TP 14160E

FEASIBILITY STUDY OF USING COMPOSITE MATERIALS FOR TRACTION MOTOR BEARINGS

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Transportation Development Centre
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National Research Council Canada
Institute for Aerospace Research
Structures, Materials and Propulsion Laboratory

August 2003

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by

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Un sommaire français se trouve avant la table des matières.

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16. Abstract

To improve train safety, a feasibility study of replacing babbitt with a composite material in traction motor journal bearings was carried out. The objective of the study was to improve train safety by minimizing the effects of lubricating system failure. It intended to take advantage of recent developments in technology of composite materials.

The project included an evaluation of the selected bearing materials using a standard Falex block-on-ring tester, as well as a full-scale experimental investigation of both babbitt- and composite material-lined traction motor bearings. The experiments were carried out on a test rig designed specifically for this study. Measurements of bearing temperature, friction torque, and bearing/shaft displacement were used to evaluate the bearing performance.

The results from this work were negative: the best performing material during the bench tests did not meet operating conditions of traction motor bearings. It was concluded that the poor heat conductivity of the material led to the bearing failures.

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16. Résumé

Une étude a été réalisée afin de déterminer la faisabilité de remplacer la régule par un matériau composite dans les paliers de moteurs de traction. Cette étude a été menée dans le but d'améliorer la sûreté des trains en minimisant les conséquences d'une défaillance éventuelle du système de lubrification. Elle visait également à tirer parti des progrès récents de la technologie des matériaux composites.

Les travaux ont comporté l'évaluation d'une gamme de matériaux sélectionnés à l'aide d'un tribomètre classique à bloc et à anneau de Falex. Des essais en vraie grandeur ont ensuite été effectués sur deux types de paliers, à garniture en régule et à garniture en matériau composite. Un banc d'essai avait été conçu expressément pour l'étude. Pour évaluer les performances des paliers, les chercheurs ont mesuré la température des paliers, le couple de frottement et le jeu palier/axe.

Ces travaux ont conduit à des résultats négatifs : le matériau jugé le plus performant au cours de l'étude tribométrique ne résistait pas aux conditions d'exploitation de paliers de moteurs de traction. Les défaillances des paliers ont été imputées à la faible conductivité thermique du matériau.

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EXECUTIVE SUMMARY

A high unit-load, relatively poor lubrication, and a harsh environment characterize operating conditions for traction motor bearings. Despite proper maintenance and condition monitoring, their failures are too frequent and usually result from failure of the lubricating system. Improving safe operation of traction motor journal bearings, and thus train safety, was the reason for this project. The hope was that a composite bearing material could not only perform well when lubricated, but also allow for a certain period of safe operation in a case of lack of normal lubricant flow in the bearing gap.

Contemporary composite materials usually consist of thermoplastic matrix and solid lubricant with a fiber providing creep resistance and strength. Evaluation of a selection of composite bearing materials was carried out on a Falex block-on-ring wear tester, in accordance with ASTM Standard G77 (Standard Practice for Ranking Resistance of Materials to Sliding Wear Using Block-on-Ring Wear Test). Out of the six materials tested, Vespel® SP214X, a polyimide resin filled with graphite, and/or PTFE, and/or molybdenum disulfide, was selected for the full-scale experimental investigation. This material is made by DuPont de Nemours.

The experimental investigation of the bearing was carried out on a test rig developed by National Research Council Canada. The rig accommodates a full-size traction motor bearing, has a maximum load capacity of 41,000 lb., and can run at speeds of up to 600 rpm. The instrumentation included 14 thermocouples monitoring the temperature of the tested bearing, eddy current proximity probes, and a force transducer recording the changes to the friction torque. As a reference for the Vespel®-lined bearing, data were collected from a babbitt-lined bearing, which was tested for a full range of operating conditions. These tests also validated the design of the test facility.

Initially, the new traction motor bearings were fabricated using Vespel® in the form of 1.25 in. wide rings. The last tested bearing was made from one cylindrical piece of Vespel®. The process of bearing fabrication included an elaborate procedure for bonding the composite material to the brass substrate.

The tests on the Vespel®-lined bearings were unsuccessful, even in the presence of lubricant. The maximum load achieved was 15,000 lb. at a speed of 106 rpm. All of the tested bearings experienced severe friction-induced vibration and damage to the bearing surface, and eventually seized on the shaft. Changes to the bonding procedure and bearing clearance did not improve the situation. In total, four bearings were tested.

It was concluded that the most likely cause for the bearing failures was that the material's thermal properties were insufficient to transfer the large amount of heat generated in the bearing. The resulting decrease in oil viscosity and thermal growth of the bearing led to the bearing seizure.

SOMMAIRE

De fortes charges, une lubrification minimale et un environnement hostile sont les conditions caractéristiques dans lesquelles sont exploités les paliers de moteurs de traction. Malgré une surveillance et un entretien rigoureux, les paliers sont trop souvent défaillants et ces problèmes sont la plupart du temps causés par une panne du système de lubrification. Le présent projet visait donc à améliorer la sûreté d'exploitation des paliers de moteurs de traction, et, partant, la sûreté des trains. On espérait plus précisément trouver un matériau composite qui pourrait non seulement fonctionner adéquatement lorsque bien lubrifié, mais aussi offrir une certaine marge de sécurité en cas de lubrification insuffisante.

Les nouveaux matériaux composites sont normalement constitués d'une matrice thermoplastique et d'un lubrifiant solide en fibres qui allie robustesse et résistance au fluage. Divers matériaux composites ont été évalués à l'aide d'un tribomètre à bloc et à anneau standard de Falex, conformément à la norme ASTM G77 (*Standard Practice for Ranking Resistance of Materials to Sliding Wear Using Block-on-Ring Wear Test*). Des six matériaux testés, le Vespel® SP214X, une résine polyimide additionnée d'une charge de graphite, de PTFE et/ou de disulfure de molybdène, a été retenu pour les essais en vraie grandeur. Ce matériau est fabriqué par DuPont de Nemours.

L'essai en vraie grandeur a été mené à l'aide d'une installation mise au point par le Conseil national de recherches du Canada. Cette installation pouvait mettre à l'essai un palier en vraie grandeur sous une charge maximale de 41 000 lb., à des vitesses pouvant atteindre 600 tr/min. L'instrumentation comportait 14 thermocouples pour la surveillance de la température du palier, des détecteurs de proximité à courant de Foucault, et un transducteur de force enregistrant les fluctuations du couple de frottement. Un palier à garniture en régule a aussi été mis à l'essai dans une vaste gamme de conditions d'exploitation. Ces essais ont permis de valider l'installation d'essai et d'obtenir des données de référence pour l'évaluation du palier à garniture de Vespel®.

Les premiers paliers de moteur de traction nouveaux étaient fabriqués à partir d'anneaux de Vespel® de 1,25 po de largeur. Le dernier palier essayé était constitué d'un seul cylindre du même matériau. Le procédé de fabrication des paliers comportait une procédure complexe de liaison du matériau composite au support en laiton.

Les essais réalisés sur des paliers à garniture de Vespel® ont échoué, même en présence de lubrifiant. La charge maximale acceptée était de 15 000 lb., à une vitesse de 106 tr/min. Pour tous les paliers mis à l'essai, les chercheurs ont enregistré des vibrations dues au frottement et des dommages à la surface du palier, et, ultimement, le grippage du palier sur l'axe. Les changements apportés à la procédure de liaison et au jeu palier-axe n'ont pas amélioré la situation. Au total, quatre paliers ont été essayés.

Les chercheurs ont évoqué les piètres propriétés thermiques du matériau comme cause la plus probable des défaillances du palier : cette lacune faisait en sorte que la forte chaleur produite dans le palier demeurait emprisonnée. Il en résultait une baisse de viscosité de l'huile lubrifiante et la dilatation thermique du palier, d'où le grippage.

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1. INTRODUCTION

Traction motor support bearings operate under difficult and severe operating conditions, usually with a limited supply of oil and a high unit-load (*p*). At times, the bearings are further handicapped by the need to operate at low axle speeds (*v*). To minimize the number of bearing failures and obtain maximum bearing life, proper maintenance is required. However, statistics [1] show that, between 1994 and 1997, there were 138 traction motor bearing failures on the Canadian National (CN), Canadian Pacific (CP) and VIA railway systems (39 in 1994; 38 in 1995; 39 in 1996 and 22 in 1997). In some cases, the bearing damage was detected at an early stage. However, other occurrences of bearing failure resulted in train derailment and even passenger mortalities. The situation is made worse by the fact that traction motors on locomotives pulling freight trains are not equipped with sensors or detectors. Sensors are fitted to locomotives pulling passenger trains, but train crews may not be acquainted with the detection system and can ignore early warning signals.

This project's aim was to assess the feasibility of replacing the conventional babbitt (lead alloy)-lined traction motor bearings with composite plastic-lined bearings as a means of increasing train safety in cases of lubrication system failure. In certain applications, composite plastics offer superior resistance to friction, heat and wear, and could help to minimize bearing failures.

The work included a full scale experimental investigation of both babbitt-and composite material-lined traction motor bearings for locomotives, at loads and speeds representing the severe operating conditions inherent to rail. These full-scale tests were carried out on a rig developed by National Research Council (NRC) specifically for this project.

The project was performed at NRC. It was initiated in the Centre for Surface Transportation Technology and completed after the Tribology Unit was transferred to the Institute for Aerospace Research.

2. LITERATURE AND PRACTICE REVIEW

2.1 Traction motor bearings

Over the past 30 years, journal bearings in traction motors have been gradually replaced by roller bearings [2][3]. However, a significant number of traction motors are still equipped with journal bearings [4]. Operating conditions of these bearings are severe, and failures of both roller and journal traction motor bearings are frequent causes of derailment [5][6]. These severe conditions are the result of [7]:

- Load concentration over a small area of the bearing, as a result of axle deflection under locomotive weight and axle misalignment because of gear reaction of the overhung pinion
- Low speed at start-up, with poor lubrication
- Extremes in ambient temperatures
- A load vector that is sometimes close to the bearing window or bearing split line

Moreover, shock loads from the interaction between wheel and rail can cause the bearing clearance to increase. This may lead to unfavourable conditions of gear engagement, and consequently to additional bearing misalignment and load [8].

It often occurs that water accumulates in the oil reservoir of the traction motor, especially in winter. Lane and Dayson [9], and later Dayson [10], investigated ingress of water to the lubricant reservoir and its effect on traction motor bearing performance. They concluded that the water supplied to the bearing clearance does not compromise bearing operation. However, a large amount of water in the reservoir can cause an excessive oil flow through the bearing clearance. These authors also showed that the wick transports oil not only from the reservoir to the bearing gap, but also in the opposite direction.

Because of difficult operating conditions, frequent inspections of the journal and bearing surfaces and the felt lubricating wick are essential. In particular, rounding of the felt wick can compromise bearing lubrication. The presence of dirt or bearing material particles deposited on the wick indicates that the bearing should be thoroughly examined.

Avery [11] analyzed the operation of traction motor bearings and emphasized the importance of bearing surface finish, because of a very thin oil-film gap between the journal and bearing surfaces.

2.2 Application of composite materials to traction motor bearings

Dry or self-lubricated bearings are used when one or more of the following situations occur [12][13]:

- A lubricating film cannot be generated or has to be supplemented by a low-friction material or a solid lubricant (low speed or oscillatory motion)
- Extreme operating conditions for the lubricant (contamination, too viscous at low temperatures, or decomposition at high temperatures)
- Lubricant may contaminate the product
- Difficult to maintain

The first two situations are common to those of the traction motor bearing application. Low friction can be achieved by using a solid lubricant such as graphite or molybdenum disulphide, or by using the low-friction properties of plastic [14]. However, poor thermal conductivity, a high coefficient of thermal expansion and a lack of strength limit straight plastics.

Contemporary composite bearing materials consist of a thermoplastic matrix and solid lubricants with a fiber added for creep resistance and strength. Recent progress in plastic materials has been reflected in a significant increase in both load and temperature capabilities [15]. However, only a few of these new materials stand a chance of being used as a replacement for the conventional babbitt-lined traction motor bearing.

E.I. Du Pont de Nemours and Company have recently been issued a patent on composite journal and thrust bearing systems [16].

