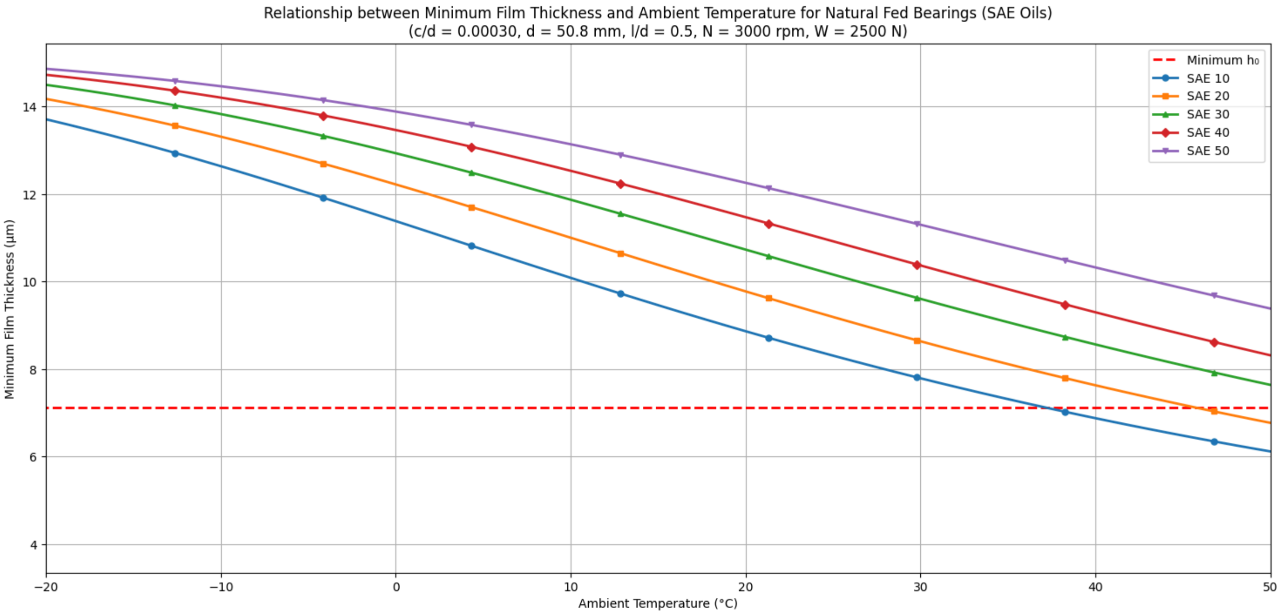


Fig 4.88:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (c/d = 0.00030, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

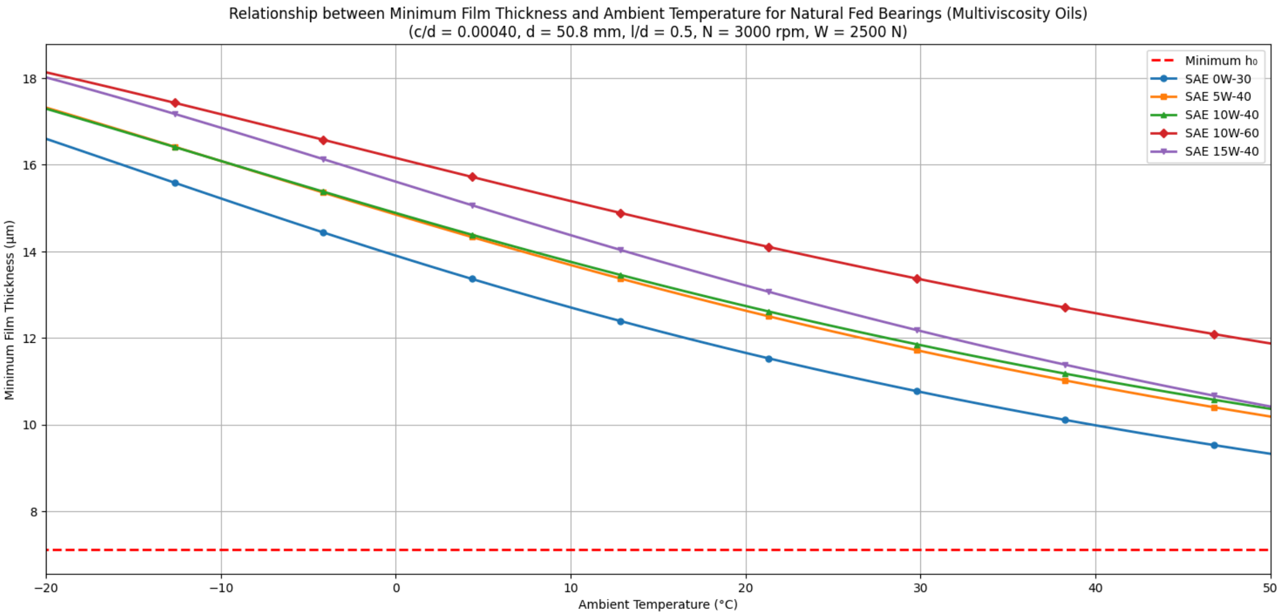


Fig 4.89:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (c/d = 0.00040, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

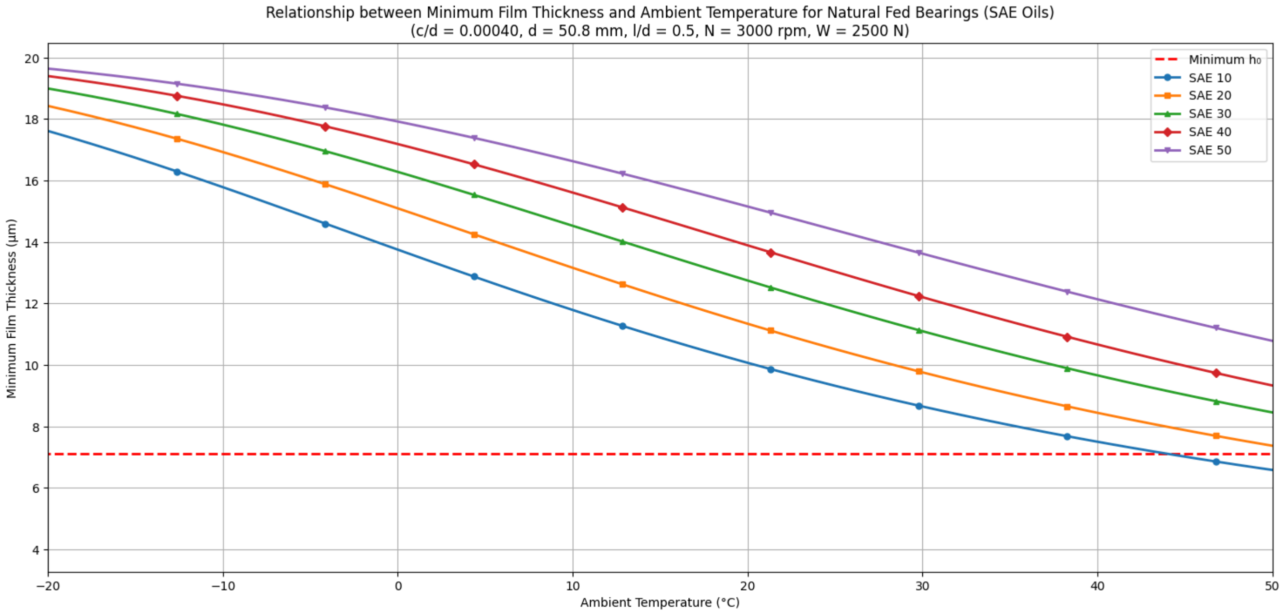


Fig 4.90:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (c/d = 0.00040, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

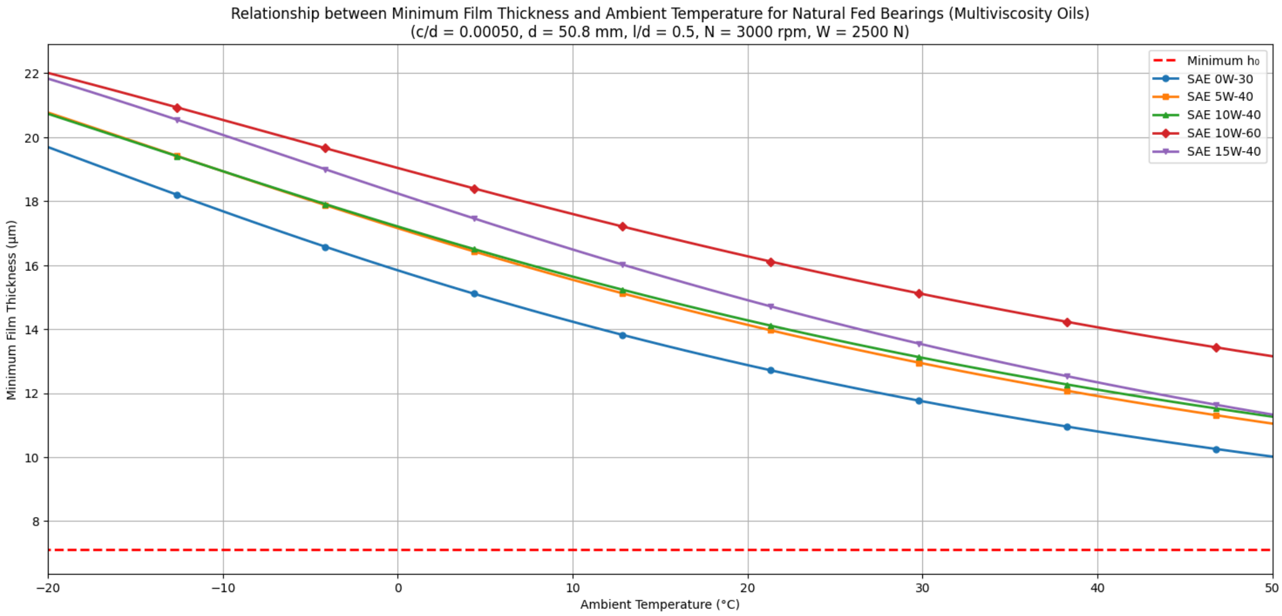


Fig 4.91Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (c/d = 0.00050, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

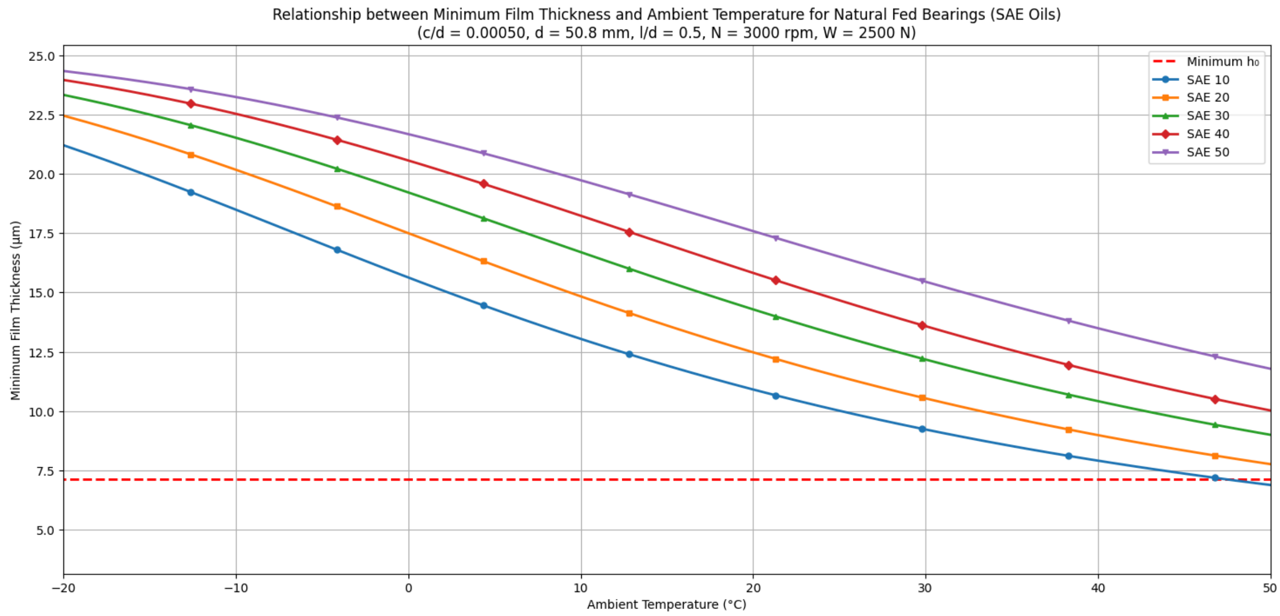


Fig 4.92:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (c/d = 0.00050, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

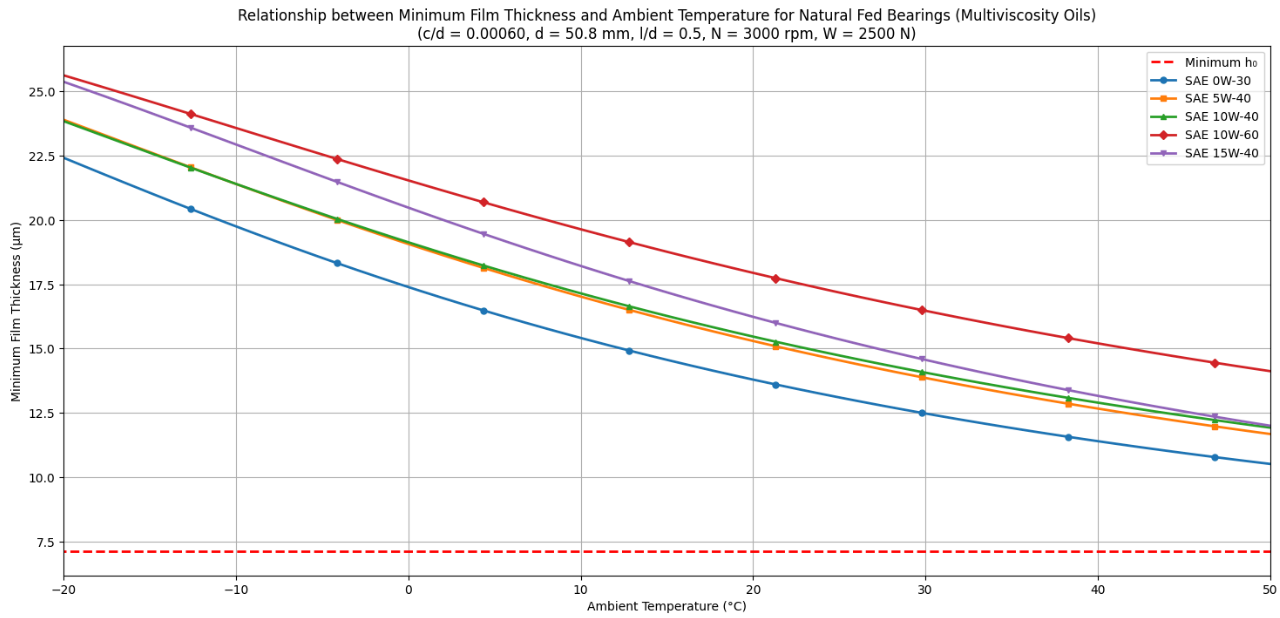


Fig 4.93:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (c/d = 0.00060, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

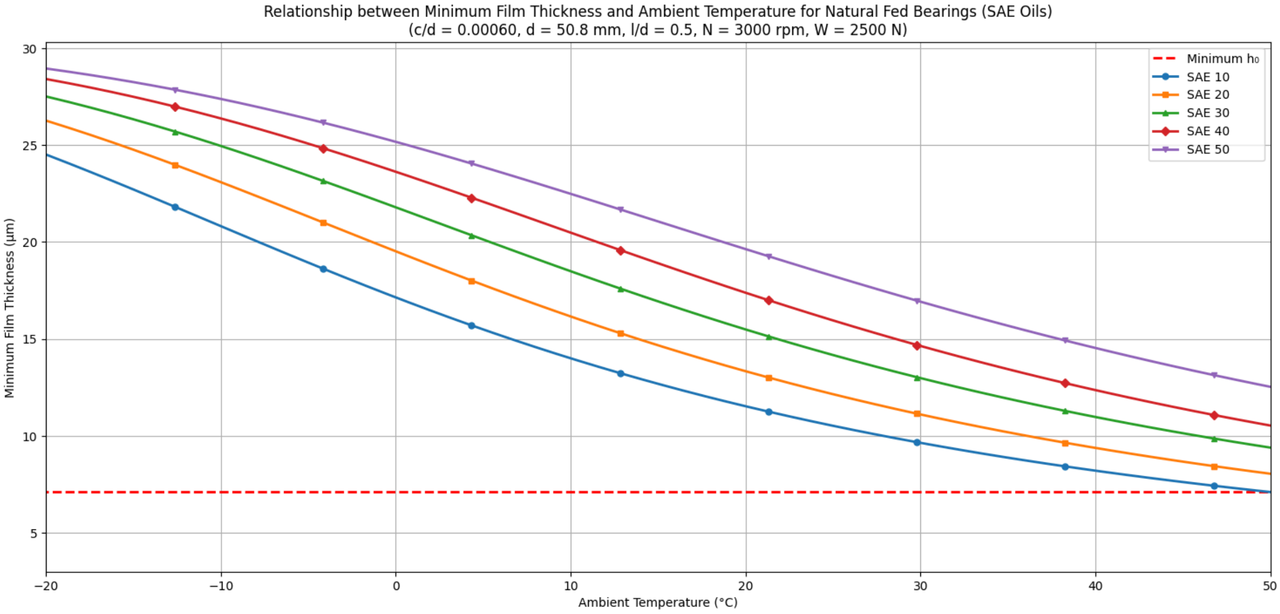


Fig 4.94:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (c/d = 0.00060, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

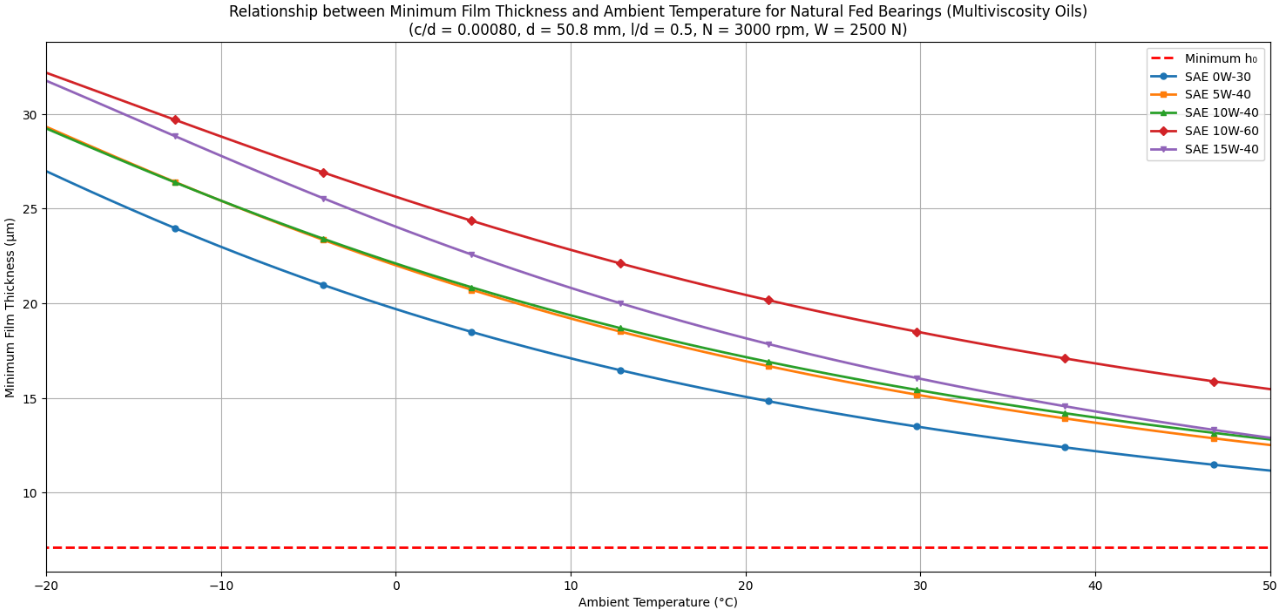


Fig 4.95:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (c/d = 0.00080, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

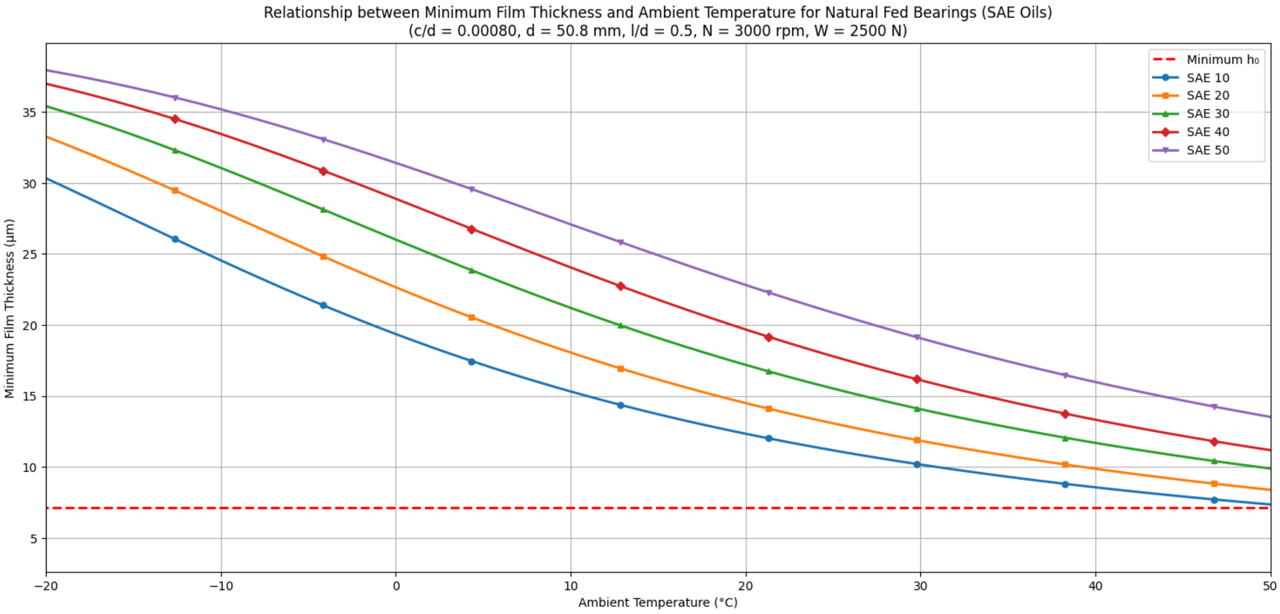


Fig 4.96:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (c/d = 0.00080, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

## 4.1.10 Effect of Varying Bearing Load (Natural Fed Bearing)

The load is varied keeping journal diameter, length to diameter ratio, rotational speed and clearance to diameter ratio constant in figure 4.97 to figure 4.108. We can see that minimum film thickness decreases with increasing load. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

