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Interactive Simulations of the Performance of Hydrodynamic Bearings in the 'Machine Design' Course*

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Bearings are very usual engineering components and are used in almost all types of machinery. They are essential for the correct operation of machines because they provide the separation between loaded surfaces in relative motion, through a hydrodynamic lubricant film. In this work, a pilot experience is set out regarding the teaching and learning process of hydrodynamic bearings in the 'Machine Design' course, through play with a helpful interactive system. Its main aim is to encourage students to deal with real mechanical engineering problems, and improve their ability to model, simulate and predict the influence of each parameter on the results, and as a consequence optimize the solution. The hydrodynamic equations are solved numerically by executing a C program, where input parameters are dimensionless minimum film thickness and the ratio of length to width of the bearing. The results obtained are the following dimensionless parameters: pressure and film thickness distributions, load capacity and bearing characteristic numbers. Finally, a post-processing using Matlab software allows deducing dimensional values of the cited parameters, and also determining other related results such as temperature, friction coefficient, lubricant flow and lubricant lateral losses. In addition, it provides a graphic representation of the results of interest. The system developed is organized in modules which are fully controlled using different applications in Matlab, which interact with the C program when necessary. Teaching and learning modules consist of interactive and parameterized applications, designed for non-expert Matlab users, which deal with practical situations. The modules developed have shown themselves to be useful for learning the mechanisms of hydrodynamic film generation and for analyzing the effects of operating variables on the performance of such films, like variations in load, velocity, eccentricity, design parameters, etc. In addition, they are suitable for deductive teaching and learning purposes.

Keywords: Journal bearing; hydrodynamic lubrication; Machine Design; learning module

1. Introduction

Following the subject 'Theory of Machines and Mechanisms', 'Machine Design' course is taught for students of Mechanical Engineering at Universidad Politécnica de Madrid. In this subject a systematic study for the calculation and optimal design of machine elements is performed. This includes the main components of machines: shafts, hub-shaft connections, hydrodynamic bearings, rolling bearings, joints, clutches, brakes, gears, gear-trains, belt drives, chain drives and others. The objective includes learning both the analysis of existing machines and the whole design stages of new machines.

In the last few years, the student and teacher surveys for 'Machine Design' have shown the need for more practical content and self-study material especially in the lessons of bearings since these machine elements have a complex behaviour, influenced by many parameters. Due to this, the interactive simulations shown in this paper discuss bearings analysis and design. They are intended not only to amend the needs detected but also to improve learning through real engineering cases.

Bearings are vital components for machines where movement is required between fixed and moving parts under load. To physically separate the two components, a thin film of lubricant is placed between them in order to reduce friction and wear. The relative movement of the two surfaces with a wedge-shaped convergent geometry causes pressure to develop at the lubricated contact, thereby providing the sufficient load capacity to separate both bodies. The most commonly used bearings in industry are the following:

- Journal bearings. They comprise a shaft that rotates inside a stationary cylindrical housing (bearing). Both the shaft and bearing are manufactured with a certain radial clearance c to provide room for a film of lubricant between both. Under operating conditions, when load and movement are applied to the shaft, the centre of the shaft and that of the bearing cease to coincide and exhibit a certain eccentricity, and generate a convergent-divergent channel. This type of bearing has numerous applications in machinery, as, for example, in engines, turbines

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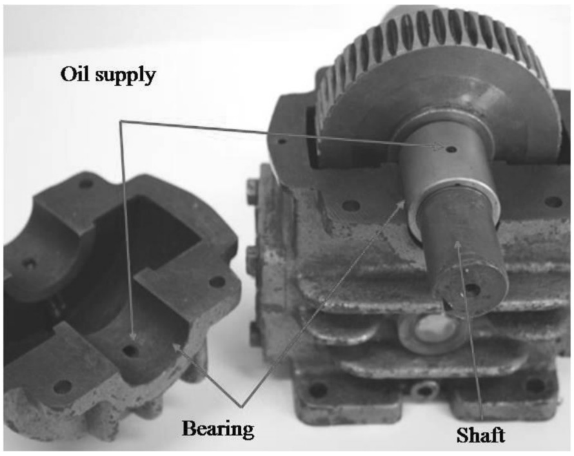


Fig. 1. Journal bearing.

and screws. Figure 1 shows a bearing in a worm gear transmission.

Sometimes the lubricant is supplied at overpressure to achieve a better flow of lubricant inside the bearing and obtain better cooling through oil flow. These devices require an external mechanism that can pump the oil under pressure.

- Pad bearings. These consist of a pad sliding over a smooth surface. Pads are employed for both tilting pad journal bearings and tilting pad thrust bearings.

Tilting pad journal bearings have the peculiarity that the bearing surface is not continuous as it is divided into a specific number of pads. Moreover, each of these pads can be oriented to better bear the load. Tilting pad journal bearings have certain technical and economic advantages over full journal bearings. They contribute stability and damping in the face of any possible perturbations, thereby enhancing the performance of the rotors. In addition, in the event of any of the pads becoming deteriorated, all that is required is to replace this component and not the complete bearing. Some of the applications using them are machine-tools and drive shafts.

Tilting pad thrust bearings allow shaft rotation under axial load. The components' own geometry creates a convergent-divergent channel. Some of their applications are telescopes, vertical lathes and cranes.

2. Modelling bearing lubrication

The Reynolds equation describes the behaviour of the pressure $p(x, y)$ of a fluid inside a wedge of variable thickness $h(x, y)$ between a minimum thickness h_1 and a maximum thickness h_2, that is influenced by the velocity field of the lubricant and by its viscosity η. Contact is produced in a two-dimensional domain Ω called a hydrodynamic wedge, which is shown in Fig. 2.

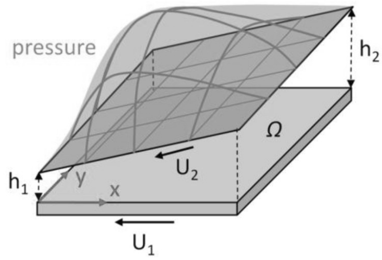


Fig. 2. Hydrodynamic wedge.

If we assume Newtonian behaviour for the lubricant, the Reynolds equation is written according to the reference [1] in the following form:

$$\frac{\partial}{\partial x}\left(h^3(x,y)\frac{\partial p(x,y)}{\partial x}\right)+\frac{\partial}{\partial y}\left(h^3(x,y)\frac{\partial p(x,y)}{\partial y}\right) - 12u_m\eta\frac{\partial h(x,y)}{\partial x}=0 \quad (1)$$

Where u_m is the mean velocity of both surfaces, U_1 and U_2. The viscosity of the lubricant depends on its temperature T, in accordance with a viscosity-temperature behaviour equation, $\eta(T)$, of the lubricant.