3. BENCH TESTS – SELECTION OF BEARING MATERIAL

Traction motor bearings operate at extremely high *pv* values and only a few of the available composite materials can withstand such operating conditions. They include:

- Vespel® – polyimide resin filled with graphite and/or PTFE and/or molybdenum disulphide (by DuPont de Nemours)
- DU® – steel backing, a layer of sintered bronze impregnated with PTFE-lead mixture, and overlay consisted of the same mixture (by Garlock Bearings Inc./Coltec Industries)
- Meldin® 2021 – polyimide resin (by Furon/Dixon Industries Corp.)
- Torlon – polyimide resin (by Johnston Industrial Plastics, Amoco, and others)
- FM3000® – carbon reinforced polyimide resine (by ICI Fiberite)

3.1 Determining test conditions

For rubbing surfaces, it is common practice to use the *pv* factor as an indicator of the operating conditions in the bearing. *p* is the unit load in the contact area, and is determined by *W/A*, where:

$$p = \frac{W}{A}$$

W load
A contact area

and *v* describes the relative velocity of the two surfaces.

Evaluation of the composite bearing materials was carried out on a Falex block-on-ring wear tester (Figure 1), in accordance with ASTM Standard G77 (Standard Practice for Ranking Resistance of Materials to Sliding Wear Using Block-on-Ring Wear Test).

A block was mounted in the tester's holder. This holder automatically aligns the block to the ring and ensures a uniform load over the contact area. A known weight was applied to the hanger, which presses the block against the rotating ring. The friction force between the block and ring was measured using a load cell.

For the purpose of these tests, the ring was made of steel while the block was made of the materials to be evaluated.

Performance of the test samples depends strongly on the area of contact. If the block is rectangular then the initial contact is linear, provided that elastic or plastic deformations do not occur. Running-in creates a rectangular contact area that increases as the wear of the block progresses. In such a case, the value of *p* decreases, and the real *pv* varies as the test progresses (Figure 2).

To provide a constant contact area during the test, the block can be machined to conform to the ring surface. In this case, the *pv* factor was maintained at a constant value throughout the duration of the test.

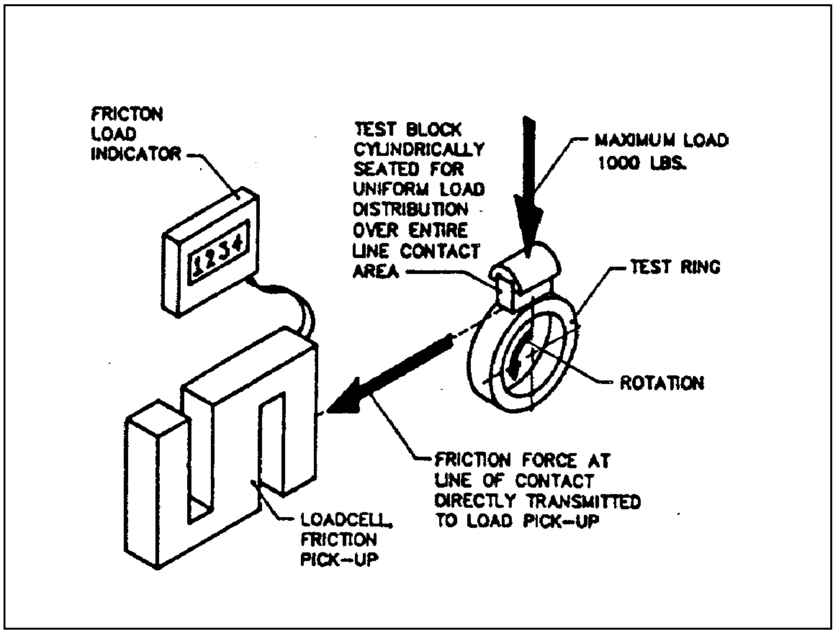


Figure 1 Schematic of the Falex block-on-ring wear tester

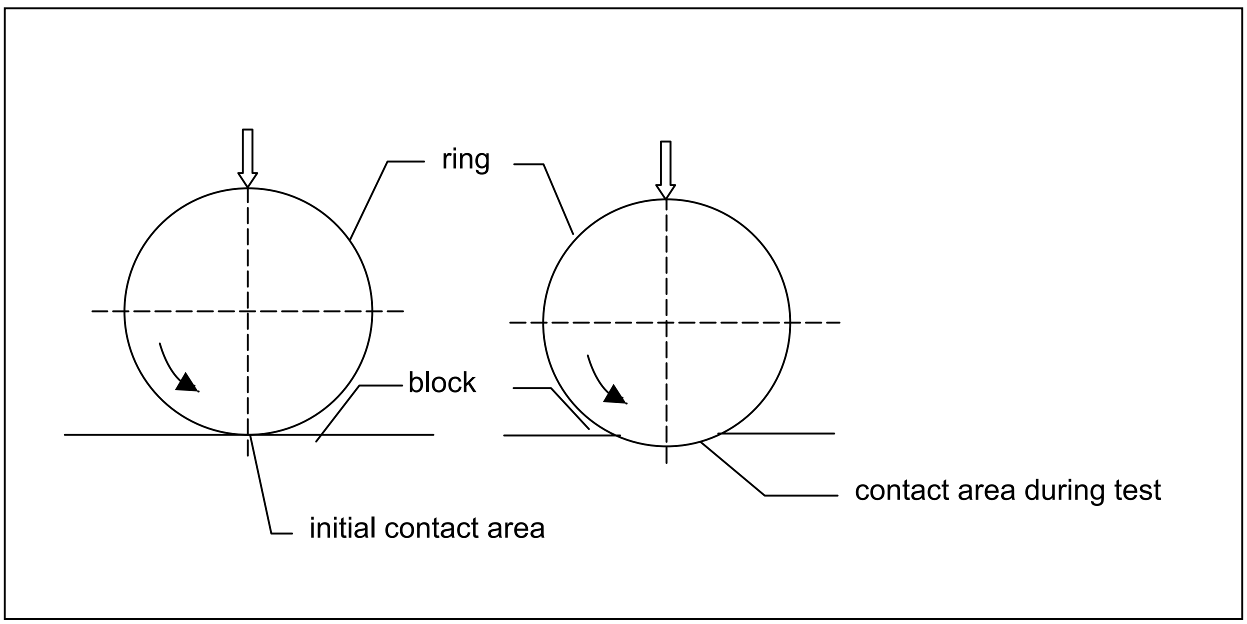


Figure 2 Contact area with block-on-ring tests

Typically, the *pv* factor for journal bearing uses a unit load based on the bearing projected area (Figure 3a):

$$p = \frac{W}{Ld}$$

where

W bearing load
L bearing width
d bearing diameter

However, the real contact area is much smaller because of the clearance in the bearing. Initially, the contact area corresponds to the elastic and plastic deformations of the block. As the test progresses, it may increase because of wear of the block, but it never reaches the value of the projected area, *L*d* (Figure 3b).

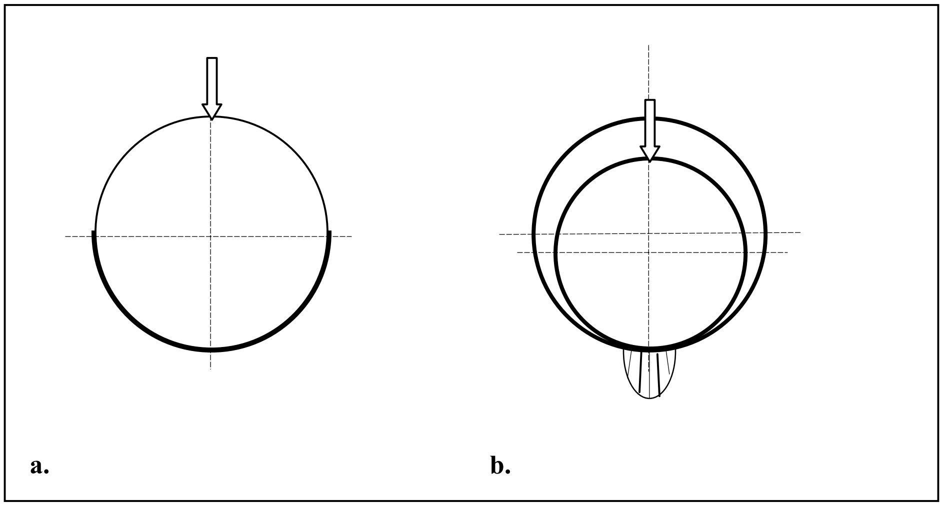


Figure 3 Contact area in journal bearing
a. assumed contact area **b.** actual contact area

The block and ring used in the Falex machine for this study are shown in Figure 4. Table 1 shows the selected test parameters and the corresponding traction motor bearing operating conditions. The contact area was assumed to be the cross-sectional area of the block.

Table 1 Test parameters for the Falex wear tester

TRAIN SPEED	PV (total)	SHAFT SPEED	BEARING LOAD	RING SPEED	LOAD ON SAMPLE
km/h (mph)	MPa*m/s (psi*ft/min)	m/s (ft/min)	kN (lb.)	rpm	kN (lb.)
24.1 (15)	5.25 (149,935)	1.34 (264)	182(40,900)	732	0.392 (88)
56.3 (35)	7.64 (218,120)	3.13 (616)	113.5 (25,500)	1,707	0.244 (55)
104.6 (65)	8.96 (255,756)	5.81 (1144)	71.6 (16,100)	3,171	0.154 (35)

Some manufacturers were not willing to provide NRC with material samples. For this reason, the bench tests were limited to the materials listed in Table 2.

Table 2 Composite materials tested

MATERIAL	MANUFACTURER	MAX. pv (dry)
Vespel® SP21 Graphite-filled polyimide	Du Pont de Nemours	200,000-300,000
Vespel® SP214X Graphite-filled polyimide	Du Pont de Nemours	600,000-700,000
FM3000® Cellulose and PTFE-filled polymer	Cytec Fiberite, Inc.	100,000
DU® Steel-backed sintered bronze filled with a mixture of PTFE and lead	Garlock Bearings, Inc (Glacier)	100,000
DP4® Steel-backed sintered bronze impregnated with polymer (acetal resin)	Garlock Bearings, Inc (Glacier)	80,000

The test procedure included the following steps:

1. Install the thermocouple in the block.
2. Clean the block and ring.
3. Weigh the block.
4. Mount the block in the tester.
5. Rotate the ring.
6. Apply the required load.
7. Run the test for 33 min. and collect measurements of block temperature and friction force.
8. Stop the test.
9. Clean the block and ring.
10. Inspect the block and ring
11. Weigh the block.
12. Measure the width of the scar.

To increase the reliability of the measurements, tests on each material were repeated once.

3.2 Test results

A summary of the test results is presented in Table 3. Three of the test materials did not perform satisfactorily at all test conditions. Examples of data collected from the failed tests on the DP4 and Vespel® SP21 materials are shown in Figure 5. Rapid growth of both the block's temperature and friction force terminated the test.

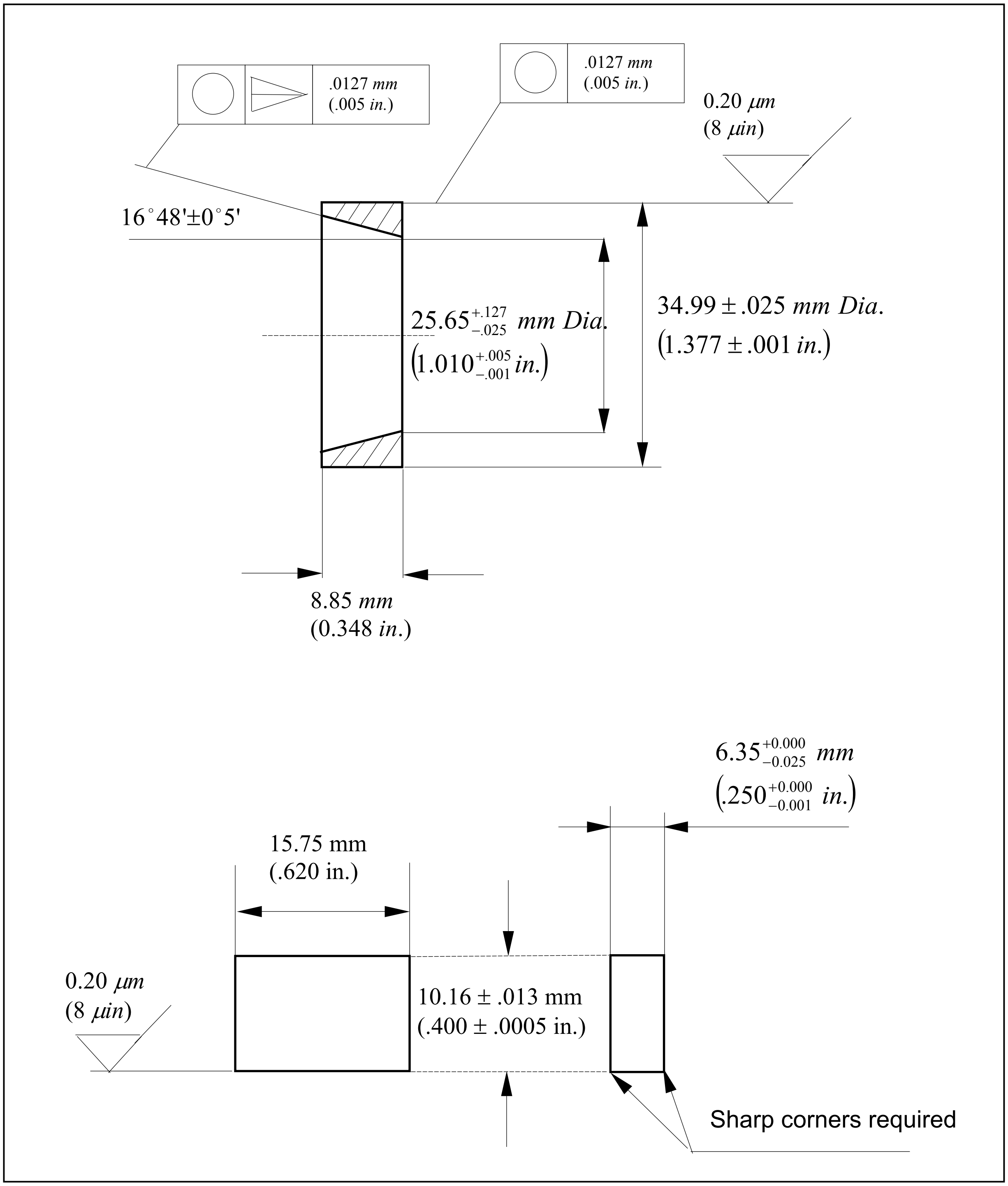


Figure 4 The ring and the block for the Falex wear tester

Vespel® SP214X and FM3000 withstood all the test conditions. Figure 6 illustrates the performance of these two materials at the highest *pv* factor. A comparison of the results from these two materials showed that the Vespel® SP214X was superior, in terms of exhibiting significantly lower block temperature, wear and friction coefficient (see Table 2 and Figure 6).