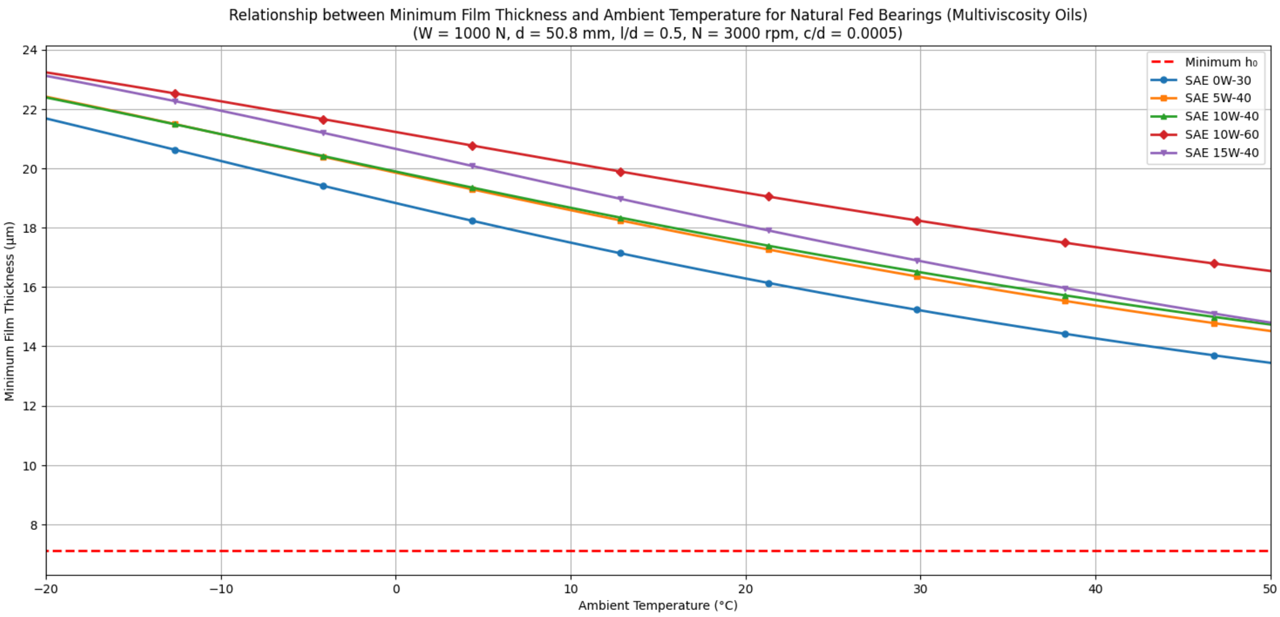


Fig 4.97:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (W = 1000 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

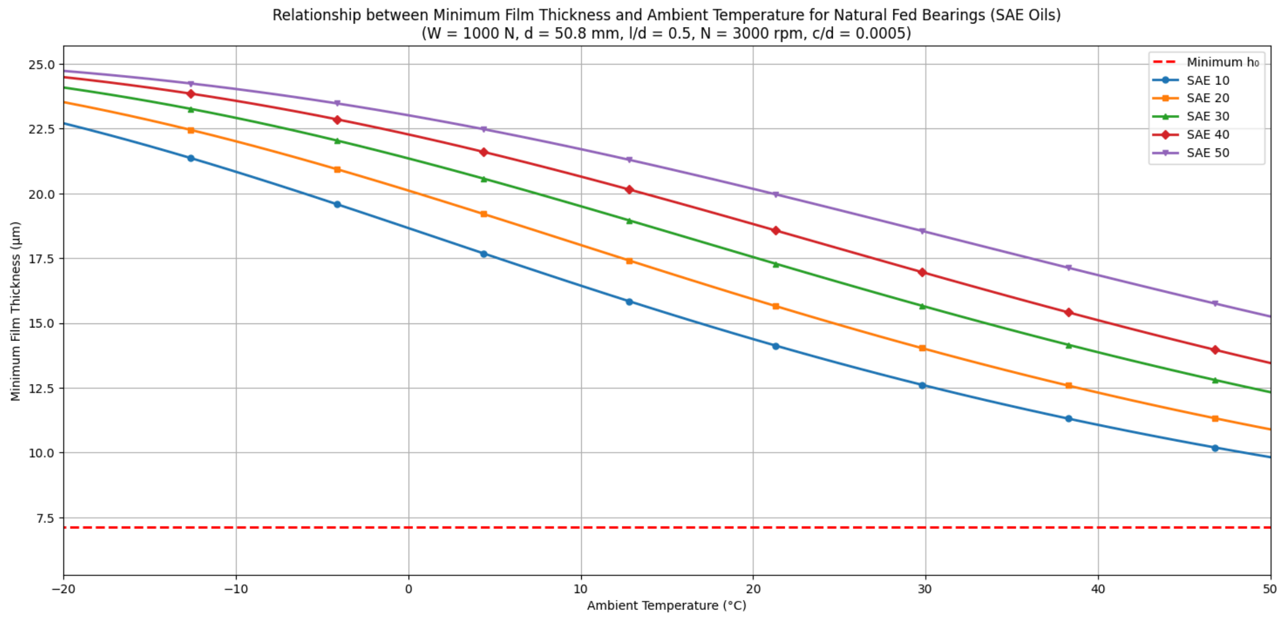


Fig 4.98:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (W = 1000 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

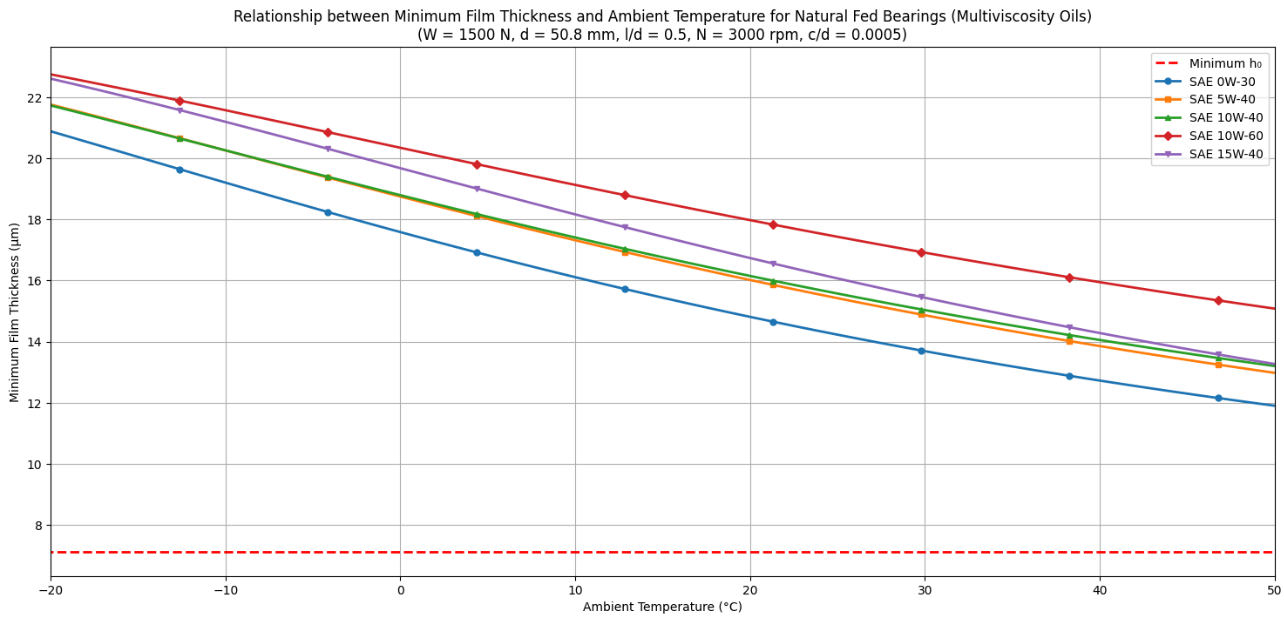


Fig 4.99:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (W = 1500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

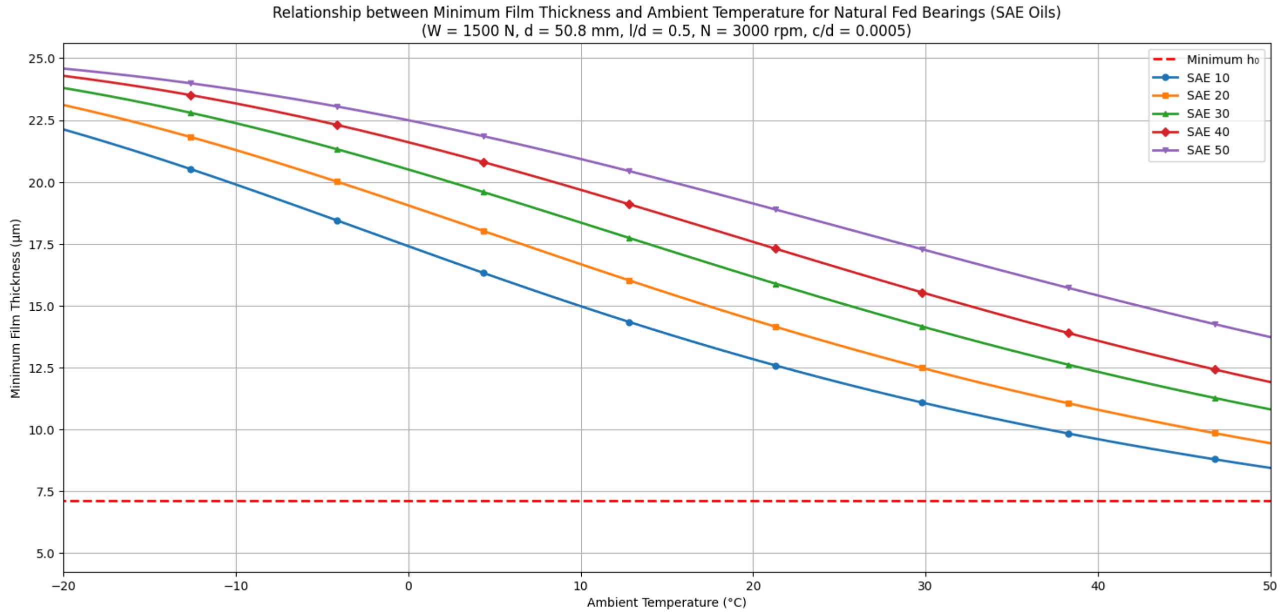


Fig 4.100:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (W = 1500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

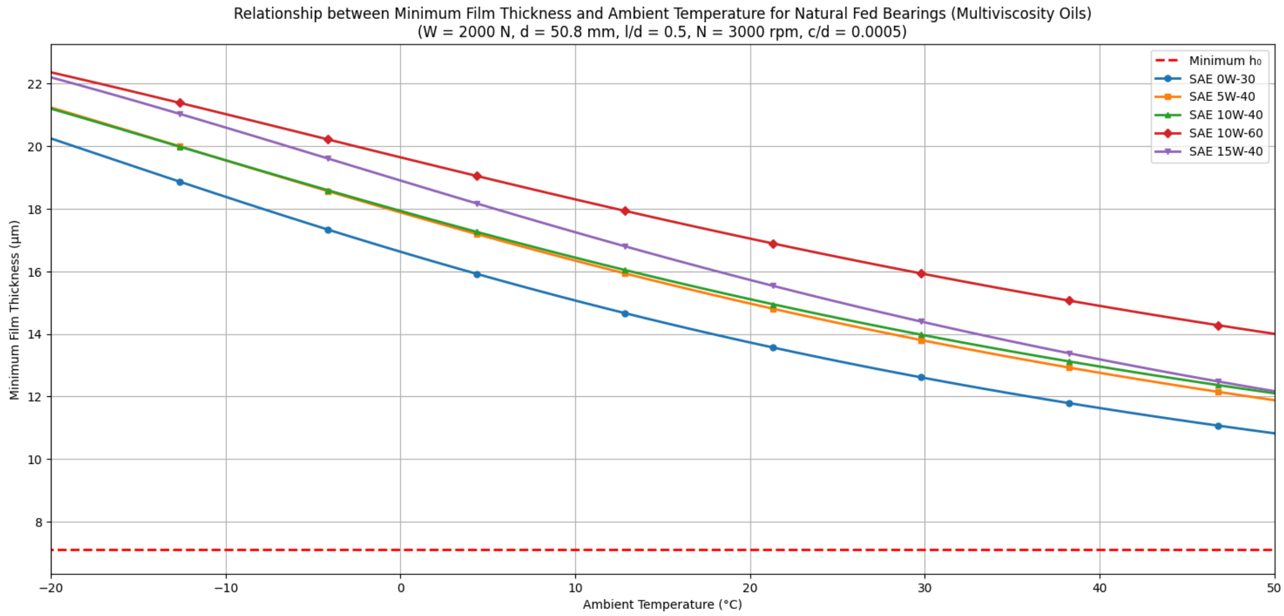


Fig 4.101:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (W = 2000 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

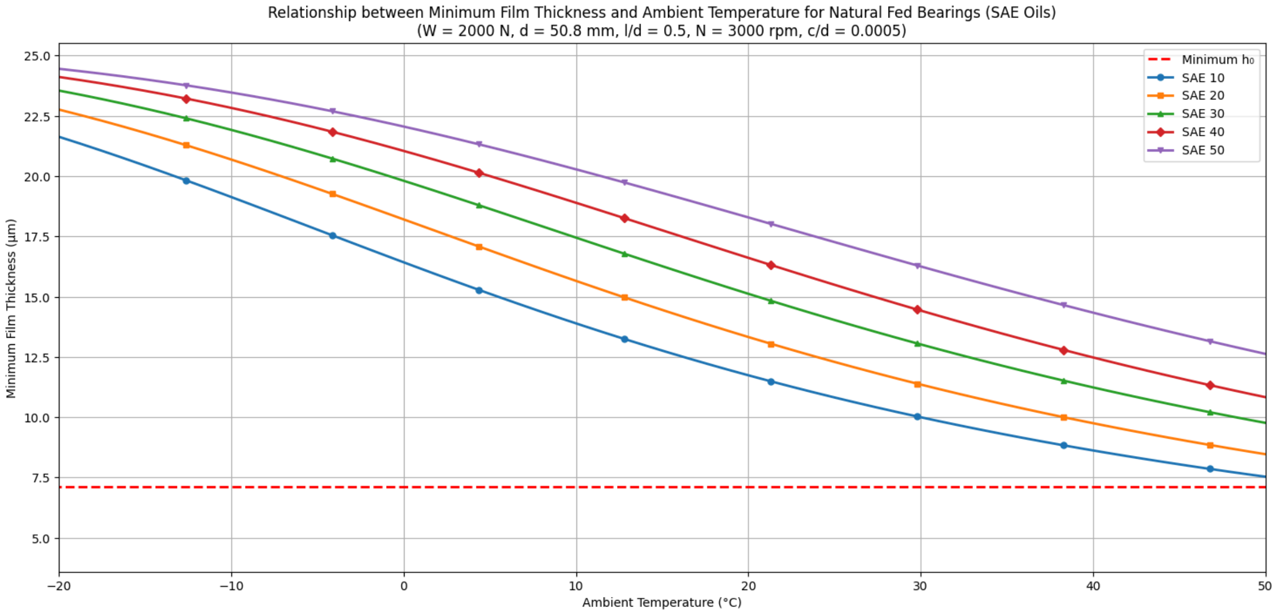


Fig 4.102:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (W = 2000 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

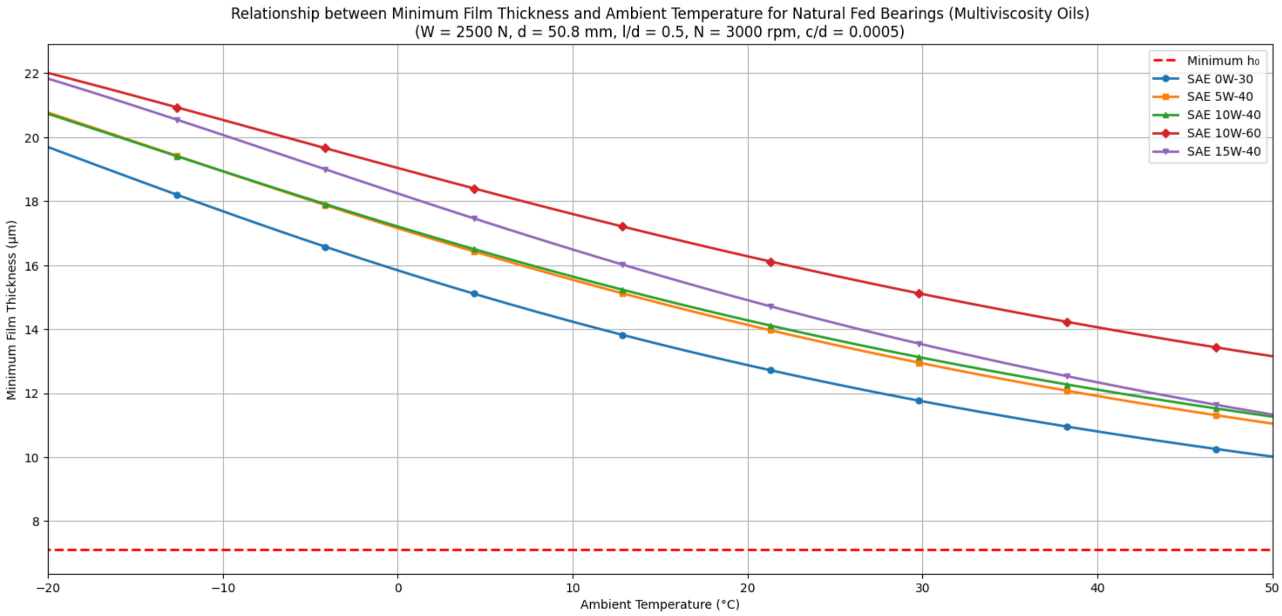


Fig 4.103:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (W = 2500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

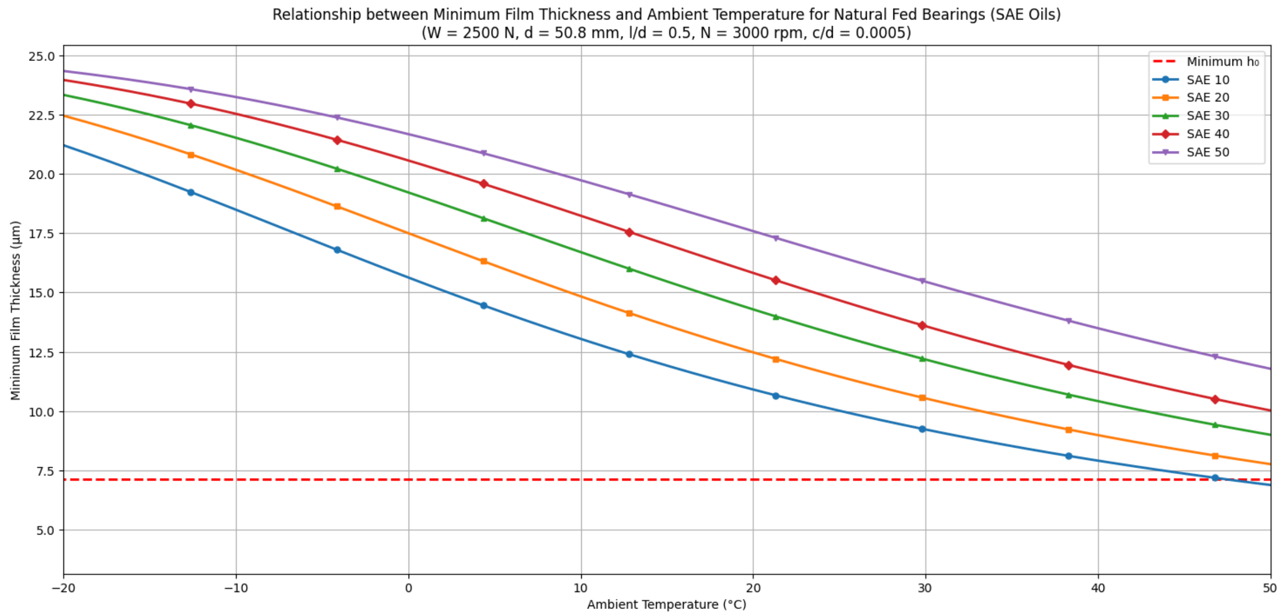


Fig 4.104:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (W = 2500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

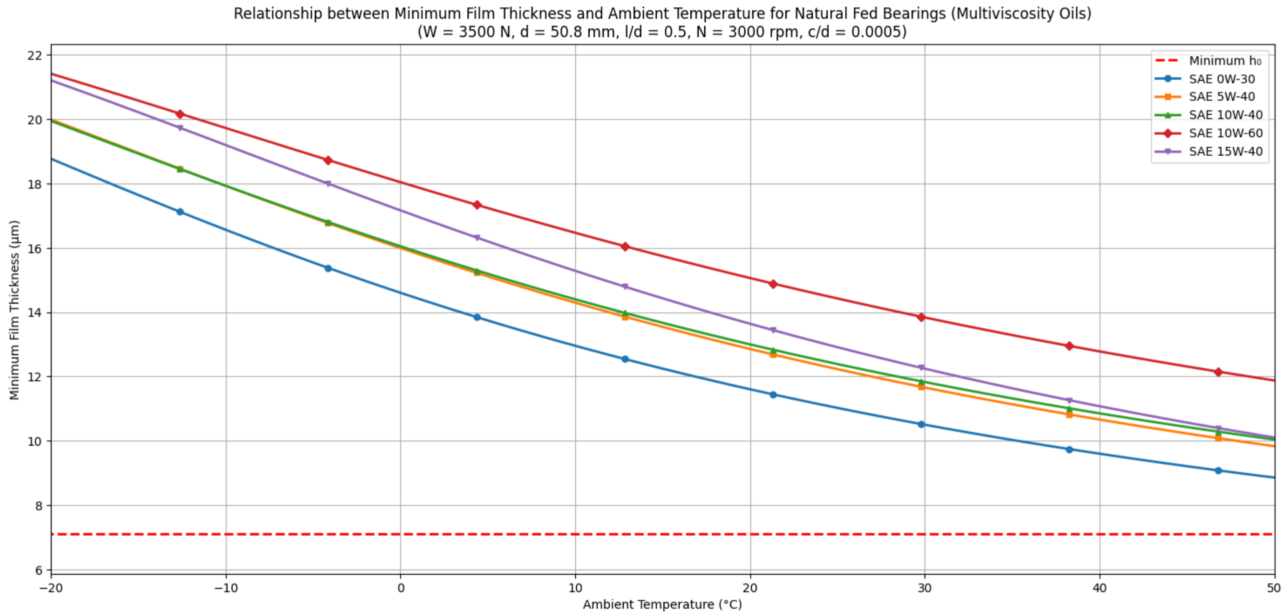


Fig 4.105:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (W = 3500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

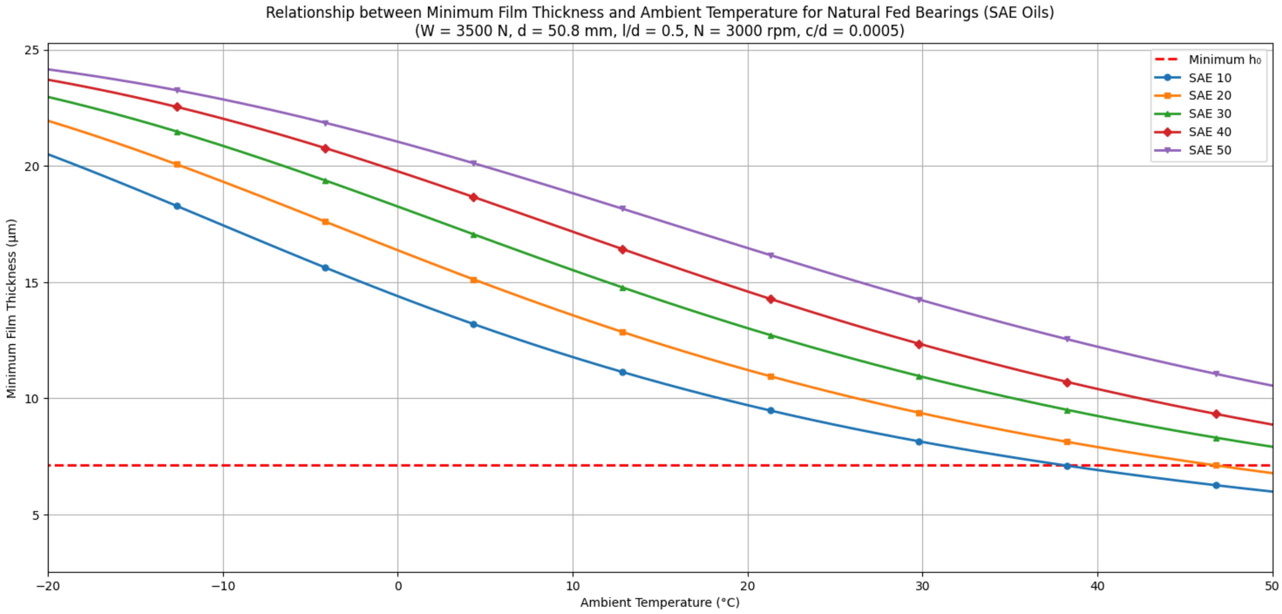


Fig 4.106:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (W = 3500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