Once the geometry of the contact components has been defined, the Reynolds equation is solved numerically and in order to achieve short computational times, 'Full multigrid' [2] techniques are used in C language. In order to simplify solving, the

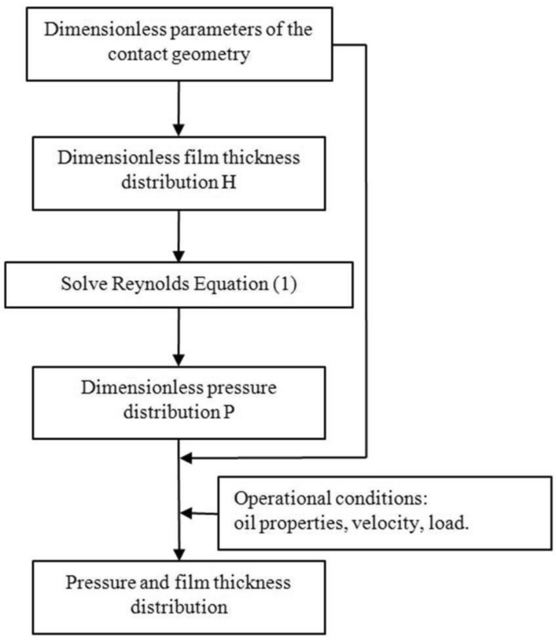


Fig. 3. Outline of the numerical solving of the Reynolds equation.

parameters (p, h, x, y, η) are non-dimensionalized as well as the Equation (1) together with its boundary conditions in the domain Ω. Thus, having found the expression for the dimensionless film thickness $H(X, Y)$, the dimensionless pressure distribution, $P(X, Y)$ can be found. Finally, in order to particularize the solution, the results obtained are dimensioned. To do this, the operating conditions of the bearing need to be known: the oil used, angular velocity and load. Each particular bearing possesses different contact geometry, and, therefore, gives different results when performing this calculation procedure, which is outlined in Fig. 3.

As a result, the velocity field of the lubricant can be found in the direction of sliding, $u(x, y, z)$, and in the direction perpendicular, $v(x, y, z)$, by using the Equations (2).

$$u(x,y,z) = \frac{1}{2\cdot\eta}\frac{\partial p}{\partial x}\left(z^2 - z\cdot h(x,y)\right) + (U_2 - U_1)\frac{z}{h(x,y)} + U_1$$

$$v(x,y,z) = \frac{1}{2\cdot\eta}\frac{\partial p}{\partial y}\left(z^2 - z\cdot h(x,y)\right) \quad (2)$$

Where z is the height of each point, which varies between zero and the film thickness h. By taking the velocity field and the viscosity η, the shear stress $\tau(x, y, z)$ in the lubricant can be calculated, which for Newtonian behaviour has the Expression (3).

$$\tau(x,y,z) = \eta\left(\frac{\partial u(x,y,z)}{\partial z}\right) \quad (3)$$

By integrating the shear stress in the contact area the value of the friction force F_r is found, which divided by the load W borne by the bearing, allows calculating the friction coefficient μ on each of the contact surfaces ($z = 0$ and $z = h$).

$$\mu = \frac{F_r}{W} = \frac{\iint \tau(x,y,z)_{\substack{z=0(\text{surface1})\\ z=h(\text{surface2})}} dx\cdot dy}{W} \quad (4)$$

When the bearing has to support heavy loads and high sliding velocities, the temperature of the lubricant can rise considerably with respect to ambient temperature, as a result of the energy dissipation, due to viscous effects. To take account of this thermal effect, the rise in lubricant temperature is calculated. This calculation can be done by using an analytical method [3], [4] or a thermal resistance model [5]. The analytical method is based on a global thermal equilibrium of the bearing. It is supposed that all the heat generated Q_g is evacuated by the oil that leaks through the lateral loss flow, q_S, which is calculated starting from the velocity field. The bearing's mean operating temperature T is considered equal to the temperature of the flow that is leaking laterally. This hypothesis represents a conservative version from a design point of view because, in actual fact, the temperature will be lower due to the effect of the fresh lubricant supply. For example, Equation (5) shows the thermal equation for a full journal bearing.

$$Q_g = \mu W N \pi D = \rho C_p q_S (T - T_0) \quad (5)$$

Where D is the diameter, N is the angular velocity (in *rev/s*) of the shaft; ρ is the density and C_p the specific heat of the lubricant. By working out the increase in temperature ΔT, in respect of the input temperature T_0, we obtain the Expression (6) for a full journal bearing.

$$\Delta T = \frac{\mu W N \pi D}{\rho C_p q_s} \quad (6)$$

To reach a more accurate solution, the thermal effects can be calculated numerically [6], by means of a thermal resistance model, in which each of these thermal resistances sets the ratio of a transmission potential (difference in temperature) to heat transfer (calorific flow) between two points. It can be of two types: conduction or convection.

A computer program developed creates a contact domain mesh and sets the thermal resistances so that a thermal circuit equivalent to an electrical one is generated. As Fig. 4 shows, the system is taken as dissipating the heat generated by the friction Q_g in each component volume to three areas: to the lubricant in motion Q_{oil}, to the shaft Q_1 and to the bearing Q_2, firstly reaching their surface and then becoming dispersed towards the inside. By solving this system using direct methods, the temperature profiles of the oil T_{oil} and the surfaces T_{s1}, T_{s2} are found.

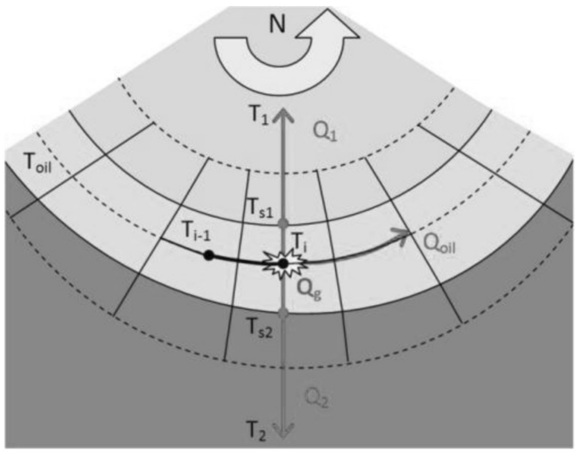


Fig. 4. Heat generation in a hydrodynamic bearing.

