Technical Development Path For Gas Foil Bearings*

by

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Seoul, Korea

*(Based upon STLE presentation (2008))

Background

NASA's missions to revolutionize aviation and explore space require the development of revolutionary, long-life, high performance, high speed rotating machinery systems.

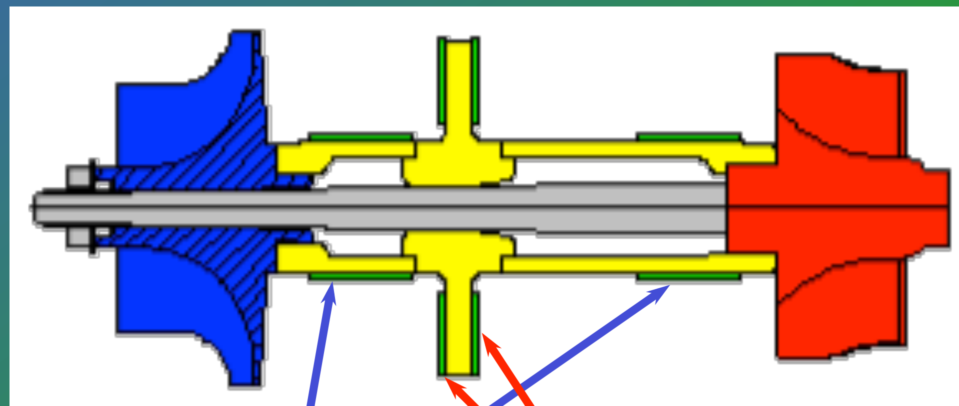
Challenge

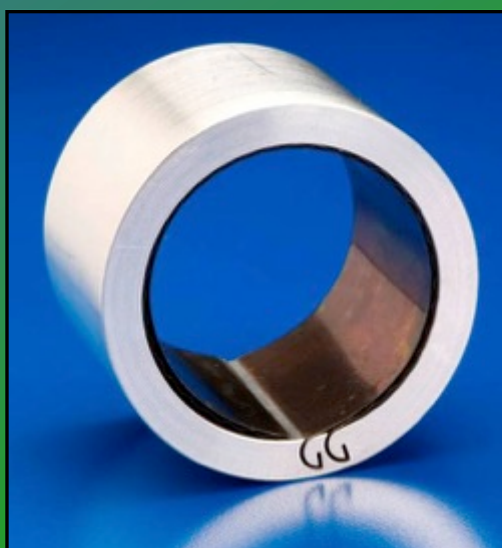
A revolution in rotating machinery performance can only be achieved through radical changes in the foundational rotor support technologies.

Approach

Combine emerging Oil-Free technologies (Foil Air and Smart Hybrid Bearings, Tribological Coatings and Analytical Modeling) to enable revolutionary Oil-Free rotating machinery systems.

Journal & Thrust Foil Bearings



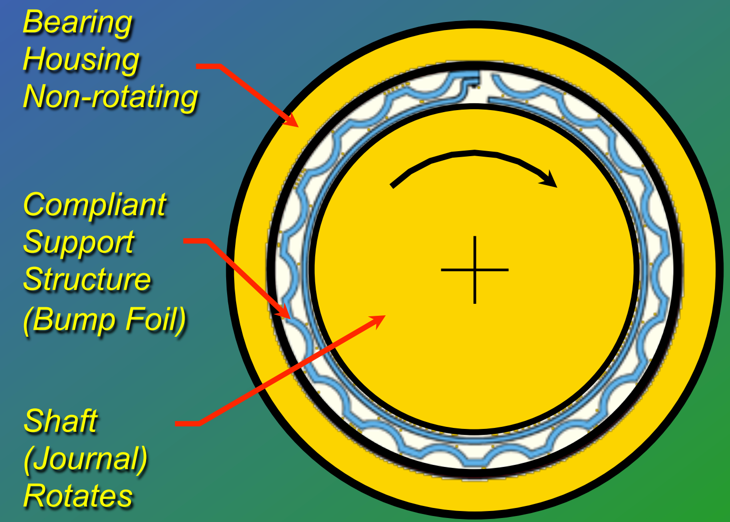


Journal Foil Bearing

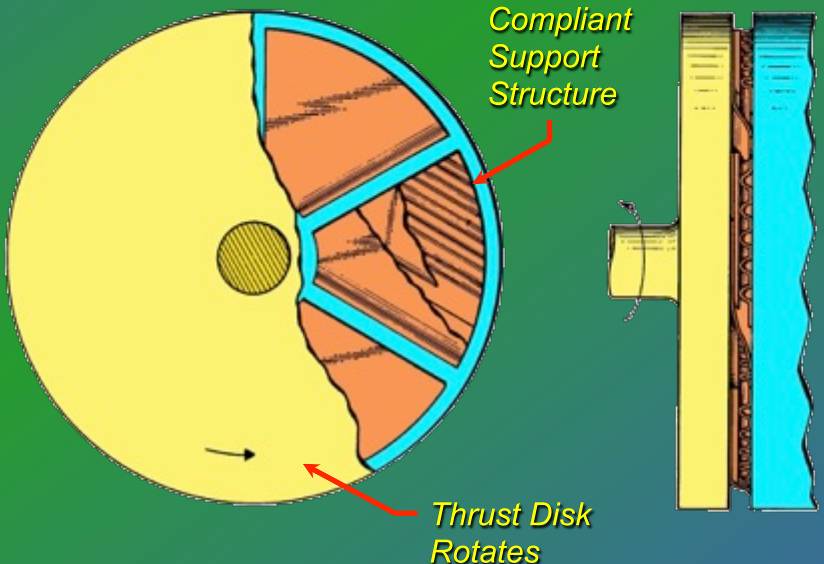
Thrust Foil Bearing

Enabling Technology: Advanced Foil Bearings

Foil Journal Bearing



Foil Thrust Bearing



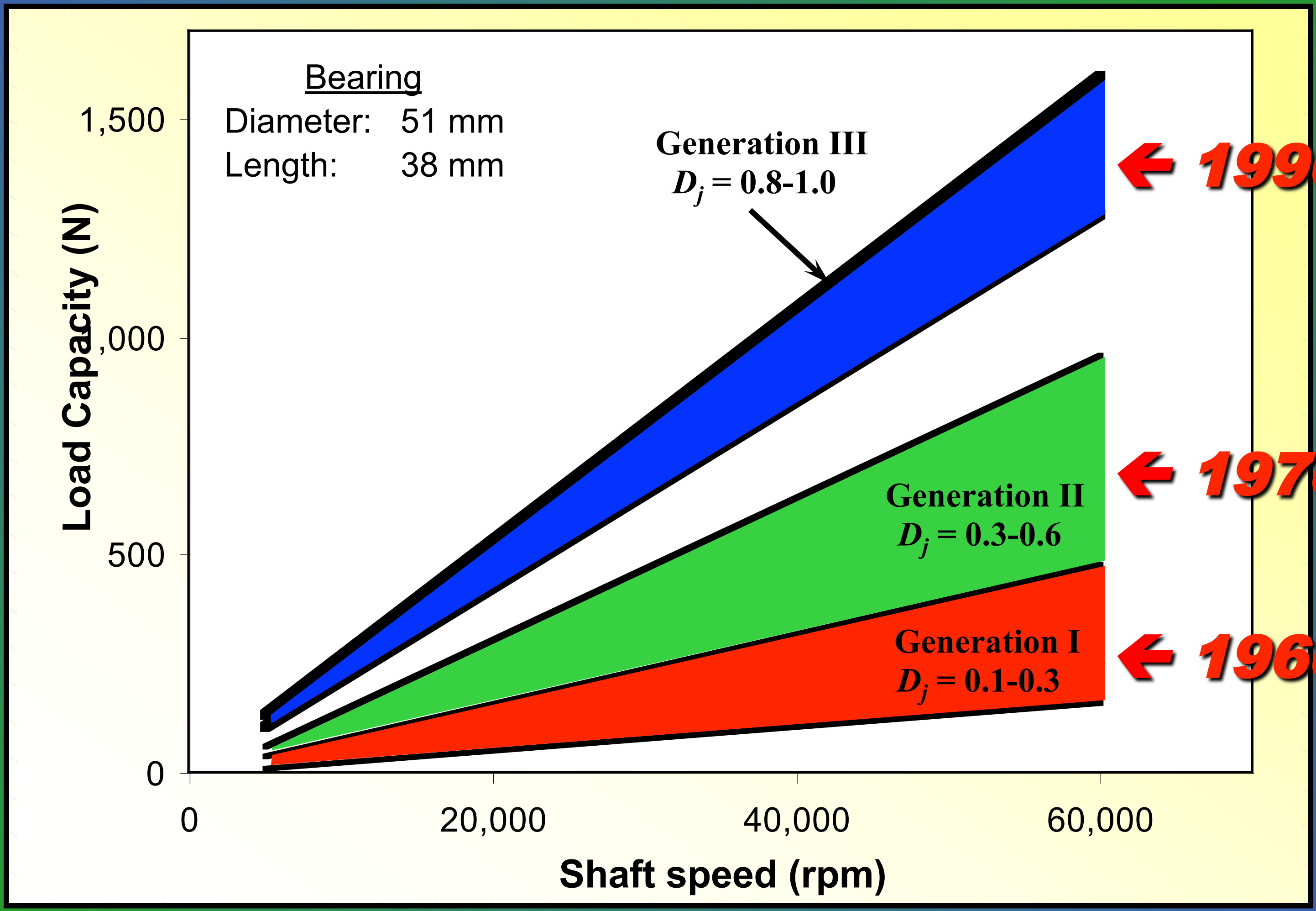
Foil Bearing Benefits:

- ✓ Self-acting hydrodynamic "float on air" ➔ No external pressurization
- ✓ No DN speed limit ➔ Higher power density
- ✓ No lube/tanks/coolers/plumbing/filters ➔ Lower weight
- ✓ Operate to 650 °C ➔ Higher efficiency
- ✓ Compliant "spring" foil support ➔ Accommodate misalignment & distortion
- ✓ No maintenance ➔ Reduce operating costs

Source, Thrust Bearing Sketch: Fortmann, US Patent #4,082,375, Apr. 4, 1978


NASA
Assembly

Foil Bearing Load Capacity – Generation I, II, & III

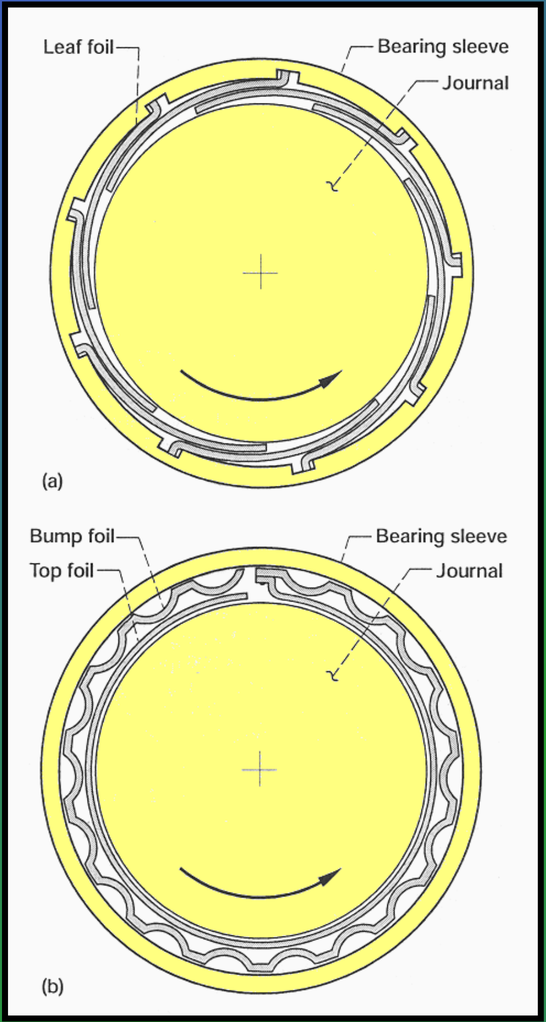


← 1990's

← 1970-80's

← 1960-70's

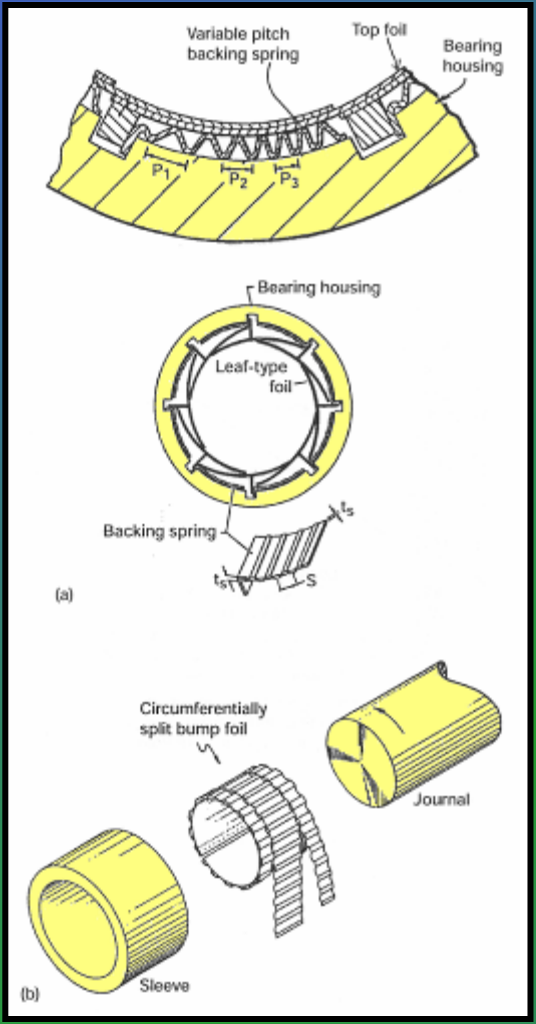
Generation I Foil Bearings *(1960's – 1970's)*