Vespel® 214X was selected as the candidate material for the full-size bearing tests.

Table 3 Block-on-ring bench tests: summary of results

	LOAD	SPEED	WEAR		FR.COEFF.	TEMP.
			WEIGHT	VOLUME		
	kN (lb.)	rpm	mg	mm^3		°C
Vespel® SP21	0.392 (88)	732	7.0	3.4	0.10	178
	0.244 (55)	1707	11.7	4.7	0.06	176
	0.154 (35)	3170				
Vespel® SP214X	0.392 (88)	732	3.5	2.4	0.04	109
	0.244 (55)	1707	7.0	3.5	0.01	149
	0.154 (35)	3170	14.2	7.9	0.01	134
FM3000	0.392 (88)	732	28.3	19.8	0.10	153
	0.244 (55)	1707	59.2	21.9	0.09	248
	0.154 (35)	3170	40.5	10.2	0.06	203
DU (dark)	0.392 (88)	732	12.0	3.0	0.07	130
DP4 (red)	0.392 (88)	732				

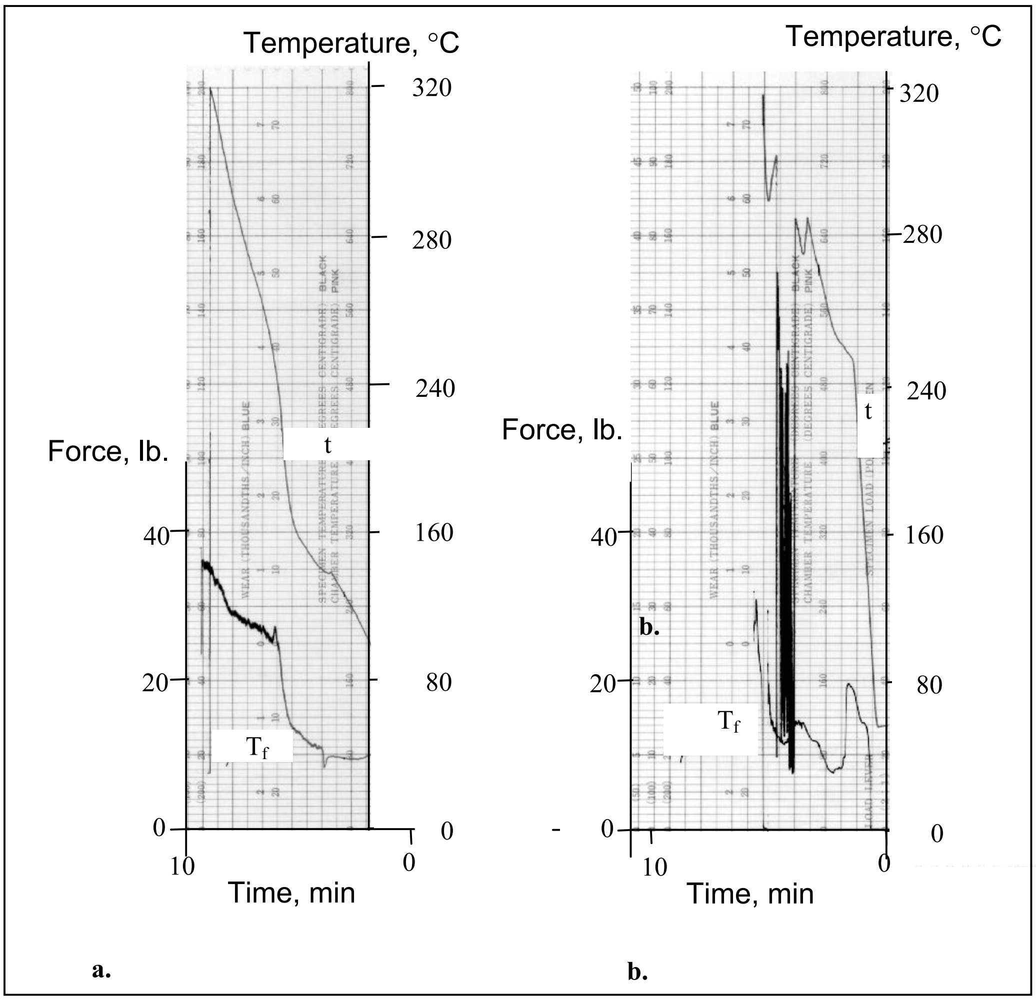


Figure 5 Data from the failed tests: T_f - friction force, t - block temperature
a. DP4, 0.392 kN (88 lb.), 732 rpm
b. Vespel® SP21, 0.154 kN (35 lb.), 3170 rpm

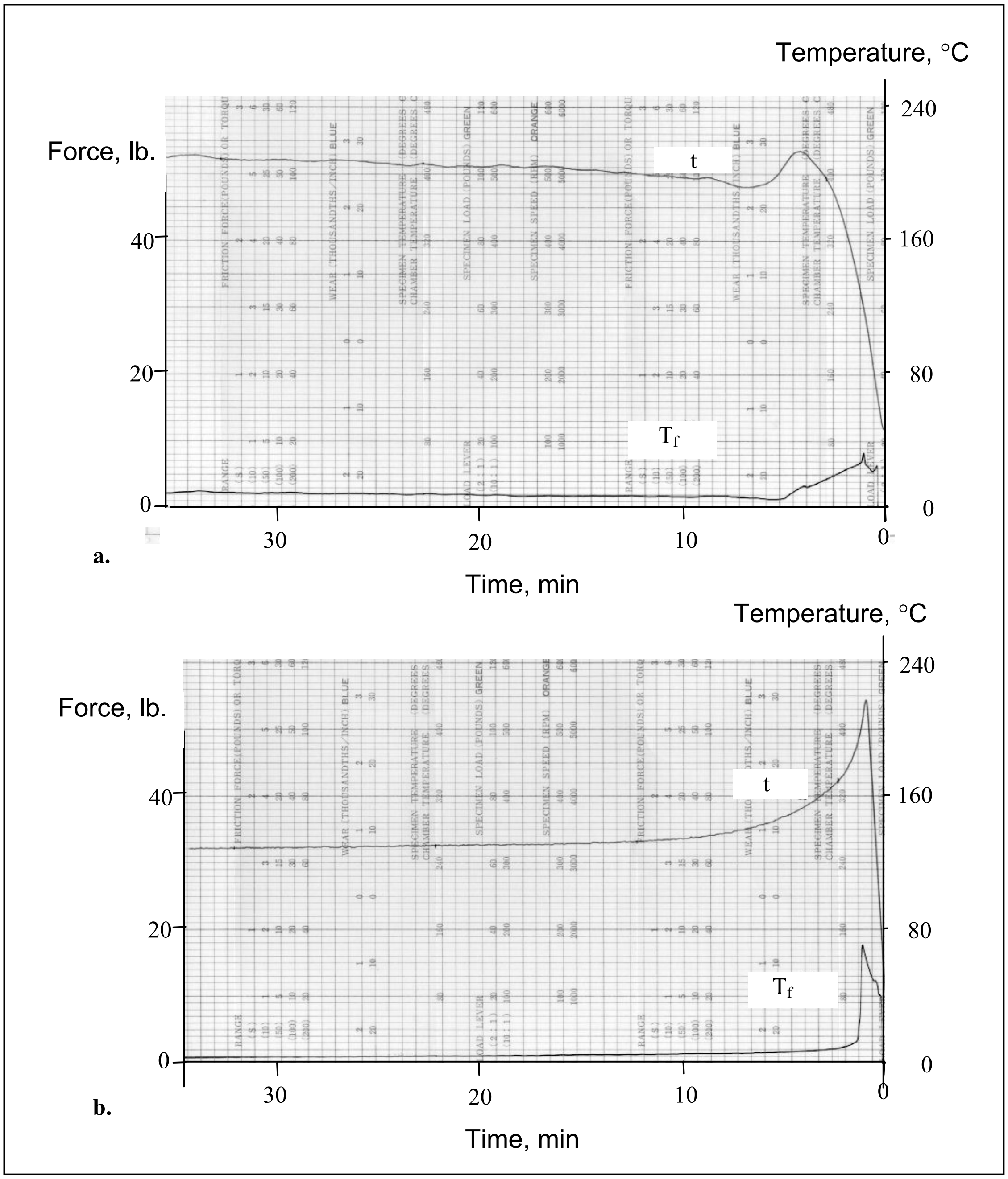


Figure 6 **Data from the tests at the highest *pv* factor (35 lb., 3170 rpm): T_f - friction force, t - block temperature**
a. FM3000 **b.** Vespel® SP214X

4 NRC BEARING TEST FACILITY

4.1 Test rig

A new experimental facility has been developed at NRC for the testing of full-size traction motor bearings under the severe operating conditions inherent in rail. The test rig is based on a design originally used to study the ingress of water into the traction motor bearing [9]. A schematic of this rig is shown in Figure 7. The rig accommodates a full-size traction motor bearing and has a maximum radial load capability of 0.185 MN (41,000 lb.). Hydraulic cylinders that are connected to the bearing housing through double flexure pivots generate this load. The flexures permit measurement of the bearing friction torque. However, the rig design does not allow for an accurate measurement of the friction torque absolute value. Instead, in this report the measured friction losses are related to those at a certain operating conditions. A 110 kW (150 hp) variable speed electric motor, driving through a speed reducing belt-pulley system, provides shaft speeds up to 10 Hz (600 rpm).