Once convergence had been achieved and all the equations involved had been solved, a program was developed to provide the basic results, in both graphic and data table form. Some of the main parameters of interest are:

- Pressure distribution in the bearing. Depending on the pressures reached, different possible materials should be used for the bearing.
- The thickness of the film separating the two surfaces. The minimum lubricant film thickness value must be three times higher than the mean surface roughness in order to minimize the contacts between the surfaces [7], and consequently the friction coefficient and the probability of wear.
- The friction coefficient. This value is directly related to the energy efficiency of the hydrodynamic bearing.
- The operating temperature reached, which is related to the lubricant's life. The lubricant will have a maximum allowable value for temperature above which it is not advisable to work, due to the fact that phenomena may arise that quickly degrade the lubricant.

As a result of this, the simulation tool developed can be used for numerous applications, such as choosing lubricants, fine-tuning optimum operating conditions and discovering the advantages and disadvantages of any possible alternatives.

3. Learning modules developed

Using C programming, a set of Matlab modules were developed to solve the different 'standard' cases in the context of teaching, but which would also be similar to the actual cases arising in an industrial context. Each case consisted of a specific problem with different input and output parameters. On occasions, finding the variables involved required performing iterative processes, which meant performing loops until convergence of the method was achieved.

A help module was available (Fig. 5) containing the basic information on the theoretical bases and the functioning of each of the modules. Block diagrams were prepared to represent the stages followed by the program in each case, in order to lead to an understanding of the initial hypotheses, the calculations to be done, the input variables to the program and the output results. The results to be presented could be chosen (pressure distribution, film thickness, temperature profile, etc) and the display format (data tables, graphs or report). Described below are the different teaching and learning modules developed.

3.1 Module 1: Characteristic behaviour curve of the lubricant

This is a very simple example, but useful for many of the modules that will be described further on. The starting point was a law on oil viscosity behaviour with temperature, $\eta(T)$, dependent on two parameters, a and b, shown in the Equation (7), where T_a is a reference temperature.

$$\eta(T) = ae^{b(T-T_a)} \tag{7}$$

With two viscosities at two different temperatures being known, the two parameters mentioned defining the characteristic behaviour of the oil according to temperature, were defined. The Graphic User Interface (G.U.I.) shown in Fig. 6a was used for data input. When the oil has been characterized, the viscosity-temperature curve can be found and used to obtain different viscosity-temperature value pairs, as depicted in Fig. 6b. Similarly, the viscosity-temperature curves of the SAE/ISO lubricants of different viscosity indexes can be drawn, as can be seen in Fig. 6a.

3.2 Module 2: Choosing the lubricant for a journal bearing of known geometry and for specific operating conditions

We have a journal bearing as shown in Fig. 7, whose geometry is defined by the length L, the diameter D and the radial clearance c. It is required to work under some specific operating conditions: angular velocity N, load W, lubricant supply temperature T_0 and eccentricity e. The eccentricity e is directly related to lubricant film thickness: $h_0 = c–e$. Often, instead of working with dimensional eccentricity e, an eccentricity factor of $\epsilon = e/c$ is worked with that is a result of dividing the eccentricity by the clearance.

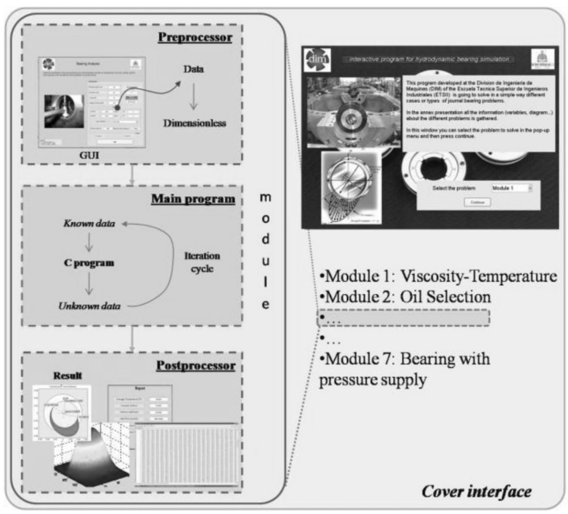
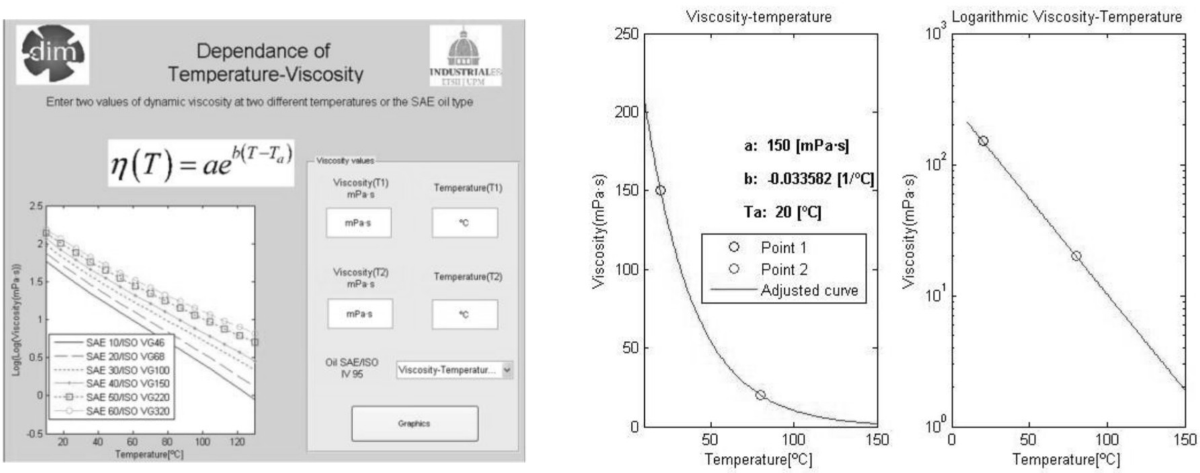


Fig. 5. Help module structure.



(a) (b)

Fig. 6. (a) Graphic user interface (G.U.I.); (b) Results of the viscosity-temperature curve, with linear and logarithmic scale.

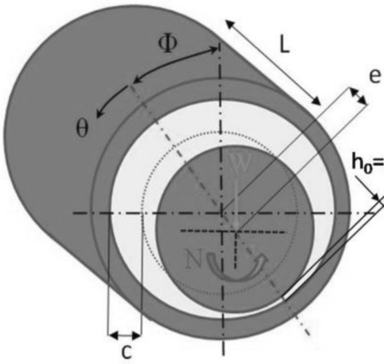


Fig. 7. Schematic diagram of a journal bearing.