- **Load capacity coefficient, D_j's, 0.1 - 0.3**
- **Foil geometry essentially uniform in both the axial and circumferential directions** *(including uniformly periodic circumferential geometry)*
- **Stiffness characteristics of the foil structure are more or less uniform**
- **Foil surface deforms due to the fluid film pressure without support structure specifically accounting for localized effects such as edge leakage, thermal gradients, heat generation and other hydrodynamic phenomena**

Source: DellaCorte & Valco, "Load Capacity Estimation of Foil Air Journal Bearings for Oil-Free Turbomachinery Applications, NASA/TM-2000-209782, ARL-TR-2334, Oct 2000

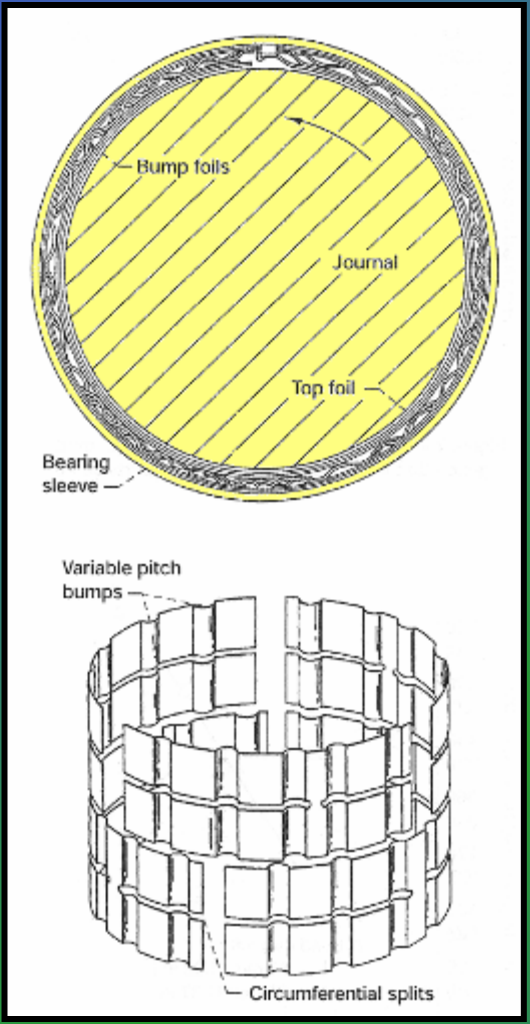
Generation II Foil Bearings *(1970's – 1980's)*



- Load capacity coefficient, D_j's, 0.3 – 0.6
- Stiffness of the foil support structure varies axially along the bearing length or in the circumferential direction, but not both
- By controlling stiffness in one dimension *(axial or circumferential)* the bearing better accommodates phenomena like edge leakage and, hence, yields improved performance
- In leaf foil bearings, use of a "stepped" backing spring
- In bump type foil bearings, bump layers are split circumferentially for axial compliance control or the bump pitch is varied for circumferential compliance control

Source: DellaCorte & Valco, "Load Capacity Estimation of Foil Air Journal Bearings for Oil-Free Turbomachinery Applications, NASA/TM-2000-209782, ARL-TR-2334, Oct 2000

Generation III Foil Bearings *(1990's)*



- **Load capacity coefficient, D_j's, 0.8 – 1.0**
- **Tailoring the foil support structure stiffness in**
  - **Axial** (L)
  - **Circumferential** (Θ)
  - **Radial** (r) *(i.e., displacement sensitive)*

  directions to enhance bearing performance

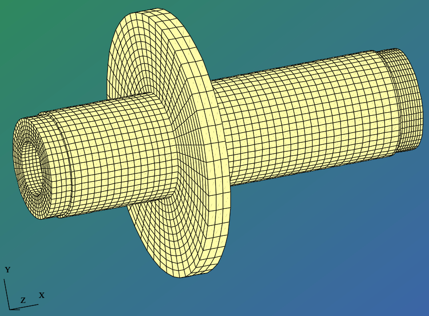
Source: DellaCorte & Valco, "Load Capacity Estimation of Foil Air Journal Bearings for Oil-Free Turbomachinery Applications, NASA/TM-2000-209782, ARL-TR-2334, Oct 2000

Enabling Technology Breakthroughs

- **Advanced Foil Bearings**
  - Load capacity has doubled

- **High-Temperature Solid Lubricant Coating**
  - NASA PS400, 100,000 start/stops, 25 °C to 650 °C

- **Analytical & Rotordynamic Modeling**
  - Less time, risk & cost from concept to application



Oil-Free Turbomachinery Technology Path

Air Cycle Machines (ACM's)

1970's

- Clean Oil-Free Cabin Air
- High Reliability
- Maintenance Free

Turbocompressors

1980's

- No Process Fluid Contamination
- Cryogenic Capability
- Long Life

Turbogenerators

1999

- Low Emissions
- Lightweight
- Maintenance Free

Turbochargers

2000

- Mounting Orientation Freedom
- No Particulate Emissions
- High Temperature

Small Gas Turbine Engines

2010

- High Speed
- Low Cost
- Maintenance Free

Mid-Range & Large Engines

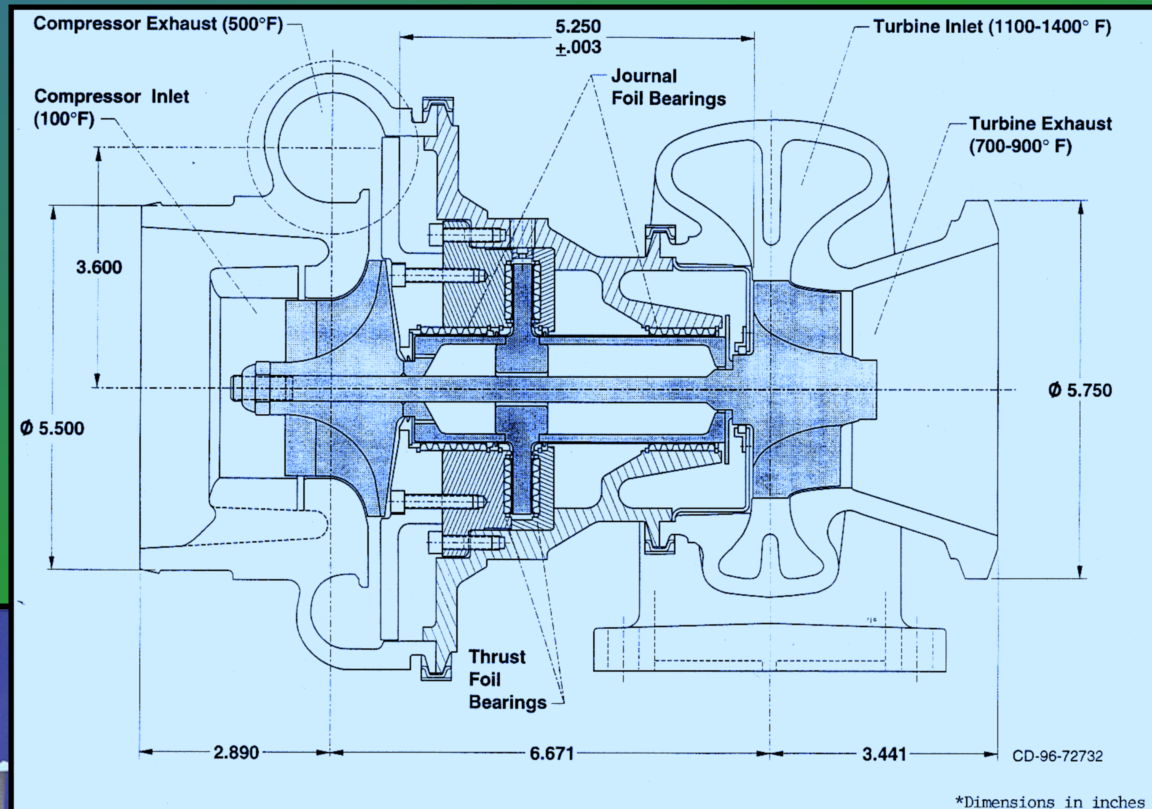
2020

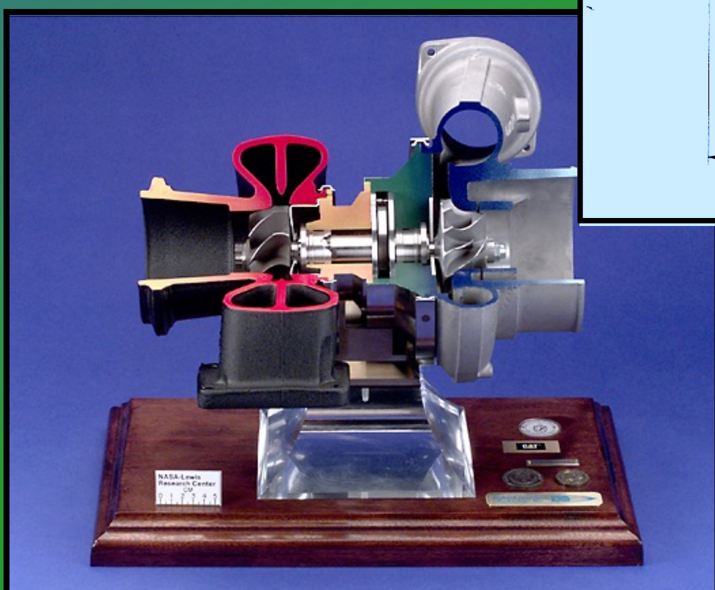
- High Temperature & High Speed
- Design Architecture Freedom
- Revolutionary Engines

...Starting a Turbomachinery Revolution...

Oil-Free Turbocharger (1999)

- 2 Journal Foil Bearings
- 2 Thrust Foil Bearings
- NASA PS304 Coating
- Rigid Rotor



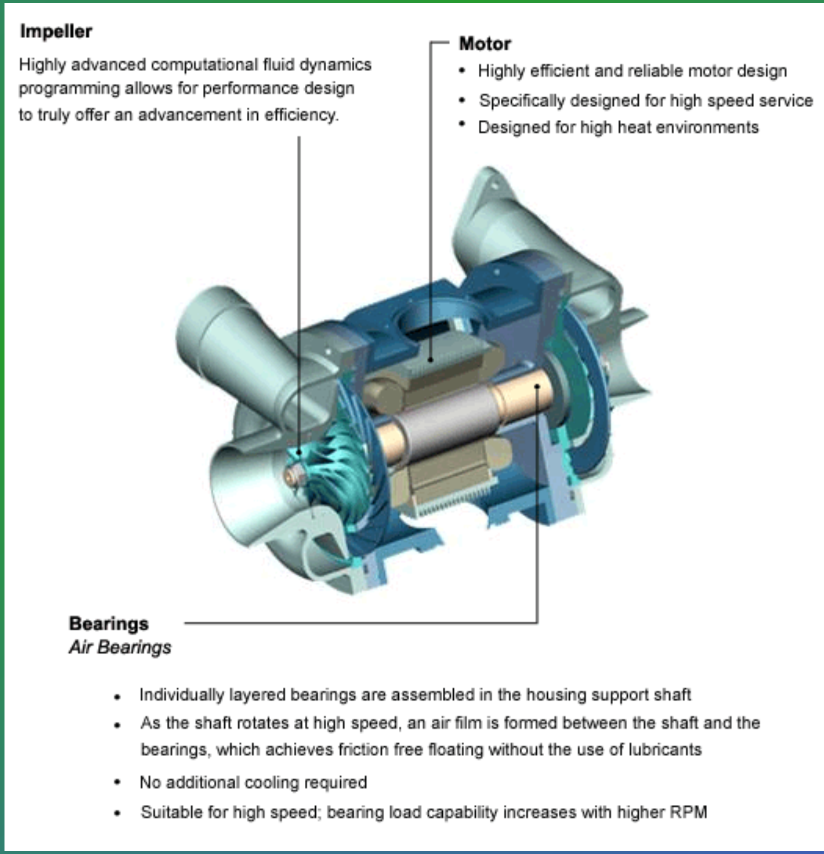


Recent history

- New foil gas bearing supported products hitting the marketplace.
    - (5-500 hp) turbocompressors, (30-200 kW) microturbine generators, automotive turbochargers, APU's and industrial blowers.
- Many of these machines use foil bearings of first and second generation designs now off patent protection.
- Korea is the most active region for Foil Bearing R&D and new product development.

Korean Foil Bearing R&D/Users

- Neuros,
- Kturbo,
- KFM (Korea Fluid Machinery)
- Samsung Techwin
- Turbomax
- KAIST
- KIST
- KIMM
- Hangwa Motors
- Hyundai
- LG
- Others



Hi-Speed Blower: Most common Korean Oil-Free product today.