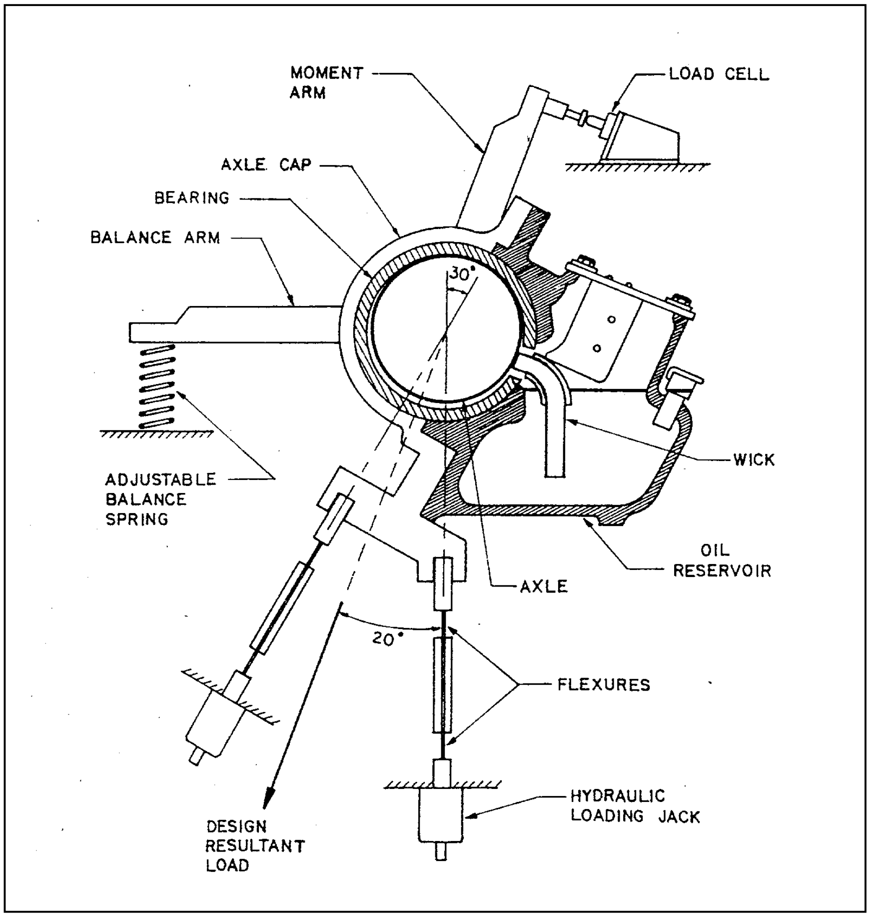


Figure 7 Schematic of the traction motor bearing test facility

4.2 Traction motor bearing

The traction motor suspension bearing comprises a split brass sleeve, approximately 200 mm (8 in.) in diameter, 310 mm (12.25 in.) long and with a 2.5 mm (0.1 in.) overlay of babbitt (lead alloy). The bearing, shown in Figure 8, is lubricated by a felt wick assembly that is held against the journal surface by spring pressure. The opposite end of the wick is immersed up to 100 mm (4 in.) in a 4.5 litre (1 gal.) capacity oil reservoir. Capillary action inside the wick draws the oil up from the oil reservoir to the journal. The two brass half shells are clamped in a housing, which is made up from the motor frame and the end cap/oil reservoir. The cover plate of the reservoir supports the springloaded wick holder. This presses the top end of the wick against the journal through a rectangular window in the wall of the bearing.

The test bearing was lubricated with an oil supplied by VIA Rail. It had viscosity approximately 7cSt at a temperature of 100°C and 43 cST at 40°C (viscosity index of 105).

Figure 8 Babbitt-lined traction motor bearing (with Vespel® ring in the foreground)

4.3 Instrumentation

To monitor bearing temperatures, 12 thermocouples were mounted in the back of the bearing, close to the bearing surface (about 0.5 mm). A further two thermocouples were mounted in the brass backing. The location of these thermocouples is shown in Figure 9. Other thermocouples measured wick and oil reservoir temperatures.

Other instrumentation included a load cell to measure bearing friction torque, and a pair of proximity probes mounted at each end of the bearing housing to measure the position of the bearing with respect to the shaft. Two probes were mounted in-line with the load vector, and two orthogonal to the load vector. Figure 10 shows a view of the test rig.

5. EXPERIMENTAL STUDY OF THE FULL-SIZE BABBITT-LINED BEARING

The purpose of the tests on the babbitt-lined traction motor bearing was to provide baseline data for the investigation of the plastic-lined bearing. The effect of bearing operating conditions such as speed, load and reservoir oil level on the following was investigated:

- bearing operating temperatures
- power loss
- relative displacement between the shaft and bearing (which determines the minimum oil-film thickness)

Three reservoir oil levels were chosen based on the GM's Maintenance Instruction [17]. In this report the recommended maximum level and minimum levels are referred to as normal and low levels, respectively. The medium level is located between these two.

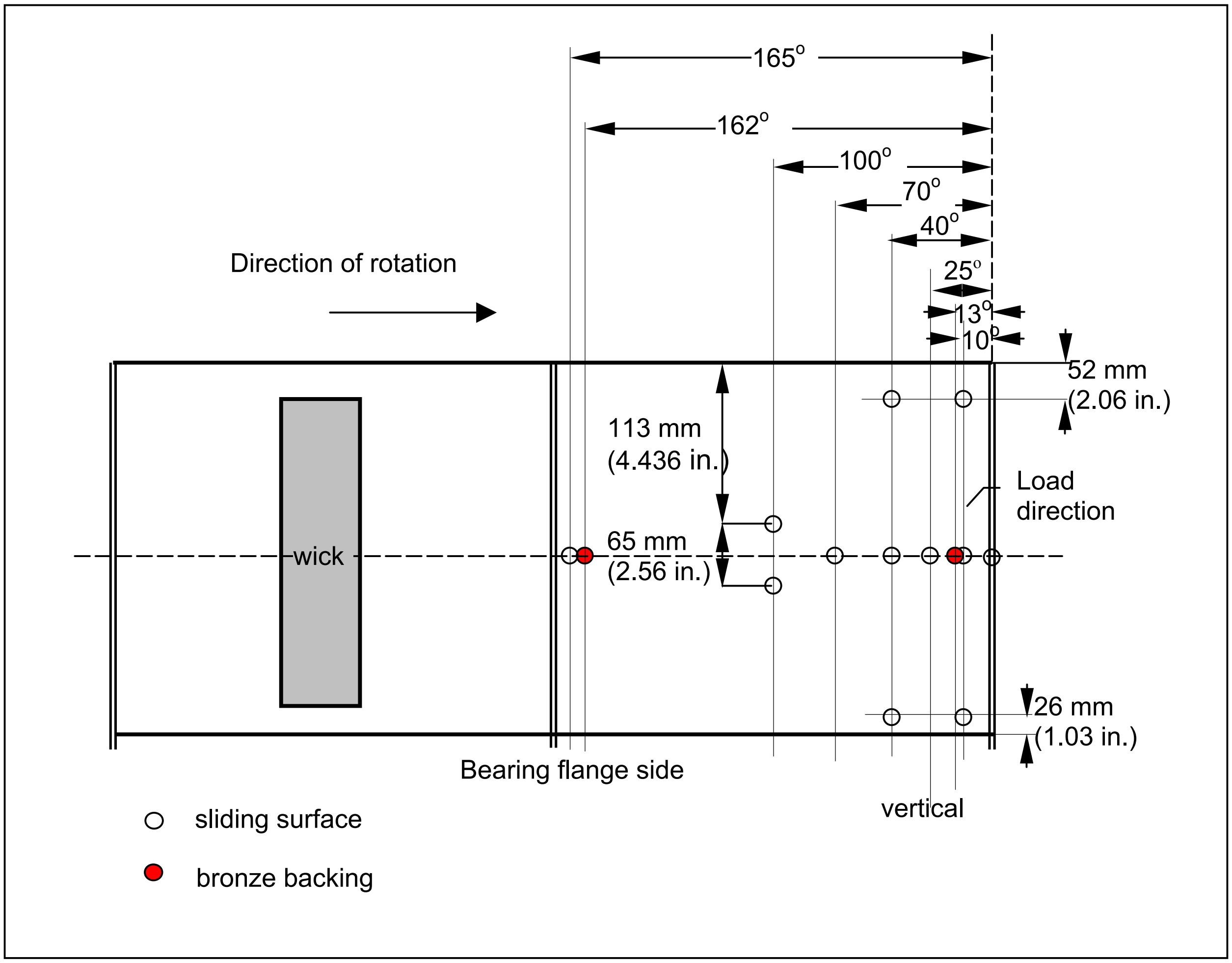


Figure 9 Thermocouple location in the test bearing

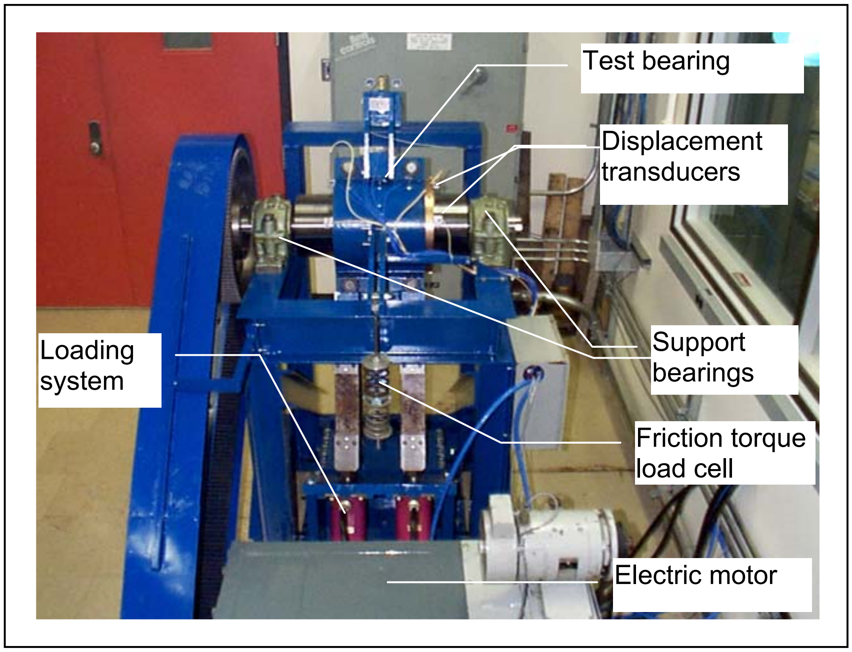


Figure 10 View of NRC's traction motor bearing test facility

5.1 Bearing temperatures

Prior to collecting data from the bearing, the oil and bearing temperatures were stabilized. This usually took between 5 and 7 hours, depending on shaft speed and load. Generally, the lower the shaft speed the longer the stabilization time. Oil level in the reservoir had a negligible effect on the stabilization time, as shown in Figure 11. Here, reservoir oil temperature is plotted against time for a speed of 538 rpm and a load of 16,100 lb.

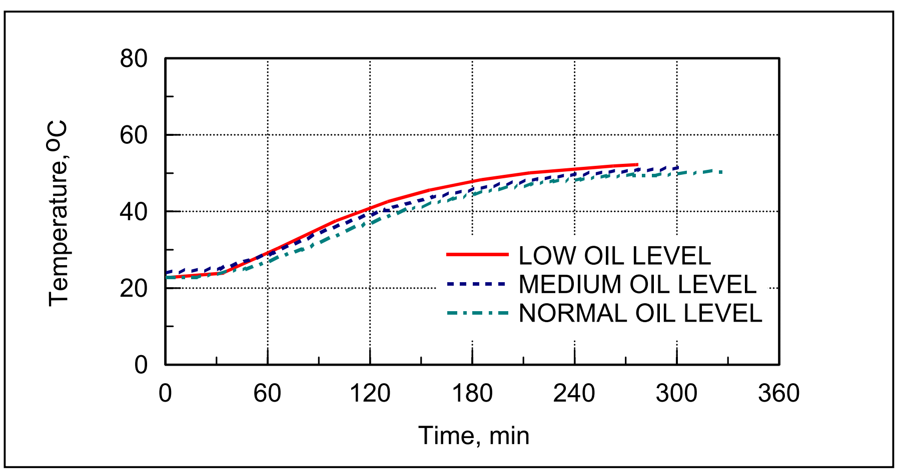


Figure 11 Stabilization of the reservoir oil temperature. Bearing load 16,100 lb., shaft speed 538 rpm

Reservoir oil temperature was mainly dependent on shaft speed, and varied from 37°C at 106 rpm to 54°C at 538 rpm. This is illustrated in Figure 12.

Consequently, reservoir oil level had little effect on bearing operating temperatures. However, bearing temperatures were more sensitive to shaft speed. A maximum bearing surface temperature of 111°C was recorded at 538 rpm and 16,100 lb., with the low reservoir oil. Figure 13 shows the recorded maximum bearing temperature for different reservoir oil levels and operating conditions, while Figure 14 compares the bearing temperature profiles for different loads and speeds with the normal oil level. At the highest shaft speed of 538 rpm, and the bearing load of 16,100 lb., the maximum bearing temperature exceeded 100°C.