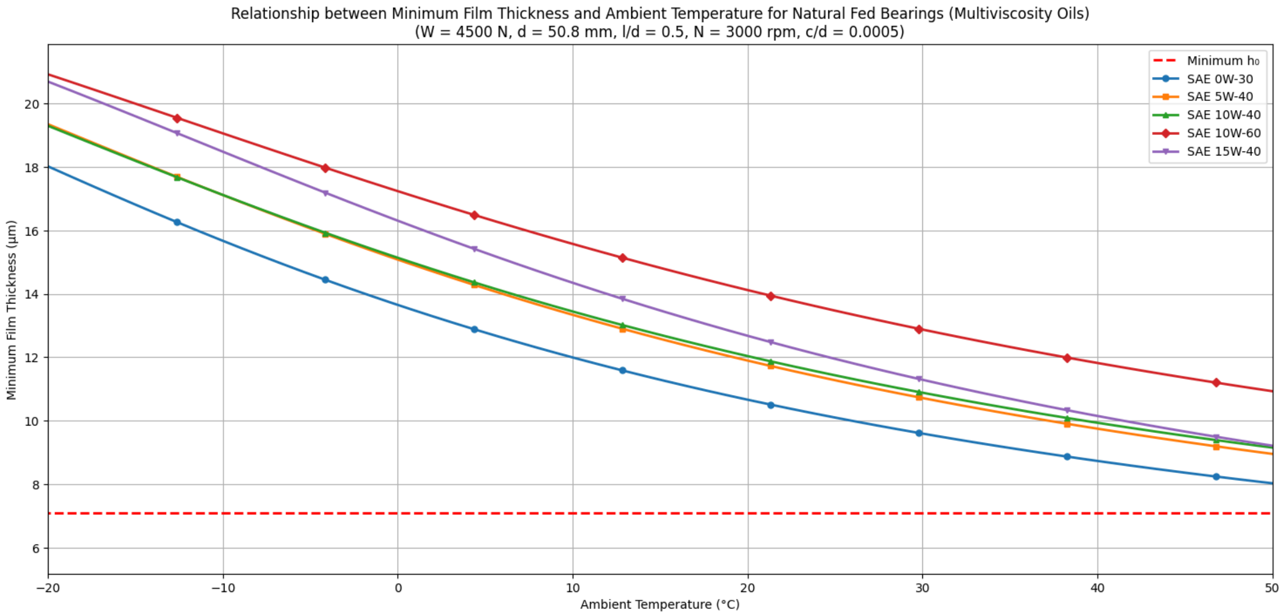


Fig 4.107:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (Multiviscosity Oils) — (W = 4500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

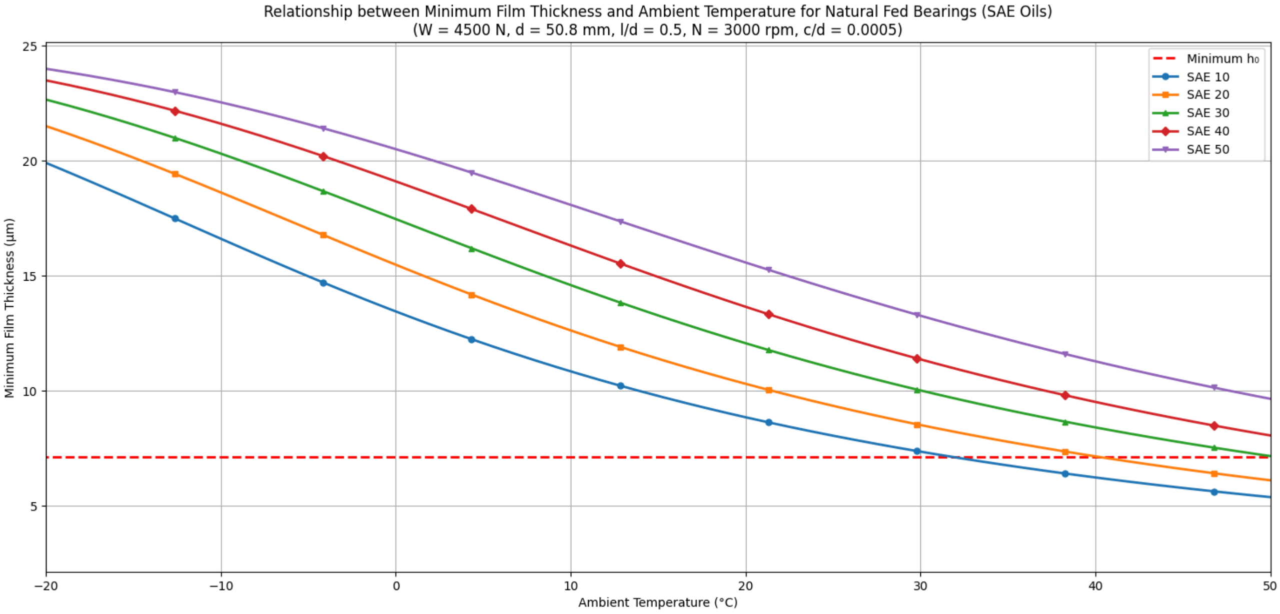


Fig 4.108:Relationship between Minimum Film Thickness and Ambient Temperature for Natural Fed Bearings (SAE Oils) — (W = 4500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

## 4.1.11 Effect of Varying Journal Diameter (Self Contained Bearing)

Figure 4.109 to figure 4.162 represents the performance of self contained bearings for different multiviscosity and SAE oils. The journal diameter is varied keeping length to diameter ratio, rotational speed, clearance to diameter ratio and load constant in figure 4.109 to figure 4.118. We can see that minimum film thickness increases with increasing journal diameter. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

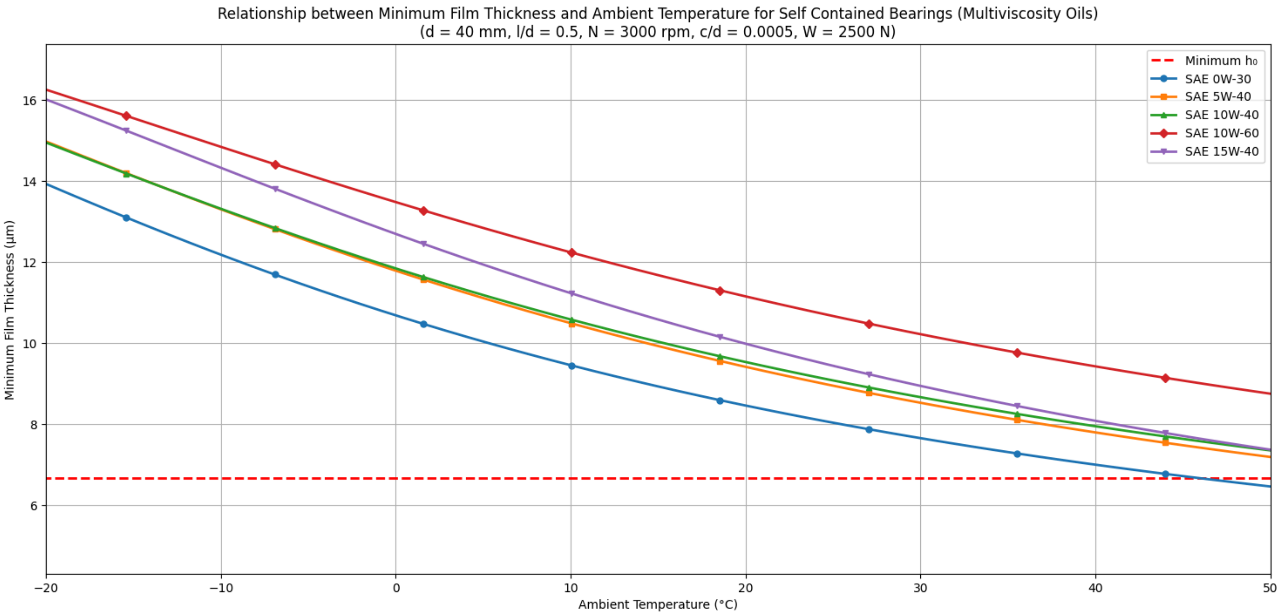

Fig 4.109:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (d = 40 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

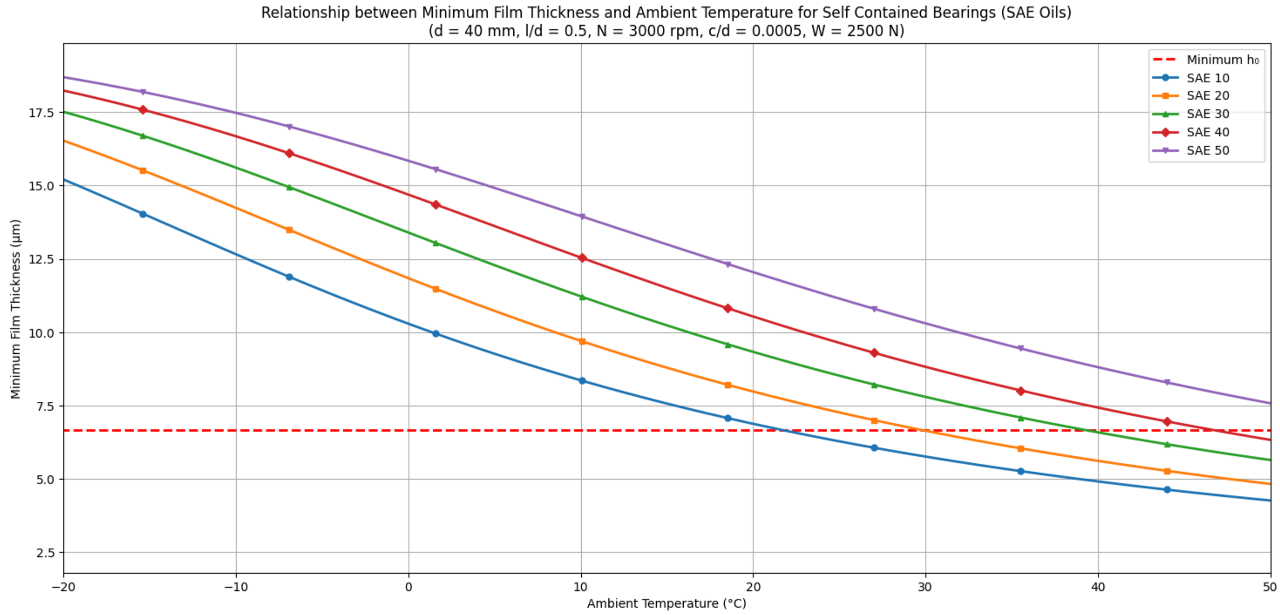


Fig 4.110:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (d = 40 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

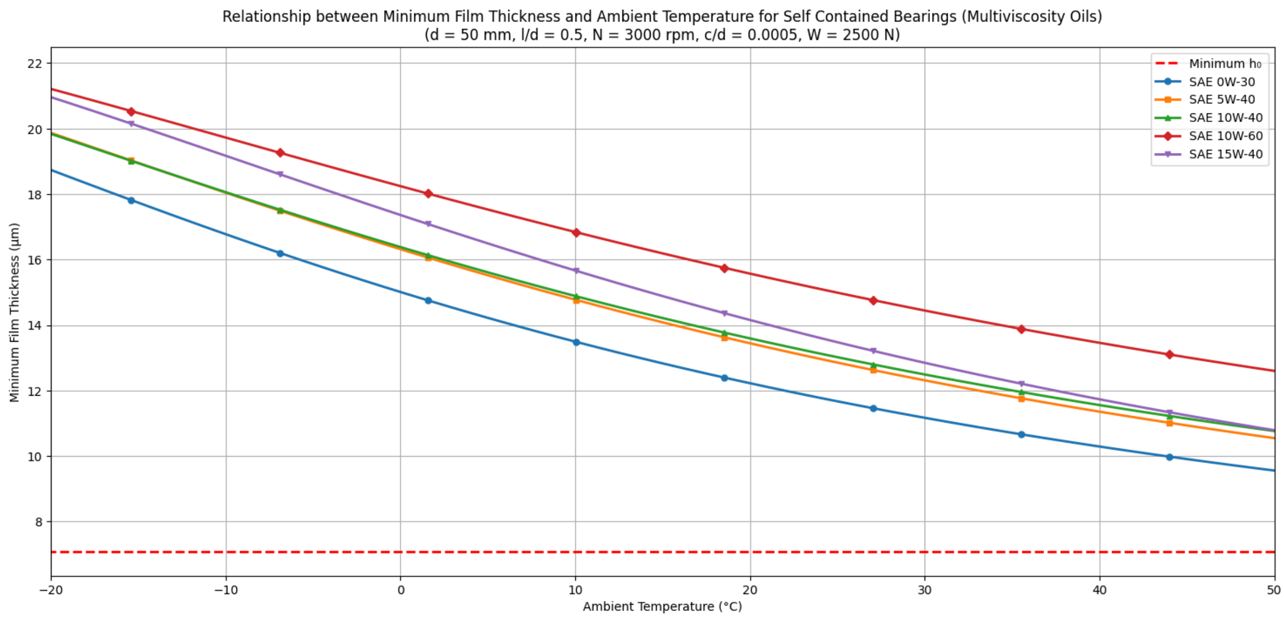


Fig 4.111:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (d = 50 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

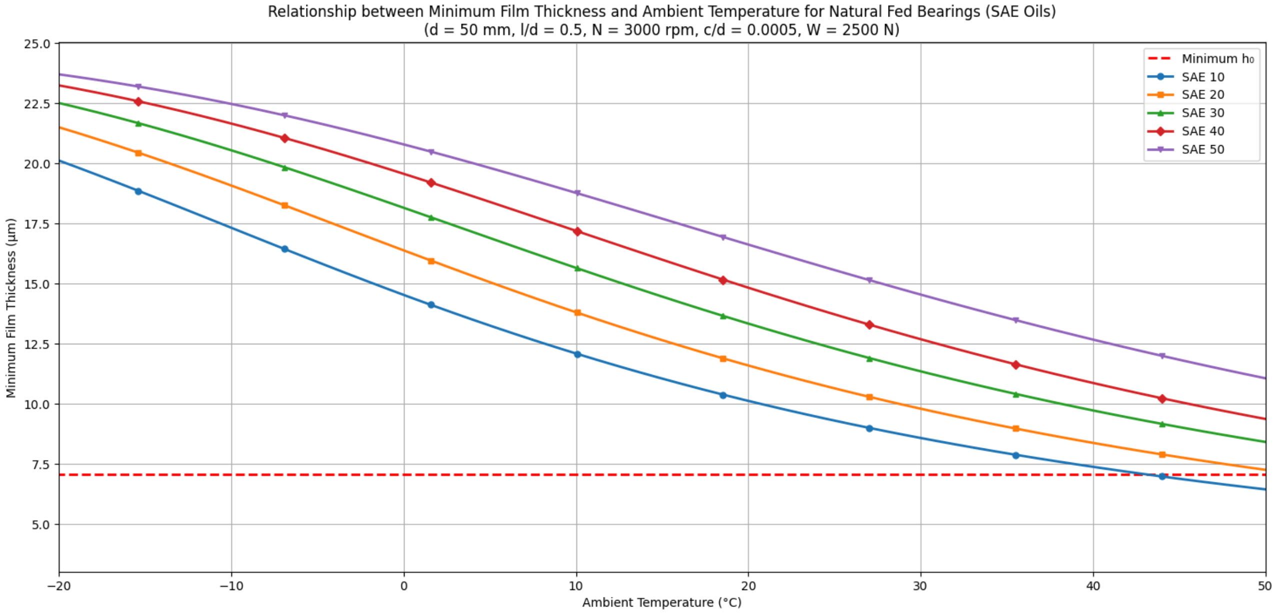


Fig 4.112:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (d = 50 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

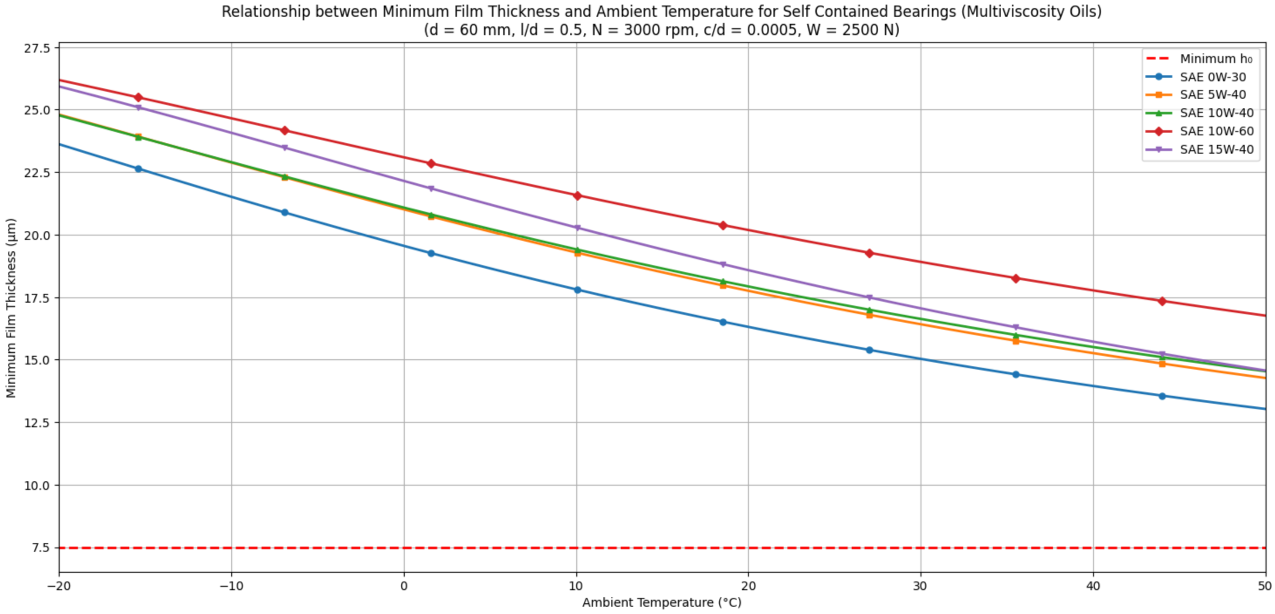


Fig 4.113:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (d = 60 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

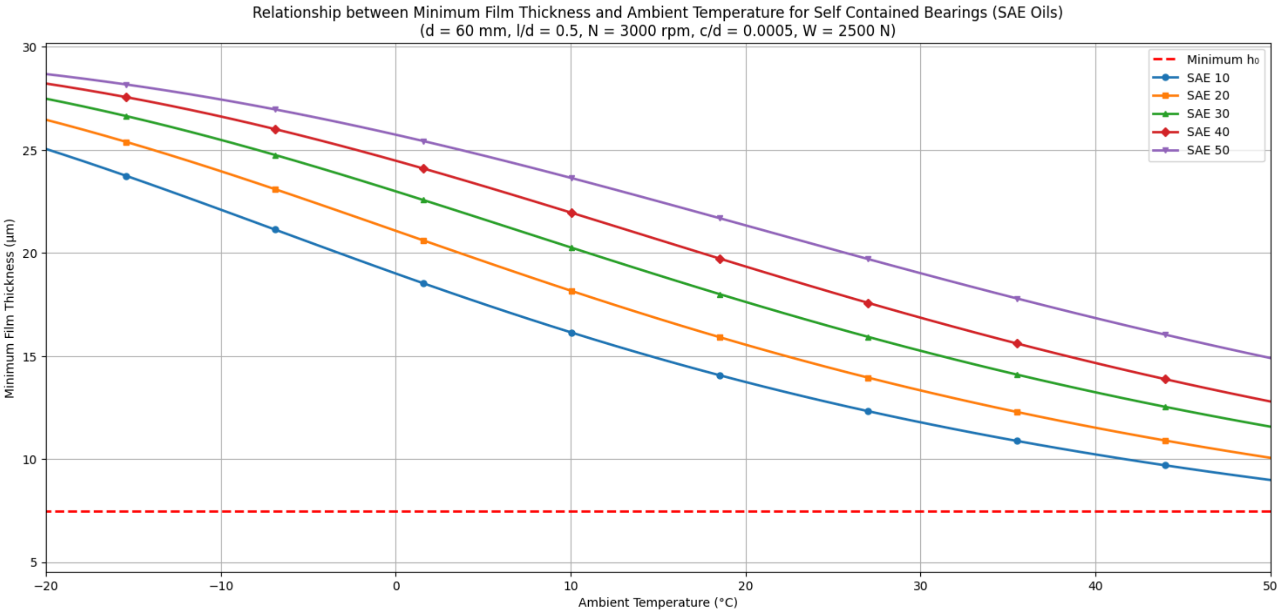


Fig 4.114:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (d = 60 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

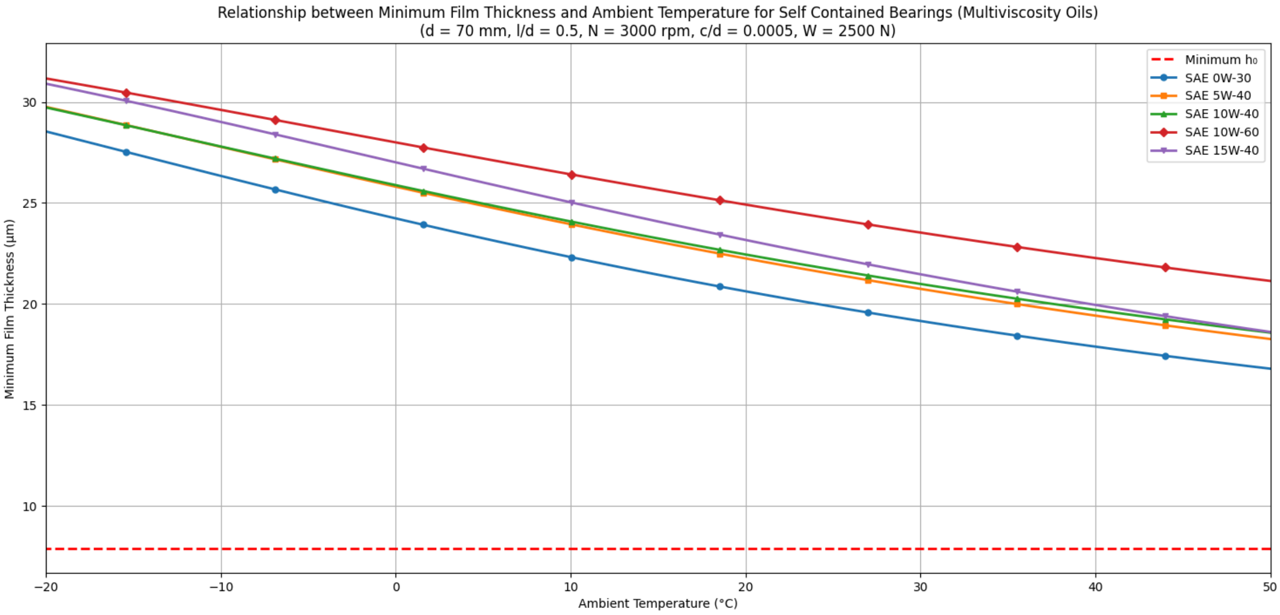


Fig 4.115:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (d = 70 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

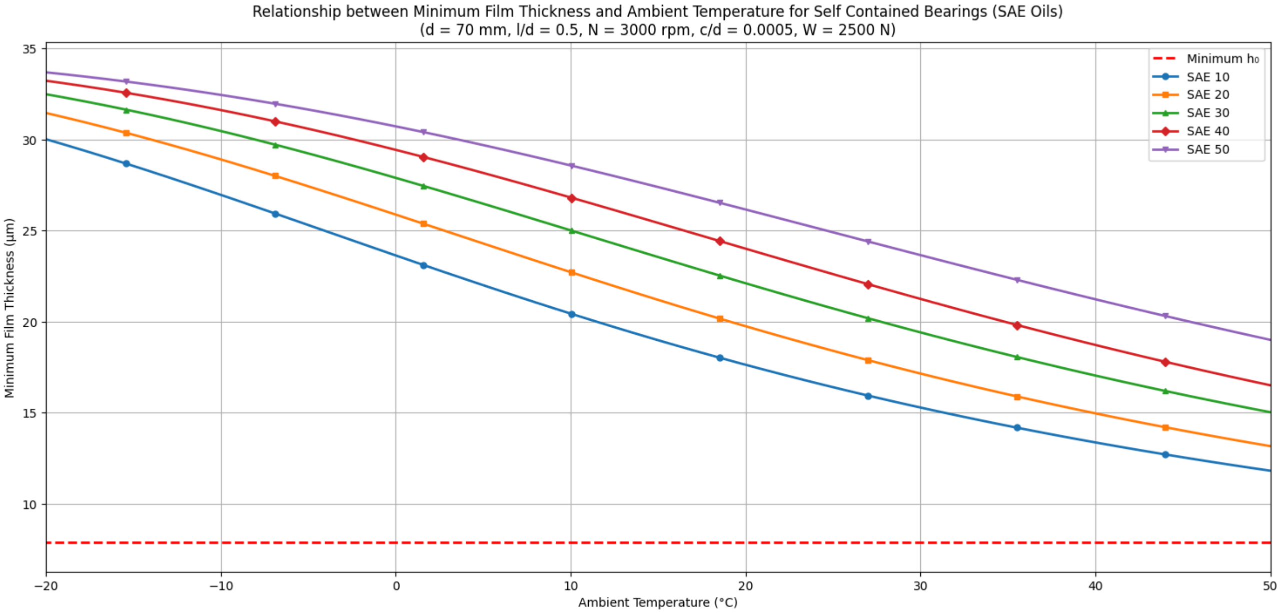


Fig 4.116:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (d = 70 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