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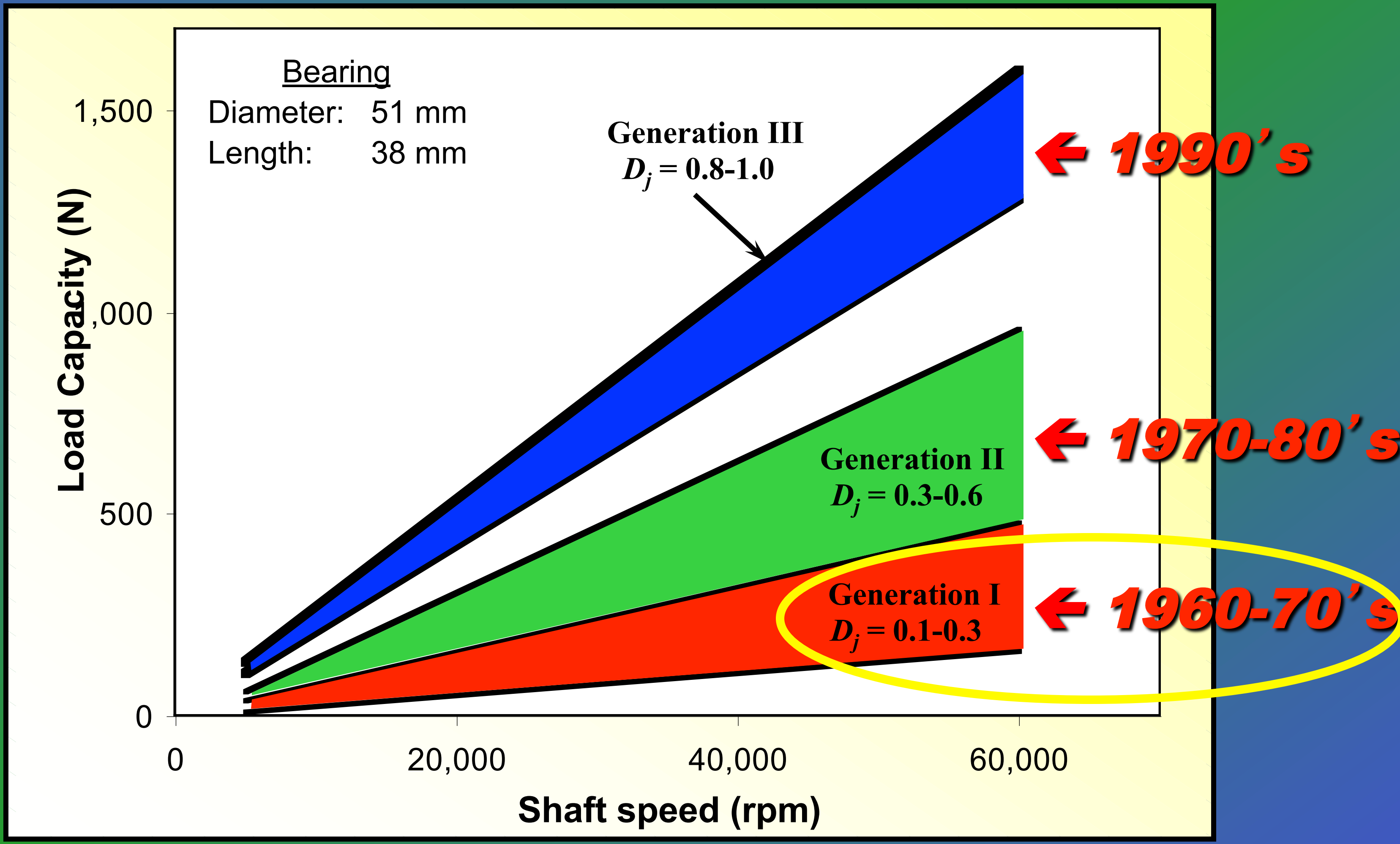
Where did these bearings come from and what can be learned from studying their history?

"So where did these foil bearings come from? How did we get to where we are? Who was responsible?"

Technological Path History: Goals

"You cannot know where you are going unless you know where you've been"

Foil Bearing Load Capacity – Generation I, II, & III



•A study of the early years provides valuable insight into foil bearings

Approach:

- Conduct literature search concentrating on earliest references to foil bearings
- Identify key breakthroughs in understanding and design.
- Carefully read publication reference lists and obscure government reports for clues describing technology dissemination and connections.

Blok and vanRossum (1953): the very first paper

THE FOIL BEARING— A New Departure In Hydrodynamic Lubrication*

by H. Blok, Technical University
Delft, Holland
J. J. vanRossum, Royal Dutch/Shell Lab.
Delft, Holland

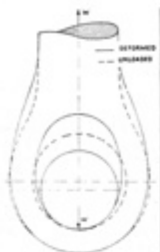
NOMENCLATURE

f	coefficient of friction
h_0	thickness of the oil film between foil and journal
h_{min}	minimum thickness of the oil film
N	rotational speed of journal
p	pressure
p_m	mean pressure per unit of projected bearing area
R	radius of journal
ΔR	radial clearance in a classical bearing
r	radius of curvature
σ	tensional force in the foil per unit of width
Z	viscosity
ZN/p_m	dimensionless parameter
U	peripheral speed of journal

INTRODUCTION: In a journal bearing running in the region of hydrodynamic lubrication, the load is borne by the pressures developed in the oil wedge that establishes itself between the surface of the journal and that of the bearing. Assuming the journal to be absolutely rigid, one can distinguish two limiting cases: (1) The bearing is likewise completely rigid. The classical bearing with which the classical theory of hydrodynamic lubrication concerns itself, is a case in point. (2) The bearing is entirely devoid of rigidity. An example is a bearing consisting of an extremely flexible foil, stretched around half the circumference of the journal. This bearing has been termed "foil bearing."

Intermediate between these limiting cases are those in which the rubbing surfaces are prone to deformation caused, either by the pressures in the oil wedge or by the external forces acting on the bearing. These deformations are of two kinds, viz.: (a) The deformations in the immediate neighbourhood of that section where the film thickness attains its minimum. They are caused by the hydrodynamic pressures, which there reach their maximum. There is an interaction between generation of hydrodynamic pressures and the elastic deformations of the surfaces. The study of this interaction is primarily of interest in relation to the lubrication of gears in which the oil pressures are of the order of the contact pressures calculated according to Hertz[1,2]. (b) Deformations of the bearing as a whole, caused by external forces that introduce circumferential bending moments into the bearing shell. Bearings are more susceptible than gears to this sort of deformation[3,4]. By way of example, Fig. 1 shows a big-end bearing with the bending deformation greatly exaggerated. In this example, the deformation causes lengthening of the active part of the oil wedge, and this promotes the hydrodynamic performance of the bearing.

The foil bearing may be considered as a bearing that offers no resistance at all to deformation. This limiting case has been realized by means of a model in which a very thin cellophane foil is stretched round a journal and loaded as in Fig. 2. From experiments with this model it has been found that the pressure and the thickness of the film separating the foil from the journal are constant except for the leading and trailing portions of the film. The foil thus causes an oil film to form that has parallel surfaces, along the whole length of which, remarkably enough, the pressure is constant. Although the classical theory of Osborne Reynolds states that such a film is possible in the case that the surfaces are parallel, with rigid surfaces this can be realized only if the pressure is maintained artificially at the trailing portion of the film, for instance by introducing oil under pressure[5].



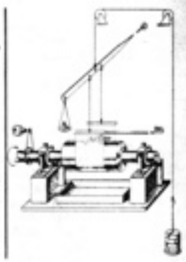


Fig. 1 (left) Flexure of a big-end bearing.

Fig. 2 (right) Sketch of a model foil bearing.

EXPERIMENTS: Friction on the foil has been measured by means of a model, using the set-up shown in Fig. 2. An electric motor drives a journal 6 cm (2.35 in.) in diameter, round which is a cellophane foil stretched by the load (see Fig. 2). The foil is 20 cm long (7.88 in.), 0.008 cm thick, (about 3 thous.) and 8 cm (3.15 in.) wide (4 cm, that is 1.58 in., in some experiments). The speed of the journal was varied between 6 and 1500 rpm, the load between 1 and 27 kg (2.2 and 60 lbs), and the viscosity of the oil between 6.5 and 100 centi-

*Paper presented before the Symposium on Engineering Studies of Bearings, organized by the Assoc. for the Advancement of Mechanical Engineering, Paris, June 23, 1952; republished by courtesy of Ingenieurs et Techniciens, see No. 51, 1953, 29/32.

- Oil-lubricated plastic film, draped over a rotating steel shaft.
- Mimics highly loaded sleeve bearings at more moderate conditions.
- First recognition that oil film pressure create the clearance.
- Resulting, more uniform and thicker film has potentially lower friction.
- No indication of use as a gas bearing.

Patel and Cameron (1957):

the second paper

Oct 1957

THE FOIL BEARING

Paper 73‡

By B. J. Patel* and A. Cameron, M.Sc., Ph.D., Assoc.I.Mech.E.†

INTRODUCTION

BLOK AND VAN ROSSUM (1953) have shown that a flexible foil, wrapped U-shaped round a shaft, can form a hydrodynamic oil film capable of carrying a load. The mechanism is that at the inlet convergent section, where the vertical leg of the U and the rotating shaft approach each other, an oil pressure is generated. This pressure continues unaltered right round the bend of the U. The pressure must be sensibly constant as the foil is flexible and the frictional force is small compared with the tension in the foil. This means that as the tension T is uniform circumferentially, the pressure p is equal to T/r, where r is the radius of the shaft.

Blok and van Rossum used a transparent cellulose foil. In this research a steel foil, 0·002 in. thick, is used.

One advantage of a metal foil is that by brazing copper wires on to the back surface of the foil, a temperature plot can easily be made. The thermo-electric characteristics of steel/copper were determined separately.

It was realized that the foil, apart from being flexible, must also stretch as a result of the pressure and tension. As the oil pressure at either edge of the foil must be atmospheric, the tension at these points must therefore be zero, assuming the foil is separated from the shaft at all points, by an oil film. This means that the pressure (and tension) will vary axially, being maximum in the middle and zero at either edge. Furthermore, the entry conditions are much more complicated than the simple theory would lead us to suppose. The foil itself is deflected by the pressure in the entry portion, and the radius of curvature alters inversely with the pressure.

The equations for both these extensions to the simple theory are set up in the appendices. The solution of these equations is not straightforward and is receiving further study.

APPARATUS

The general arrangement of the foil bearing test rig can be seen in Fig. 73.1. The journal (3 in. diameter) was driven by a ¾-h.p. shunt wound variable speed motor whose range was 100 to 1600 rev/min. The foil was stretched round half the circumference of the journal and held in a block, which allowed the friction torque to be measured and a uniform load to be applied to the foil. The load was produced by a simple lever mechanism of ratio 12/1. The frictional torque was measured by a spring balance at the end of the torque arm. The oil was supplied from a gravity tank and was pumped back from a sump. For normal running the temperatures of the oil were measured in the centre of the foil at 2 points, 60° and 120° from the inlet. The mean of these two temperatures was used to obtain the viscosity.

Less scatter was found on the $\mu \rightarrow (\eta N/p_m)$ curve if the speed was kept constant and the load varied. The reason was that there is a bigger variation of temperature round the foil with speed than with load.

EXPERIMENTAL RESULTS

Temperature Measurement

The temperature was measured round the foil by brazing to it copper wires at various points. These junctions acted as the hot end of a thermocouple, the cold junction being formed by brazing a similar copper wire near the holding block. A mercury-in-glass thermometer at this point gave the datum temperature. This thermocouple pair was calibrated and a potential difference of 5·35 $\mu V/^{\circ}C$ was found.

The distribution of temperatures round the foil was measured at various speeds and loads. Typical results are shown in Fig. 73.4. The variation of maximum temperature with speed at different loads is shown in Fig. 73.2. Maximum variation of temperature with load, within the rubbing portion of the foil is shown in Fig. 73.3, for various speeds. Ordinary straight S.A.E. 10 oil was used for these tests, with viscosity of 31·7 cS at 100°F (38°C) and 5·2 cS at 210°F (99°C).

Measurement of Coefficient of Friction

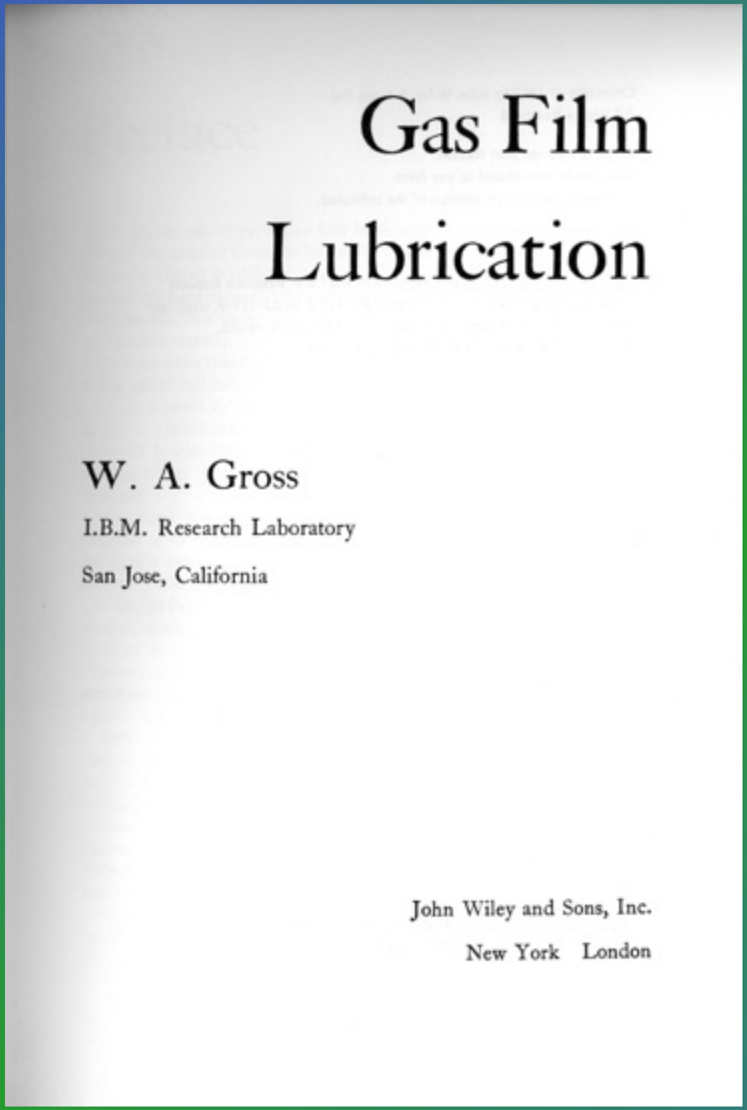
The graph of coefficient of friction for a wide range of $\eta N/p_m$ is shown in Fig. 73.5, where η is viscosity in lb sec/in² (reyns), N speed in rev/s and p_m load in lb/in².