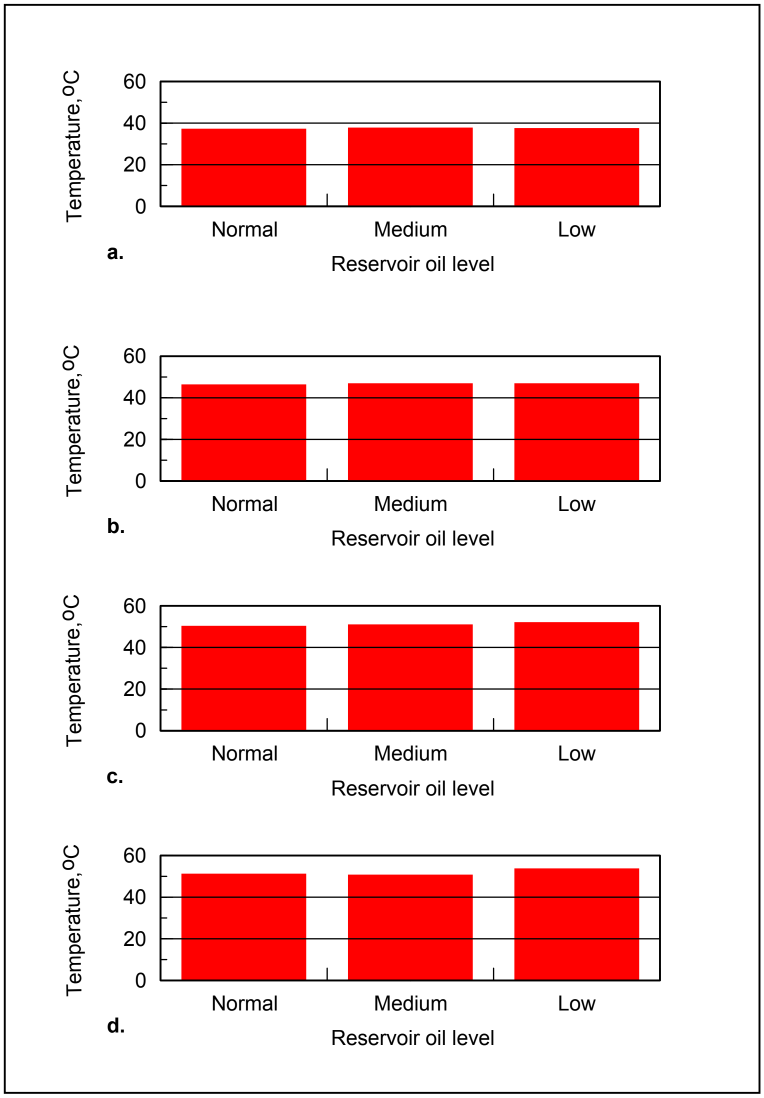


Figure 12 **Effect of bearing operating conditions and reservoir oil level on oil reservoir temperature**
a. 106 rpm/40,900 lb. **b.** 295 rpm/25,500 lb.
c. 458 rpm/18,100 lb. **d.** 538 rpm/16,100 lb.

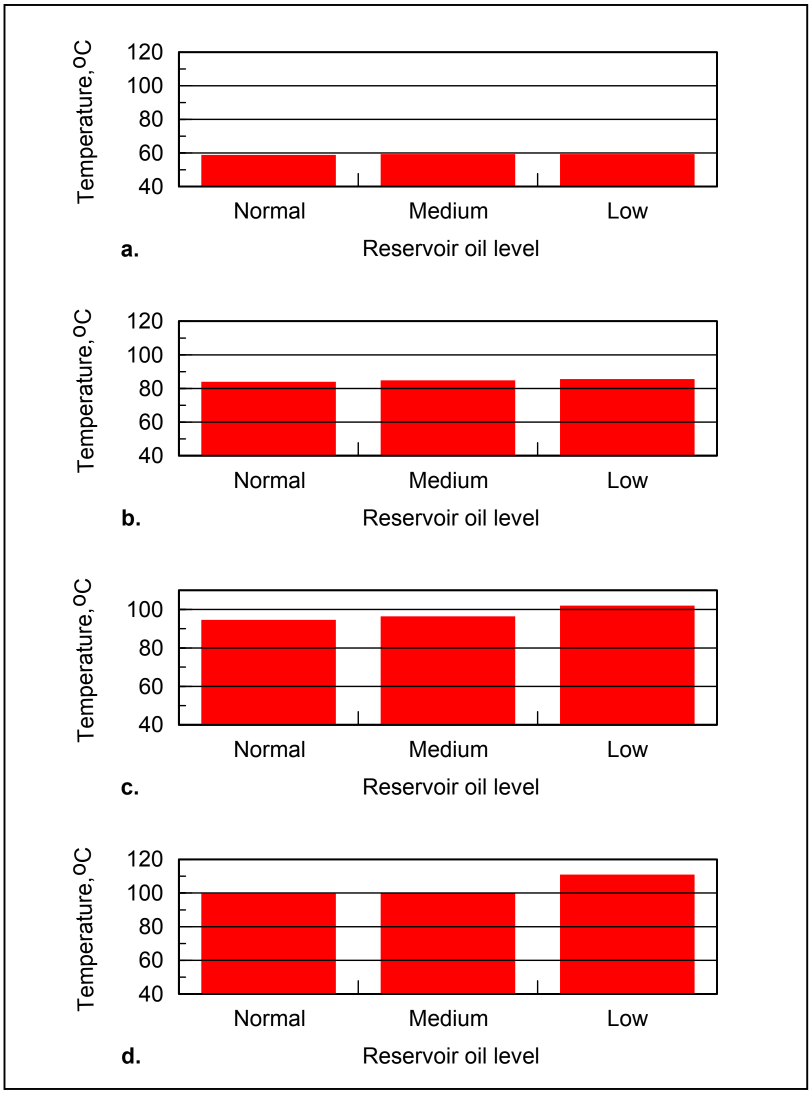


Figure 13 Effect of bearing operating conditions and reservoir oil level on maximum bearing temperature
a. 106 rpm/40,900 lb. **b.** 295 rpm/25,500 lb.
c. 458 rpm/18,100 lb. **d.** 538 rpm/16,100 lb.

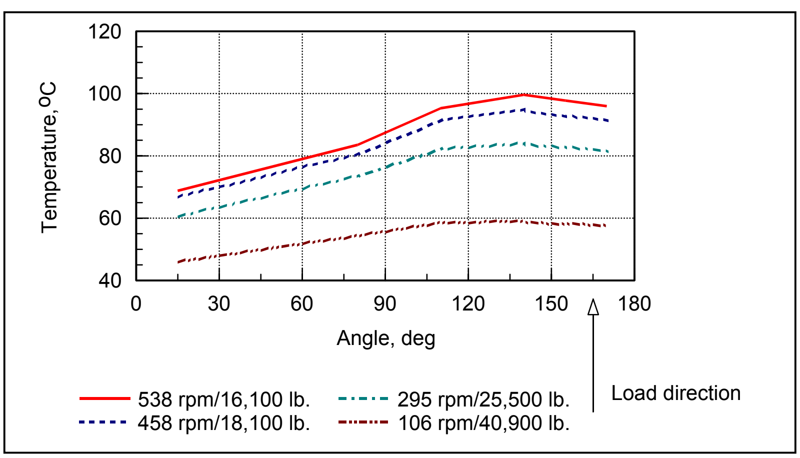


Figure 14 Bearing temperature profiles. Normal reservoir oil level

5.2 Power loss

Figure 15 presents a comparison of the experimentally obtained power loss data. Each measured power loss is related to the highest power loss, measured at 538 rpm and 16,100 lb. with the low oil level in the reservoir. The data show strong dependence of power loss on the operating conditions, and a rather weak dependence on reservoir oil level.

5.3 Shaft – bearing displacements

A relatively weak dependence of reservoir oil level on the displacement measurements was also observed. Displacement measurements for both the normal and low oil levels are illustrated in Figure 16. It can be concluded that, within the range of the tests conducted in this study, oil level has little or no effect on the thickness of the oil film in the bearing.

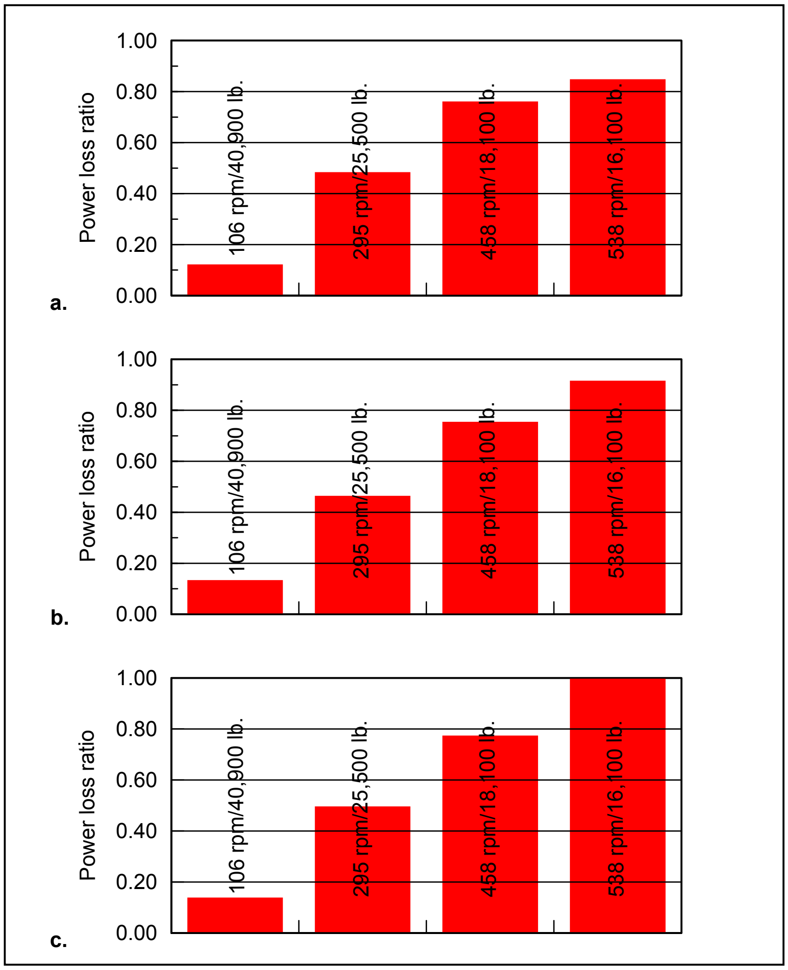


Figure 15 Comparison of bearing power loss for different reservoir oil levels
a. normal level **b.** medium level **c.** low level

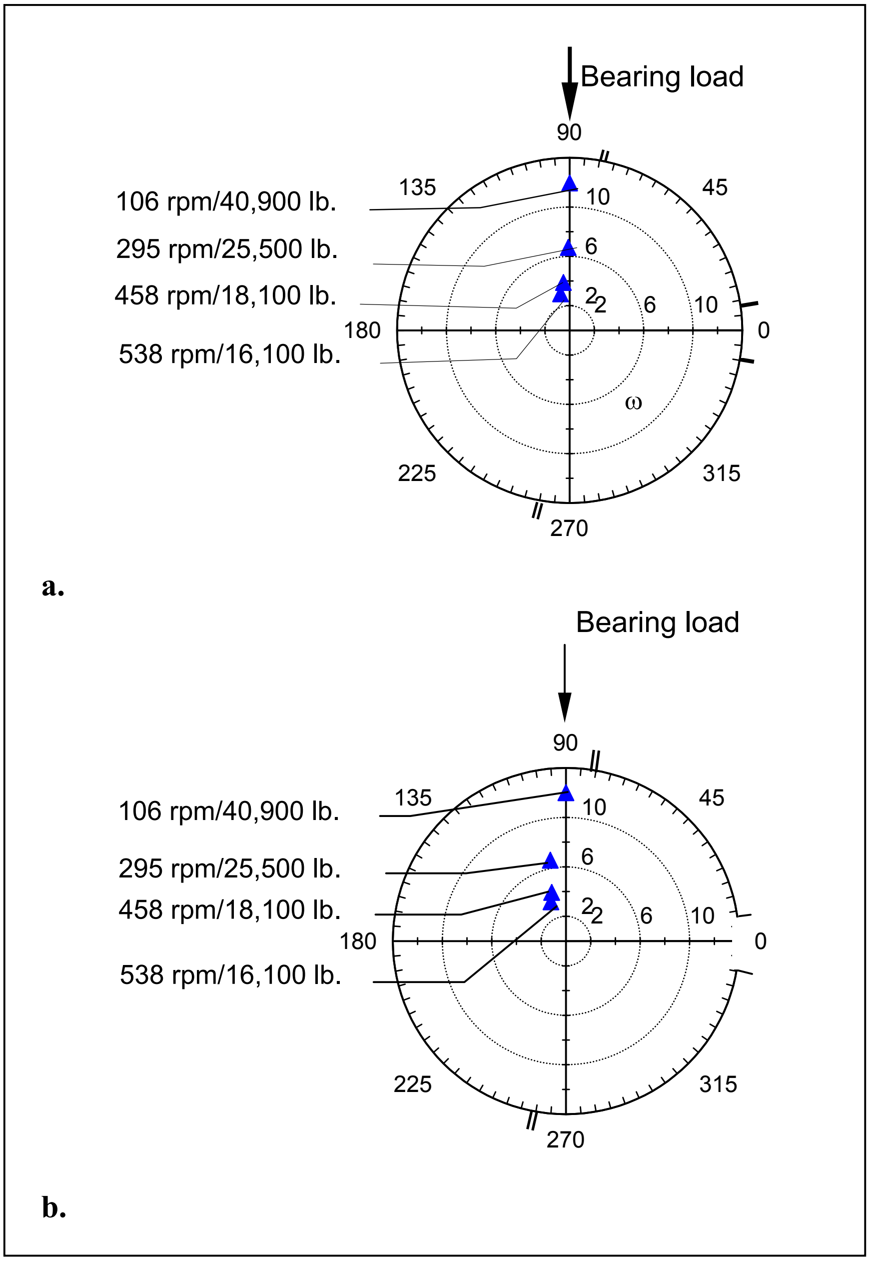


Figure 16 Journal centre location within the bearing clearance circle (10^{-3} in.)
a. normal reservoir oil level **b.** low reservoir oil level

6. EXPERIMENTAL INVESTIGATION OF FULL-SIZE VESPEL®-LINED BEARINGS

The Vespel® material was supplied by DuPont in a form of rings of 8.345 in. in inner diameter, 8.768 in. in outer diameter, and 1.25 in. in width, as illustrated in Figure 8.