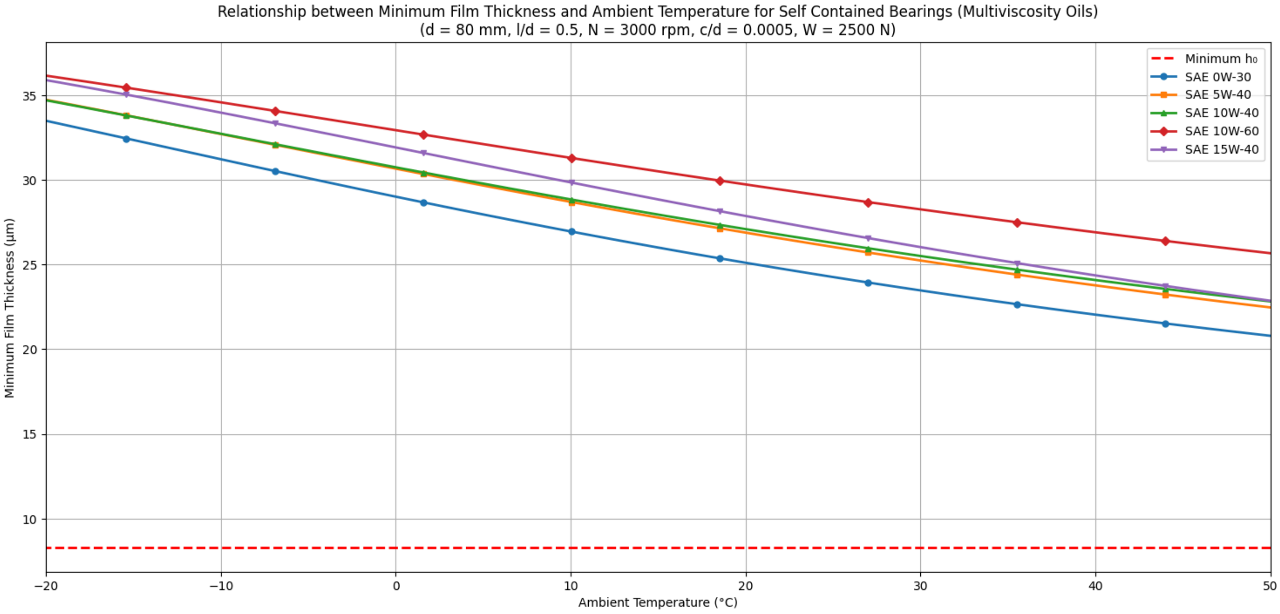


Fig 4.117:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (d = 80 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

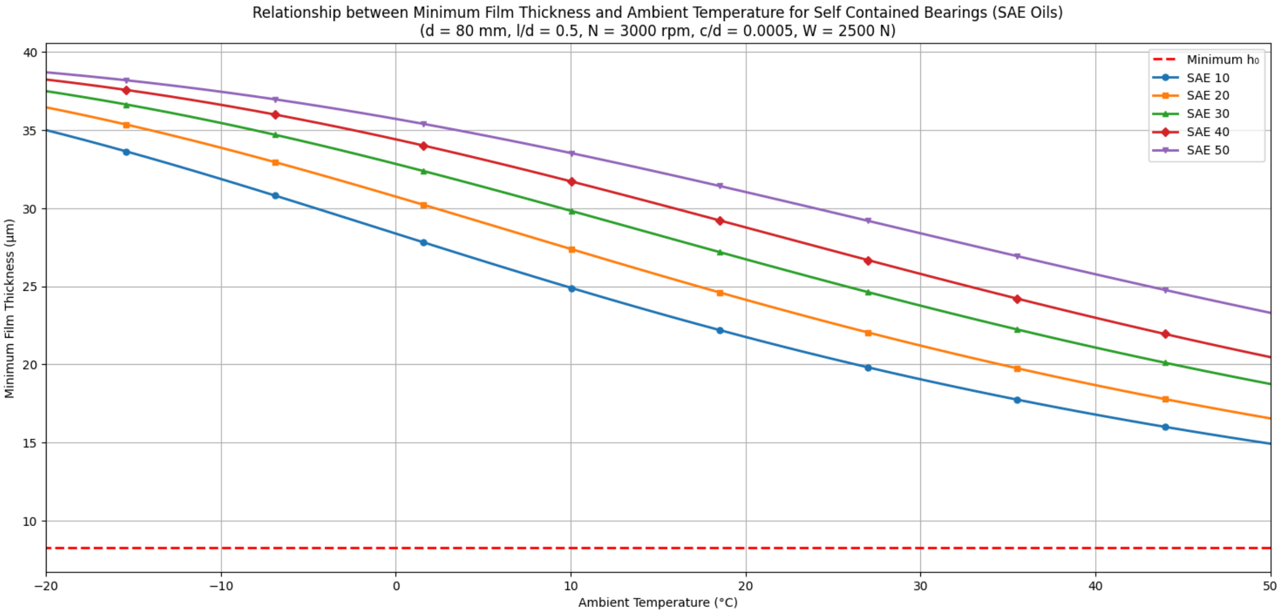


Fig 4.118:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (d = 80 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

## 4.1.12 Effect of Varying Length to Diameter (Self Contained Bearing)

The length to diameter ratio is varied keeping journal diameter, rotational speed, clearance to diameter ratio and load constant in figure 4.119 to figure 4.126. We can see that minimum film thickness increases with increasing length to diameter ratio. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

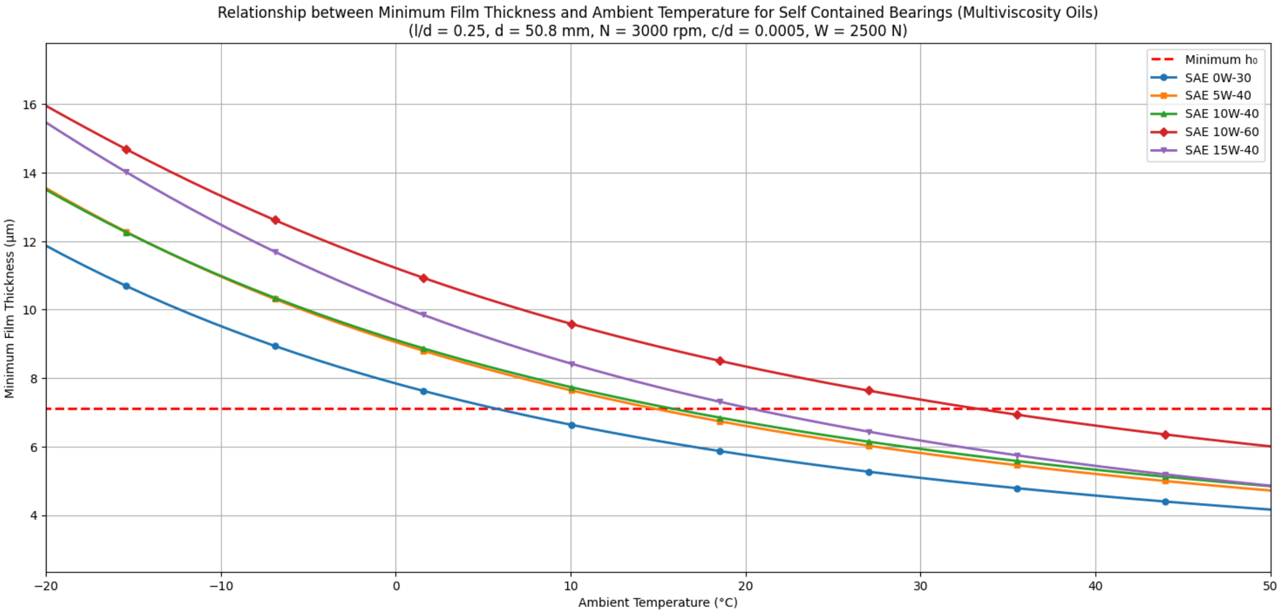


Fig 4.119:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (l/d = 0.25, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

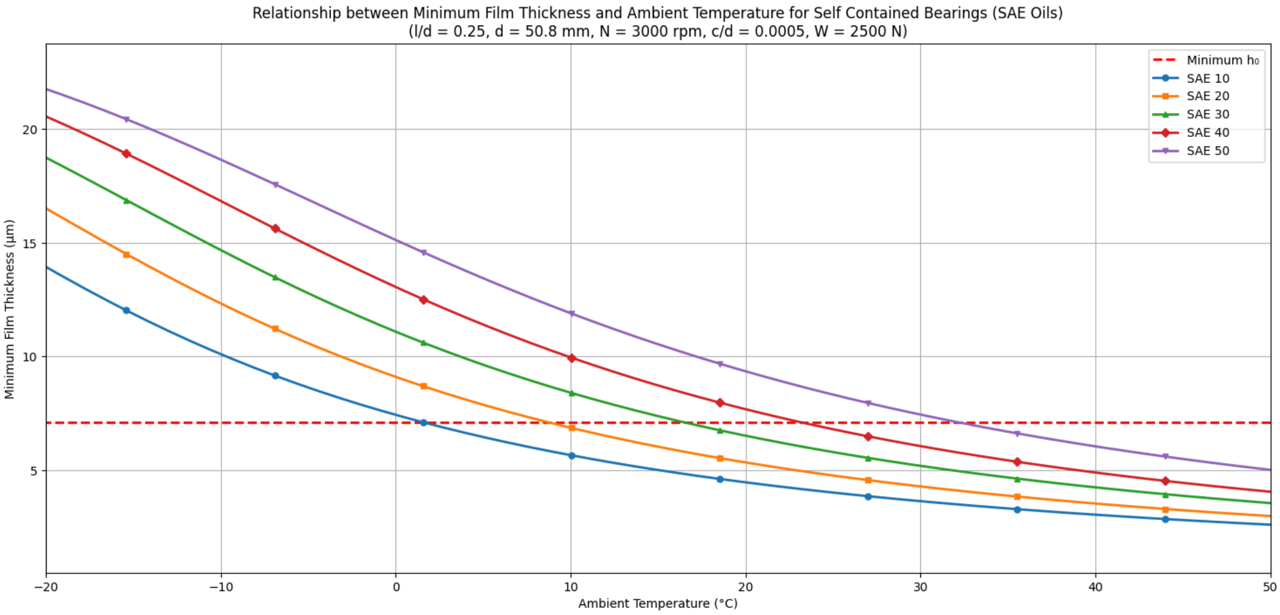


Fig 4.120:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (l/d = 0.25, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

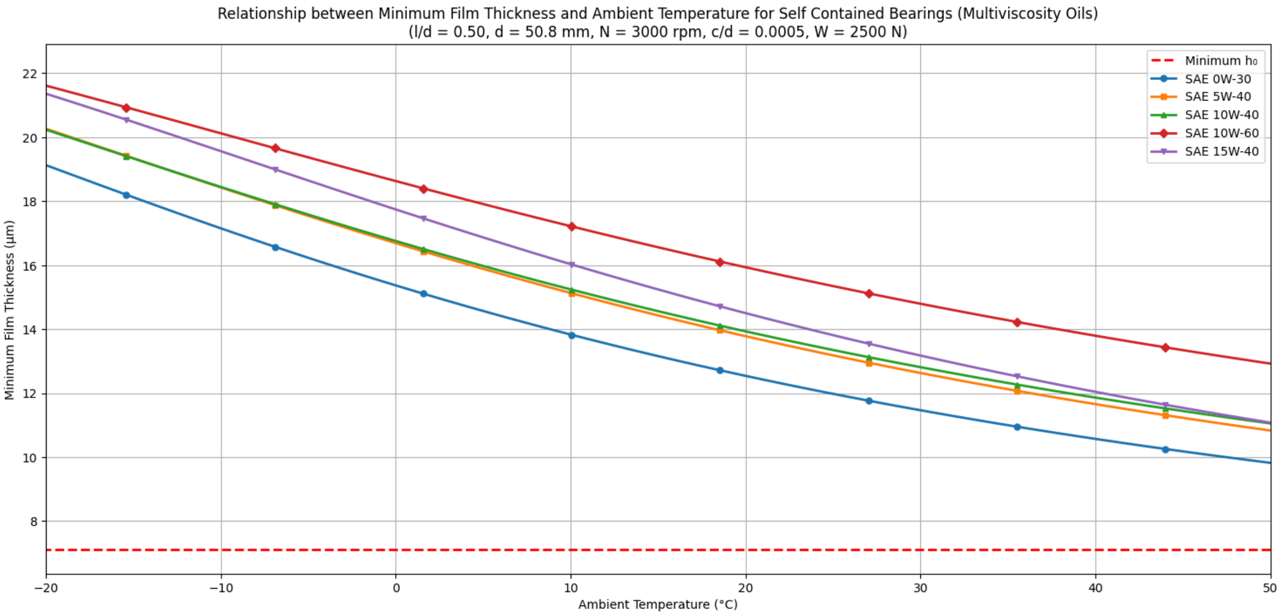


Fig 4.121:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (l/d = 0.50, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

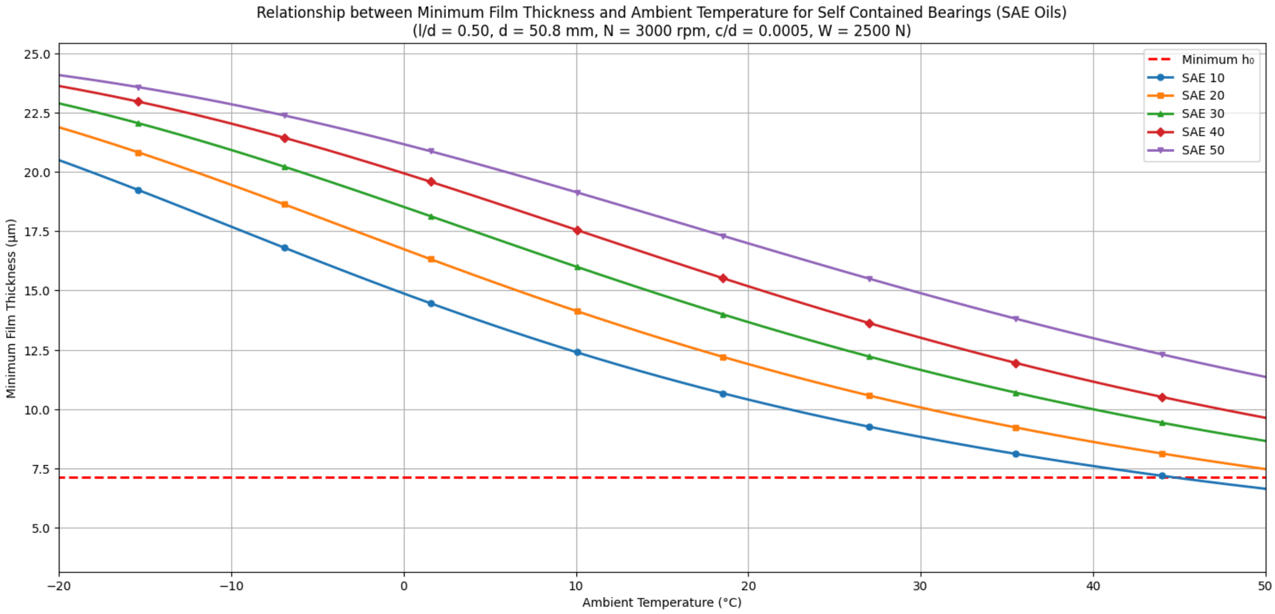


Fig 4.122:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (l/d = 0.50, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

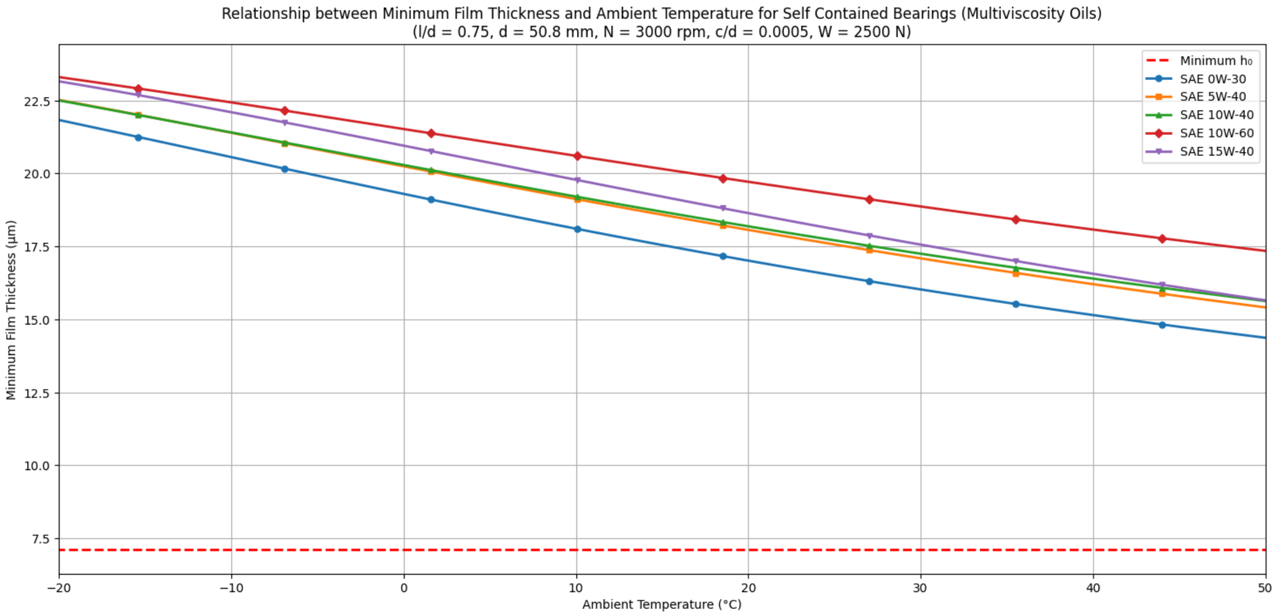


Fig 4.123:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (l/d = 0.75, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

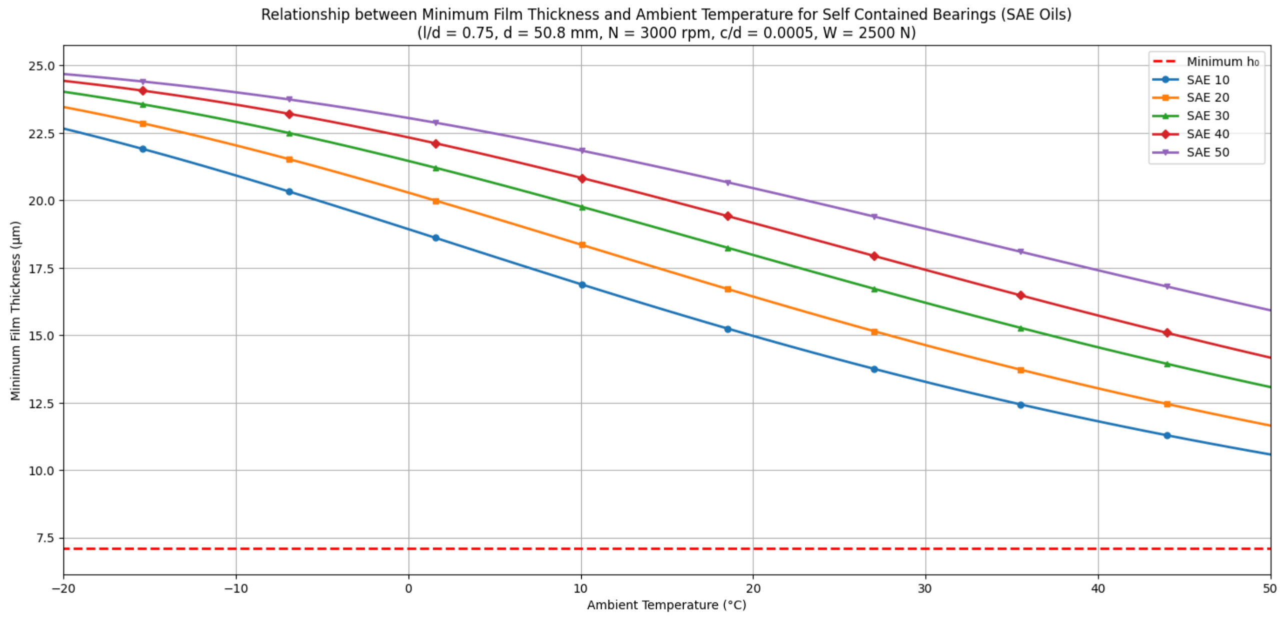


Fig 4.124:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (l/d = 0.75, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

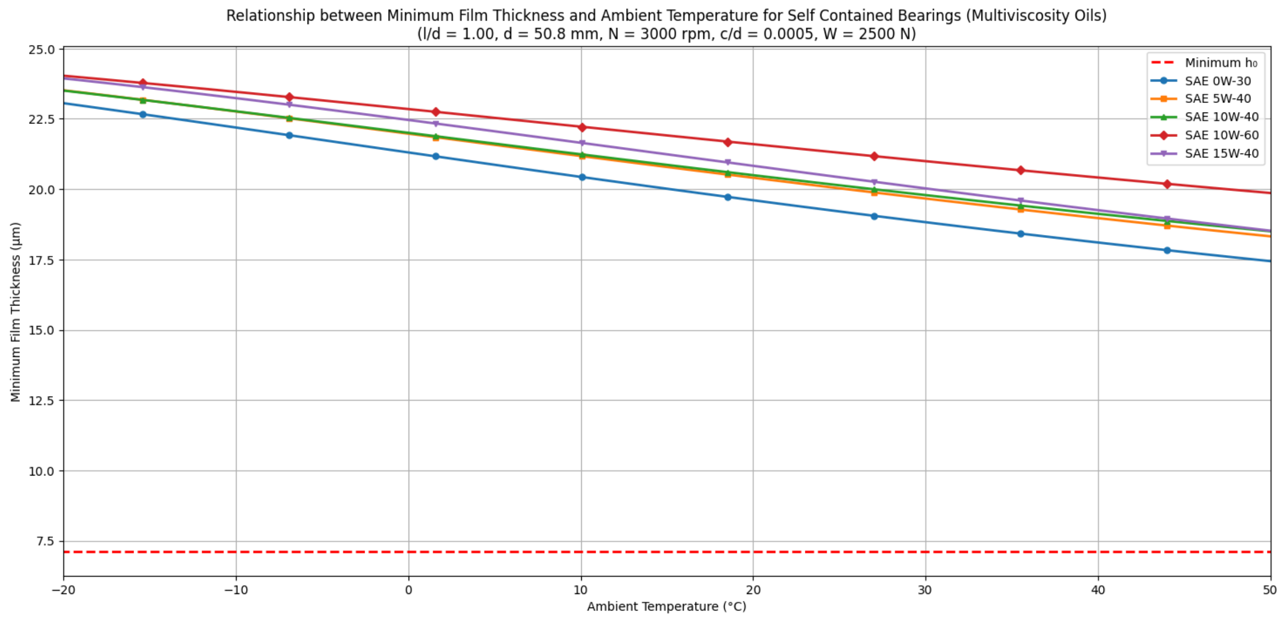


Fig 4.125:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (l/d = 1.00, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