The M.S. of this paper was received at the Institution on 30th March 1957.
* *Research Student, City and Guilds College, S.W.7*
† *Lecturer, City and Guilds College, S.W.7.*
‡ *Discussed on pp. 201, 743, 833.*

H*

- Oil-lubricated metal foil, wrapped over a rotating steel shaft.
- Foil backside covered in thermocouples to map temperature distribution.
- First use of metal foil and engineering loads.
- Pressure distribution speculated as varying axially and circumferentially.
- Modeling assumed to be complex.
- No indication of use as a gas bearing.

W.R. Gross (1962): the third well known reference


Gas Film Lubrication

W. A. Gross

I.B.M. Research Laboratory

San Jose, California

John Wiley and Sons, Inc.

New York London

- The classic, comprehensive book on gas bearings.
- Heavily based upon mathematics.
- Originally written as internal IBM reports (chapters) to guide corporate development of disk drives and high speed machinery.
- Appears to be the first written acknowledgment for a gas foil bearing.
- Led to Gross being recognized as the source for foil gas bearings though the book vaguely suggests others.

W.R. Gross (1962): pages 138-140.

same radial clearance is 0.711, (0.234 for a similar Type 2 bearing). The appreciable reduction in friction makes this type of bearing especially valuable for high-speed applications.

The lemon or elliptical finite bearing with liquid lubricant was analyzed by Pinkus [1956a, b] who used a digital computer to solve the finite difference equations involved. The results agreed well with experiment. Pinkus [1958] used a similar technique to obtain computer solutions for the three-section finite bearing.

3.10 Foil Bearing

It is possible to use a flexible band as a bearing for a rotating shaft. Blok and Van Rossum [1953] first demonstrated the feasibility of this configuration, which they called a foil bearing. Since then, Patel and Cameron [1957] among others have experimented with this type of bearing.

When a foil is used as a bearing for a rotating shaft, the angle of wrap is likely to be approximately 180°. A variant of this configuration is employed in magnetic-tape machines in which a tape glides over a recording magnetic head; under these conditions, the angle of wrap may be anywhere between 1 and 180°. When the angle of wrap is very small, the slenderness ratio may be large enough that sufficient accuracy is obtained by setting $L = \infty$, thus enormously simplifying the analysis. Furthermore, velocities are often low enough that only solutions for $\Lambda \to 0$ are necessary.

Attention must sometimes be directed to the elasticity of the band because some axial deflection is present. Since the pressure is ambient at the edges and above ambient under the band, the band tends to form a containing pocket for the lubricant. Furthermore, dynamic instability will appear unless L/B is sufficiently large.

Analysis of a foil bearing is complicated by the fact that the film thickness cannot be specified a priori. It is necessary to obtain an additional equation to express the band dynamics. Refer to Fig. 3.10.1, which illustrates a differential section of an infinitely long foil bearing. The foil density is σ per unit arc length, the velocity is U, the pressure is p, and the frictional resistance is F. The radial coordinates of the solid body and band at this section are r_1 and r, respectively, the curvature of the band is $\varkappa$, and the film thickness is h. Thus $r = r_1 + h$.

By restricting this analysis to the infinitely long bearing, axial variations may be ignored. For steady conditions, therefore, equilibrium of forces on a differential unit section of arc with coordinate s requires

$$\frac{dT}{ds}\,ds = F ds, \qquad (3.10.1)$$

in which $ds = d\beta/\varkappa(\beta)$, T is the band tension, and F is the friction. In addition,

$$T d\beta = \frac{p}{\varkappa}\,d\beta + \sigma U^2\,d\beta. \qquad (3.10.2)$$

By virtue of (3.10.1), the change in tension along the film must equal the frictional resistance. It has been observed in a lubricating film with rigid boundaries that F/W is comparable to h/B. The tension change in

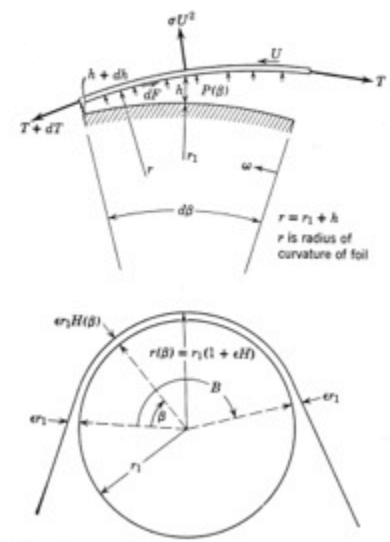


Fig. 3.10.1. Foil bearing over a circular cylinder and differential element of foil bearing.

a foil bearing is usually small enough to be ignored. The curvature of the band is given by

$$\varkappa = \frac{2r'^2 - rr' + r^2}{(r'^2 + r^2)^{3/2}}, \qquad (3.10.3)$$

in which the prime is used to represent differentiation with respect to the angle, i.e., $r' = dr/d\beta$.

"…others have experimented with this type of bearing."

Oil-Free Turbomachinery Program

W.R. Gross (1962): pages 138-140.

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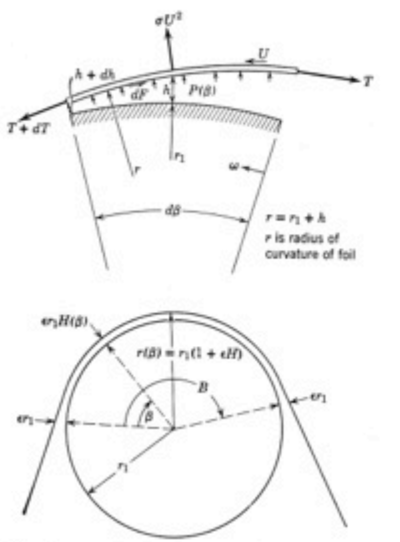

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What does this mean???

W.R. Gross (1958): Original internal IBM Chapter 5. (Courtesy IBM Archives)

RJ 117-5
October 10, 1958

FILM LUBRICATION

V. Infinitely Long Incompressible Lubricating Films of Various Shapes

by

W. A. GROSS

This is the fifth of a series of reports which deal with the theoretical aspects of hydrodynamic film lubrication. The aim is to aid understanding of the properties of gas lubrication.

One of nine internal IBM reports written between 1957 and 1960 that formed the basis of the 1962 book

W.R. Gross (1958): Chapter 5, page 70

sections. In these regions, the pressure may be determined by Equation (5k-7). An approximation of the extent of the constant film thickness region is then obtainable from equilibrium considerations.

Equation (5k-13) provides a satisfactory approximation to both lead in and lead out regions of an air film subject to the restrictions that $H \to 1$ and the bearing number is sufficiently low. At very large bearing numbers, Equation (4b-3), in combination with Equation (5k-7), results in an untabulated elliptical form for the pressure.

A liquid lubricating film may be approximated by ignoring the lead out region.

Application of Laplace transforms to Equation (5k-13) yields

$$\Delta = \mathcal{L}^{-1} \frac{\Delta(0)\, s^2 + \Delta_\theta(0)\, s}{(s+\lambda)(s-\lambda/2+i\sqrt{3}\lambda/2)(s-\lambda/2-i\sqrt{3}\lambda/2)},$$

in which the Laplace operator is s, and the film thickness variable Δ is measured from $\theta = 0$, at the origin of the outflow region for positive angles. It may also be applied for negative angles from $\theta = 0$, at the termination of the inflow region. In each case, $\theta = 0$ occurs where $H = 1$, $P = P_M$. By taking the inverse transform,

$$\Delta = H - 1 = \frac{1}{3}\Big\{\left(\Delta(0) - 3\Delta_\theta(0)\lambda^{-1}\right)e^{-\lambda\theta} + e^{\lambda\theta/2}\Big[\left(2\Delta(0) + \Delta_\theta(0)\lambda^{-1}\right)\cos(\sqrt{3}\lambda\theta/2) + \left(\sqrt{3}\Delta_\theta(0)\lambda^{-1}\right)\sin(\sqrt{3}\lambda\theta/2)\Big]\Big\}. \quad (5k\text{-}14)$$

The result is that the inlet section is governed largely by the exponential term, and the outlet section by the expanding sinusoidal terms. The boundary conditions, $\Delta(0)$ and $\Delta_\theta(0)$ determine the nature of the coefficients. For very small bearing numbers, $\Delta_\theta(0)$ is of primary importance compared to $\Delta(0)$. The reverse is true for large bearing numbers.

Baumeister, working with V. Nejezchleb has observed good correlation between experimental film thickness determinations and those obtained by integration of Equation (5k-13).

Pressure

The pressure may be determined by combining Equations (5k-7) and (5k-14).

"Baumeister, working with Nejezchleb has observed good correlation between experimental film thickness determinations and those obtained by integration of equation (5k-13)."

J.T.S. Ma (1965): Ampex Corporation

An Investigation of Self-Acting Foil Bearings

J. T. S. MA
Member of Research Staff, Ampex Corporation, Redwood City, Calif. Now with International Business Machines Corporation, Los Gatos, Calif. Assoc. Mem. ASME

Experimental results on the interior and the exit region film-thickness measurements of self-acting foil bearings are presented and discussed. These measurements were made with capacitive sensors and conductive foils. The measured and predicted values agree very well within the range of nondimensional parameters, h_0/R, from 10^{-4} to 10^{-3} and, $T/\mu U$, from 10^4 to 10^6. The agreement deviates for values beyond these ranges. Empirical expressions for predicting the constant and minimum film thickness applicable beyond these ranges are also presented. They are valid within the range of h_0/R from $5(10)^{-6}$ to 10^{-3} and $T/\mu U$ from 10^4 to 10^8. Growing sinusoidal film thickness in the exit region was also observed, measured, and checked with theoretical predictions.

Introduction

In most applications of fluid-film lubrication, the lubricant is contained between two rigid, or nearly rigid, surfaces. Blok and Van Rossum [1][1] first demonstrated the feasibility of using a flexible foil as a bearing for a rotating shaft and termed this the foil bearing. In their pioneering investigation, they showed that it could form a self-acting oil film capable of carrying a load. They found experimentally that the film thickness is approximately constant over most its interior region and that cavitation can occur in the exit region of the oil film. Later, Patel and Cameron [2], also using oil as a lubricant, measured the temperature distribution round the foil for different speeds and loads. In both experiments, no film thickness measurements were made.

Theoretical investigations of the self-acting foil bearing problem have been performed by several authors using perturbation theory. Baumeister [3] used numerical method in his solution the nominal clearance of the foil bearing in terms of other parameters. Langlois [4] developed a method for determining, to the first order, the deflection from a straight path of a perfectly flexible tape moving near a rigid cylinder. All the analyses of self-acting foil bearings have been directed toward incompressible fluids, infinitely wide, and perfectly flexible foils with effects of fluid friction assumed to be negligible. They agreed on an approximately constant film thickness in the interior region, as did Blok and Van Rossum. Langlois, in his solution, considered the first integral of the lubrication equation and showed that there is a region where the pressure under the foil is subambient; in this region the foil is concave away from the cylinder. Gross [5] showed that the film thickness in the entrance region is determined largely by an exponential term in his equation while in the exit region, it is determined by an exponentially growing sinusoidal term. Recently, Barlow [6] solved for the film thickness in exit region with gap discontinuity and obtained an expression for the minimum film thickness in terms of the constant film thickness. It is significant that he also obtained the theoretical wavelength of the sinusoidal waveform at the exit region. To the best of our knowledge, there have been no experimental investigations published to confirm these findings.

Because of the complexities involved and the simplifying assumptions needed in solving the nonlinear, high order, partial differential foil bearing equation, a complete theoretical analysis is difficult. Therefore, the foil bearing problem lends itself readily to experimental investigation. This report describes experiments on the measurements of foil bearing film thicknesses by means of capacitance probes intended to provide confirmation for theoretical results previously available. It also describes attempts to verify the existence of the three regions of the film under the foil. This report presents the complicated case of the exit region, particularly in regard to the measurements of the minimum film thickness, and the wavelength of the sinusoidal waveform. We were especially interested in these latter experiments since they had not been investigated before.

The aim of this report, aside from confirming theoretical analyses, is to stimulate interest and provide physical data useful both in the solution of the complete self-acting foil bearing problem and in practical design applications.

Description of the System

Basic Equations. A typical configuration of the foil bearing is shown in Fig. 1. Various combinations of relative motions between the foil and the read-write head are possible. A rotating read-write head is usually in the form of a cylinder or drum; otherwise, any configuration will suffice. The wrap angle β may be any value between zero and 360 deg. In this report, we shall be concerned exclusively with the case in which the rigid surface is circular. For this case, it has been found [1, 3] that the pressure and the thickness of the film separating the foil from the cylinder are approximately constant, except for the leading and the trailing portions of film. The explanation for this is that in the entrance convergent section, where the foil and the rotating circular

[1] Numbers in brackets designate References at end of paper.