It was wished to choose the most appropriate lubricant for the given conditions. In addition, the SAE/ISO graphs were included, so as to be able to choose a lubricant from within the range.

We were dealing with a direct application that did not require an iterative process as all the parameters were known, which were entered directly into the calculation program using the graphic interface in Fig. 8a. When the program is up and running a window appears to summarize all the results of the bearing (Fig. 8b), as well as the pressure distribution graphs (Fig. 8c) and the film thickness (Fig. 8d). Figure 8e represents the work point in a viscosity-temperature graph, which allows choosing the most suitable lubricant for the case study. Figure 8f shows a schematic diagram of the operating position of the bearing under the operating conditions studied.

3.3 Module 3: Analysis of the main operating parameters for a specific geometry of a journal bearing, a given lubricant and known load and velocity conditions

We took a journal bearing defined by its geometric parameters (*L, D, c*) and of usage (*W, N*) and the lubricant to be used (η, ρ, C_p), and calculated the operating parameters under which it was working: the distributions of temperature in the lubricant and the contact pressure (Fig. 9a), the operating position (Fig. 9b), the friction coefficient, the eccentricity factor, the lubricant flow circulating in the bearing and the lateral loss flow.

Figure 10 shows some additional details of the results, such as the variation in pressure in the central section of the bearing throughout the contact (Fig. 10a), the temperature in the lubricant and on the surfaces (Fig. 10b), as well as the calculation error and the dimensionless eccentricity compared to the number of iterations (Figs. 10c and 10d).

Taking this information, we can analyse whether the pressure borne by the bearing is allowable, analyse the likelihood of any metal contact between the shaft and the bearing, and determine the oil's maximum operating temperature. Aspects such as the bearing's energy efficiency can be studied and the effect to be had by using another lubricant.

This module is useful for understanding the mechanics of Hydrodynamic film generation and for analysing the effects of the operating variables on the behaviour of this film, such as the variations in load, velocity, eccentricity, design parameters, etc. Figure 11 shows an example of the analysis of the influence of eccentricity on pressure distribution and the corresponding load capacity.

3.4 Module 4: Study of the operating limits of a journal bearing as a function of design

This study was conducted to find the most suitable fit according to the system of tolerances ISO in the shaft-hole pair during the design of a bearing that was to work at a known velocity *N* and under a known load *W*. To this end, we studied the influence of the parameters that define the contact, on varying

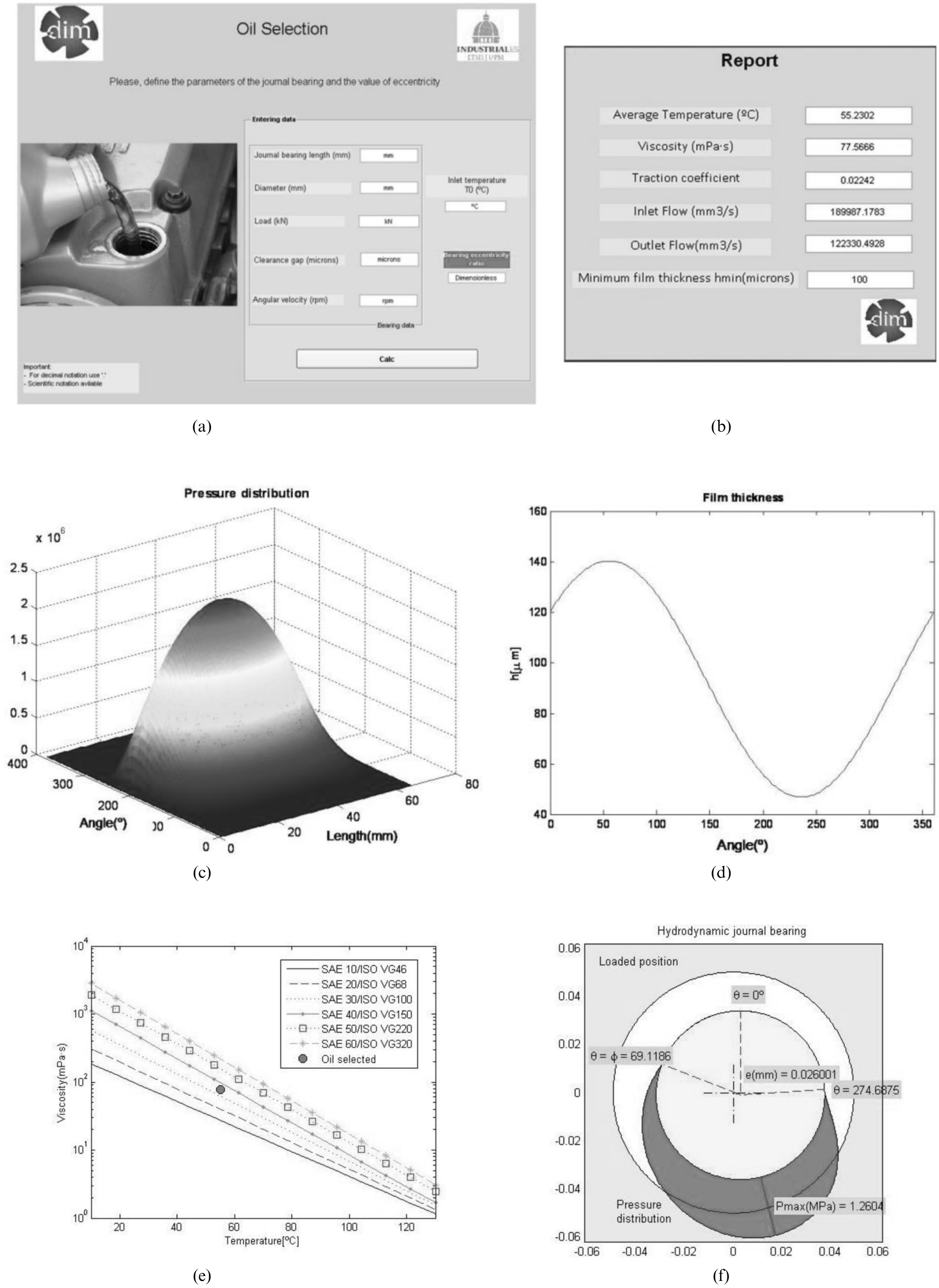


Fig. 8. (a) G.U.I. of the module; (b) Results summary window; (c) Pressure distribution in 3D; (d) Film thickness; (e) Graph for choosing a SAE/ISO oil; (f) Schematic representation of the operating position.