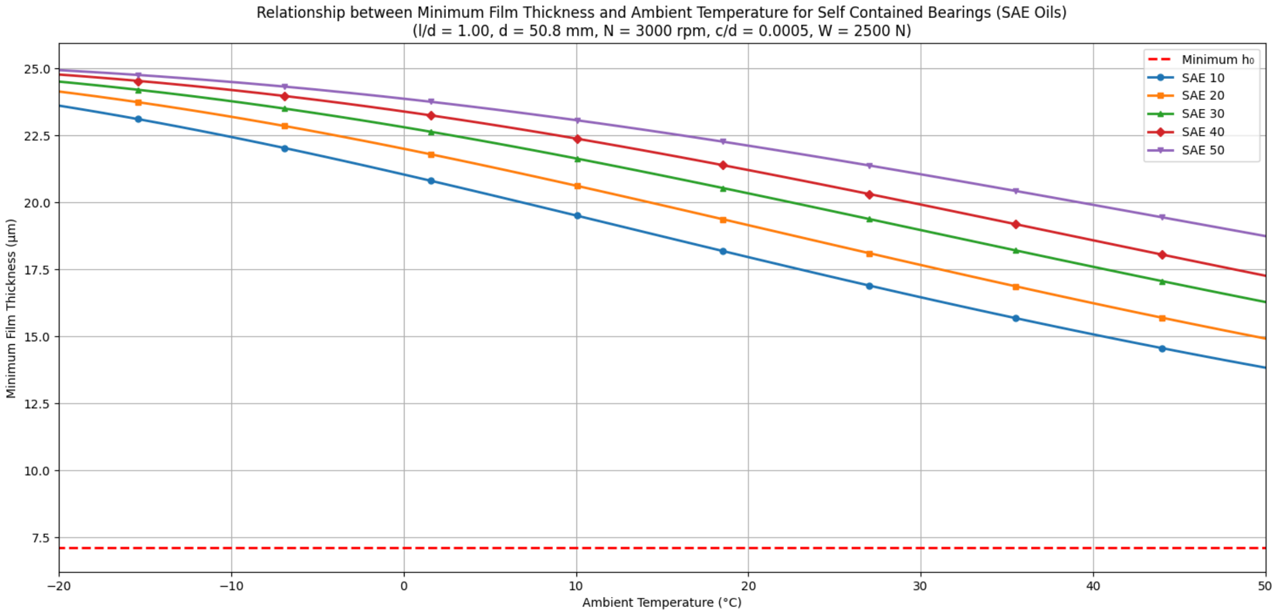


Fig 4.126:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (l/d = 1.00, d = 50.8 mm, N = 3000 rpm, c/d = 0.0005, W = 2500 N)

## 4.1.13 Effect of Varying Rotational Speed (Self Contained Bearing)

The clearance to diameter ratio is varied keeping journal diameter, length to diameter ratio, rotational speed and load constant in figure 4.127 to figure 4.140. We can see that minimum film thickness increases with increasing clearance to diameter ratio. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

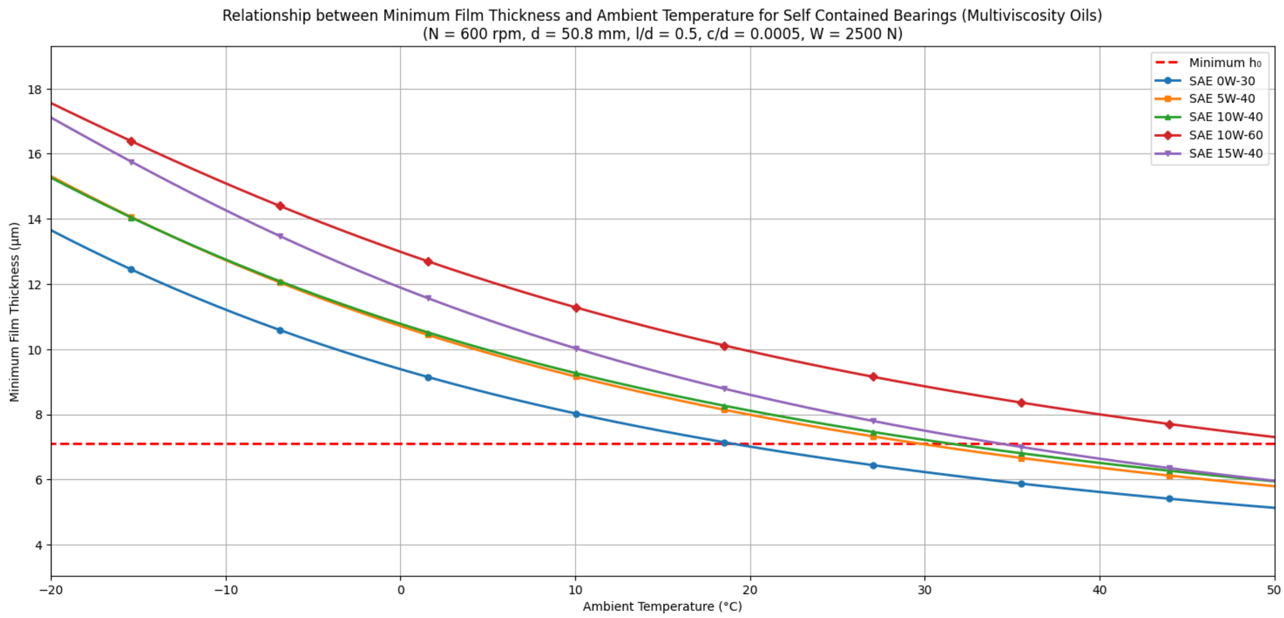

Fig 4.127:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (N = 600 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

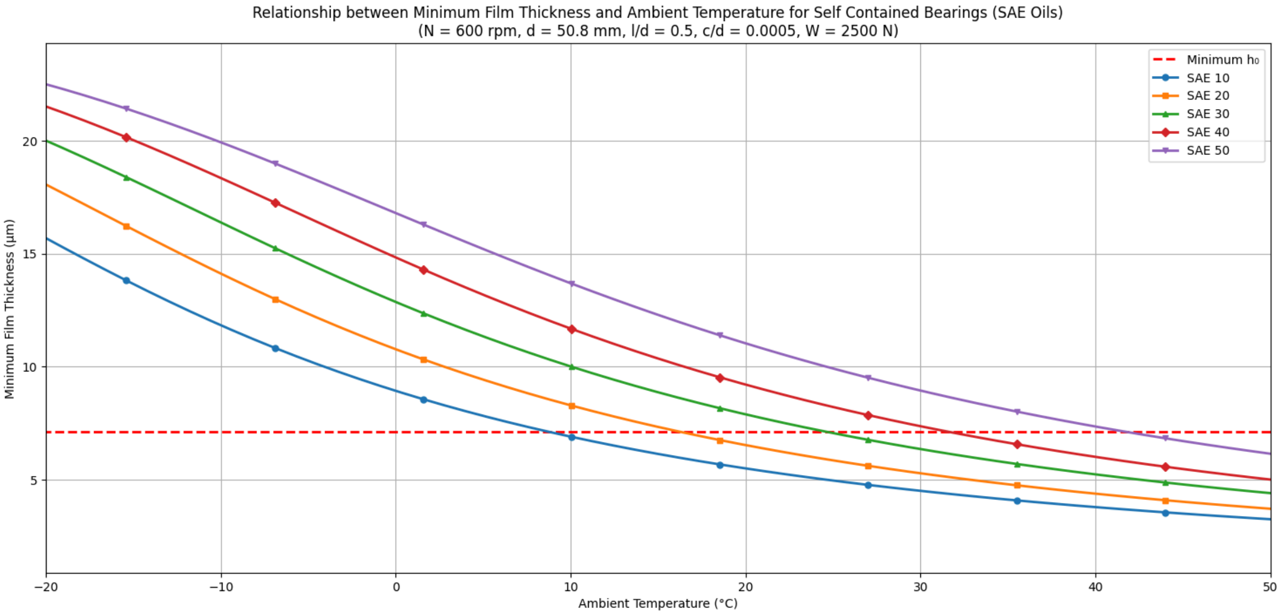


Fig 4.128:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (N = 600 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

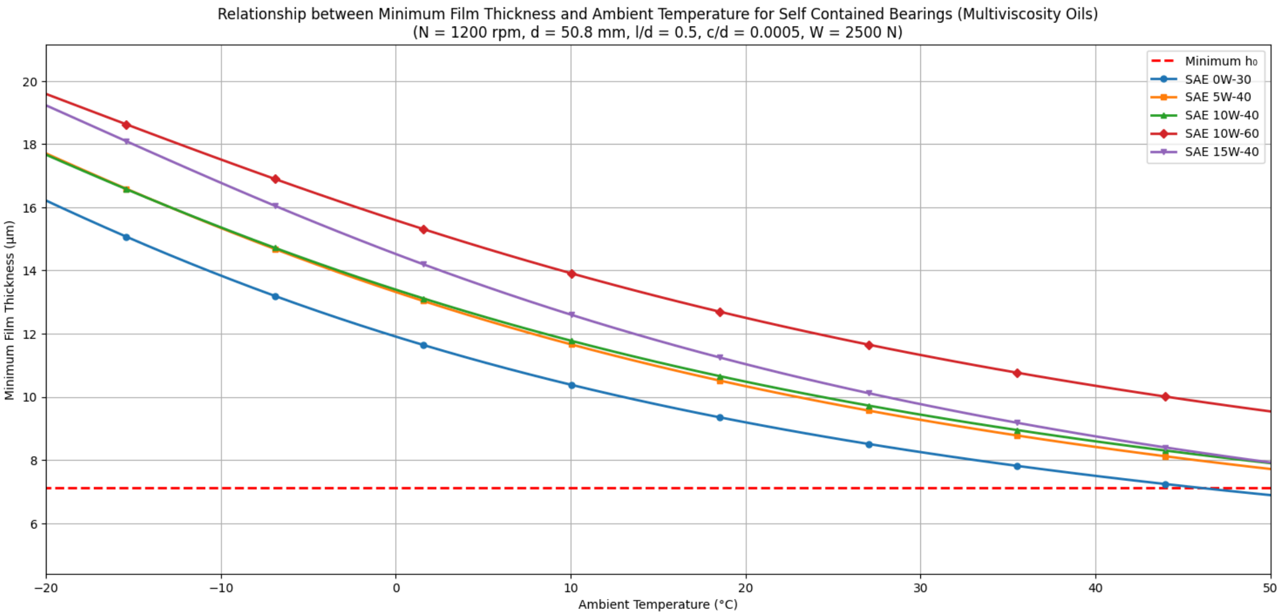


Fig 4.129:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (N = 1200 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

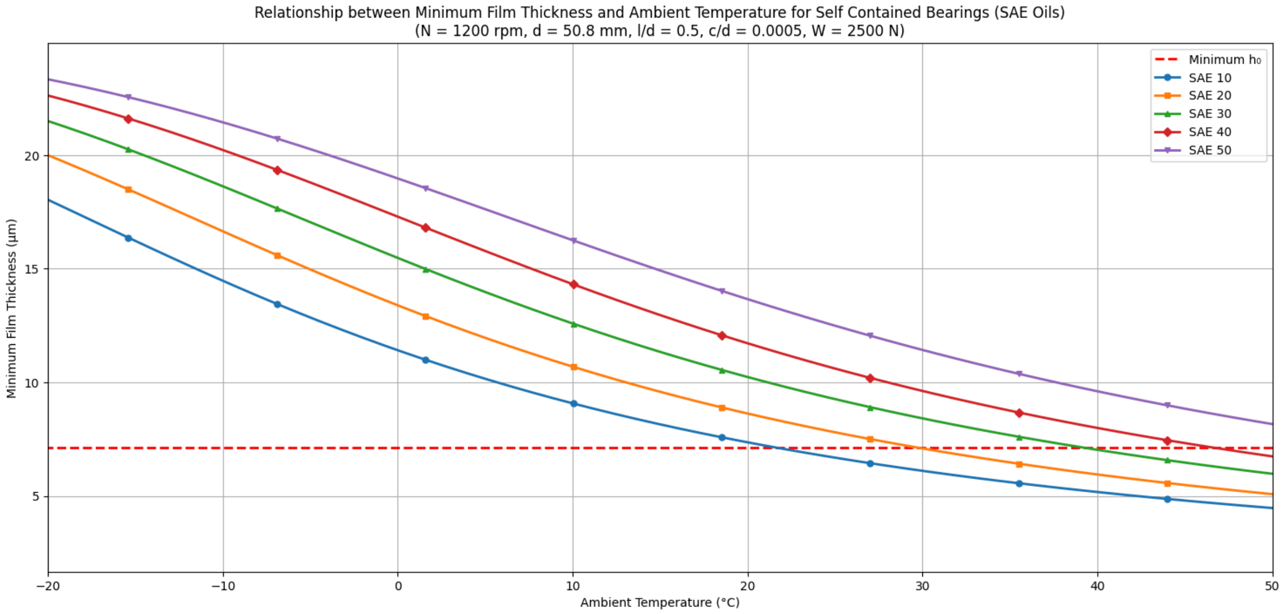


Fig 4.130:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (N = 1200 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

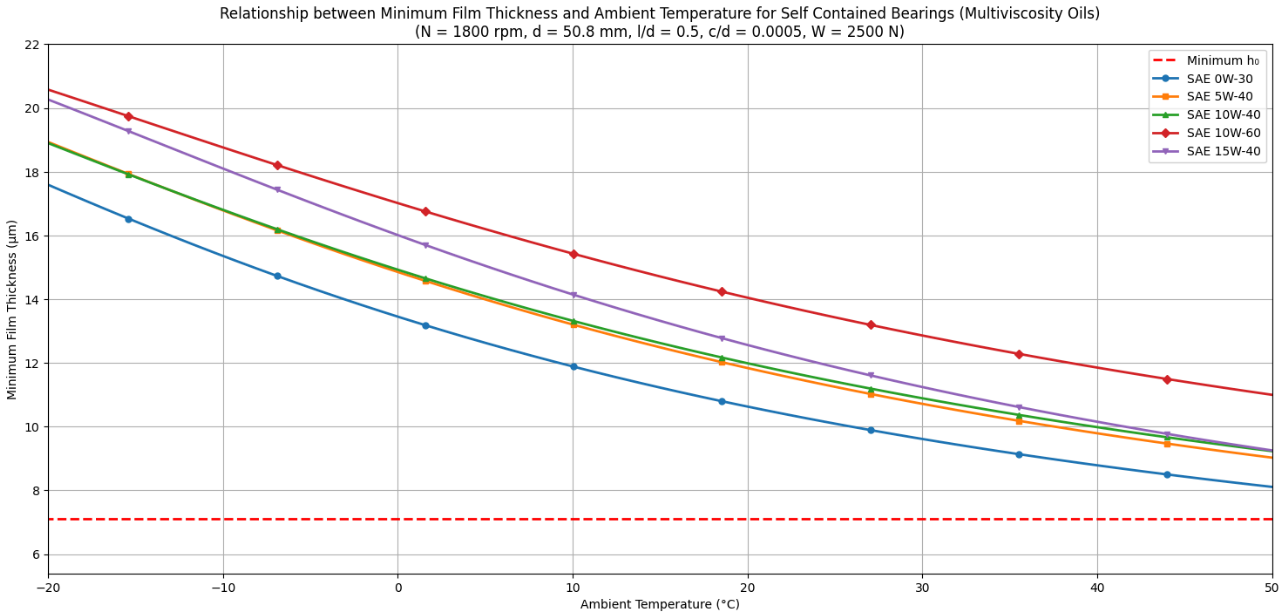


Fig 4.131:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (N = 1800 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

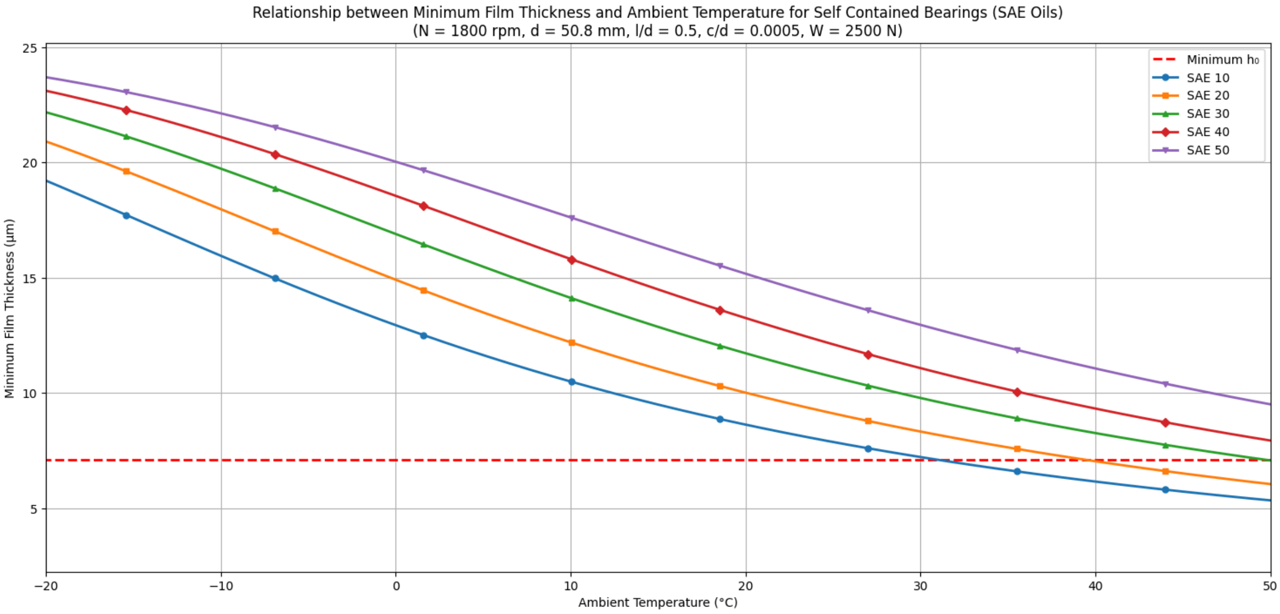


Fig 4.132:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (N = 1800 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

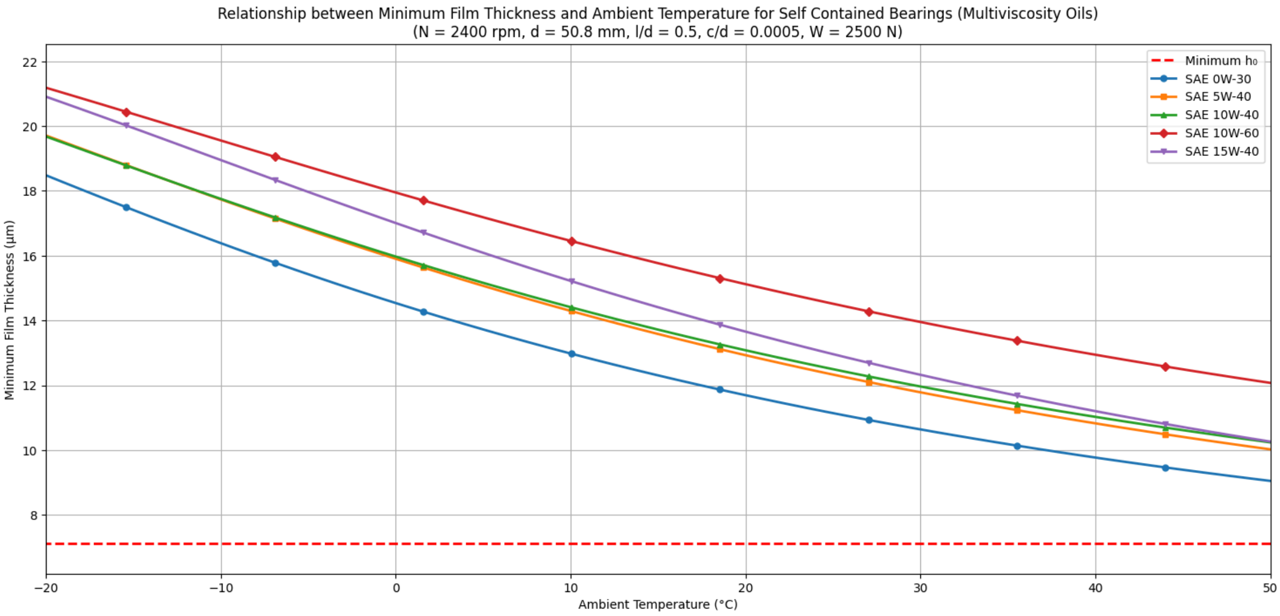


Fig 4.133:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (N = 2400 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

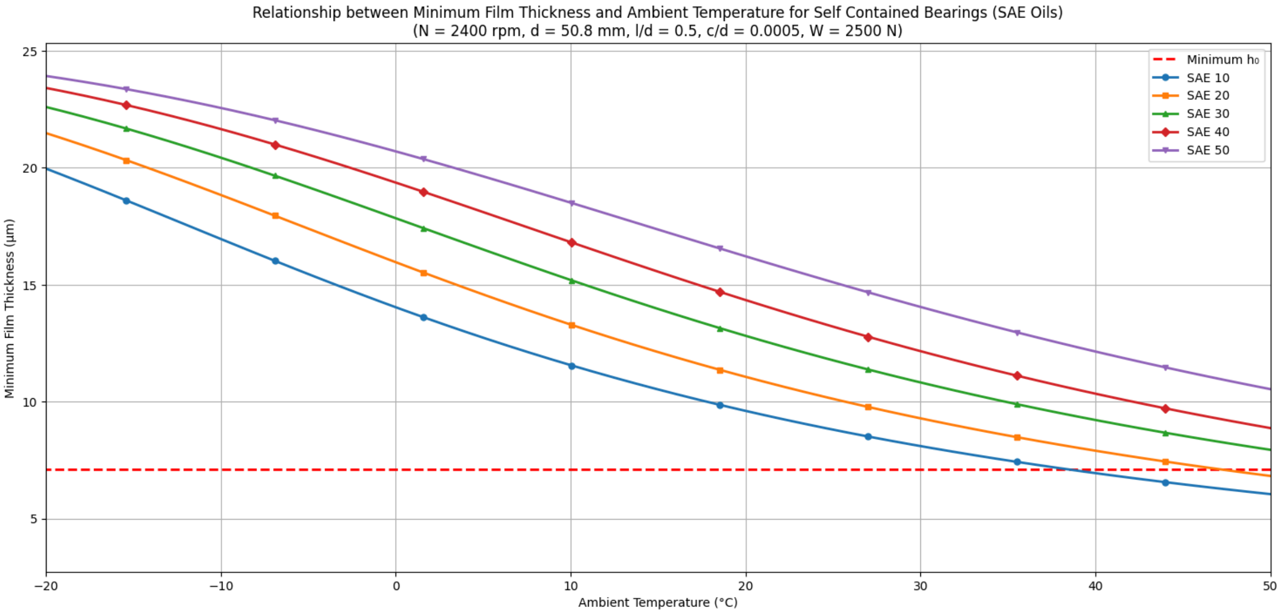


Fig 4.134:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (N = 2400 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