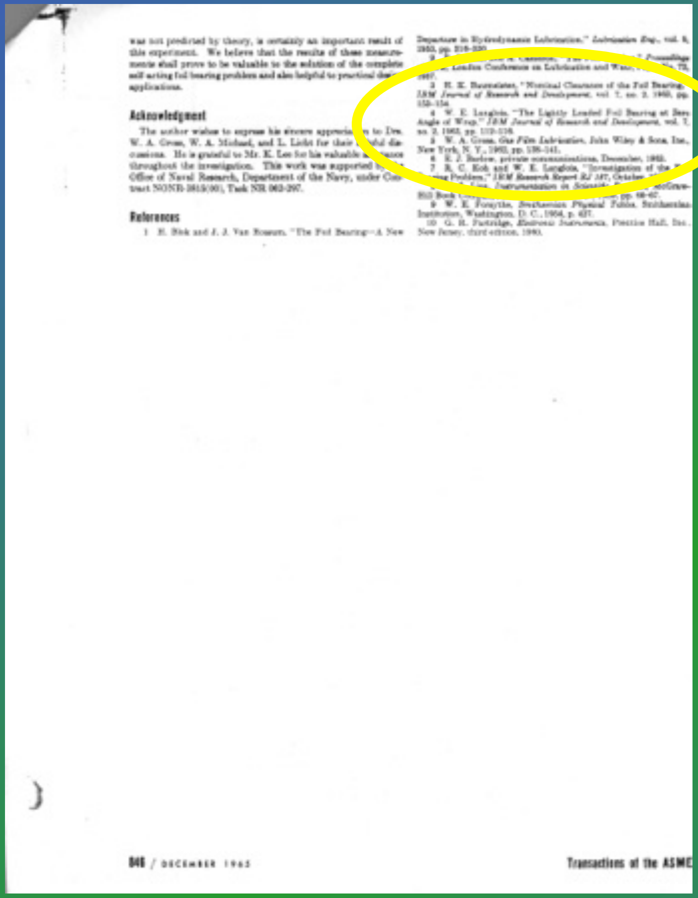
Contributed by the Lubrication Division and presented at the Lubrication Symposium, New York, N. Y., June 6-9, 1965, of The American Society of Mechanical Engineers. Manuscript received at ASME Headquarters, March 3, 1965. Paper No. 65—LubS-4.

Nomenclature

- A = capacitor area, in.^{2}
- B = wrap angle, deg
- C = capacitance
- C_p = parasitic capacitance, f
- C_r = capacitance between tape and drum, f
- C_s = capacitance between probe and tape, f
- C_t = total capacitance, f
- H = nondimensional film thickness $= h/h_0$
- K, k = constant
- p = bearing pressure, lb/in.^{2}
- P_a = ambient pressure, lb/in.^{2}
- R = radius, in.
- T = foil tension, lb/in.
- U = foil speed, in/sec
- h = film thickness, in.
- h_0 = average film thickness, in.
- h_m = minimum film thickness, in.
- s = length coordinate, in.
- Δ = change in gap from nominal, $(H - 1)$
- Σ = dielectric constant
- β = running angular coordinate s/R, rad
- δ = constants determined by boundary conditions
- ϵ = perturbation parameter
- κ = radius of curvature, 1/in.
- λ = constant $\frac{1}{\kappa}\left[\frac{T}{6\mu U}\right]^{1/3}$
- μ = viscosity (lb-sec/in.^{2})
- ρ = density of lubricant, slug/in.^{3}
- σ = density of tape
- τ = viscous drag, lb/in.^{2}
- ω = angular velocity, rad/sec

J.T.S. Ma and the AMPEX Corporation were relatively new players in the early foil bearing history investigation.

J.T.S. Ma (1965): Ampex Corporation

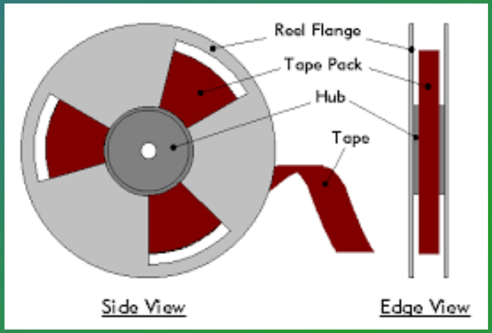


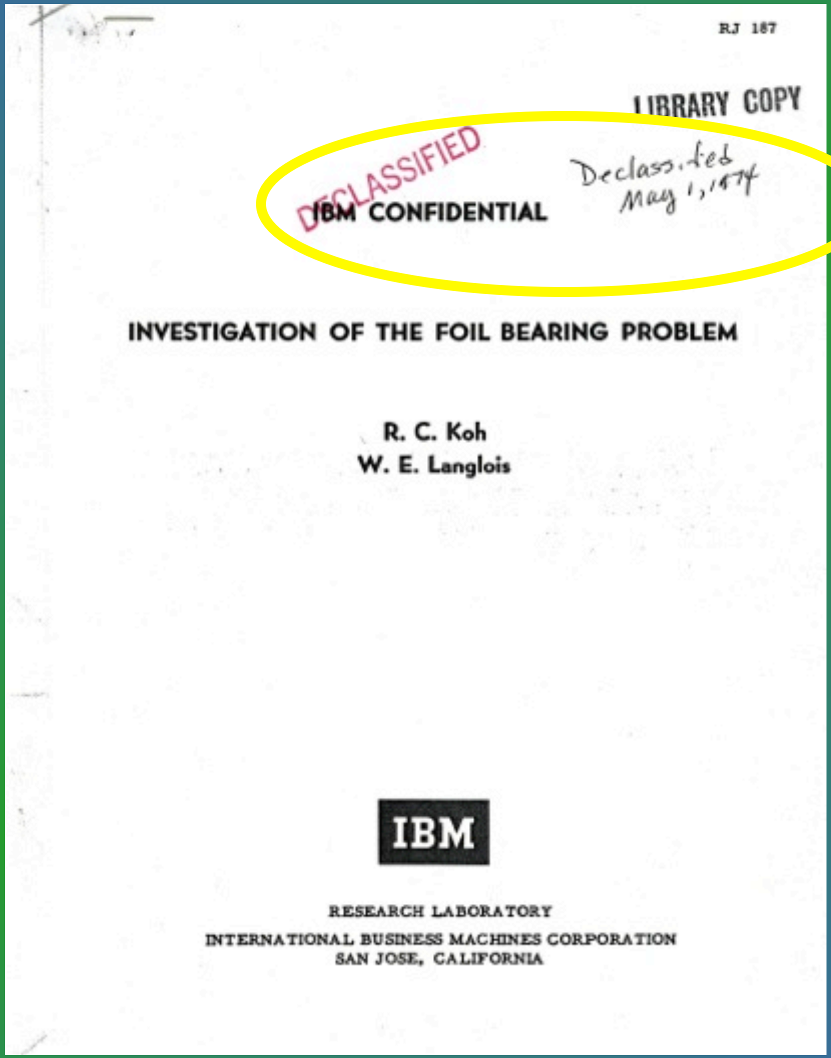
Reference 7:

R.C. Koh and W.E. Langlois, "Investigation of the Foil Bearing Problem," IBM Research Report RJ 187, October 1960."

IBM research reports are not public archival documents. Fortunately, IBM still maintains a library…

A reel-to-reel tape drive [photo courtesy of The Computer Museum]





RJ 187

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Declassified May 1, 1974

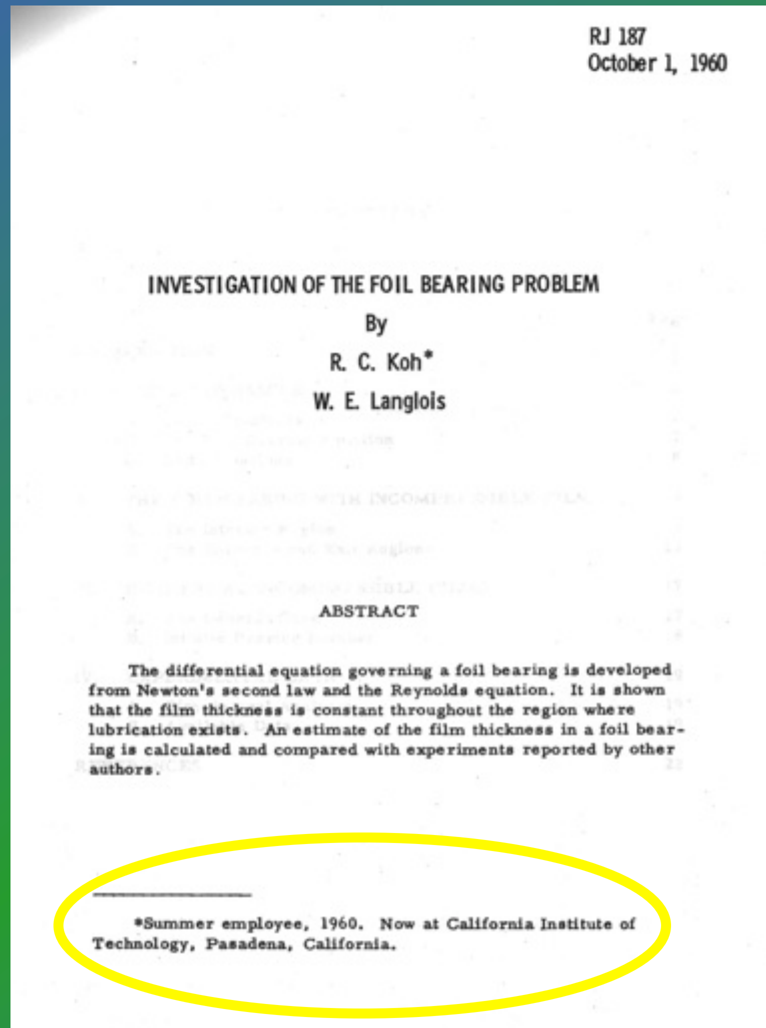
INVESTIGATION OF THE FOIL BEARING PROBLEM

R. C. Koh
W. E. Langlois

IBM

RESEARCH LABORATORY
INTERNATIONAL BUSINESS MACHINES CORPORATION
SAN JOSE, CALIFORNIA

Classified until 1974, this report offers a clear picture of early foil bearing discovery.



RJ 187
October 1, 1960

INVESTIGATION OF THE FOIL BEARING PROBLEM

By

R. C. Koh*

W. E. Langlois

ABSTRACT

The differential equation governing a foil bearing is developed from Newton's second law and the Reynolds equation. It is shown that the film thickness is constant throughout the region where lubrication exists. An estimate of the film thickness in a foil bearing is calculated and compared with experiments reported by other authors.

*Summer employee, 1960. Now at California Institute of Technology, Pasadena, California.

Written by R.C. Koh, an IBM "Summer Employee" in 1960 who went on to CalTech.

INVESTIGATION OF THE FOIL BEARING PROBLEM

INTRODUCTION

Blok and Van R[illegible] have [illegible] possible to develop [illegible] film between a rotating journal and a flexible foil [illegible] ca[illegible] the weight of the journal. Within IBM, Baumeister[2] has pointed out that a lubricating air film is sometimes built up when a flexible tape passes over a rigid object. He notes that the theoretical analysis of a foil bearing, as a system producing this effect is called, is quite different from the analysis of a bearing whose surfaces are all rigid. The Reynolds lubrication equation is strongly coupled to the differential [illegible]uations governing the tape dynamics. In all, there are three dependent vari[illegible]: the film pressure, the film thickness, and the tape tension

The situation is [illegible] of a journal or slider bearing in which the pressure is high enough to cause elastic deflection of the bearing surfaces--in this case, the Reynolds equation is coupled to the equations of elasticity theory.

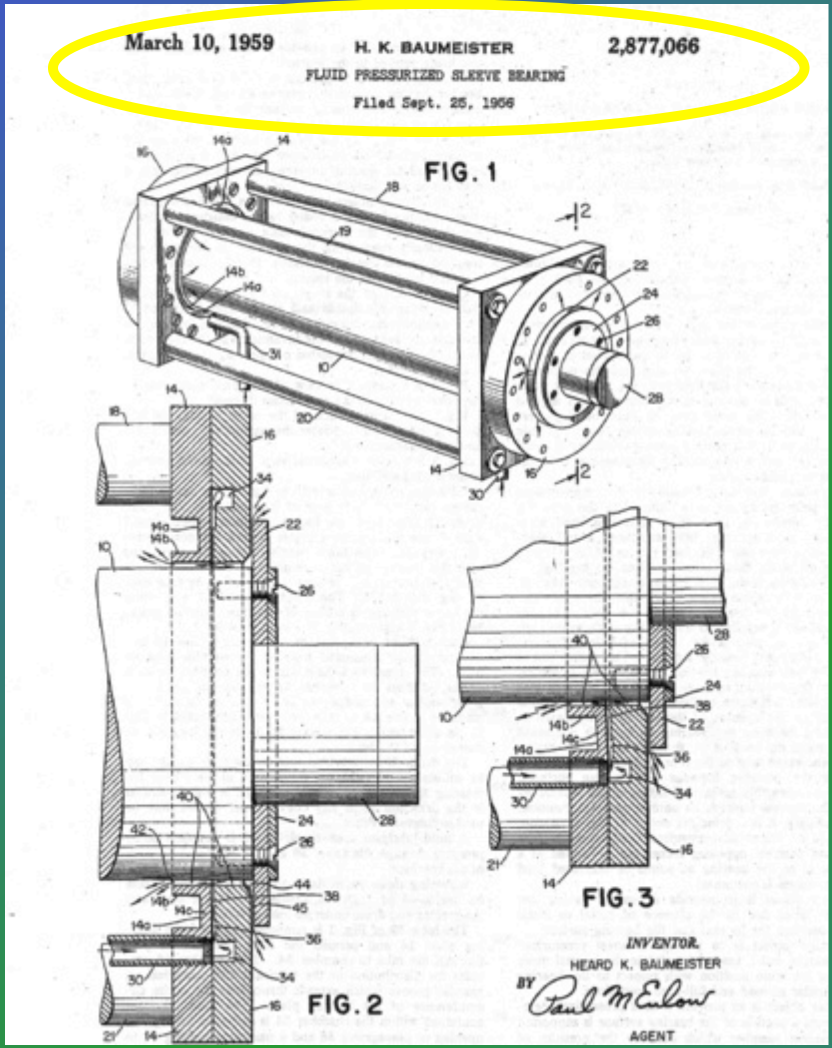
In some foil bearing problems, it is reasonable to assume, as Baumeister did, that the tension is constant. When friction is significant, the variation of tension cannot be ignored.