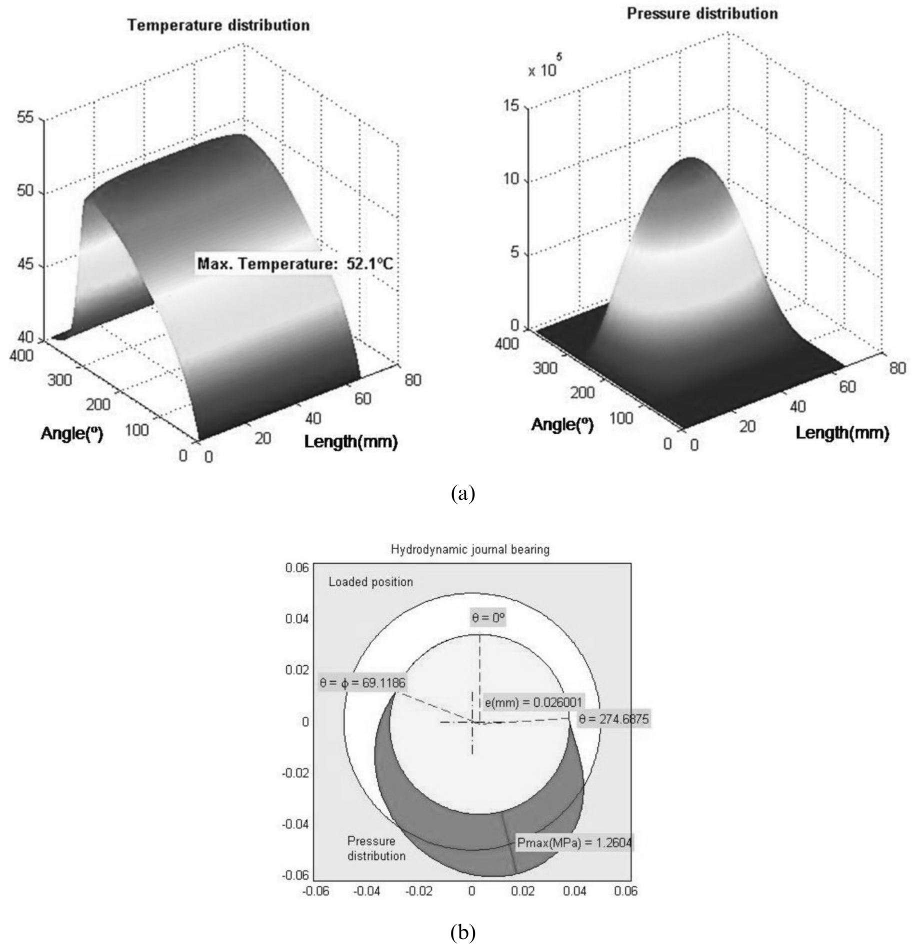


Fig. 9. (a) Results obtained for the distribution of pressure and temperature in 3D; (b) Schematic diagram of the operating position of the bearing.

the clearance *c* between the shaft and the hole, both with the same nominal diameter *D*.

Figure 12 shows a comparison of two fits: *H7-d9* and *E8-h6*, showing the influence exerted by the fit in some decisive operating parameters, such as lubricant minimum film thickness at the contact, the operating temperature, and the friction or traction coefficient.

In this way, the operating zone can be optimized, ensuring the film thickness is sufficient to prevent any surface roughness from coming into contact with the ensuing wear, while at the same time allowing the minimum possible friction coefficient, in order to enhance energy efficiency. The lower limit of the clearance is also conditioned by the operating temperature so that it will not be greater than the maximum admitted by the lubricant.

3.5 Module 5: Raimondi-Boyd graphs

By using the numerical program, the classic Raimondi-Boyd graphs can be obtained [8]. These are habitually used for manually calculating journal bearings from the Reynolds Equation (1). The Raimondi-Boyd graphs, like that depicted in Fig. 13a, show the dimensionless parameters resulting from integrating the Equation (1) as a function of the dimensionless length (*L* compared to the diameter *D*) and the number of Sommerfeld *S* defined in the Equation (8), according to the reference [8].

$$S = \left(\frac{D}{2c}\right)^2 \frac{\eta N}{p_m}, \text{ with : } p_m = \frac{W}{DL} \qquad (8)$$

Where p_m is the mean pressure and *W* is the load. The module prepared to obtain the Raimondi-Boyd graphs (Fig. 13b) lets the results obtained be compared by using the developed numerical model and the classic method. The discrepancies between both methods can be analysed and the hypotheses used contrasted for each of the cases.

3.6 Module 6: Analysis of tilting pad journal bearings

We take the length *L* and diameter data *D* of the bearing, the number of pads and their position,