The bearings were fabricated using the following sequence of operations:

1. Cut Vespel® rings into two halves
2. Machine bore the brass bearing to 8.346 in. in diameter

3. Clean housing surface with MEK and then grit blast using Al-oxide 220 grit and dry nitrogen gas
4. Bond the bearing into the housing using one layer of FM400NA film adhesive
5. Ultra-sonic C-scan the housing to determine bond integrity
6. Remove any excess Vespel® from the joints
7. Machine Vespel® lining to the required bore size

Figure 17 shows one of the bearings in the process of being machined.

Figure 17 Machining Vespel®-lined bearing

6.1 Mark 1 Vespel®-lined bearing

The diameter of the Mark 1 bearing was $8.039^{+0.002}$ in. The nominal diameter of the shaft was 8.000 in. thus the radial clearance was 0.020 in.

Tests on the first Vespel®-lined bearing were not concluded successfully. The bearing failed after only 11 minutes of operation, when the shaft speed was 538 rpm and the bearing load was 5000 lb. Figure 18 shows the main test parameters plotted against time (friction loss is related to the average loss for the load of 5000 lb.). At the time of the failure, concern was expressed about the large amount of torque applied to the housing bolts during the assembly procedure. This puts a high hoop strain on the bearing. While this is an acceptable practice in the case of the conventional bearing, it was speculated that this might have caused the Vespel® rings to separate from the brass backing.

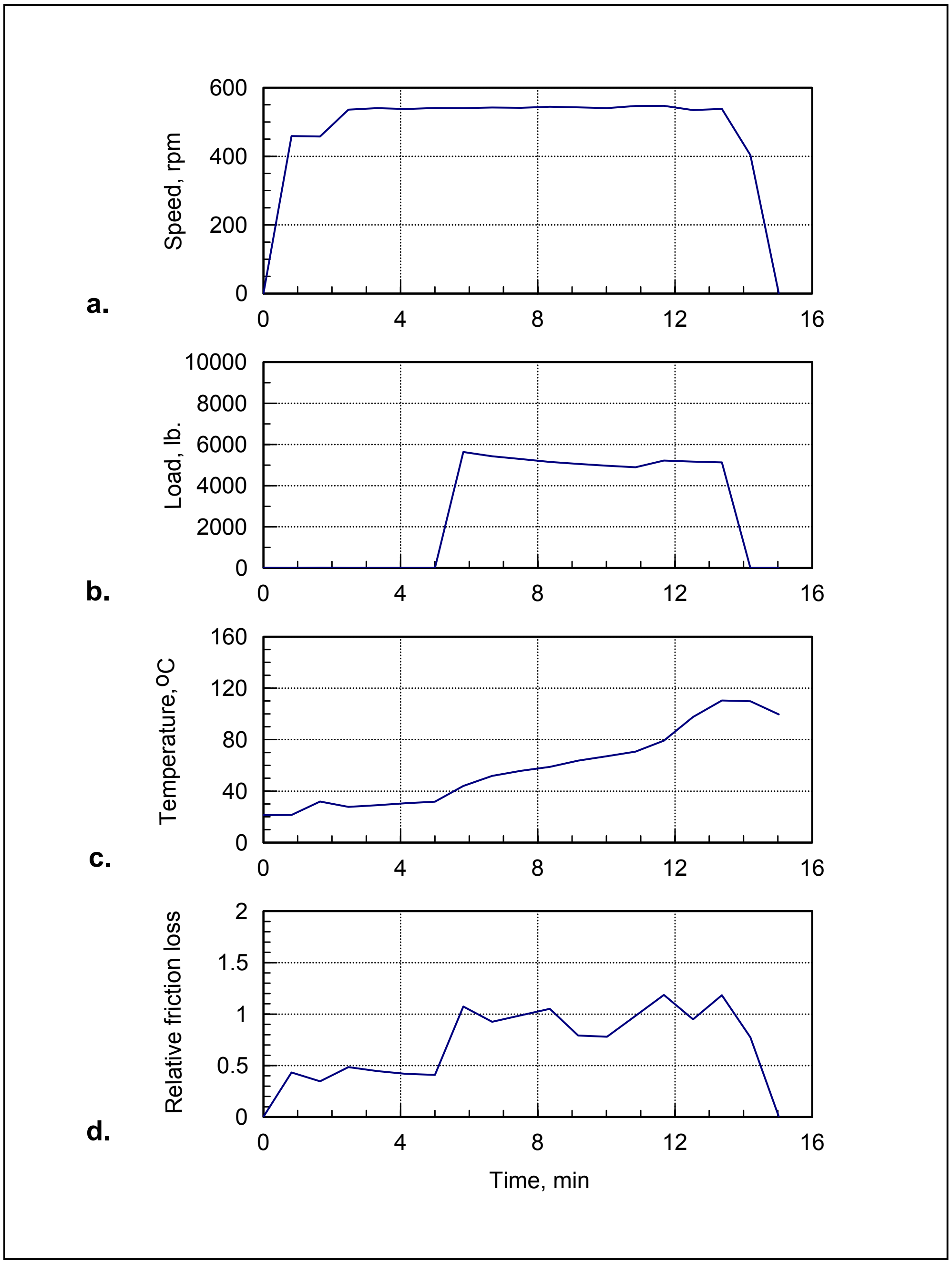


Figure 18 Test results. Mark 1 Vespel®-lined bearing
a. shaft speed **b.** bearing load
c. maximum bearing temperature **d.** relative friction loss

Figure 19 shows the heavily rubbed areas and their location in relation to the load line. The shaft after this test is shown in Figure 20.

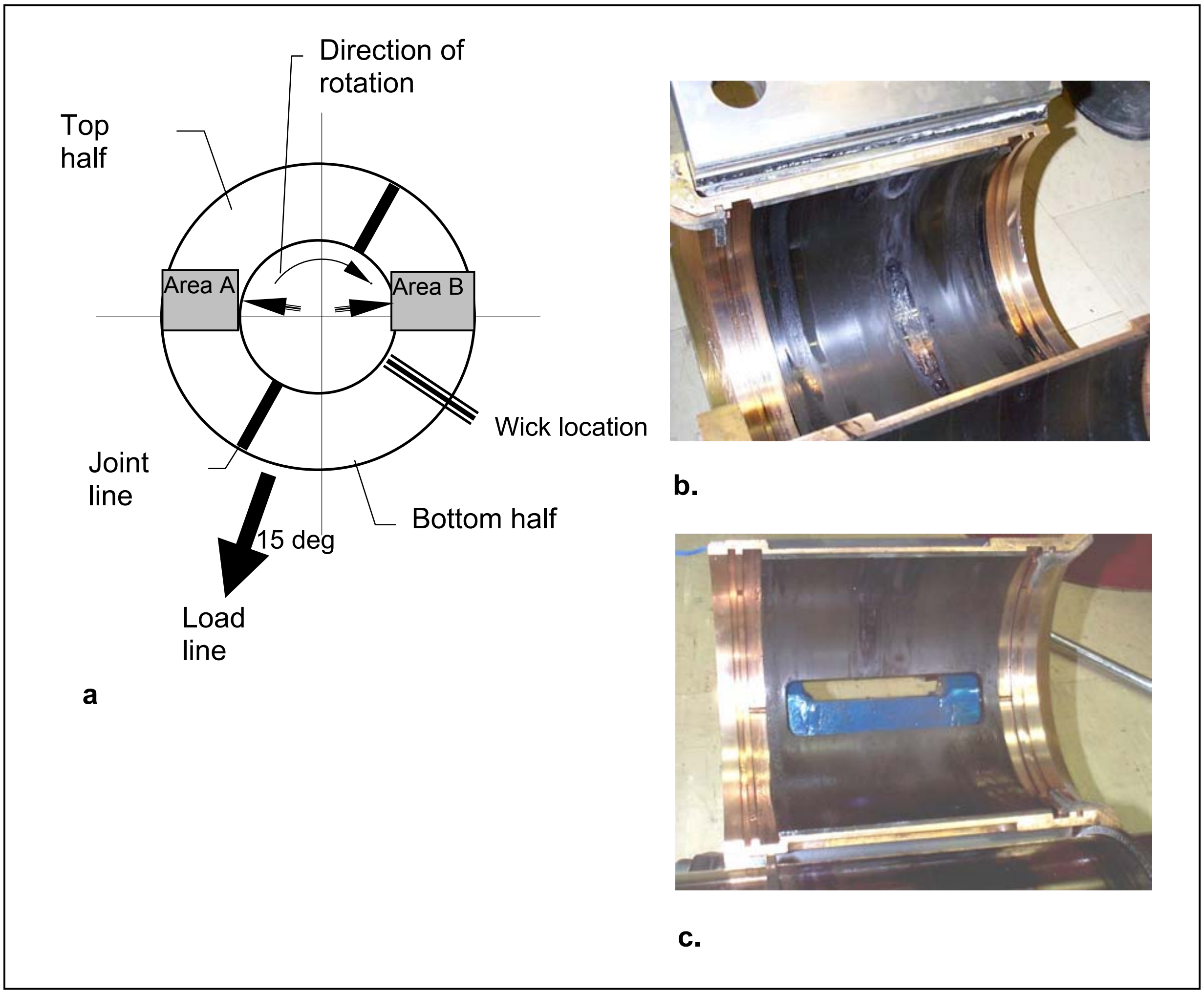


Figure 19 **Mark 1 Vespel®-lined bearing after failure**
a. damaged sites **b.** top half **c.** bottom half

Figure 20 **Shaft after failure of the Mark 1 Vespel®-lined bearing**

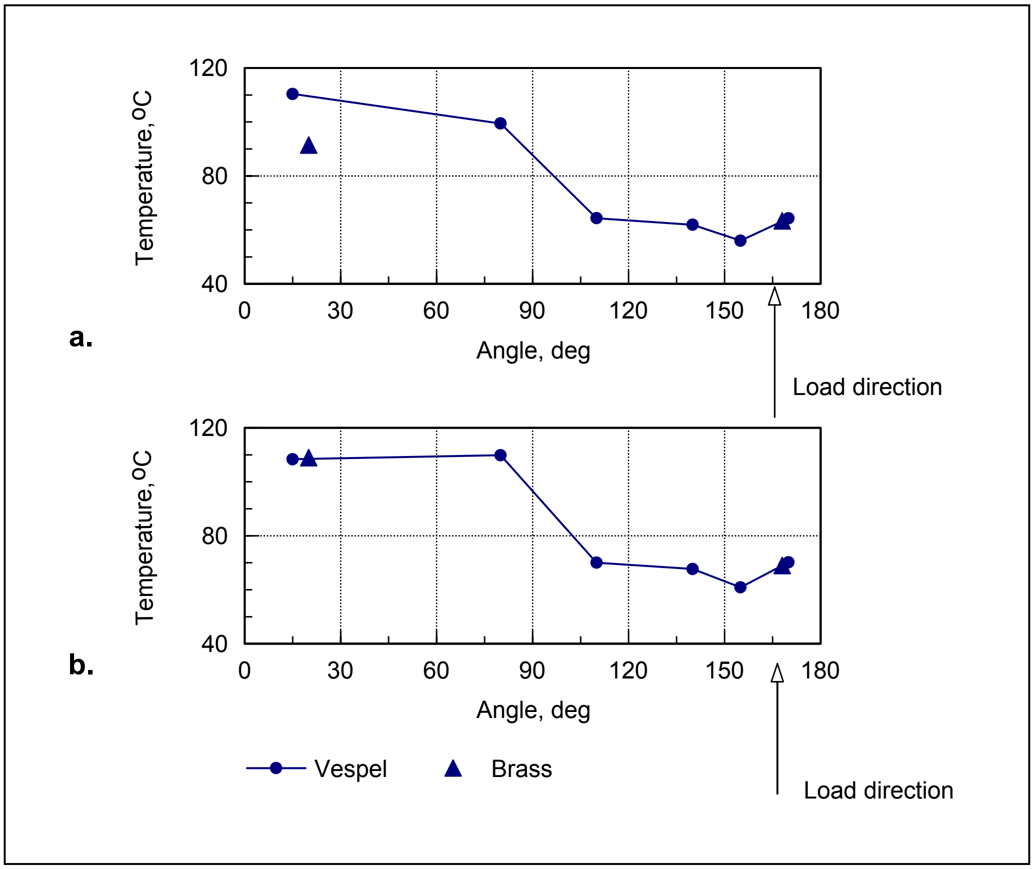


Figure 21 Centre line temperature profile: Mark 1 Vespel®-lined bearing
a. at time of failure **b.** immediately after removing 5000 lb. load

Measured centre-line temperature profiles from the Vespel®-lined bearing, just at the point of failure and shortly after the load was removed, are shown in Figure 21. Points of interest arising from these plots are as follows:

- The maximum recorded bearing temperature was approximately 110°C.
- At the time of the failure, a thermocouple in the brass housing, which was very close to the location of the maximum bearing temperature, recorded a temperature of about 90°C. However, by the time the load had been removed, this temperature had quickly risen to 110 °C (the same temperature as the bearing).
- At the time of the failure, the maximum temperature was recorded by the thermocouple located approximately 155 ° from the load line.