This report treats in detail several aspects of foil bearing theory. In Section I A we establish the differential equations which govern the motion of a lubricated tape. The dynamical equations governing the motion of an inextensible tape passing close to a circular cylinder are derived from Newton's second law. These equations, together with an appropriate form of Reynolds lubrication equation and a relationship between viscous drag and film thickness, are combined in Section I B to yield a fourth-order differential equation. This equation and the appropriate boundary conditions form the mathematical basis of foil bearing theory.

In Section II we consider the foil bearing with an incompressible film and show that, for cases of practical interest, there is an interior

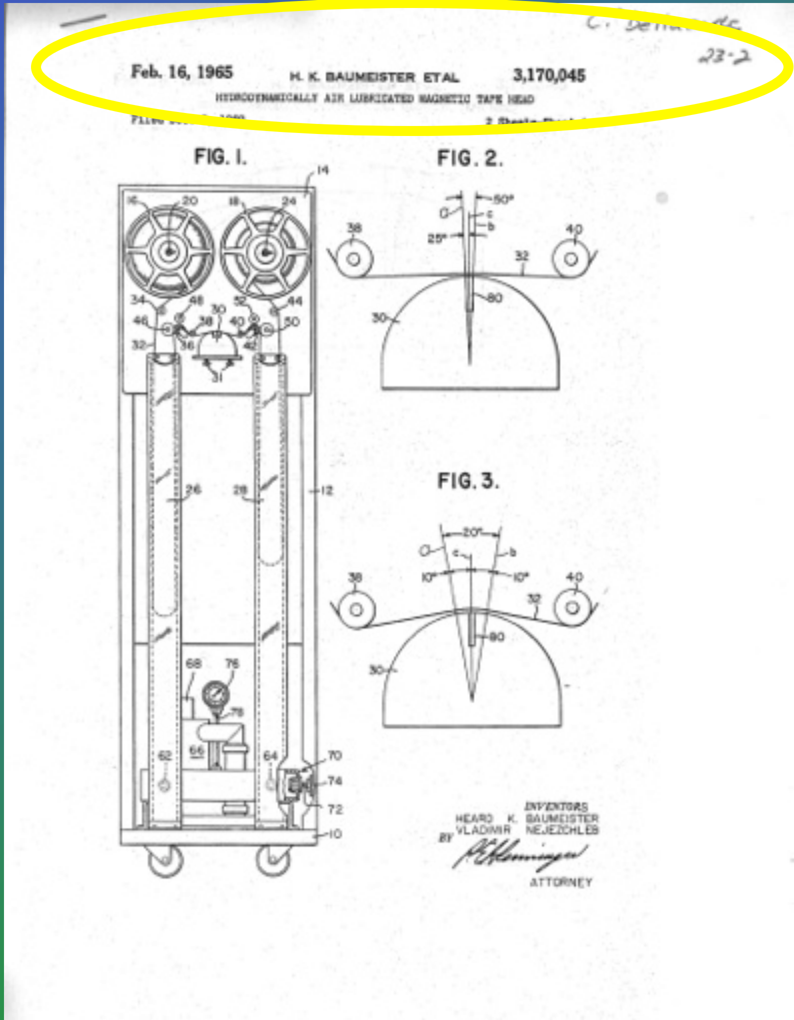
- Koh credits Baumeister for recognizing recording tape transport phenomena as a type of foil bearing.
- Baumeister also identified strong coupling of Reynold's Eq. with tape dynamics.
- Remainder of paper combines analyses from Gross and Elrod with experimental results from Baumeister-Nejezchleb's to develop analytical models for tape behavior.

In 1960, the foil bearing phenomenon was a "problem" for IBM and other tape drive companies.



- Baumeister actively working on gas bearings in mid-1950's, was well aware of lubrication field and the potential for gas bearings within recording technology.
- Patent literature does not show that IBM had any interest in foil bearings for shaft support.
- Foil bearing study limited to improving tape transport.
- Gas bearing work also focused on disk drive-read/write head lubrication.

Baumeister published several foil bearing briefs on modeling but moved to other technologies by 1970.

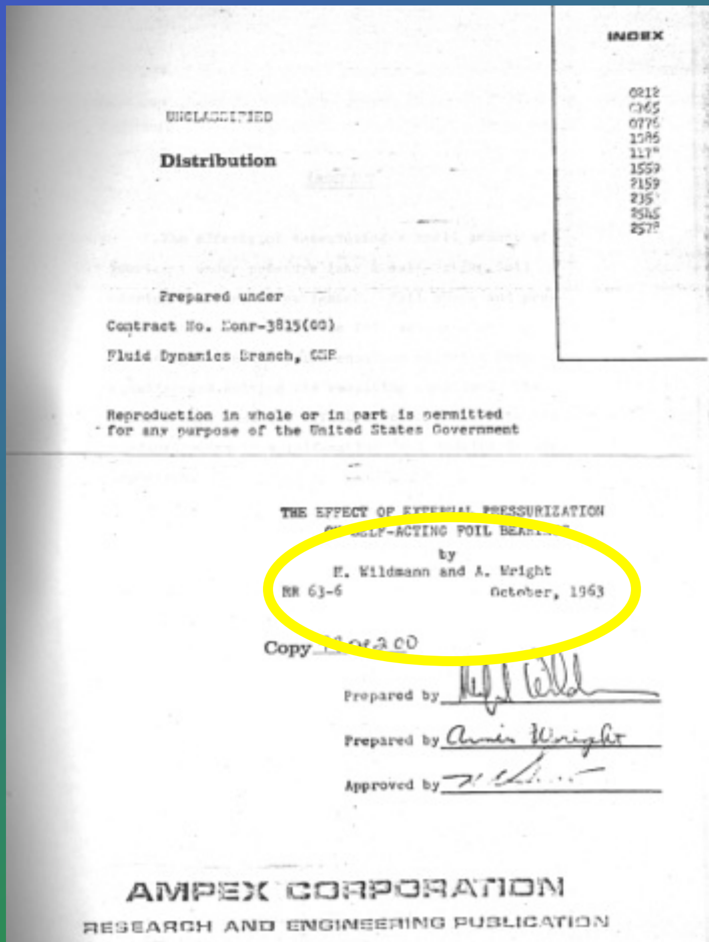


- IBM's foil bearing study limited to improving tape transport.
- Gas bearing work that continued, especially that in California, focused on disk drive-read/write head lubrication.
- W.R. Gross, inspired by Baumeister's results envisioned a practical rotor support system based on foil bearings.

Baumeister deserves credit for being the first to make the connection between tape motion and foil bearings.

"So how did the early foil bearing investigations move from IBM to AMPEX and beyond?"

"With W.R. Gross"



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THE EFFECT OF EXTERNAL PRESSURIZATION
ON SELF-ACTING FOIL BEARINGS
by
H. Wildmann and A. Wright
RR 63-6 October, 1963

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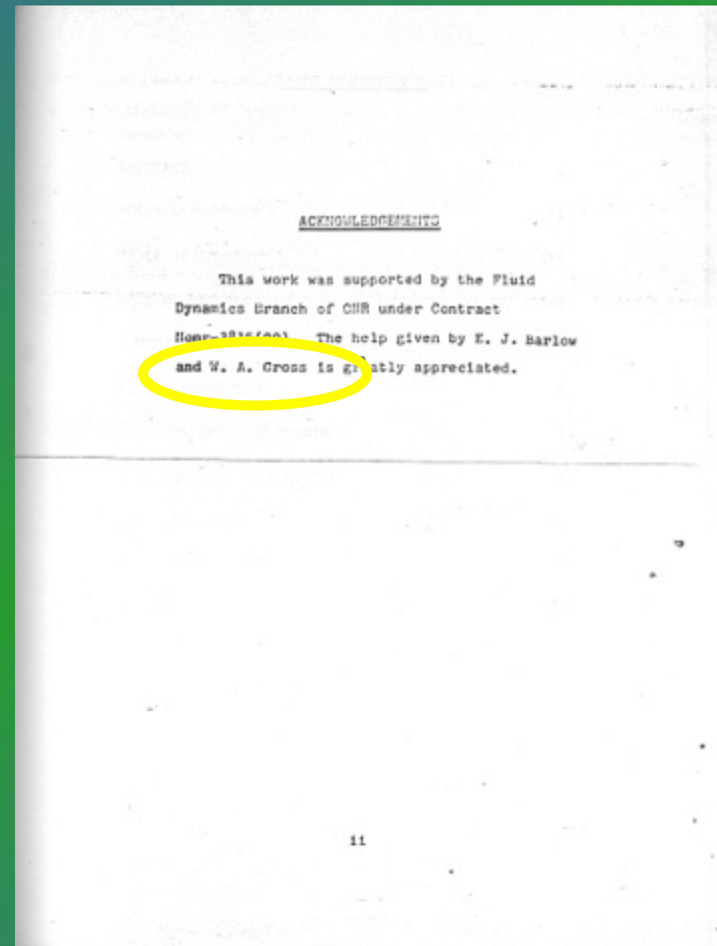

ACKNOWLEDGEMENTS

This work was supported by the Fluid Dynamics Branch of ONR under Contract Nonr-3815(00). The help given by E. J. Barlow and W. A. Gross is greatly appreciated.

ii

In 1963, W.R. Gross moved to AMPEX to join his colleague (Wildman) and further develop foil bearings as a shaft support technology under a NAVY funded bearing program.

Oil-Free Turbomachinery Program

AMPEX

Contract No. NAS3-13482

STUDIES OF FOIL JOURNAL-BEARINGS FOR
BRAYTON CYCLE TURBOMACHINERY

FINAL REPORT

RR 71-02 February 16, 1971

Prepared by

Ampex Corporation, Research Department
401 Broadway, Redwood City, California 94063

for

National Aeronautics and Space Administration
Lewis Research Center
21000 Brookpark Road, Cleveland, Ohio 44135

Prepared by: L. Licht, Principal Investigator

Approved by: P. Szego, Manager, Mechanics Section

RP 71-02 iii

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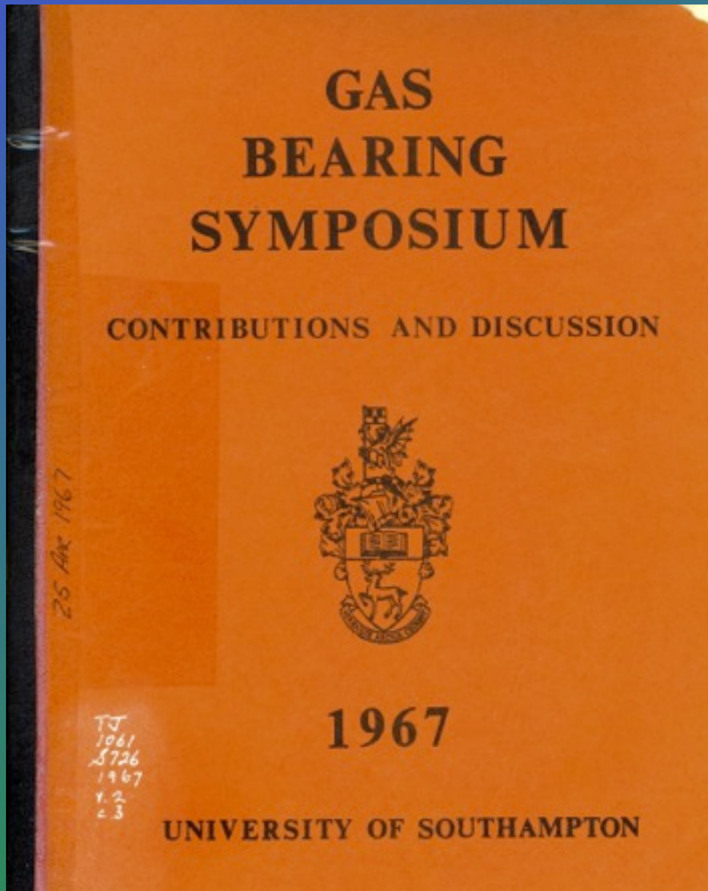
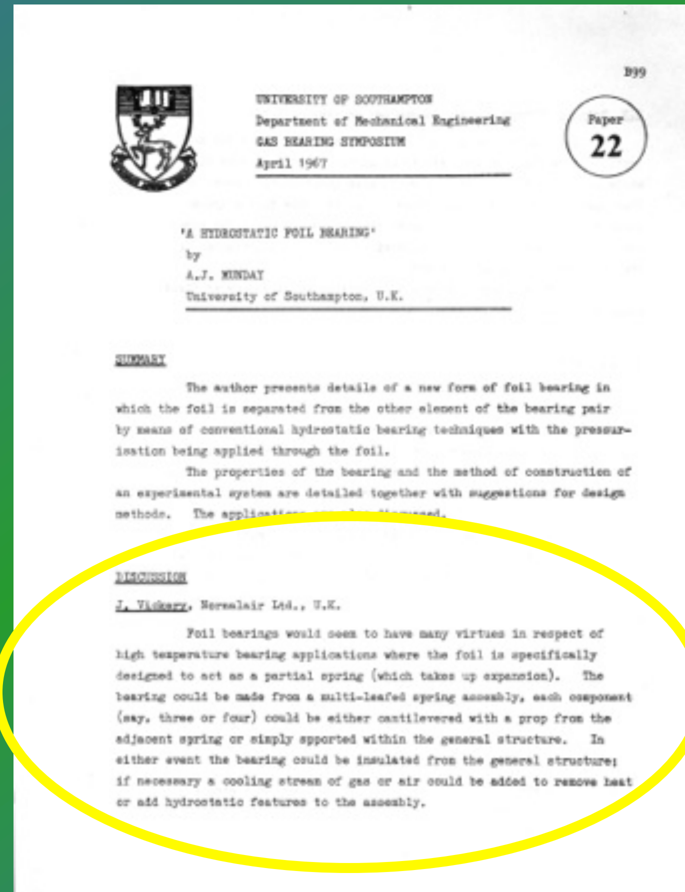
These US government funded efforts, managed by NASA resulted in widely distributed reports.