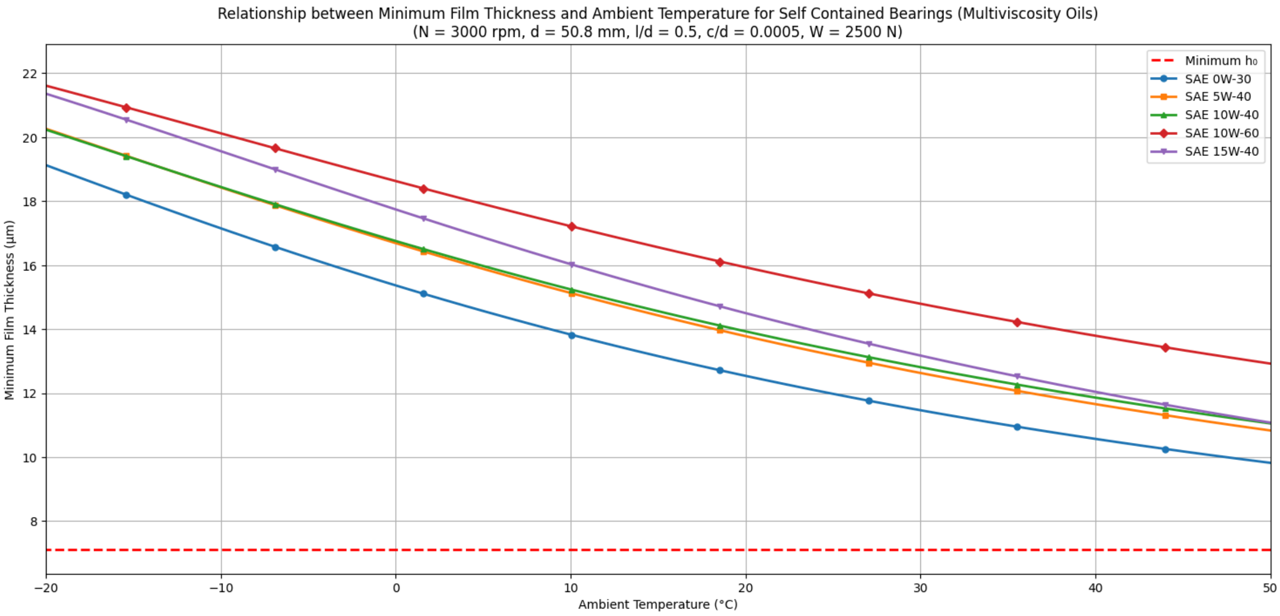


Fig 4.135:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (N = 3000 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

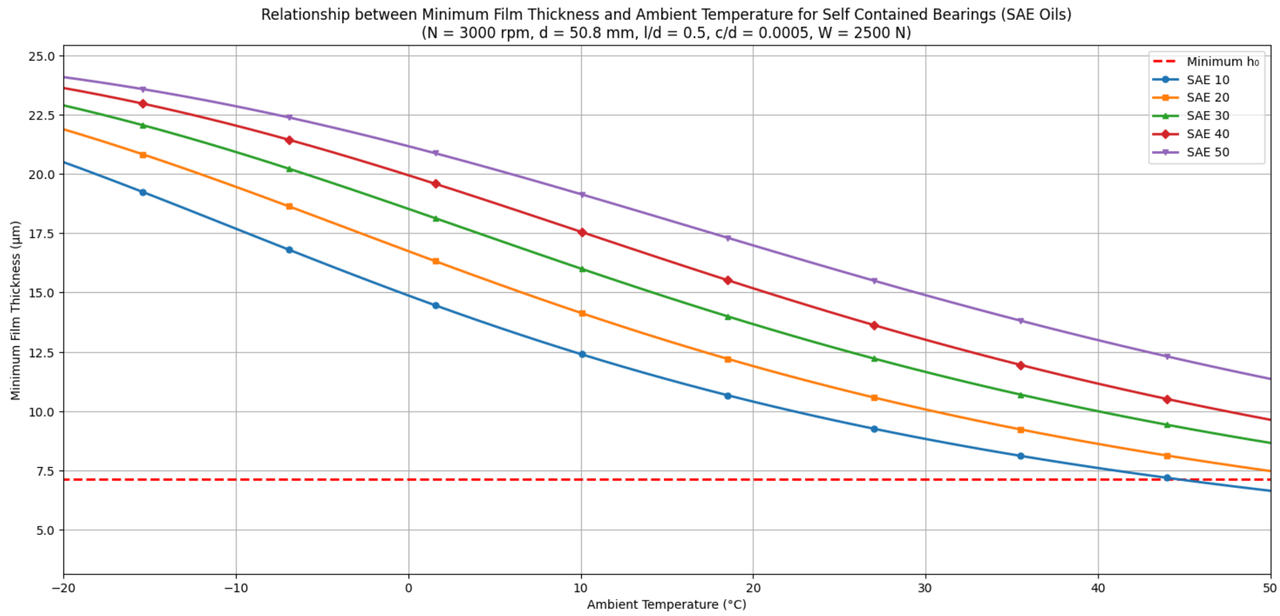


Fig 4.136:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (N = 3000 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

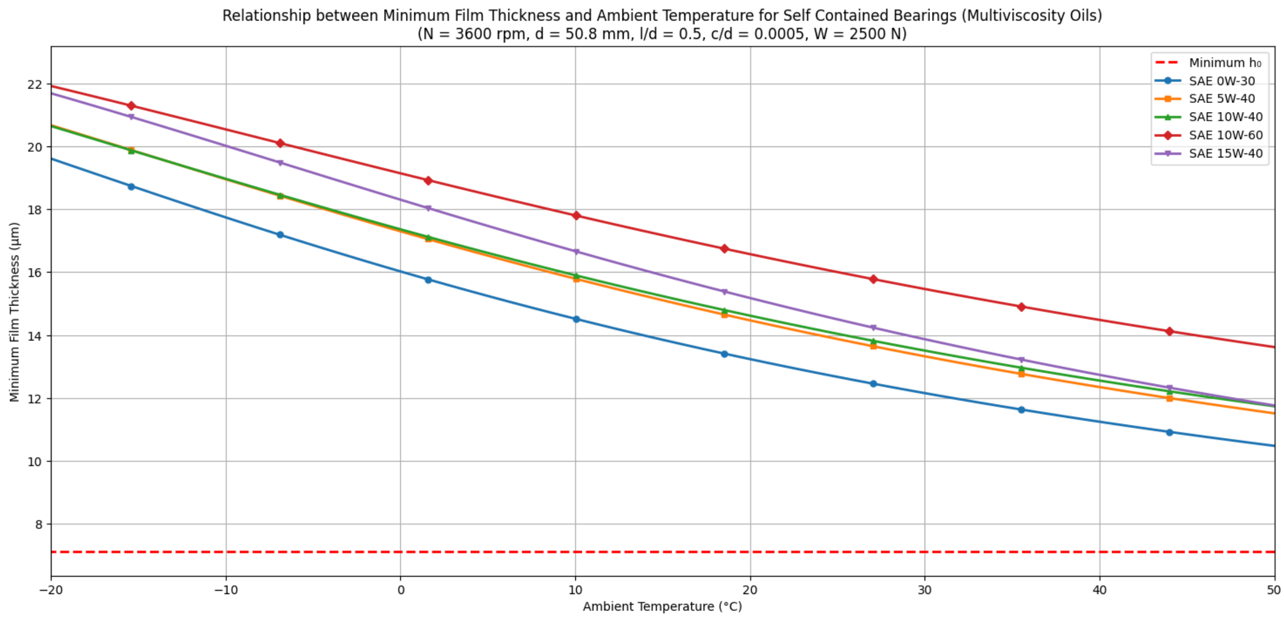


Fig 4.137:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (N = 3600 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

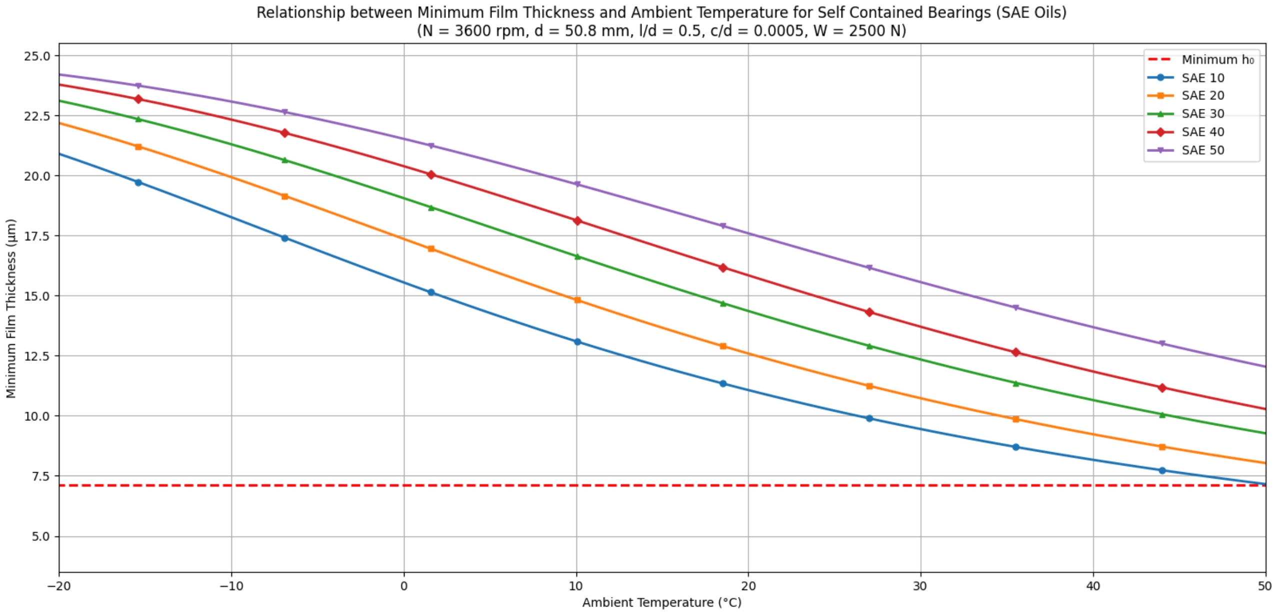


Fig 4.138:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (N = 3600 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

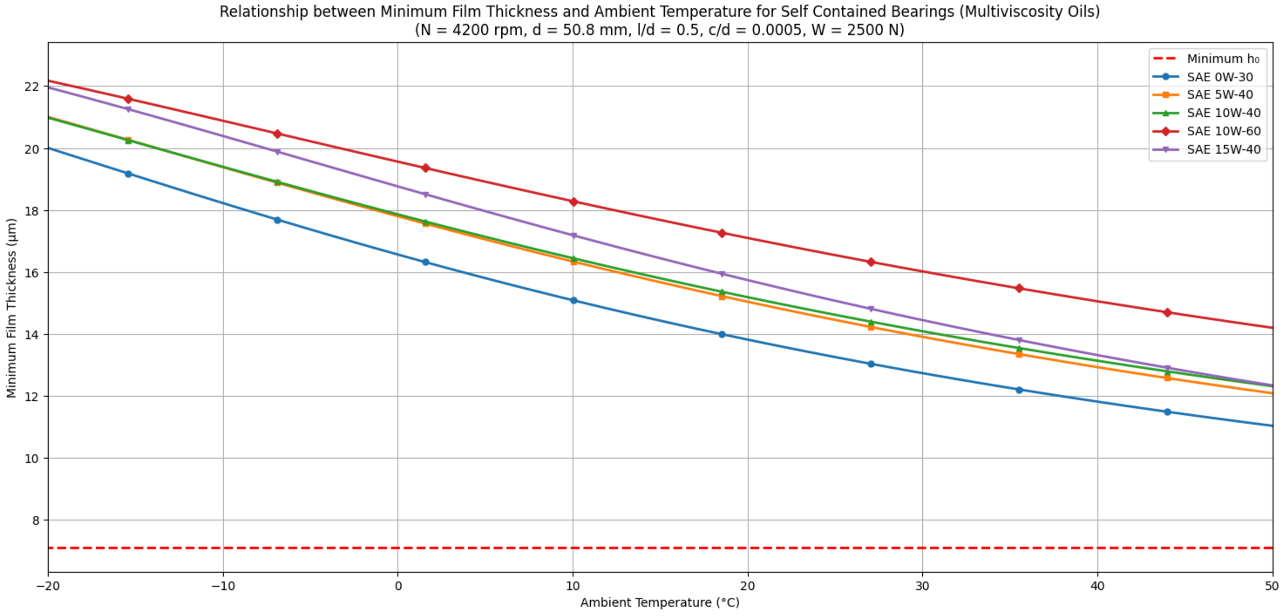


Fig 4.139:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (N = 4200 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

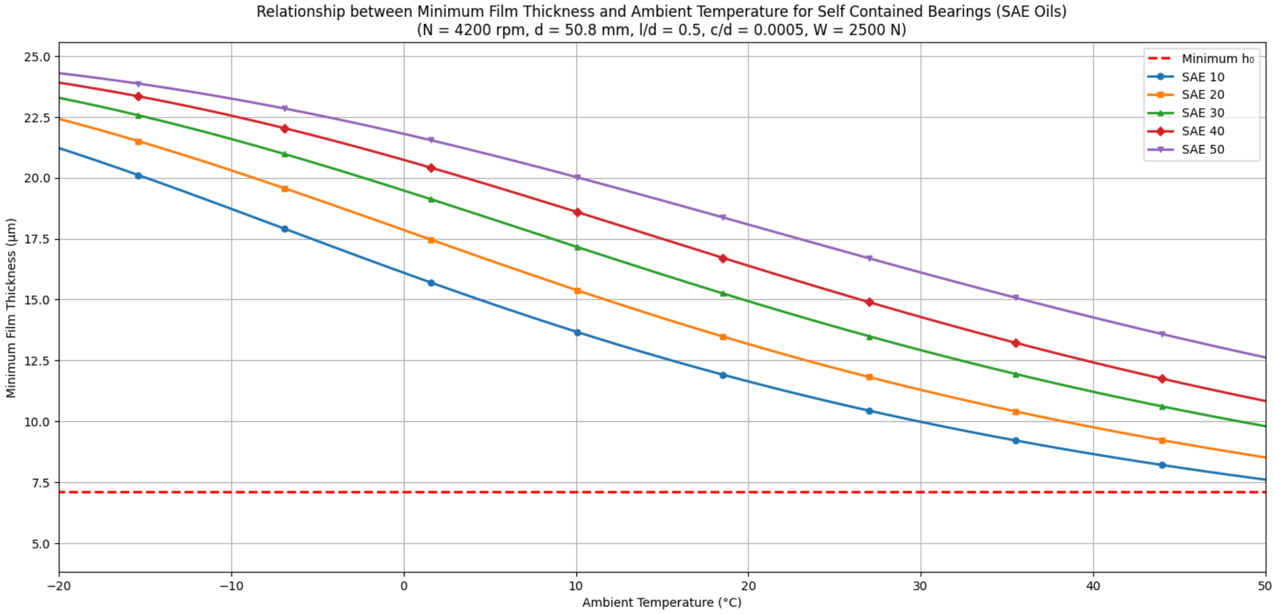


Fig 4.140:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (N = 4200 rpm, d = 50.8 mm, l/d = 0.5, c/d = 0.0005, W = 2500 N)

## 4.1.14 Effect of Varying Clearance Ratio (Self Contained Bearing)

The clearance to diameter ratio is varied keeping journal diameter, length to diameter ratio, rotational speed and load constant in figure 4.141 to figure 4.152. We can see that minimum film thickness increases with increasing clearance to diameter ratio. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

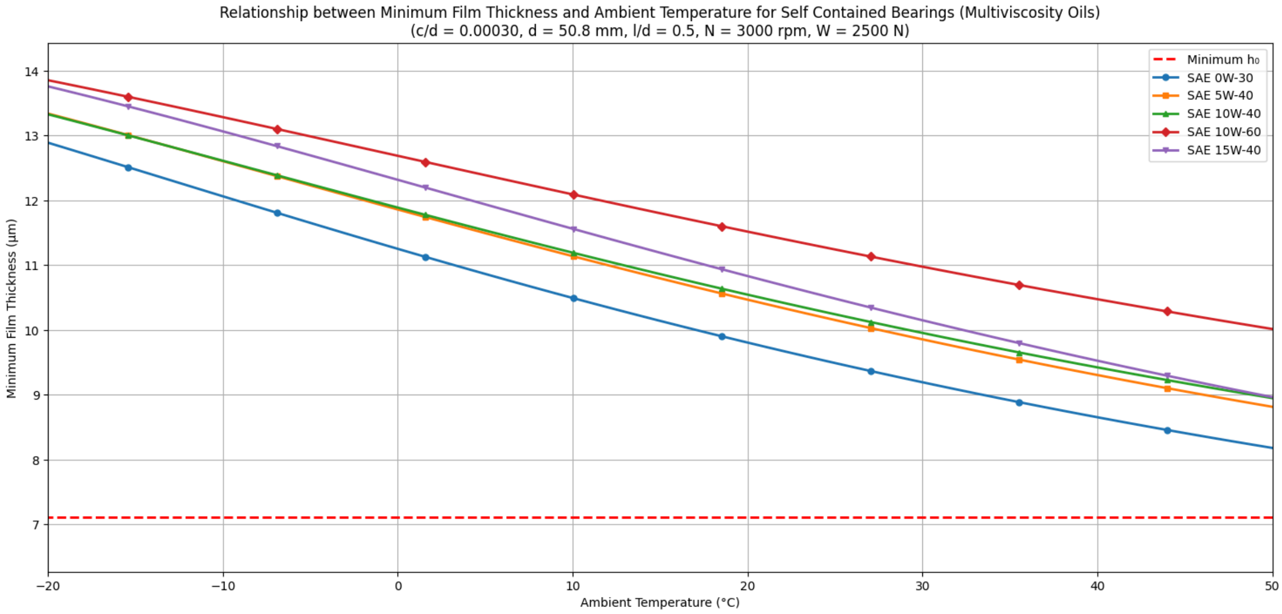

Fig 4.141:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (c/d = 0.00030, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

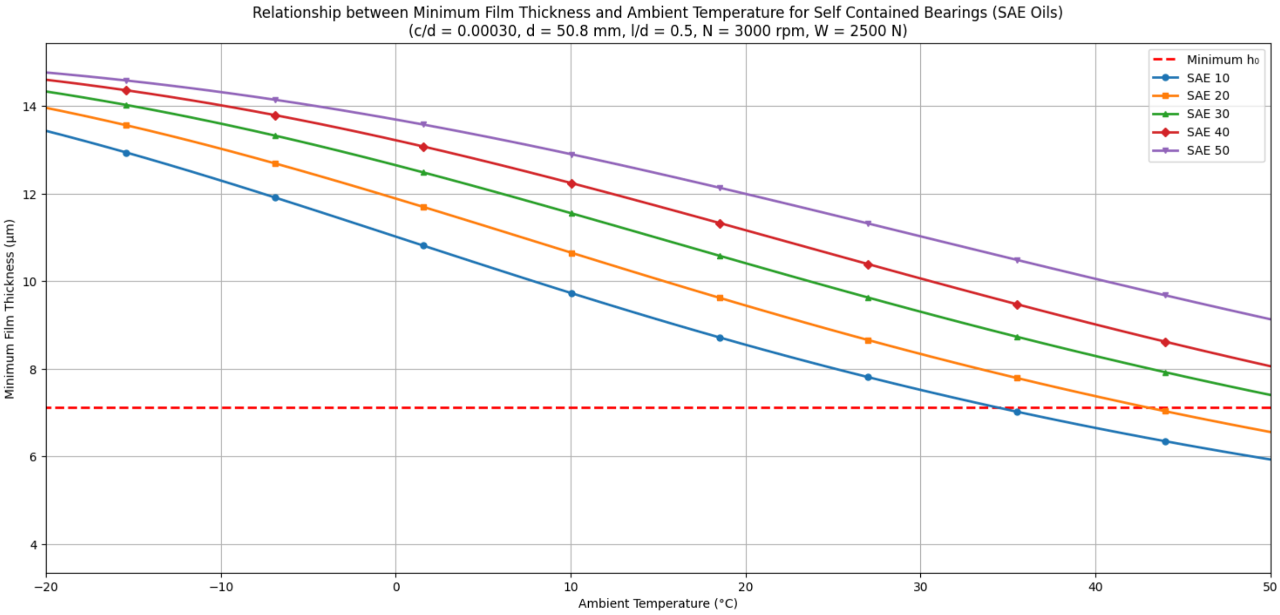


Fig 4.142:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (c/d = 0.00030, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

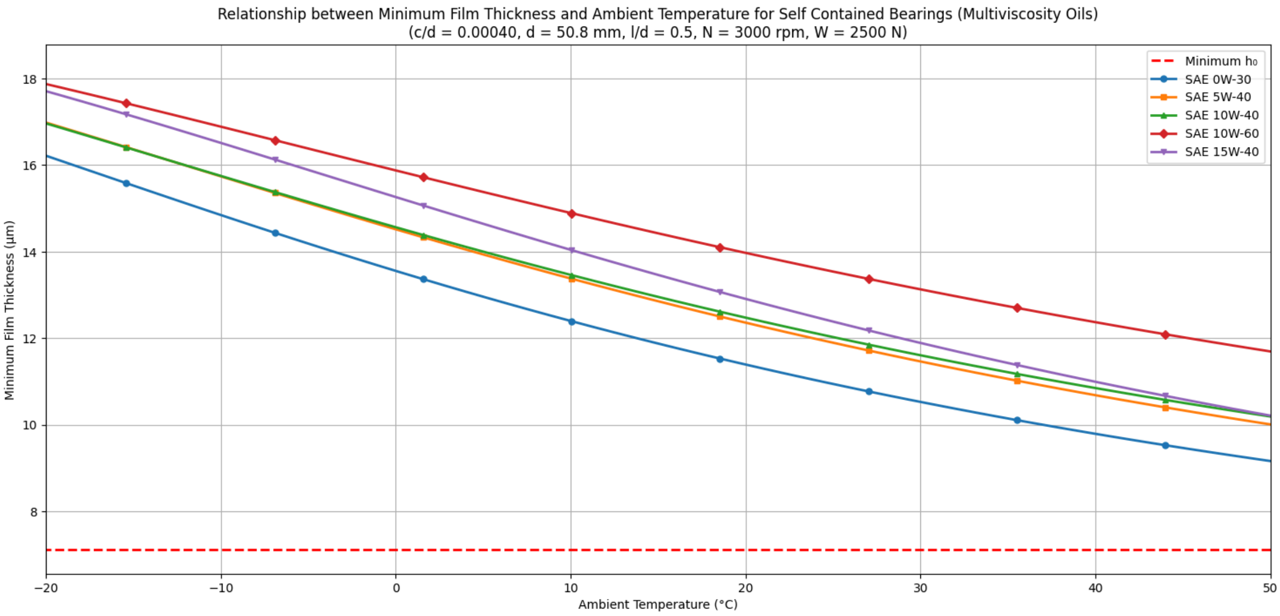


Fig 4.143:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (c/d = 0.00040, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

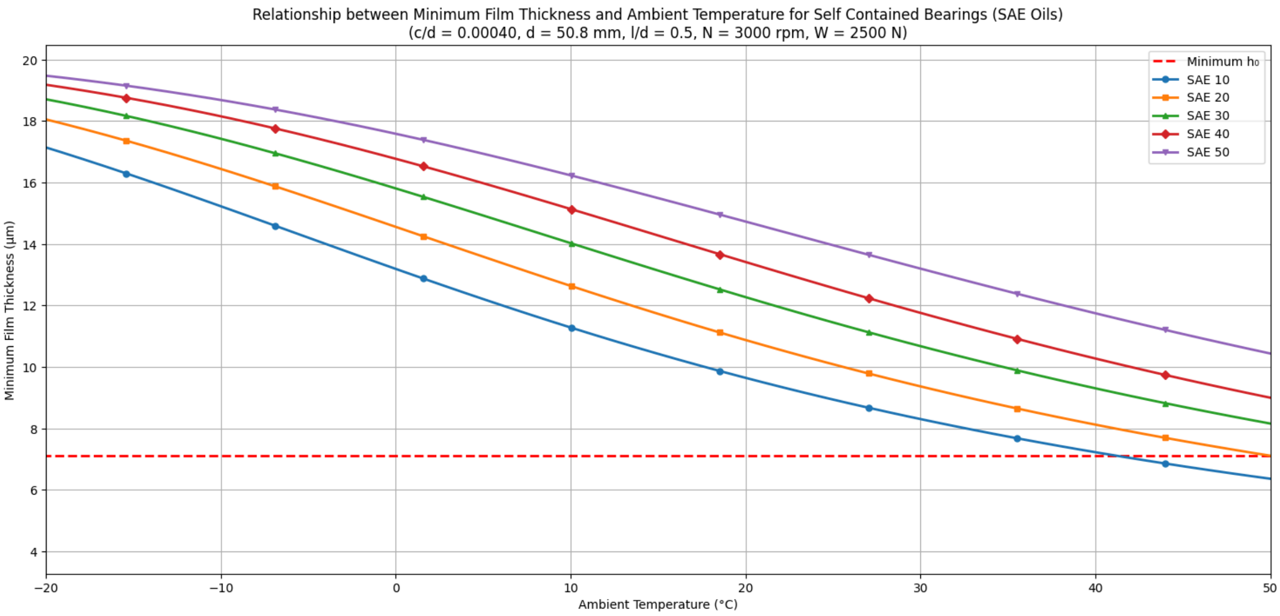


Fig 4.144:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (c/d = 0.00040, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