6.2 Mark 2 Vespel®-lined bearing

Initially, the Mark 2 bearing had a nominal radial clearance of 0.020 in., which was the same as that of the Mark 1 bearing. In a series of tests involving the Mark 2 Vespel®-lined bearing, the housing bolts were torqued to a much lower level.

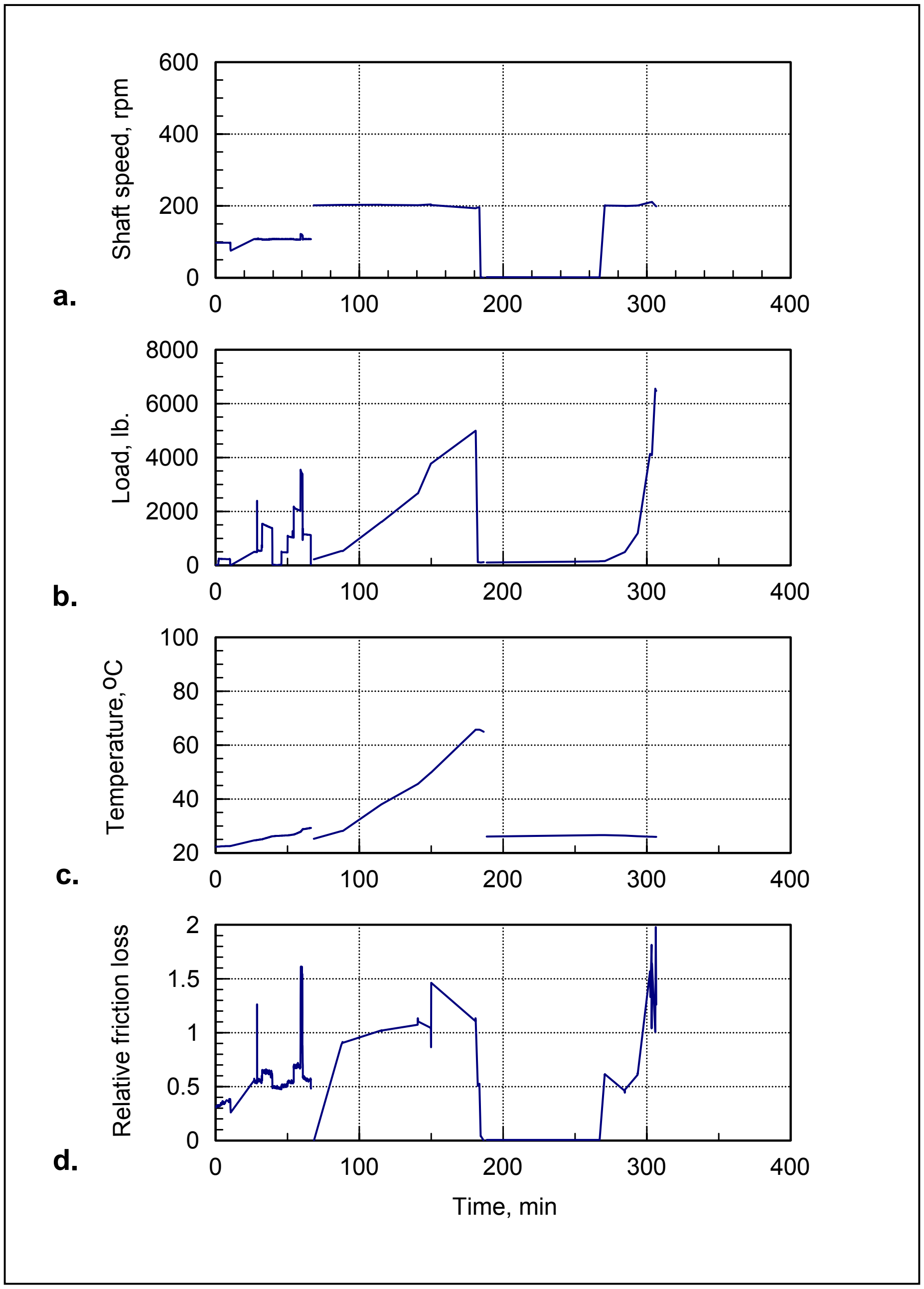


Figure 22 **Test results: Mark 2 Vespel®-lined bearing (original clearance)**
a. shaft speed **b.** bearing load
c. maximum bearing temperature **d.** relative friction loss

The following preliminary tests were conducted on the Mark 2 Vespel®-lined bearing for a total running time of 12 hours:

- 5.4 hrs @ 200 rpm, 0 to 5000 lb.
- 4.25 hrs @ 100 rpm, 0 to 3500 lb.
- 2 hrs @ 200 rpm, 0 to 6500 lb.

The measured test data are illustrated in Figure 22 (friction loss is related to that for the load of 4000 lb.).

Measured bearing center-line temperature profiles from the first and second tests are shown in Figure 23.

In the course of each of these tests, very heavy rubbing sounds were heard.

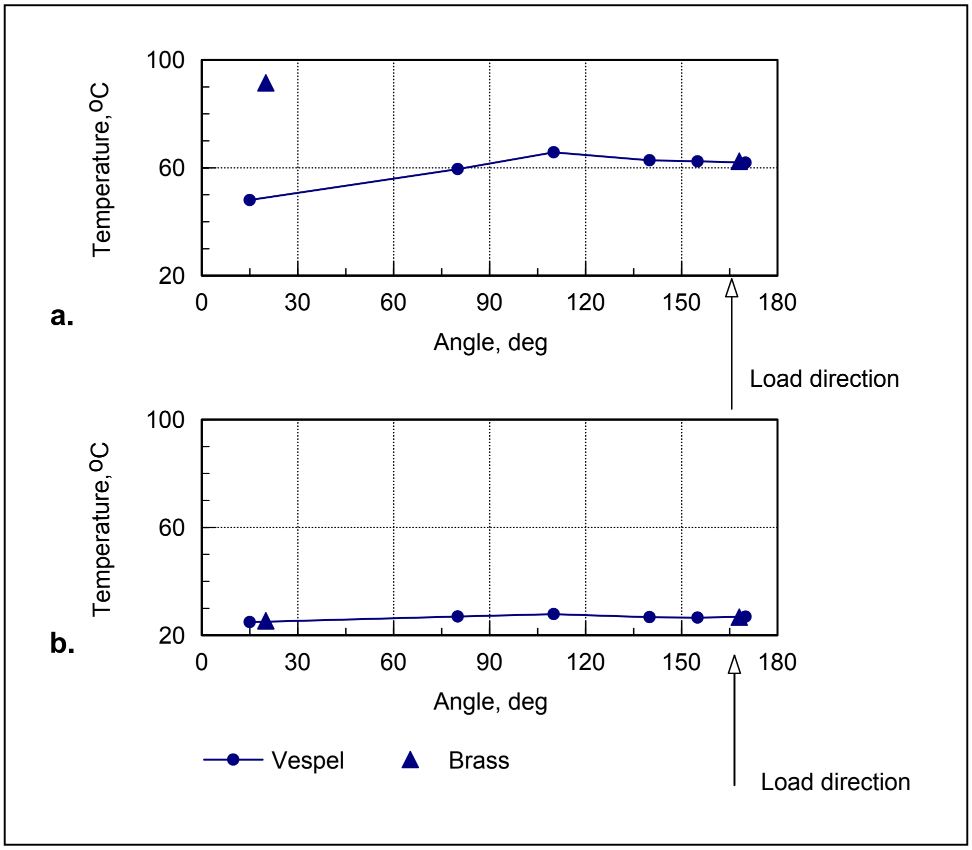


Figure 23 Centre line temperature profiles: Mark 2 Vespel®-lined bearing
a. 200 rpm, 5000 lb. **b.** 100 rpm, 3500 lb.

The bearing was removed from the rig and inspected. It was found that certain areas of the bearing bore had sustained quite heavy wear, as shown in Figure 24. The wear pattern became more clearly evident after the bearing had been machined and 0.005 in. was removed from the bearing bore. This damage is mainly confined to three major areas, as shown in Figure 25.

a.

b.

Figure 24 Damage to the Mark 2 Vespel®-lined bearing prior to rebore
a. bottom half **b.** top half

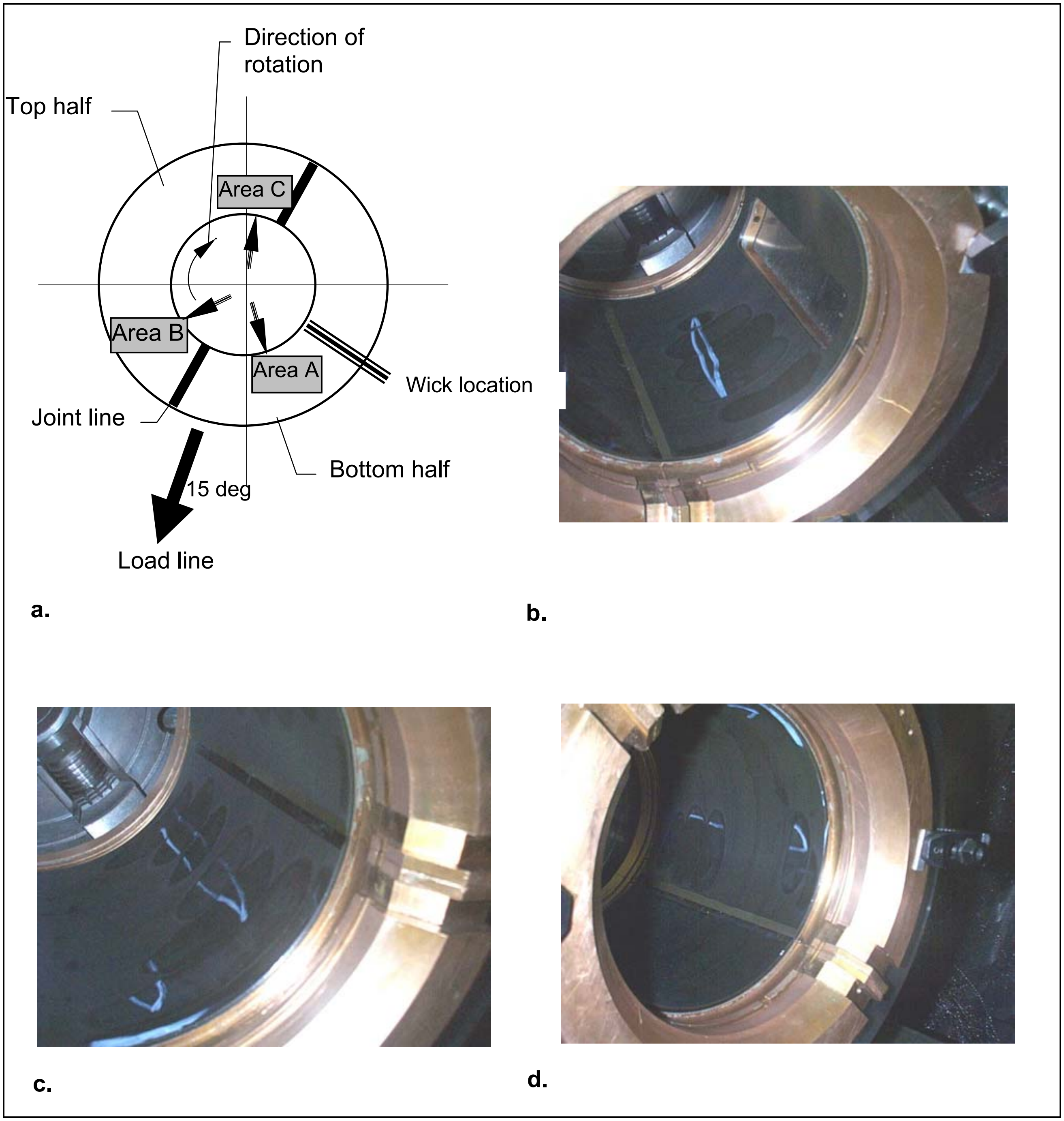


Figure 25 Damaged sites (after 0.0025 inch machining operation): Mark 2 Vespel®-lined bearing

a. damaged sites **b.** area A
c. area B **d.** area C

The bearing bore was cleaned up until there were no further signs of damage. As a result, the bore size increased from the design value of 8.039 in. to the re-machined size of 8.067 in. (radial clearance of 0.039 in.). Thus, it may be concluded that some of the wear scars were as much as 0.014 in. deep.