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UNIVERSITY OF SOUTHAMPTON
Department of Mechanical Engineering
GAS BEARING SYMPOSIUM
April 1967

Paper 22

'A HYDROSTATIC FOIL BEARING'
by
A.J. MUNDAY
University of Southampton, U.K.

SUMMARY

The author presents details of a new form of foil bearing in which the foil is separated from the other element of the bearing pair by means of conventional hydrostatic bearing techniques with the pressurisation being applied through the foil.

The properties of the bearing and the method of construction of an experimental system are detailed together with suggestions for design methods. The applications are also discussed.

DISCUSSION

J. Vickery, Normalair Ltd., U.K.

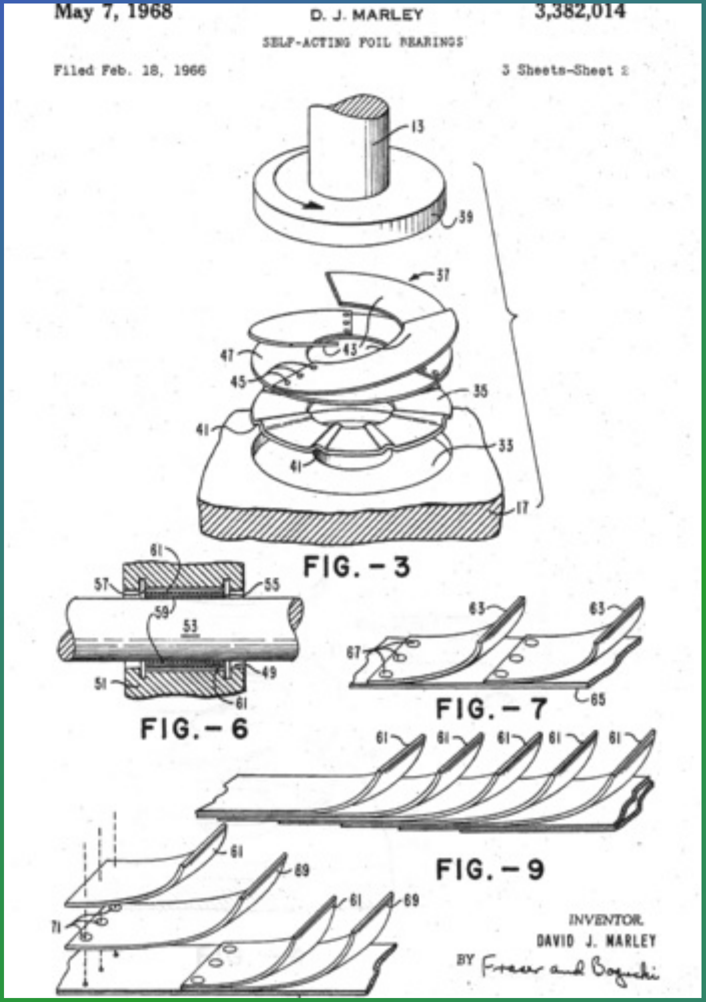
Foil bearings would seem to have many virtues in respect of high temperature bearing applications where the foil is specifically designed to act as a partial spring (which takes up expansion). The bearing could be made from a multi-leafed spring assembly, each component (say, three or four) could be either cantilevered with a prop from the adjacent spring or simply spported within the general structure. In either event the bearing could be insulated from the general structure; if necessary a cooling stream of gas or air could be added to remove heat or add hydrostatic features to the assembly.

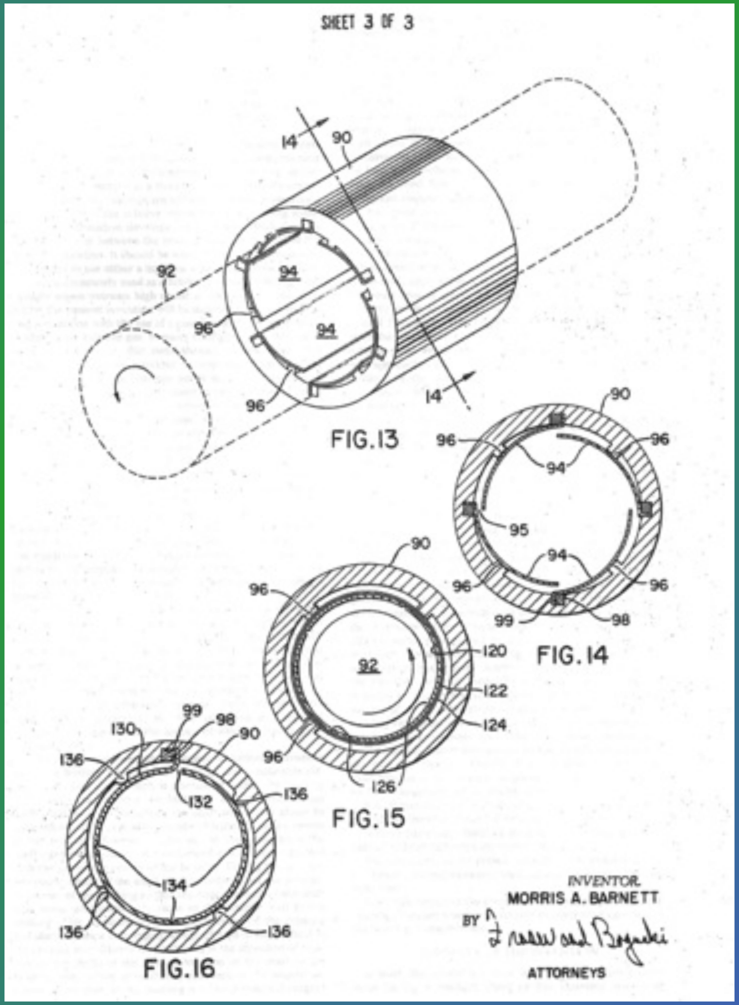
The technology also quickly spread to the UK as evidenced by this discussion of foil bearings suggesting a leaf type bearing in early 1967.

“Patent filings for all kinds of foil bearings followed…and the rest is technical history…”

Oil-Free Turbomachinery Program

The leaf foil bearings: (~1966 and beyond Marley, Barnett and Silver at AiResearch)





The bump foil bearings: (~1971 and beyond Cherubim, Gray and Shapiro at MTI)

United States Patent [19]
Cherubim

[11] **3,809,443**
[45] **May 7, 1974**

[54] **HYDRODYNAMIC FOIL BEARINGS**

[75] Inventor: **Justin Lawrence Cherubim,** Flint, Mich.

[73] Assignee: **Mechanical Technology Incorporated,** Latham, N.Y.

[22] Filed: **Aug. 5, 1971**

[21] Appl. No.: **169,372**

[52] **U.S. Cl.** **308/9**
[51] **Int. Cl.** **F16c 17/16**
[58] **Field of Search** 308/160, 121, 9, 73

[56] **References Cited**

UNITED STATES PATENTS

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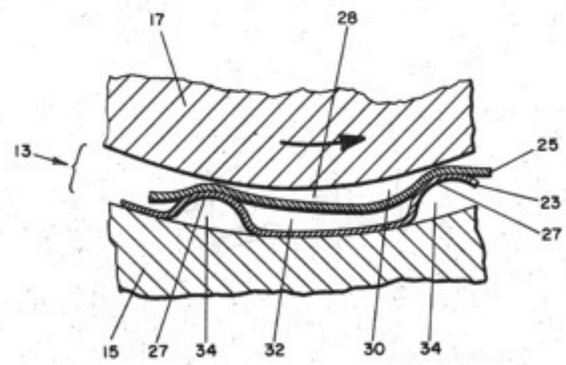
3,382,014	5/1968	Marley	308/160

Primary Examiner—Charles J. Myhre
Assistant Examiner—Frank Susko
Attorney, Agent, or Firm—Joseph V. Claeys; Charles W. Helzer

[57] **ABSTRACT**

A resilient hydrodynamic bearing wherein a resilient bearing insert made up of two separate bearing element members arranged in laminate relationship and anchored together to allow a limited amount of relative movement therebetween is disposed within the spacing defined by the confronting surfaces of the opening in a supporting structure and a shaft or other movable member received within the opening and anchored to the supporting structure to allow for a limited amount of relative movement and operative under dynamic conditions to establish a hydrodynamic fluid film support for the movable member. One bearing element member is disposed adjacent the movable member and presents a surface area thereto and the other bearing element member includes a plurality of spaced-apart resilient surface elevations formed therein which under dynamic conditions are operative to frictionally contact and resiliently support the said one bearing element member and cause it to deflect between adjacent resilient surface elevations to create the load supporting hydrodynamic fluid film:

23 Claims, 10 Drawing Figures



"By 1972, the first generation foil bearings (leaf and bump type) were entering production use in air cycle machines (ACM)' s."

Notable Names in Foil Bearings

- Blok and vanRossum
- Patel and Cameron
- H.K. Baumeister
- W.R. Gross
- Marley, Barnet and Silver
- Cherubim, Shapiro and Gray
- Miller
- Hehsmat
- Alston Gu, Marshall Saville
- Kang
- Bosley and Weissert

Where is Oil-Free Turbomachinery Headed?

Oil-Free Turbomachinery Technology Path

Air Cycle Machines (ACM's)
1970's
- Clean Oil-Free Cabin Air
- High Reliability
- Maintenance Free

Turbocompressors
1980's
- No Process Fluid Contamination
- Cryogenic Capability
- Long Life

Turbogenerators
2000
- Low Emissions
- Lightweight
- Maintenance Free

Turbochargers
2015
- Mounting Orientation Freedom
- No Particulate Emissions
- High Temperature

Small Gas Turbine Engines
2020
- High Speed
- Low Cost
- Maintenance Free

Mid-Range & Large Engines
20?0
- High Temperature & High Speed
- Design Architecture Freedom
- Revolutionary Engines

...Starting a Turbomachinery Revolution...

OIL-FREE TURBOMACHINERY PROGRAM

Propulsion for the 21st Century

NASA Glenn Research Center

Automotive OEM's are going Oil-Free

TOYOTA CRDL., INC.
TECHNICAL NEWS

Air Bearing for Automotive Turbocharger

Minoru Ishino

1. Introduction

The application of oil-free bearings such as air or magnetic types to automotive turbochargers is expected to realize a reduction in mechanical losses while eliminating oil consumption. As a result, engines fitted with oil-free turbochargers will offer an improved response and lower fuel consumption and exhaust emissions. We have designed compliant foil air bearings with uniquely shaped dampers for the journal and thrust bearings of small-sized turbochargers (**Figs. 1** and **2**).

2. Method

First, we undertook a rotational test of a prototype turbocharger with air bearings. The results of this test revealed the need to increase the load capacity of the thrust air bearing relative to that of the journal bearing. To solve this problem, we used 3D computational fluid dynamics to analyze the effects of the film thickness distribution between the bearing and runner surfaces of the thrust air bearing on the generated pressure distribution, so as to increase the load capacities of the bearing.

3. Results and conclusion

Numerical analyses revealed three effective methods of increasing the load capacity, namely, increasing the size of the fluid charge in the bearing, generating the maximum pressure at the center of the bearing surface, and preventing the leakage of the fluid in the radial direction of the bearing surface. To realize these three improvements, we devised a new thrust bearing design with a shallow squared groove leading from the leading edge to the center of the bearing surfaces of the topmost foil. **Figure 3** shows the calculated pressure distributions on the surfaces of the thrust bearings both with and without the groove. The grooved bearing allows fluid to enter the bearing surface over a wider area and increases the amount of fluid by 70% relative to that without the groove. Because the groove does not link the circumferential edges or interrupt the center of the bearing surface, there is basically no radial leakage of fluid in the groove and the maximum pressure is generated at the center of the bearing. Numerical analyses with the grooved foil bearing indicated a 1.5-times increase in the maximum pressure and a 2.5-times increase in the load capacity, relative to the conventional bearing. **Figure 4** shows a trial version of the improved bearing together with a conventional version, both of which were installed in a turbocharger and then evaluated experimentally. A turbocharger with the improved bearing has been run at a rotational speed up to 200,000 rpm.

(Report recieved on Jul. 3, 2006)

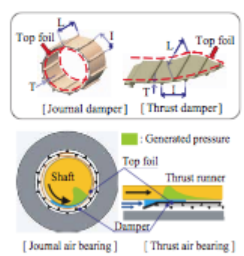


Fig. 1 Compliant foil bearings.

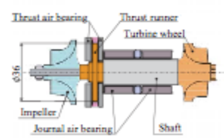


Fig. 2 Construction of air bearings for turbocharger.

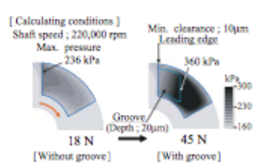


Fig. 3 Comparison of load capacity (calculated).

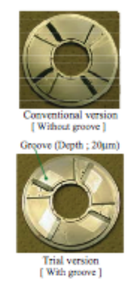


Fig. 4 Test peaces of compliant foil thrust bearing.

This news is reported on R&D Review of Toyota CRDL Vol.41 No.3.

Oil-Free Turbomachinery Program

- Turbochargers are emerging
- High volume application will improve manufacturing
- Competition will drive technology forward.
- New bearings designed for high volume and low cost.
- Significant development for Oil-Free turbomachinery.