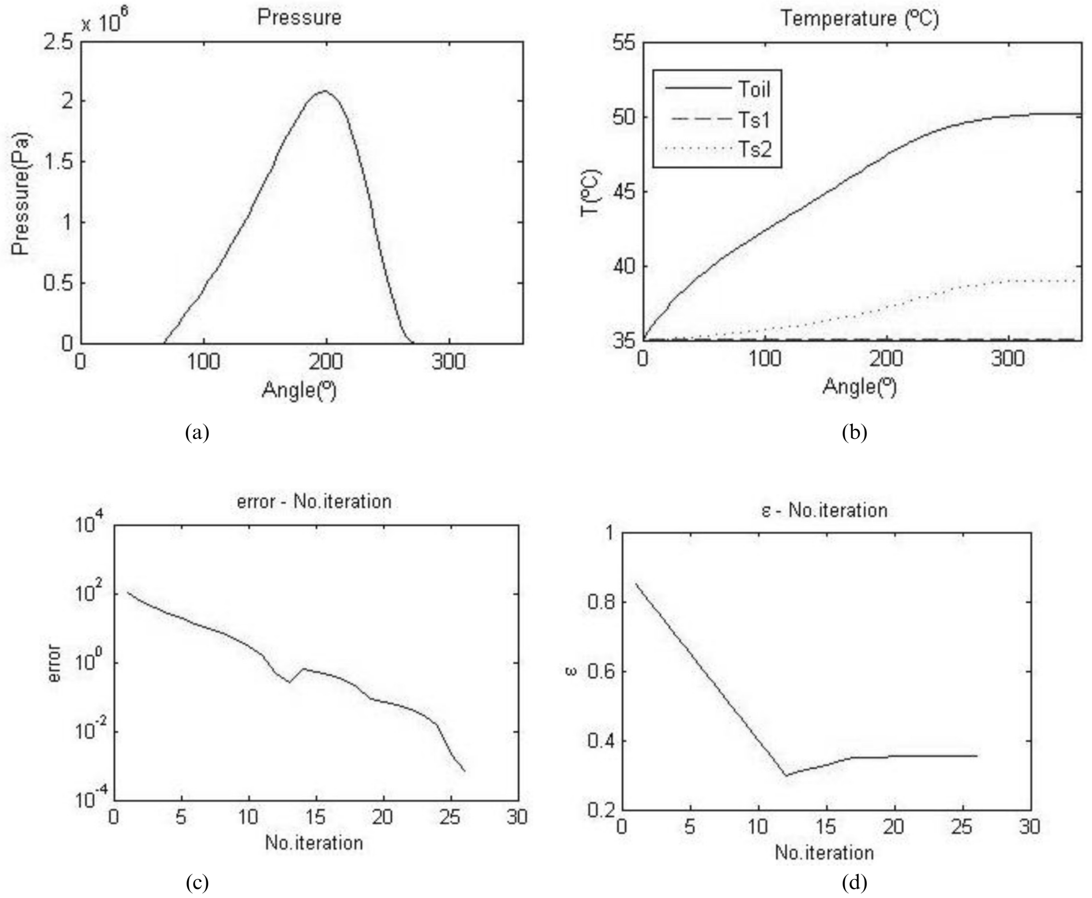


Fig. 10. (a) Central section of the pressure at each point; (b) Oil and surface temperatures; (c) Evolution of error with the number of iterations; (d) Evolution of eccentricity factor with the number of iterations.

shaft angular velocity, the lubricant used and the input temperature. Following a very similar method to that used for module 3, an analysis is made of the operating conditions of this type of bearing in each of the pads of which it is comprised according to

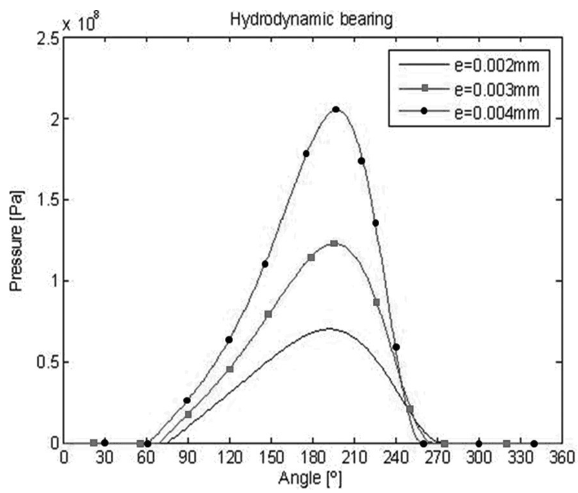


Fig. 11. Example of analysis of the influence of eccentricity on pressure distribution and the corresponding load capacity.

design parameters, like the radial clearance c and other operating parameters like the film thickness h or the eccentricity factor ϵ, calculating the load capacity in each case (Fig. 14a). Figure 14b shows the operating conditions of the film for different clearances, while Fig. 14c reflects pressure distribution in the central section in a pad bearing of known clearance and eccentricity, and under a given load.

3.7 Module 7: Analysing bearings with overpressure

These bearings have the peculiarity of the oil being introduced with a pressure above atmospheric pressure, mainly to facilitate the circulation of the lubricant and reduce the oil's operating temperature. Now knowing the bearing data, the lubricant, the oil's operating conditions and its temperature limits, the supply pressure required is calculated together with the remaining bearing parameters, as for modules 3 and 6, already presented, but bearing in mind the lateral losses of lubricant due to its forced circulation. To the results obtained in this module we add the overpressure calculation

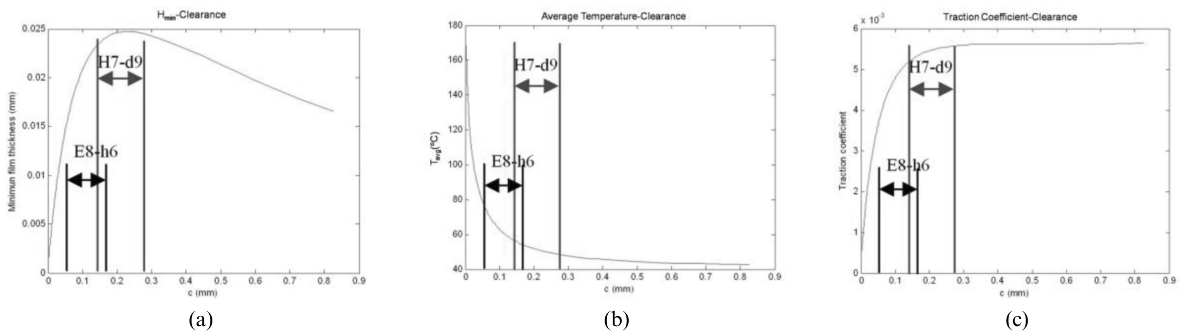


Fig. 12. Variation in different parameters with the radial clearance; (a) Minimum film thickness; (b) Lubricant temperature; (c) Friction coefficient.

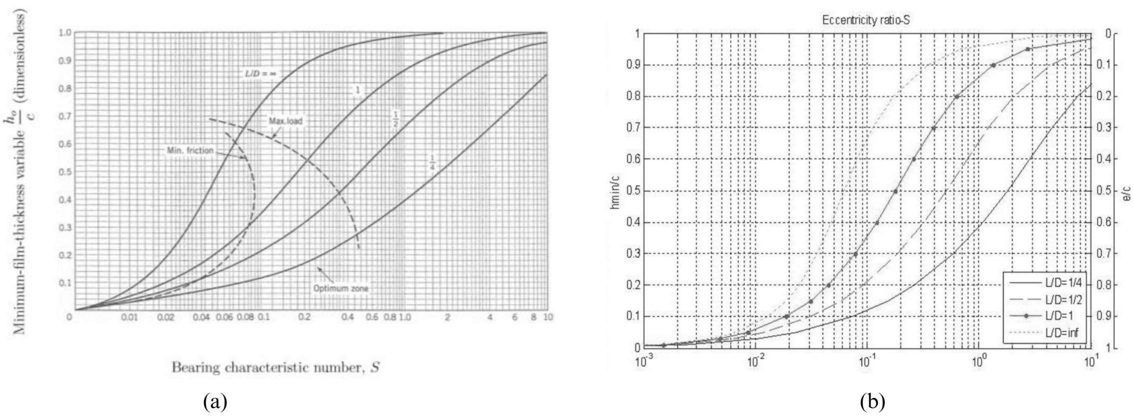


Fig. 13. (a) Results obtained by Raimondi and Boyd for the dimensionless film thickness h_0/c and eccentricity factor ϵ = e/c; (b) Results obtained with the numerical model.

needed to work with the lubricant at the required operating temperature.