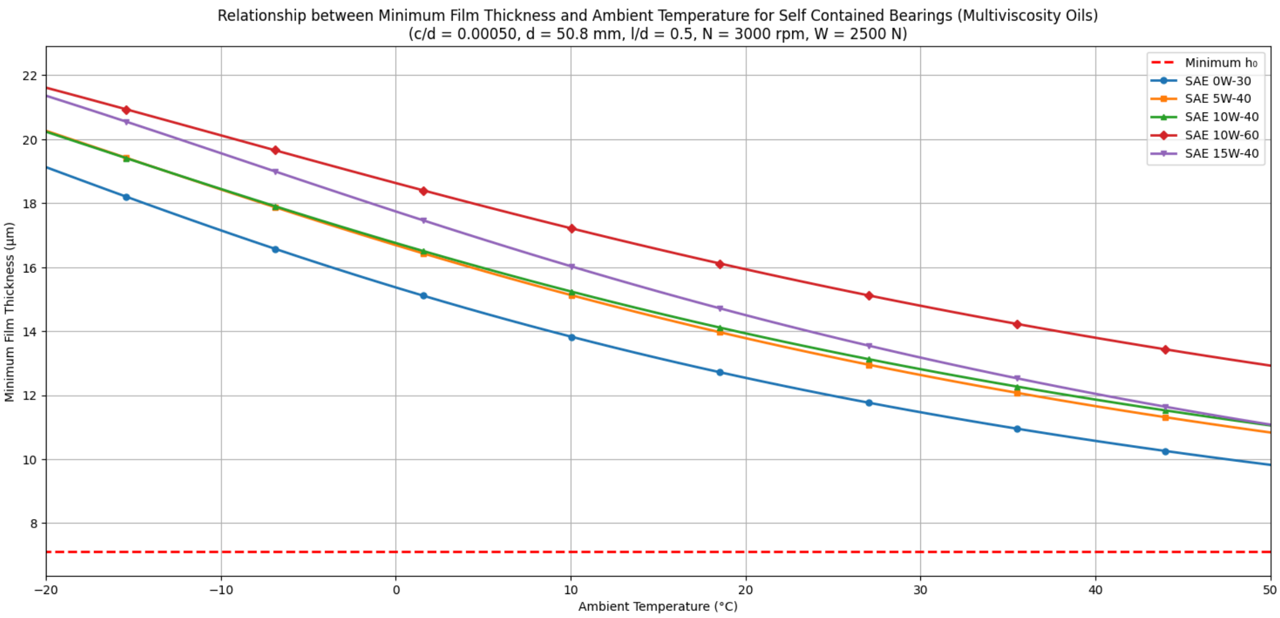


Fig 4.145:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (c/d = 0.00050, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

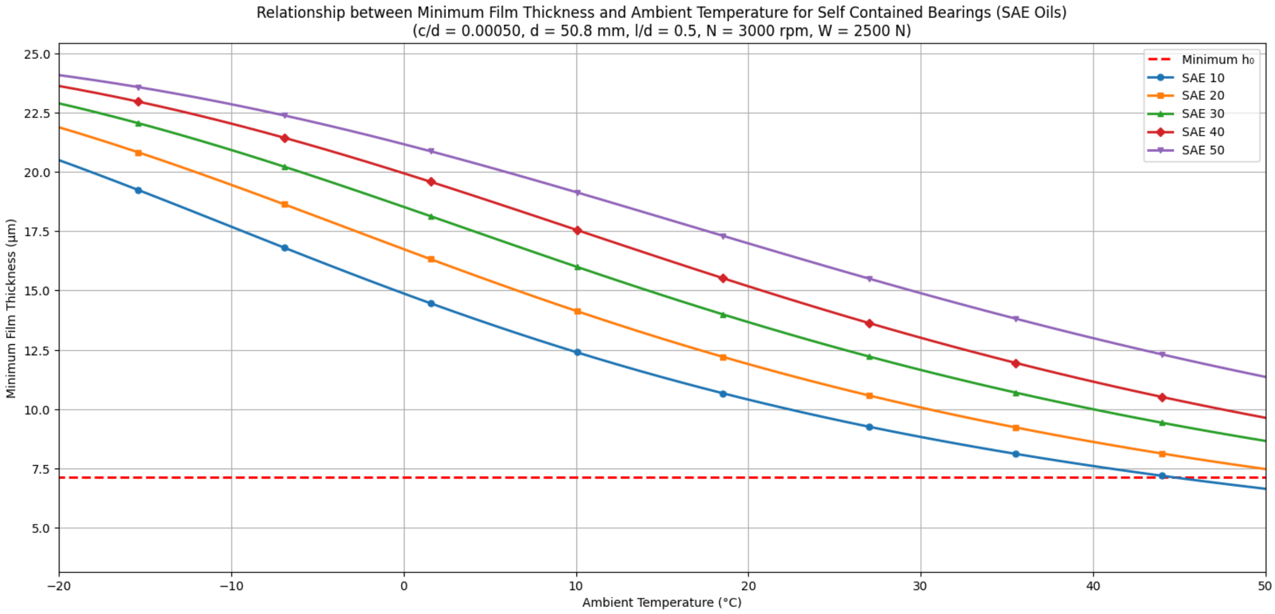


Fig 4.146:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (c/d = 0.00050, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

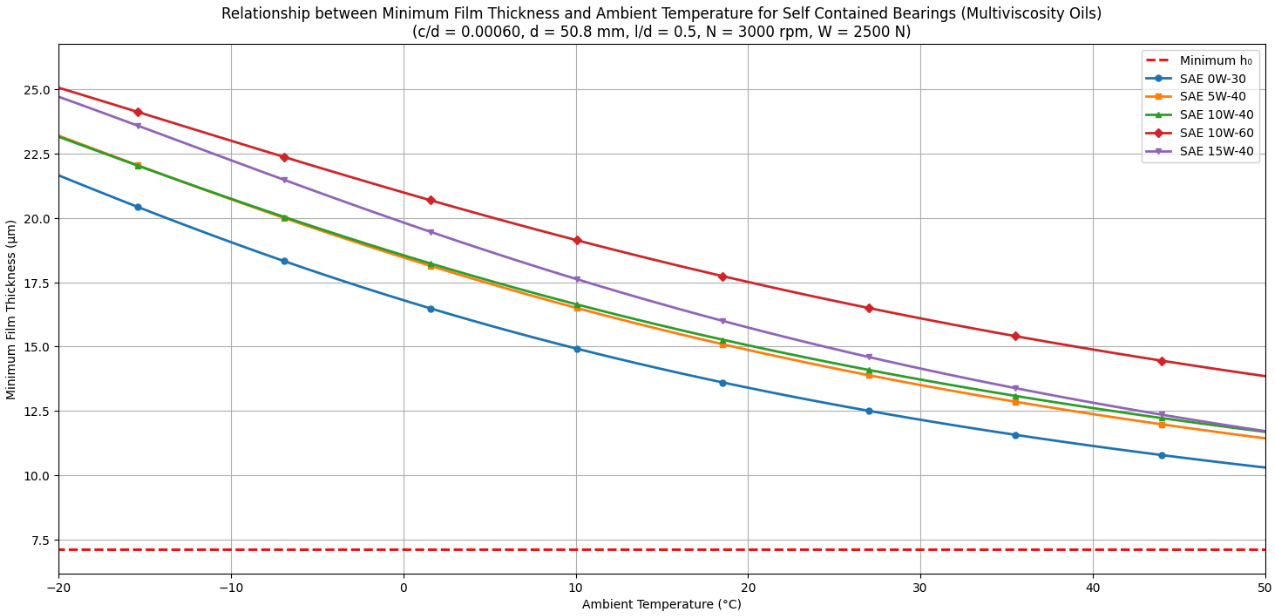


Fig 4.147:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (c/d = 0.00060, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

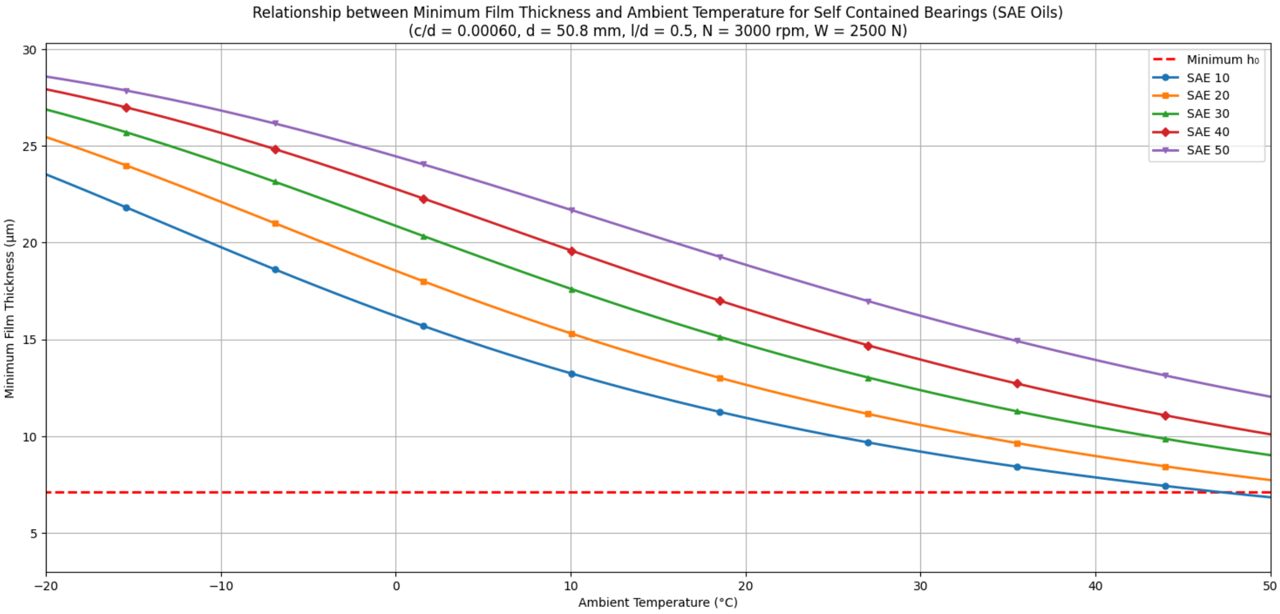


Fig 4.148:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (c/d = 0.00060, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

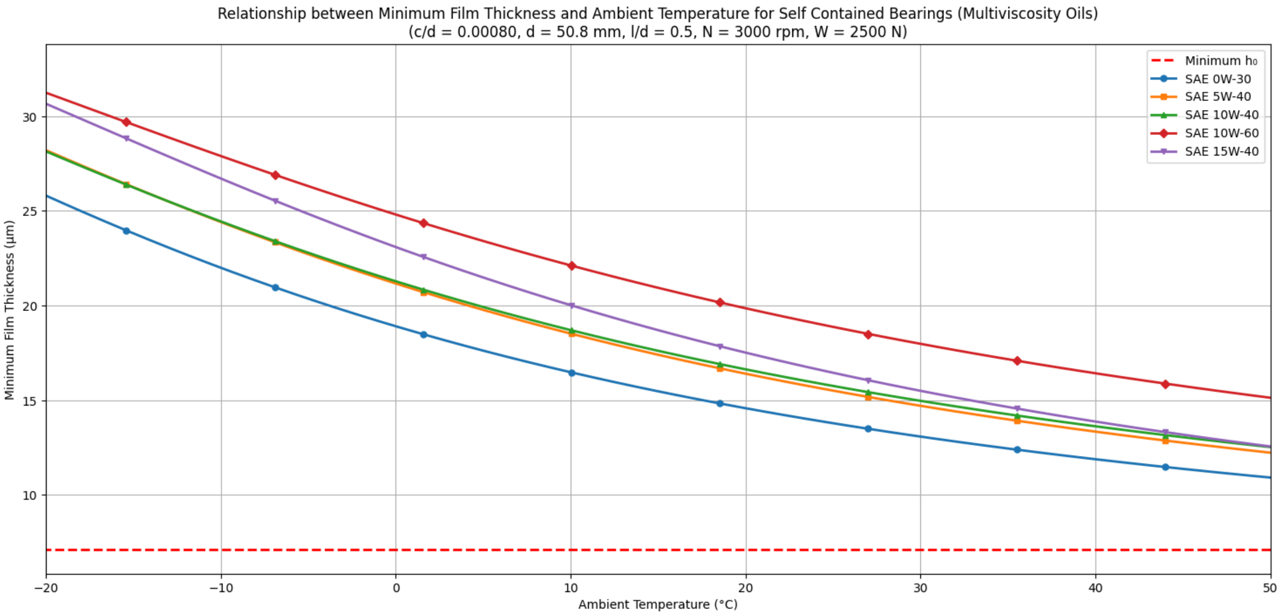


Fig 4.149:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (c/d = 0.00080, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

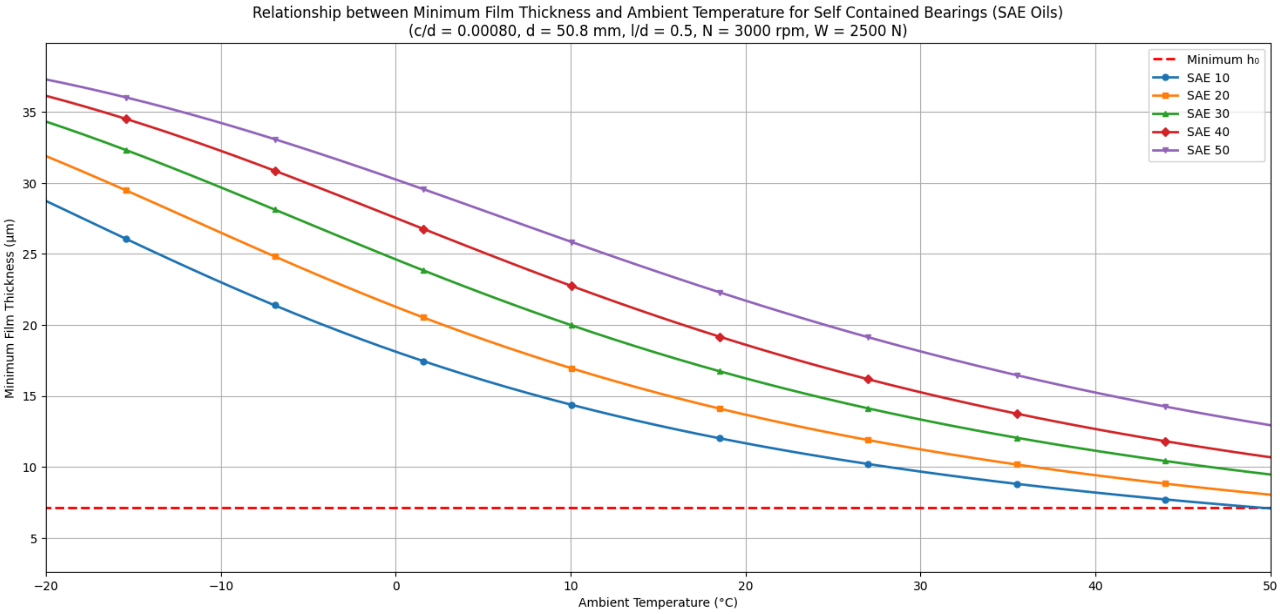


Fig 4.150:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (c/d = 0.00080, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, W = 2500 N)

## 4.1.15 Effect of Varying Load (Self Contained Bearing)

The load is varied keeping journal diameter, length to diameter ratio, rotational speed and clearance to diameter ratio constant in figure 4.151 to figure 4.162. We can see that minimum film thickness decreases with increasing load. We can also see that multiviscosity oils perform better than SAE oils in high ambient temperatures but SAE oils perform better than multiviscosity oils in low ambient temperatures.

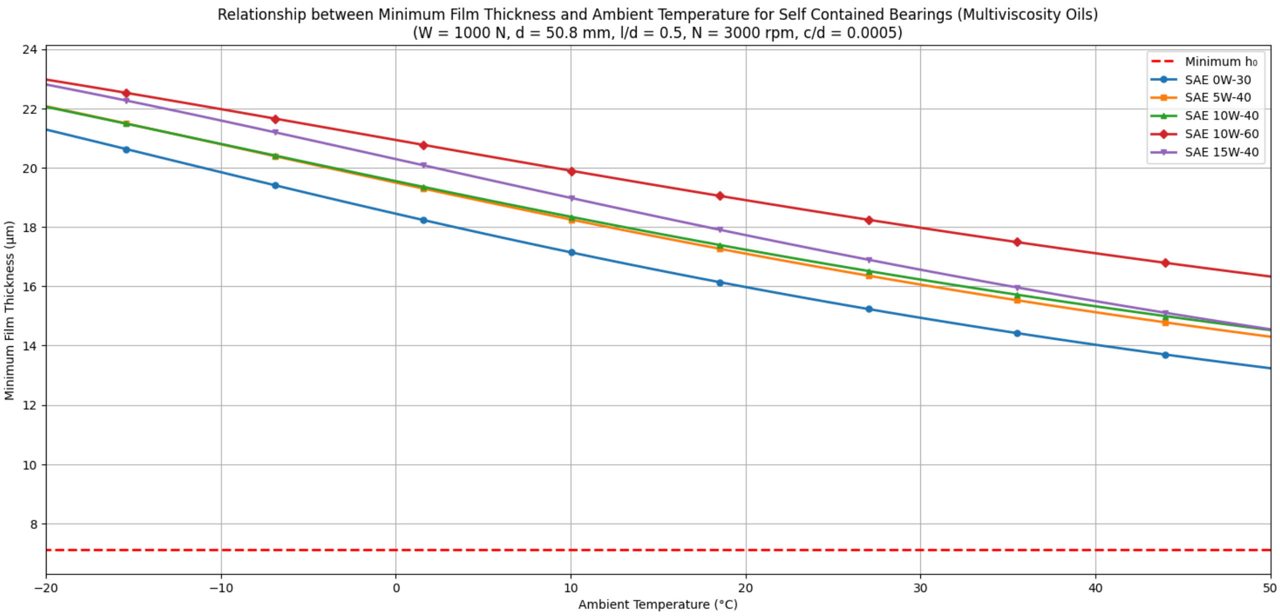

Fig 4.151:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (W = 1000 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

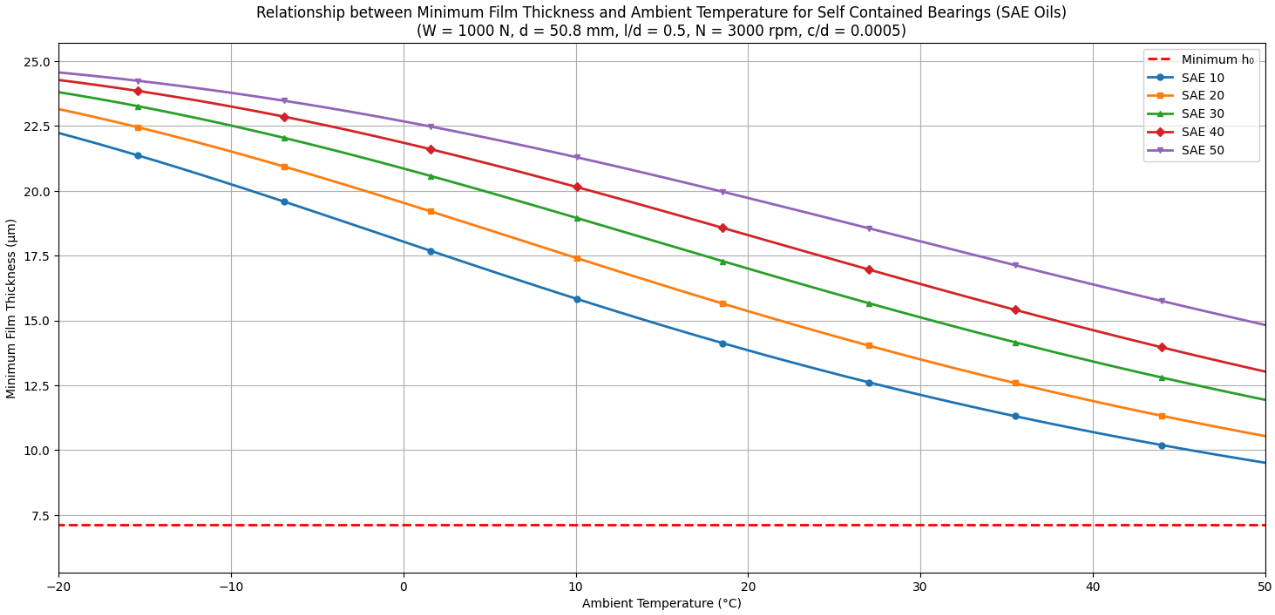


Fig 4.152:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (W = 1000 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

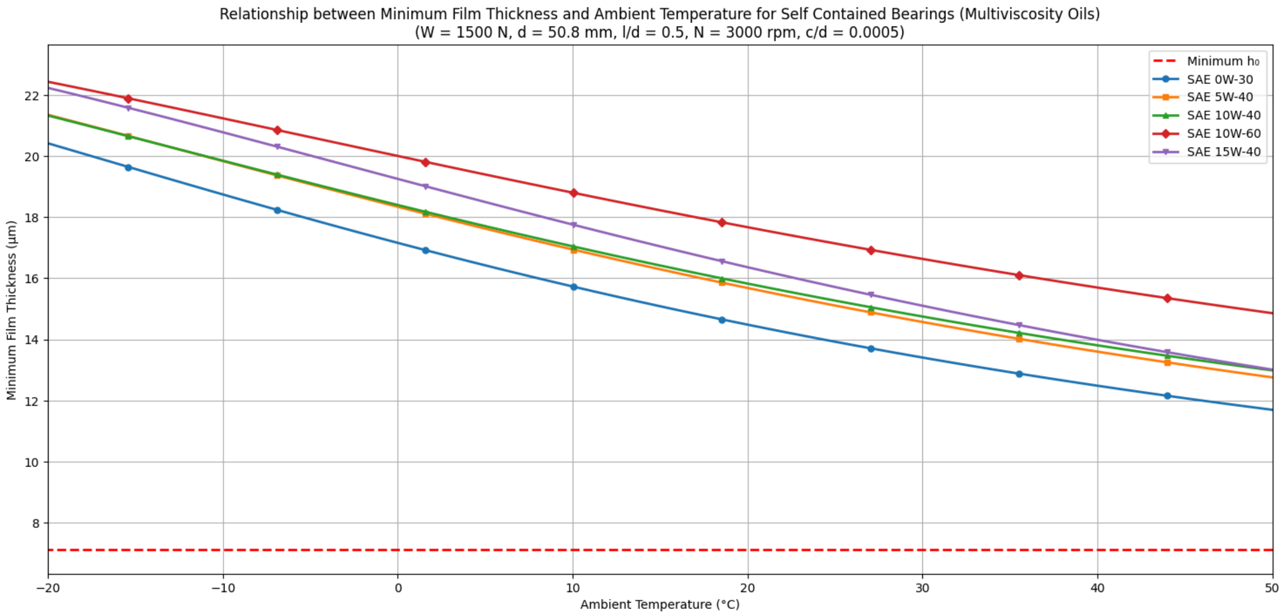


Fig 4.153:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (W = 1500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