US007108488B2

(12) **United States Patent**
Larue et al.

(10) **Patent No.: US 7,108,488 B2**
(45) **Date of Patent: Sep. 19, 2006**

(54) **TURBOCHARGER WITH HYDRODYNAMIC FOIL BEARINGS**

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(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

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(51) **Int. Cl.**
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F02B 35/00 (2006.01)
B61F 17/00 (2006.01)

(52) **U.S. Cl.** **417/407**; 60/605.1; 384/103; 384/105; 384/106

(58) **Field of Classification Search** 417/407; 384/103–106, 535, 119, 160; 123/572, 559.2; 60/684, 605.1
See application file for complete search history.

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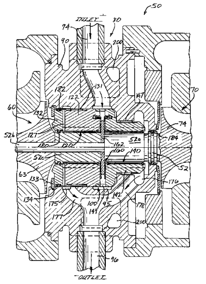
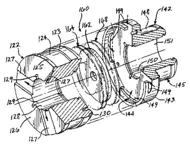
(57) **ABSTRACT**

A turbocharger includes a foil bearing assembly mounted in a center housing between a compressor and a turbine of the turbocharger. The bearing assembly forms a unit installable into the center housing from one end thereof, and the center housing is a one-piece construction. The bearing assembly includes a foil thrust bearing assembly disposed between two foil journal bearings. The journals foils are mounted in annular bearing carriers fixedly mounted in the center housing. A radially inner portion of a thrust disk of the thrust bearing assembly is captured between a shaft and a shaft sleeve of the turbocharger. The center housing defines cooling air passages for supplying cooling air to the foil bearings, and optionally includes a water jacket for circulating engine coolant through the center housing.

10 Claims, 4 Drawing Sheets





Oil-Free Turbomachinery Program

- Capstone C30 turbine generator integrated with full size minivan
- Plug-in hybrid approach (batteries, controls, regenerative braking) yields impressive performance
- First Oil-Free car, a sign of the future, never needs service.

Photo Release -- Capstone C30 Successfully Integrated Into Ford Vehicle by Langford Performance Engineering Ltd. 6/11/09 2:15 PM

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Capstone

Source: Capstone Turbine Corporation

Photo Release -- Capstone C30 Successfully Integrated Into Ford Vehicle by Langford Performance Engineering Ltd.

CHATSWORTH, Calif., June 11, 2009 (GLOBE NEWSWIRE) -- Capstone Turbine Corporation (www.capstoneturbine.com) (Nasdaq:CPST), the world's leading clean technology manufacturer of microturbine energy systems, today announced that its C30 liquid fueled microturbine has been successfully integrated into a Ford S-Max people carrier in the United Kingdom.

A photo accompanying this release is available at http://www.globenewswire.com/newsroom/prs/?pkgid=6263

To see a promotional video of the "Whisper" please click on the following link: http://www.capstoneturbine.com/whisper_promo.wmv

Langford Performance Engineering (www.lpengines.com), headquartered in Wellingborough England, designed and modified the Ford S-Max seven seat crossover vehicle into a series hybrid plug in vehicle with a C30 under the hood as an electric range extender. Langford reports that the "Whisper Eco-Logic" car gets up to 80 mpg in early stage demonstration testing.

"The Ford modified by Langford is an extremely practical solution and one that Langford has been working on for over two years," said Jim Crouse, Capstone's Executive Vice President, Sales and Marketing. "The design characteristics of Capstone's turbine permits ultra low emissions, high fuel economy, multi fuel capability, no coolants or lubricating oil, and little to no maintenance in an automotive application," added Crouse.

"Our Whisper Eco-Logic vehicle is a plug in electric car with an on board turbine generator to keep the batteries charged and extend the range of the car beyond that of a typical electric vehicle," said Dick Langford, Langford's Founder and Managing Director. "This sets it apart from the hybrids now available such as the Lexus and Toyota which use conventional 4 stroke engines to provide both vehicle drive and battery charging. In early demonstration testing the car is getting up to 80 miles per gallon and travels 40 miles on electric power before the Capstone turbine generator starts up and charges the lithium ion batteries," added Langford.

"Capstone was founded on the concept of a C30 powering hybrid vehicles so it is extremely gratifying to see the Langford Ford with a C30 under the hood," stated Darren Jamison, Capstone's President and Chief Executive Officer. "Langford did an exceptional job integrating the turbine, power electronics and batteries into the vehicle without impacting any of the seven seats or increasing the overall vehicle weight," added Jamison.

Langford Engineering will be marketing and demonstrating the plug in hybrid vehicle in hopes of further developing this concept with a suitable automotive partner who could commercialize the product for U.S. use and capitalize on a portion of the Obama administration's $2.4 billion outlined in the stimulus fund to get more electric vehicles on U.S. roads.

Images

Capstone Powered Hybrid Electric Vehicle

Other Company Press Releases

Capstone Signs First C1000 Factory Protection Plan Pushing Long Term Service Backlog Over $11 Million - Jun 9, 2009

Capstone Expands Southeast Asia Distribution -- Names Aqua Nishihara Its Distributor in Thailand - Jun 2, 2009

Capstone Turbine to Announce Fourth Quarter & Fiscal Year 2009 Results On June 15, 2009 - Jun 1, 2009

Capstone Completes Underwriters Laboratories (UL) Scheduled Testing of the C200 Product - Jun 1, 2009

Capstone Receives Order for C200 Microturbines for

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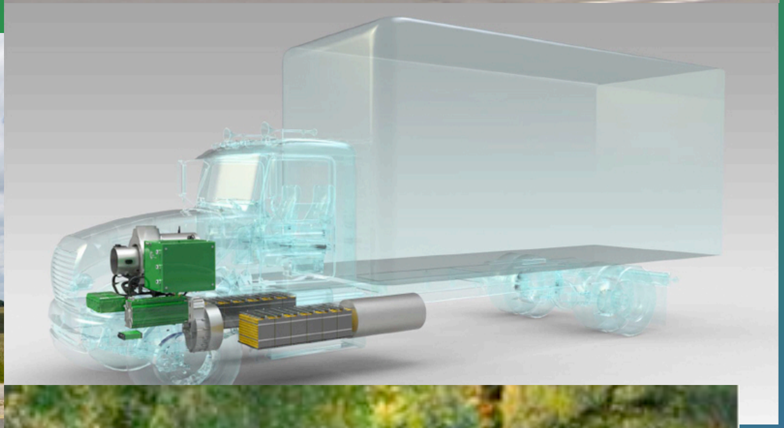
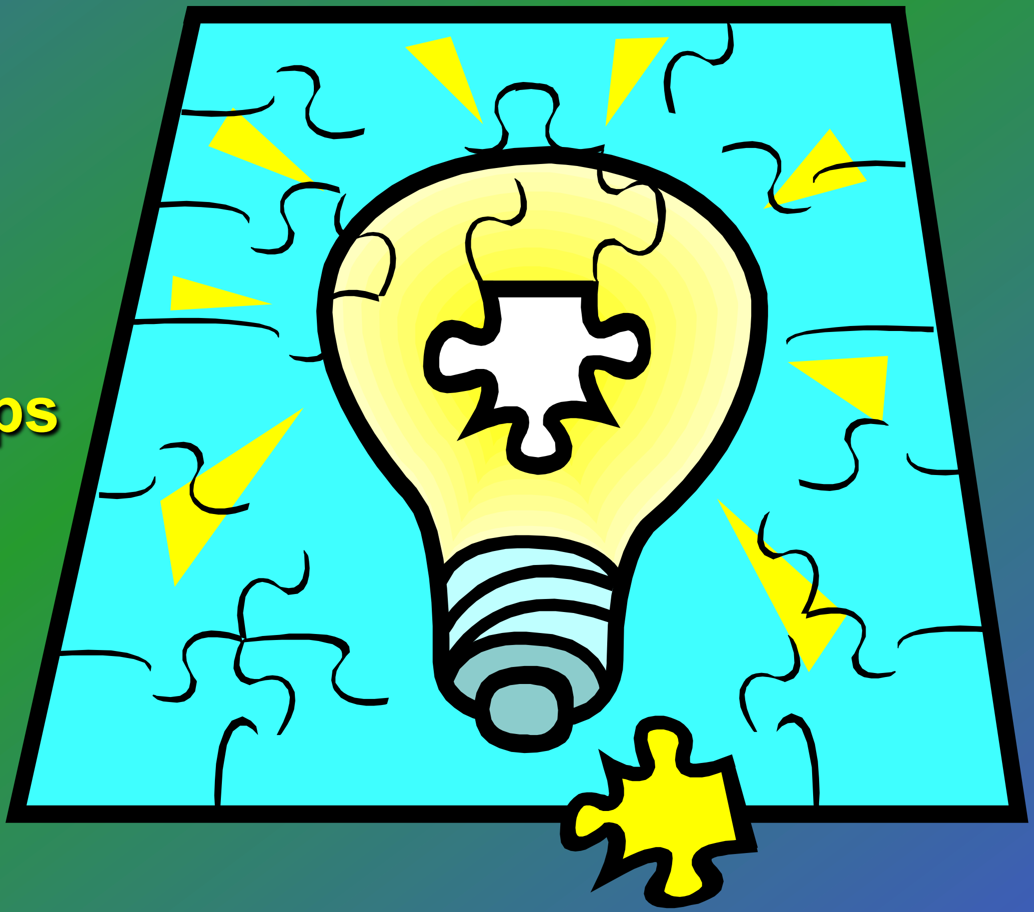
Oil-Free Turbomachinery Program

- Maintenance-Free hybrid electric vehicle demonstrators in all markets.

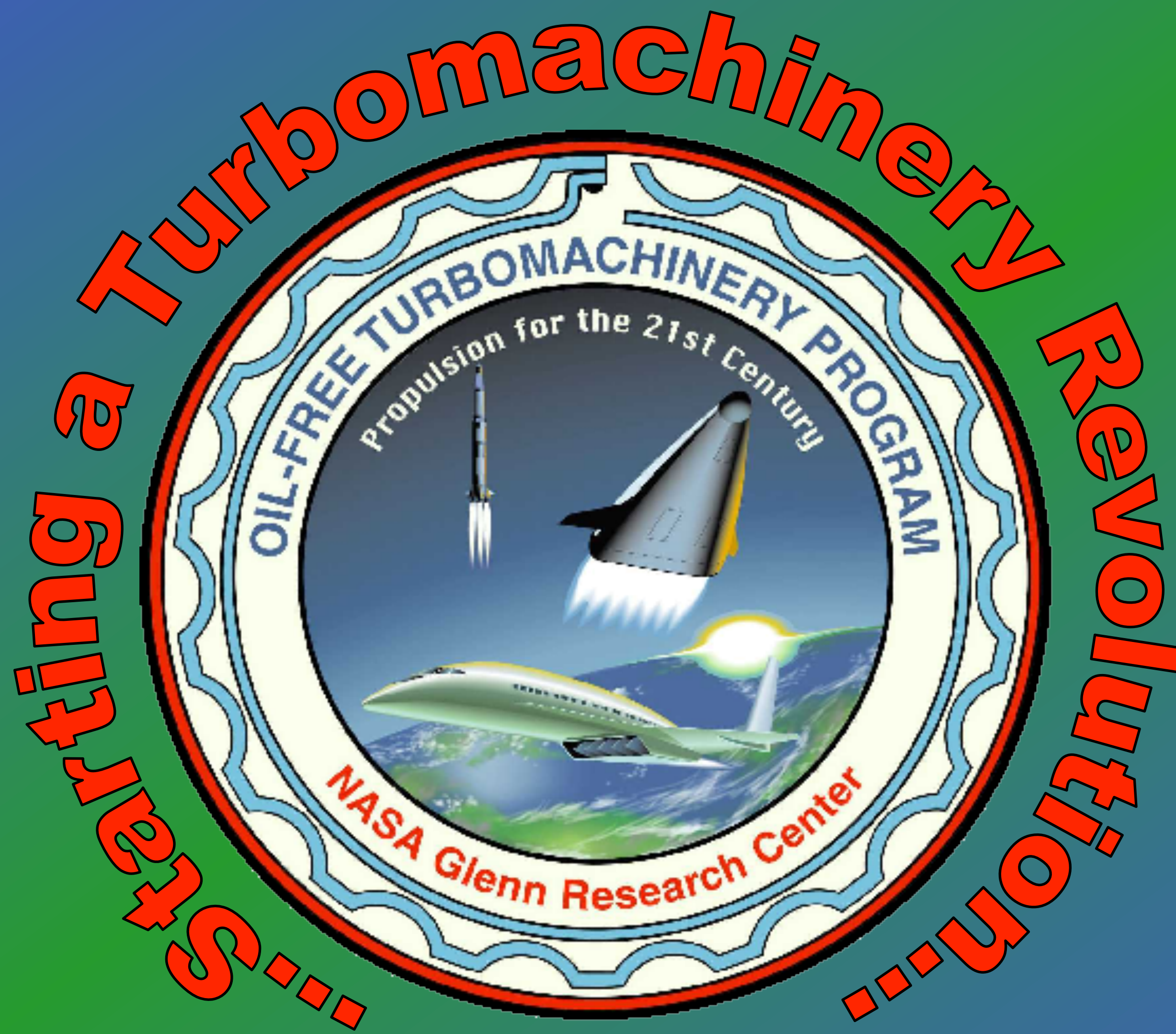


Oil-Free Turbomachinery: A Large and Complex Puzzle

- Foil bearing design
- Solid lubricants
- Materials science
- Rotordynamic models
- Thermal management
- Bearing performance maps
- Bearing modeling
- Bearing design tools
- Bearing test facilities



With many exciting career opportunities!



www.grc.nasa.gov/WWW/Oilfree