4. Implementing the modules

The programs developed were placed at students' disposal as well as other simulation tools developed by the same Educational Innovation Group GIE-DIM [9]. The Aulaweb computer platform [10] was used (http://aulaweb.etsii.upm.es), because both students and teachers are very familiar with it.

Regarding the implementation of the programs presented, there are two different stages:

- In the first stage, the theoretical content is explained and some practical cases are solved in the classroom. Exercises are also set for the students to solve by approximate methods using classic Raimondi-Boyd graphs. But these activities have their limitations due to their requiring a lot of manual calculation that is very tedious and of little added value for the subject.
- In the second stage, the simulators developed for students' use are activated. These enable information concerning any problems that arise to be exchanged with the teacher in real time. The teacher can monitor student learning in Aulaweb while the students are working on the subject.

The main aim is to improve understanding of the behaviour of mechanical contacts with hydrodynamic lubrication, while helping to develop the transversal competencies needed by an engineering student, such as the ability to design and improve machines by using analytical and computational techniques, the application of scientific knowledge to engineering practice and the incorporation of new tools to professional practice.

At the same time it boosts problem-solving skills that integrate multidisciplinary knowledge embracing fluid mechanics, heat transmission and mechanical engineering.

On the other hand, it is important to point out that any design and analysis process in engineering [11] begins with approximate preliminary calculations and ends with more complex and exact calculations. In the same way, the teaching/learning process set out has an initial manual calculation stage so that students will first get to know every problem in depth. They then go on to use the interactive system, developed along the lines already known by the students, so that they will reach a

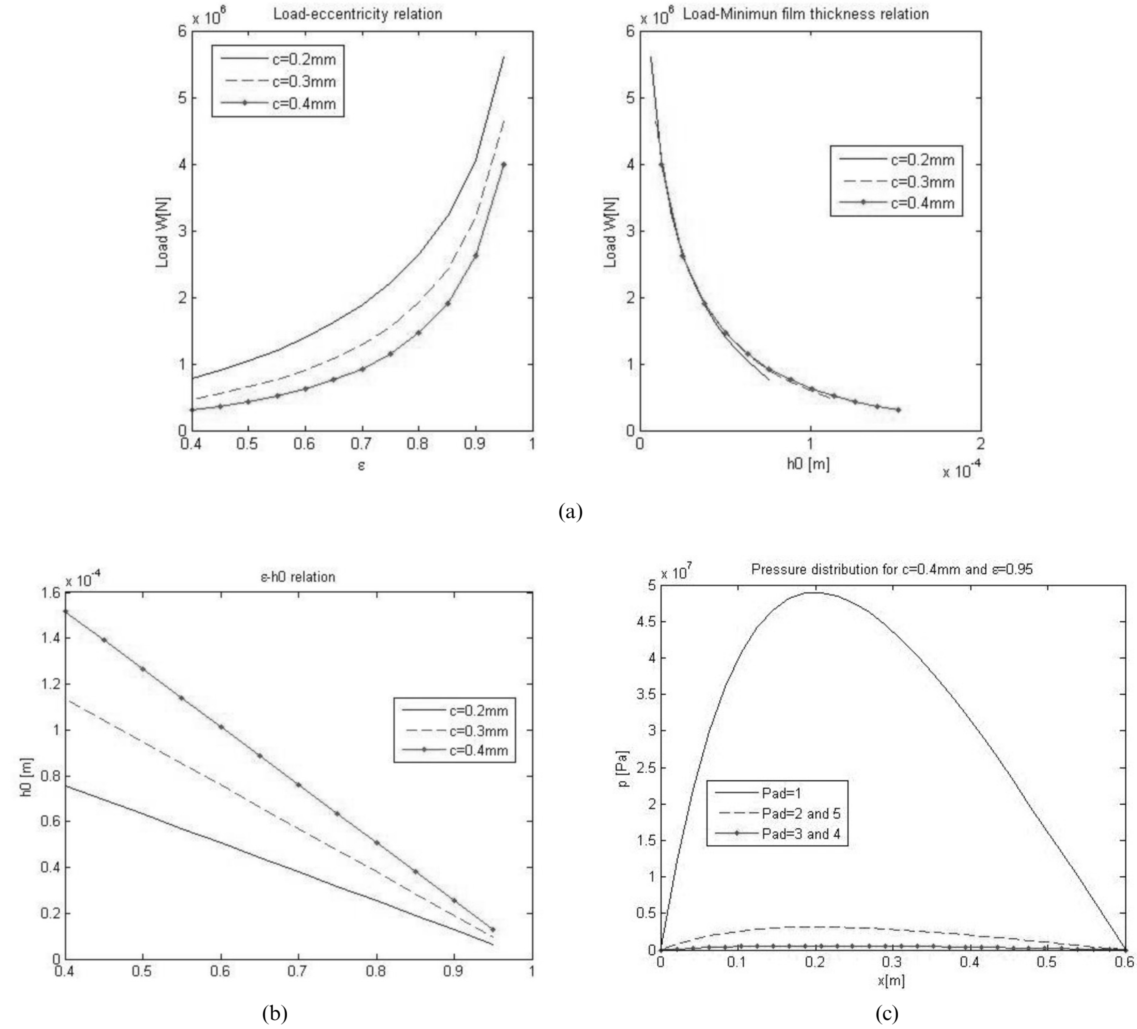


Fig. 14. (a) Load evolution with the eccentricity factor ϵ and with the minimum film thickness h_0, for different clearances; (b) Lubricant minimum film thickness for different clearances; (c) Pressure distribution for a five-pad bearing.

better understanding of how it can be used and its full potential for different situations. At this point, the simulator is useful to analyse the effect of each parameter involved on the results obtained, providing an easier interpretation of the results.

Each module includes a case study presentation showing the initial data and the results needed to solve the problem. The corresponding block diagram is included, showing the calculation procedure to be followed in each module until the solution sought is found. Students then pass on to the screen of each program where the case in question is solved and the selected results are presented.

Thus, the modules presented are used to supplement the theory and problem classes in the subject, with the purpose of expanding the knowledge acquired and for quickly checking the results that have come out of the individual problems set for each student in the subject.