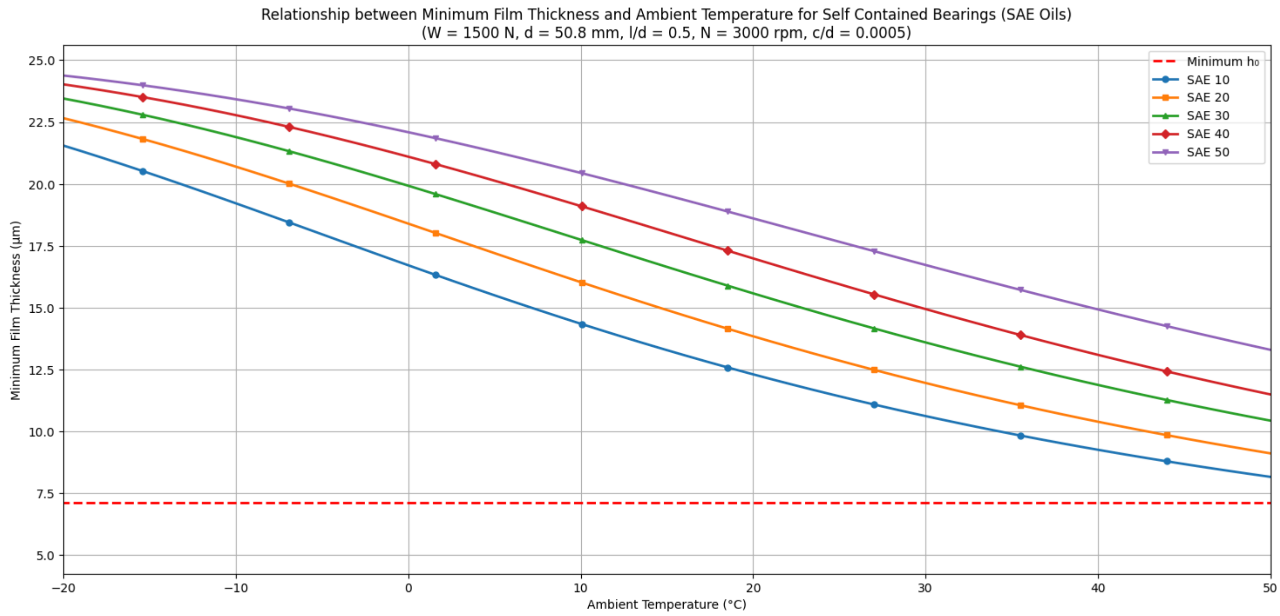


Fig 4.154:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (W = 1500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

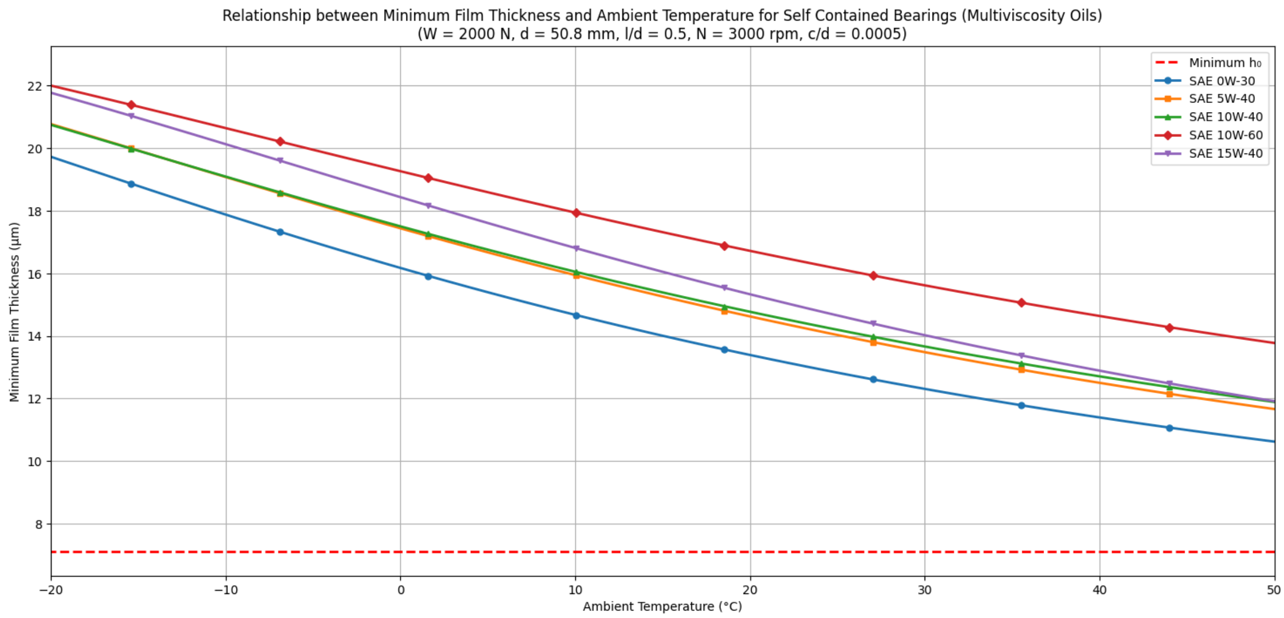


Fig 4.155:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (W = 2000 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

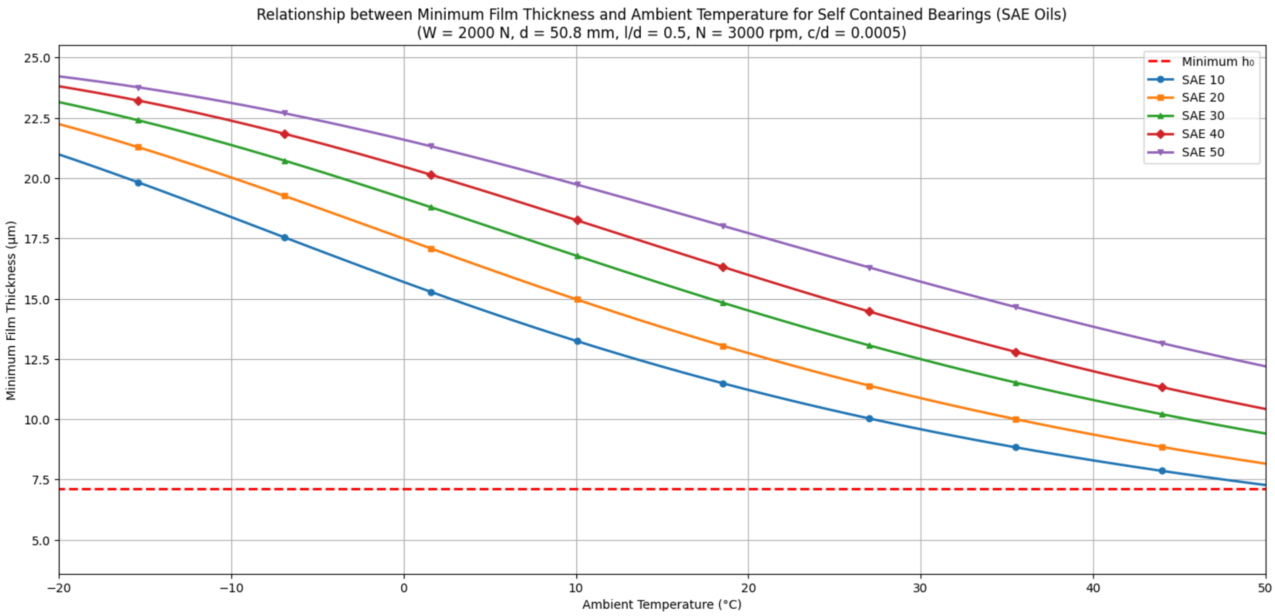


Fig 4.156:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (W = 2000 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

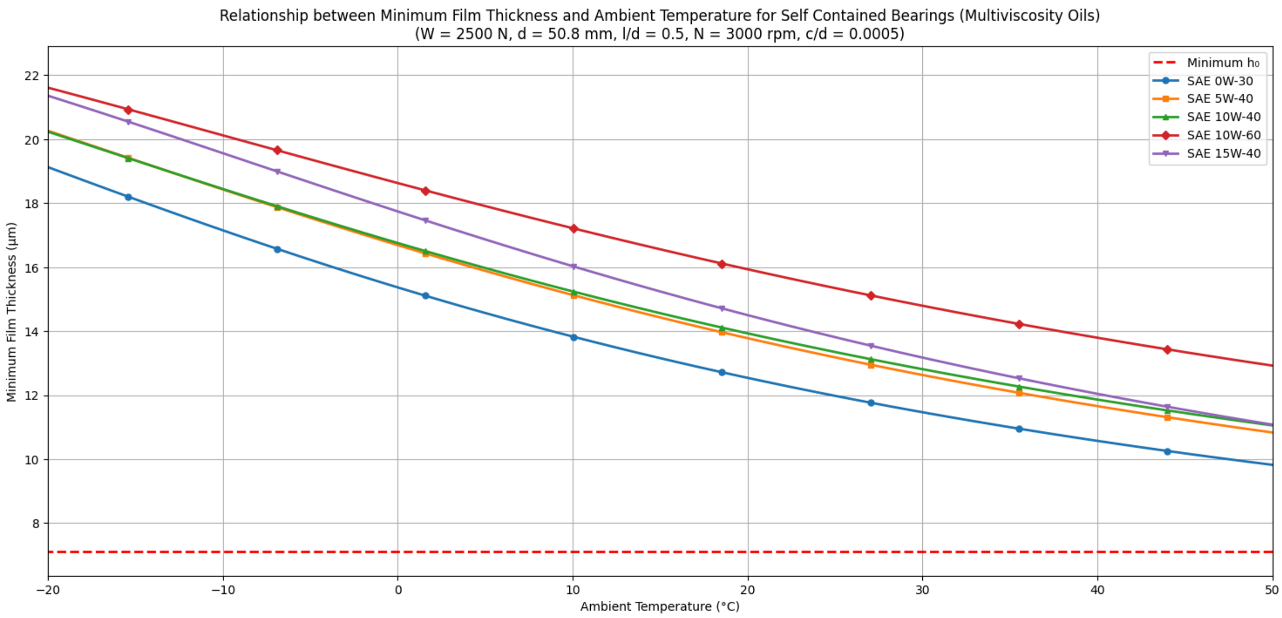


Fig 4.157:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (W = 2500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

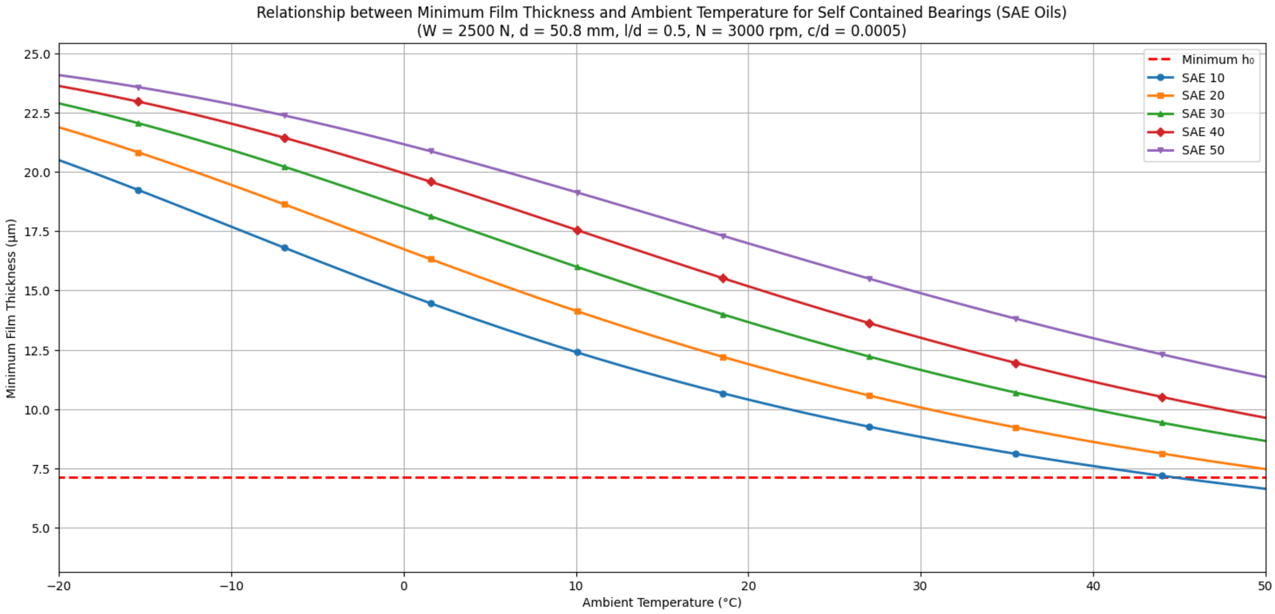


Fig 4.158:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (W = 2500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

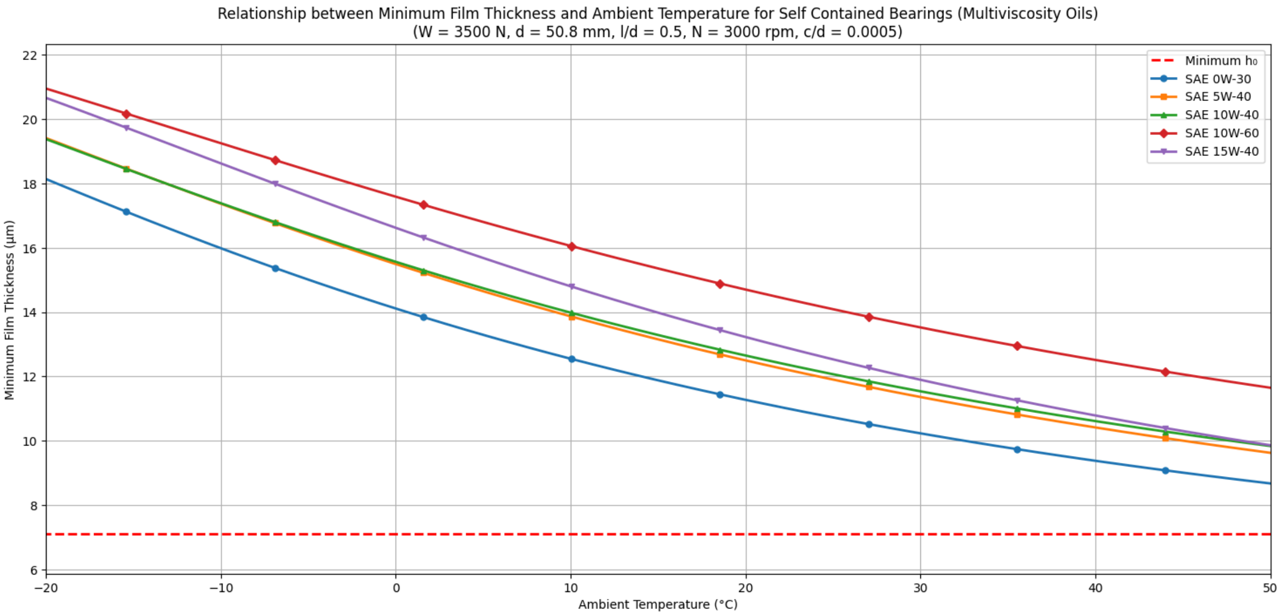


Fig 4.159:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (W = 3500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

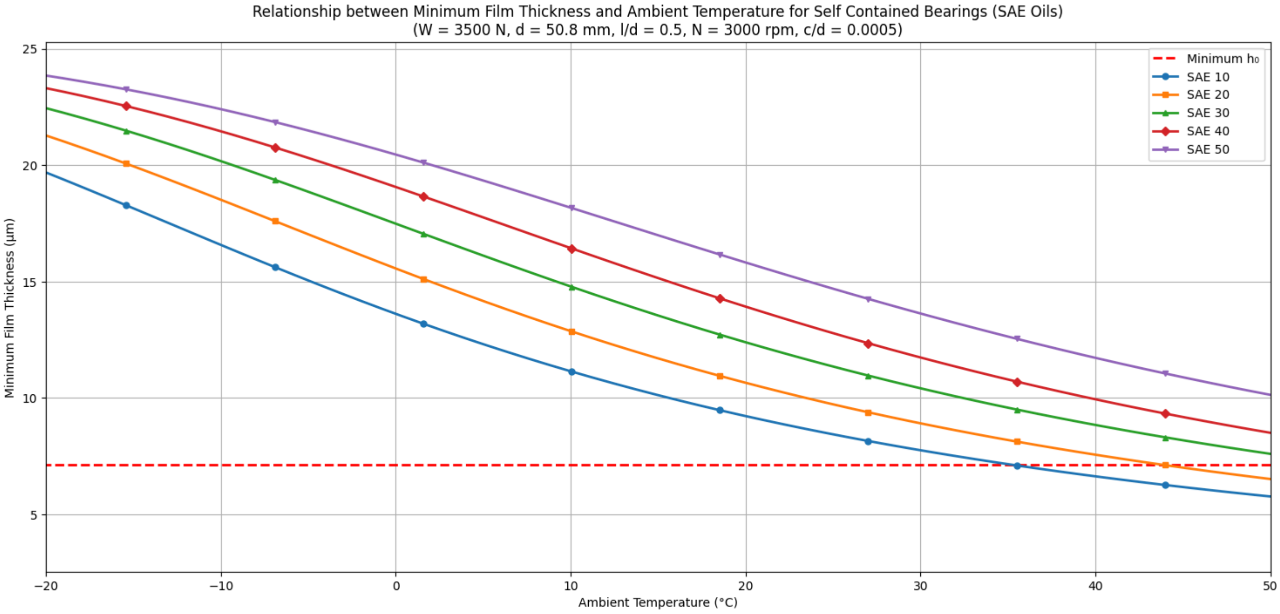


Fig 4.160:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (W = 3500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

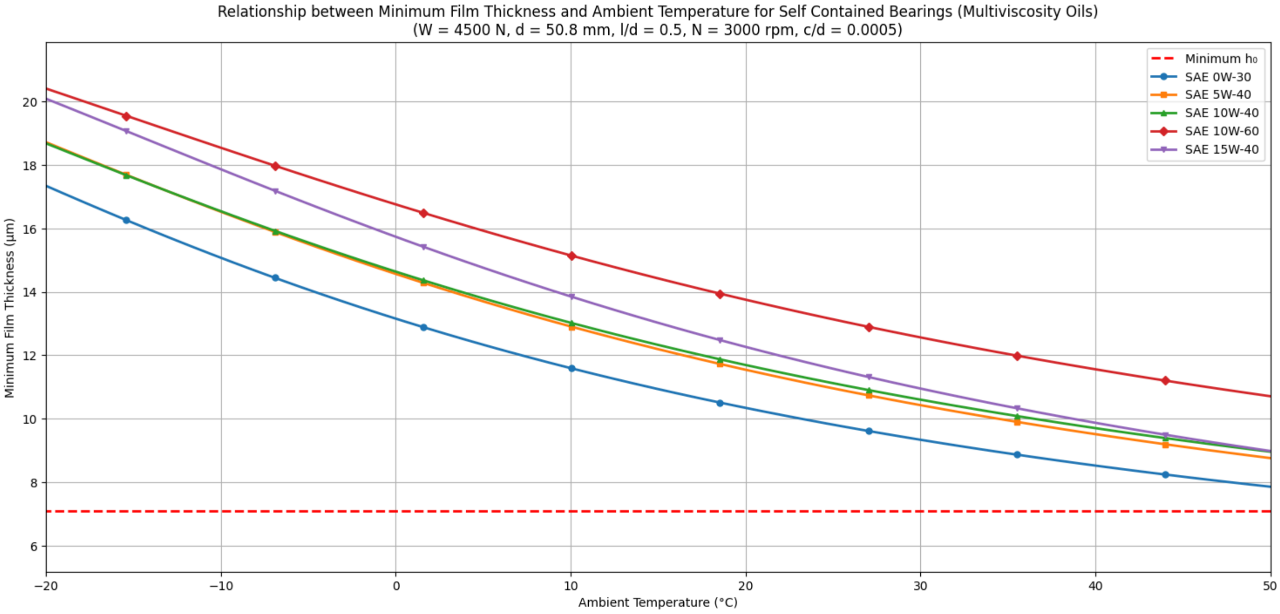


Fig 4.161:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (Multiviscosity Oils) — (W = 4500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

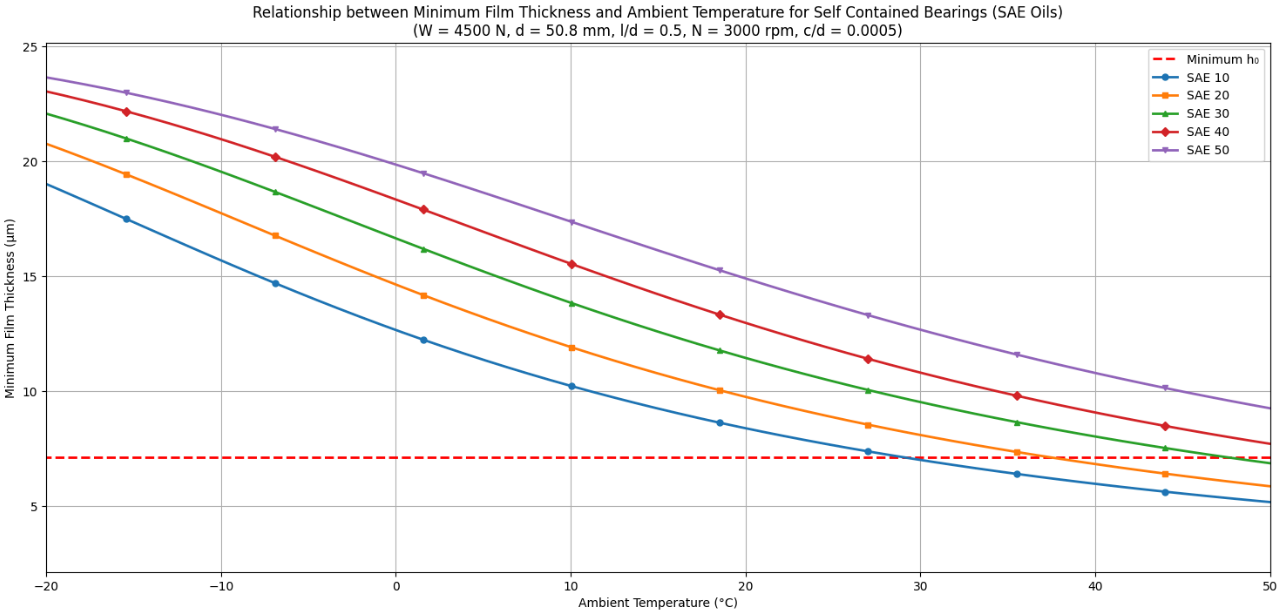


Fig 4.162:Relationship between Minimum Film Thickness and Ambient Temperature for Self Contained Bearings (SAE Oils) — (W = 4500 N, d = 50.8 mm, l/d = 0.5, N = 3000 rpm, c/d = 0.0005)

### 4.1.16 Comparison of Performance among Pressure Fed, Natural Fed and Self Contained Bearings

By observing lubricating oil's performance in pressure fed bearings (figure 4.1 to figure 4.54), natural fed bearings (figure 4.55 to figure 4.108) and self contained bearings (figure 4.109 to figure 4.162), we can say that when all other parameters are constant, lubricating oils perform worst in pressure fed bearings and best in natural fed bearings for a given ambient temperature.

# CHAPTER 5

# CONCLUSIONS

## 5.1 Conclusions

We have seen that there is a positive correlation of minimum film thickness with journal diameter, length to diameter ratio, clearance ratio and rotational speed. We have also observed that there is a negative correlation of minimum film thickness with ambient temperature and bearing load for all three studied cases. We have also observed that SAE oils perform better at low ambient temperature and multiviscosity oils perform better at high ambient temperature for all three cases. It is also evident that lubricating oils perform best for a given ambient temperature in natural fed bearing and worst in pressure fed bearings.

We can easily see from our study that the lubricating oil recommended by the machinery manufacturer may fulfill the condition of minimum allowable oil film thickness in the machinery manufacturer's condition but may fail to fulfill the same condition in the machinery user's condition due to different ambient temperatures. We do not claim that the failure to meet the criteria of minimum allowable oil film thickness would cause metal to metal contact since the manufacturers apply a factor of safety while testing for minimum oil film thickness in their conditions [48] along with conducting standardized tests. But we surely do claim that the ambient temperature being different reduces the factor of safety in the machinery user's ambient temperature. This reduction signifies the importance of testing the performance of different lubricating oils in the machinery user's ambient condition.

## 5.2 Limitations and Future Recommendations

In this study, we were only concerned with minimum film thickness as the performance parameter. In reality, many other parameters are tested by the machinery manufacturer in their ambient condition such as oil film pressure [48], friction torque and friction coefficient [49], oil supply flow and leakage flow [50] etc. Moreover, only the effect of local ambient temperature was studied. Other ambient condition parameters like humidity, ambient pressure were not considered. So there is a potential to study the mentioned parameters as well as minimum film thickness under varying ambient condition parameters mentioned before. SAE oils having higher minimum film thickness in low temperatures compared to multiviscosity oils and lower minimum film thickness in case of high temperatures is closely related to the viscosity of oils. The interdependence of viscosity and minimum film thickness is not studied here and it demands a separate study in future.

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