As for the programs, we aim for them to be interactive and easy to use and that they should also enable the time devoted to intermediate calculations to be reduced, particularly when a design problem is set that requires evaluating the effect caused by each decision made. The aim is to show bearing behaviour graphically, rapidly and intuitively as well as its sensitivity to any changes to the input parameters, so that the program will become a powerful tool for learning through play.

5. Evaluating the activity

The learning models developed were widely accepted by the students in the subject, with 88% of students participating in the different learning modules developed.

A questionnaire was sent to students who had interacted with the programs designed, in order to

Table 1. Students' personal scores

Questions	Score (1–5)	Standard deviation
On the Importance of Simulation Tools in general		
For understanding the theoretical problems of engineering	4.5	0.55
For solving real technical problems in engineering	4.5	0.55
As an active learning tool	4.3	0.63
On the Environment used		
Ease of use	4.5	0.84
Versatility for studying different physical phenomena	3.8	0.41
Rating of Matlab compared to other simulation tools	4.2	1.17
On the programs developed for hydrodynamic bearings		
They are related to the theoretical content set in class	4.7	0.52
They improve learning as a supplement to theory and learning classes	4.7	0.52
Their framework is appropriate	4.5	0.84
They are complete (they enable every possible situation to be studied)	4.3	1.03
They help the understanding of theoretical problems in the subject	4.5	0.55
Applicability to solving real engineering problems	4.2	0.75
They help to understand the influence of each parameter involved	4.3	0.82
They allow optimizing problem-solving	4.2	0.75
They have a degree of interactivity that leads to 'learning through play'	4.5	0.84
Students are better motivated	4.2	0.98
They can be handled without being an expert Matlab user	4.7	0.52
General score for the programs available	4.7	0.52

know their opinions about the relevant aspects concerning the use of simulation tools in engineering, the environment used and the individual programs prepared. Table 1 summarizes the responses received on a scale of 1 (minimum) to 5 (maximum).

As can be seen from Table 1, in general, students consider the simulation tools to be highly useful, not only for active learning but also, and mainly, for their future work as engineers. The environment used is considered to be appropriate, mainly for its ease of use, which, combined with the interactive nature of the programs developed, has led to students showing a great interest in them. The surveys show that student motivation increases considerably if they are able to learn by playing.

An analysis of opinions also reveals that the simulators have helped students to understand basic concepts more easily, while letting the usefulness of what was learned be demonstrated in a more practical manner. They have improved their understanding of the importance of each parameter involved and the connection between these parameters, in order to determine if the contact under hydrodynamic lubrication performs properly and under the best conditions.

Thus, we are moving forward towards optimizing the operating conditions in lubricated bearings, which is related to numerous current lines of research. The energy efficiency of the lubricated contacts can be improved as well as the likelihood of the appearance of failures due to wear [12]. Moreover, the effect of any operative malfunction can be analysed from the results obtained, enabling the safety of the design to be known under unfavourable operating conditions that may lead to in-service failures.

The following are some of the comments made by students: 'Getting results by using the program is much more flexible and dynamic', 'It leads to an understanding of the different real problems from a global perspective'.

Interest in the learning modules and their wide acceptance among students has had a considerable impact on improving student's marks in hydrodynamic bearings compared to other parts of the subject and compared to the results of previous courses. Figure 15 shows the average marks (on a scale of 0–10), obtained by students in the two courses prior to using the tool (2007–2008 and 2008–2009) and the course on which the simulators were used (2009–2010). It is demonstrated that the marks achieved in the subject increased in the last course, particularly in the part on bearings. Using the developed modules over the coming years will enable these preliminary results to be compared.

The analyses presented in this work are in line with the results obtained with the use of simulators in mechanical engineering [13–14]. This demonstrates the benefits of these tools for optimizing teaching-learning strategies within the framework of the European Space for Higher Education. The link between several real applications and theory show the developed modules to be useful for deductive learning applications [15] planned to be developed as a future activity.

Moreover, it must be emphasized the tool itself is very flexible and can be used to model and solve other problems of mechanical contacts under hy-

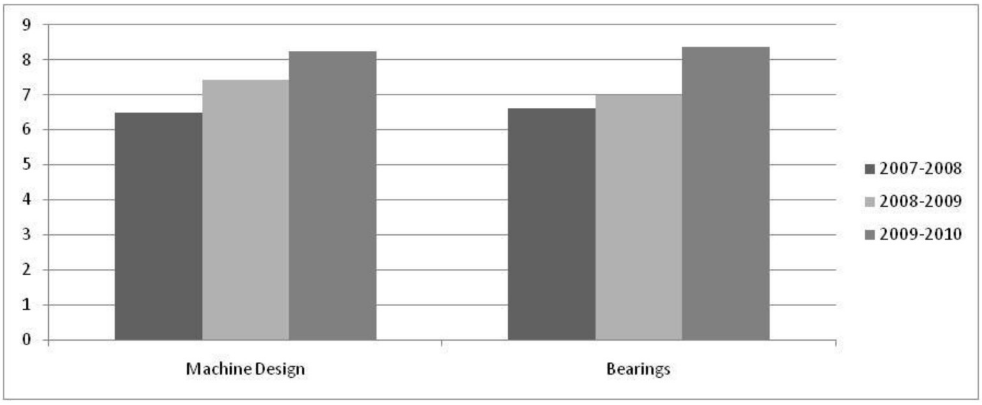


Fig. 15. Evolution of students' marks in 'Machine Design' and in the Bearings part of this subject.

drodynamic lubrication. For it to be applied to the development of new modules it is not necessary to understand C code. All that is required is to be able to handle a simpler, more widely-used program in mechanical engineering, as is Matlab.

6. Conclusions

Several lubricated bearing simulators were developed to be representative of real engineering problems, such as choosing the best lubricant for an application, the problems of analysing the working conditions and the design of bearings in order to be able to optimize their operating conditions, from an energy efficiency point of view, to prevent any possible wear or obtain an appropriate operating temperature for the lubricant.

A pilot scheme was designed to implement the self-learning modules in a 'Machine Design' course. The system was widely-accepted and well-evaluated by students who had an excellent opinion of its simplicity of use and its applicability for gaining a better understanding of the theoretical aspects governing the real engineering problems of bearing lubrication. As a result, the system has proved itself to be feasible for active learning, especially for learning through play.

In addition, its use increased the motivation of students who were able to obtain good results without having to devote time to tedious manual calculations, enabling them to improve their ability to model, simulate and predict the influence of each parameter on the results, and as a consequence, learn to optimize the solution to a mechanical engineering problem, which is directly related to the aims of the subject 'Machine Design'.

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