The re-machined bearing was reinstalled in the test rig, and attempts were made to bring the bearing up to the first test condition (538 rpm/16,100 lb.). After approximately 30 min of operation, when the load had reached approximately 12,000 lb. and the shaft speed was 500 rpm, smoke started to issue from the test bearing. At this time, the test was terminated. The results from this test are shown in Figure 26 (friction loss is related to that for the load of 12,000 lb.). It should be pointed out that quite significant rubbing sounds were heard during the course of this test. The measured bearing centre-line temperature profile, just at the point of failure, is shown in Figure 27.

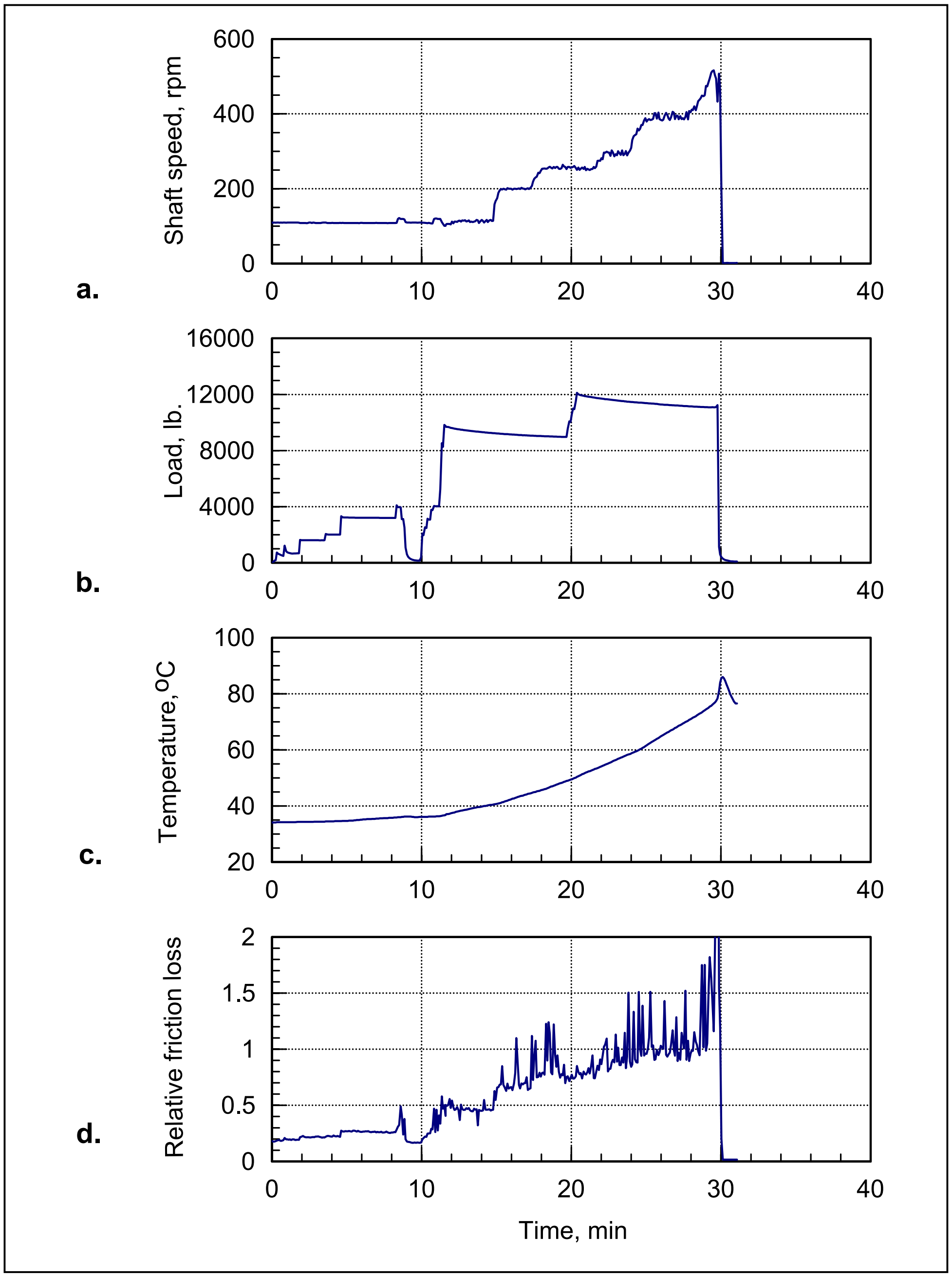


Figure 26 **Test results: Mark 2 Vespel®-lined bearing (larger bore)**

a. shaft speed **b.** bearing load

c. maximum bearing temperature **d.** relative friction loss

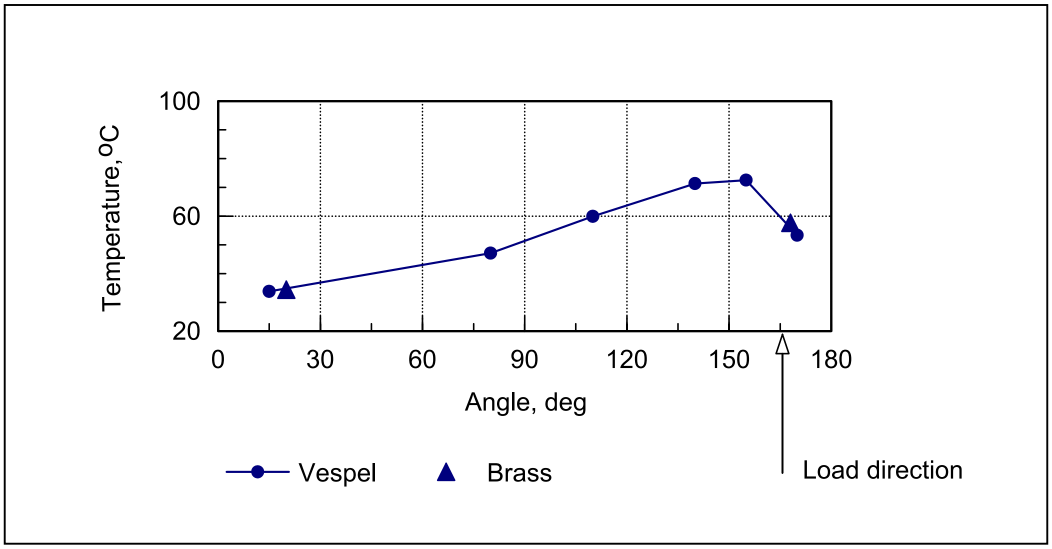
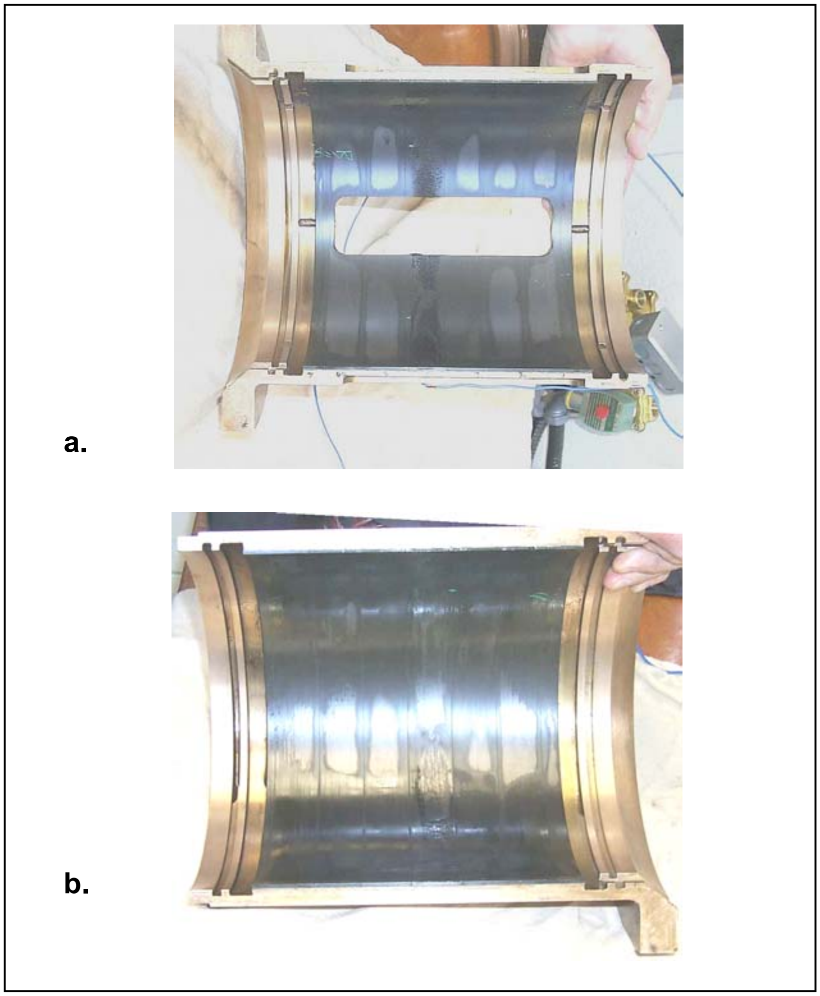


Figure 27 **Centre line temperature profile at point of failure 500 rpm, 11,000 lb. load: Mark 2 Vespel®-lined bearing (larger bore)**



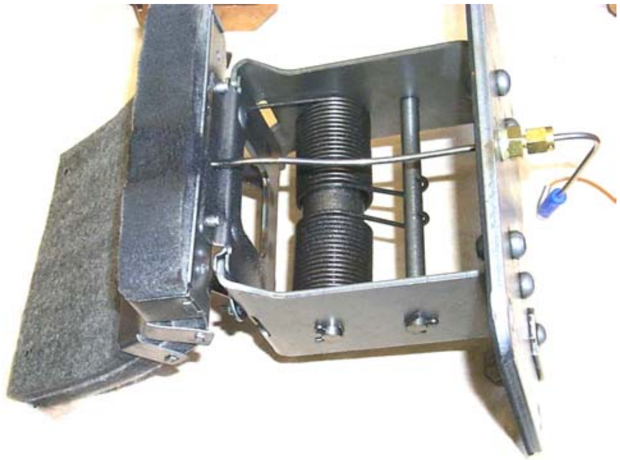
a.

b.

Figure 28 **Damaged Mark 2 Vespel®-lined bearing (larger bore)**
a. bottom half **b.** top half

The bearing was removed from the test rig. Photos showing the damaged Mark 2 Vespel®-lined bearing are presented in Figure 28. The wick and shaft after this test are shown in Figure 29.

a.



b.

Figure 29 Damaged Mark 2 Vespel®-lined bearing (larger bore)
a. shaft **b.** wick

6.3 Mark 3 Vespel®-lined bearing

The Mark 3 bearing was made using a modified procedure for bonding the Vespel® to the brass substrate:

1. Clean bearing and housing surfaces with MEK and then grit blast using Al-oxide 220 grit and dry nitrogen gas
2. Apply a silane (coupling agent) to the housing surface
3. Apply a BR27 primer onto the housing surface
4. Bond the bearing into the housing using one layer of FM400NA film adhesive
5. Ultra-sonic C-scan the housing to determine bond integrity

Another important difference was that the high temperature tape used in the Mark 1 and 2 bearings to prevent the ingress of adhesive between the Vespel® strips during the bonding process was not used in the Mark 3 bearing. There had been concern that this tape affected the bond strength between the Vespel® material and the brass substrate.

Following DuPont's recommendations, the bearing clearance was reduced to 0.020 in. diametrally (bearing bore of 8.020 in. diameter).

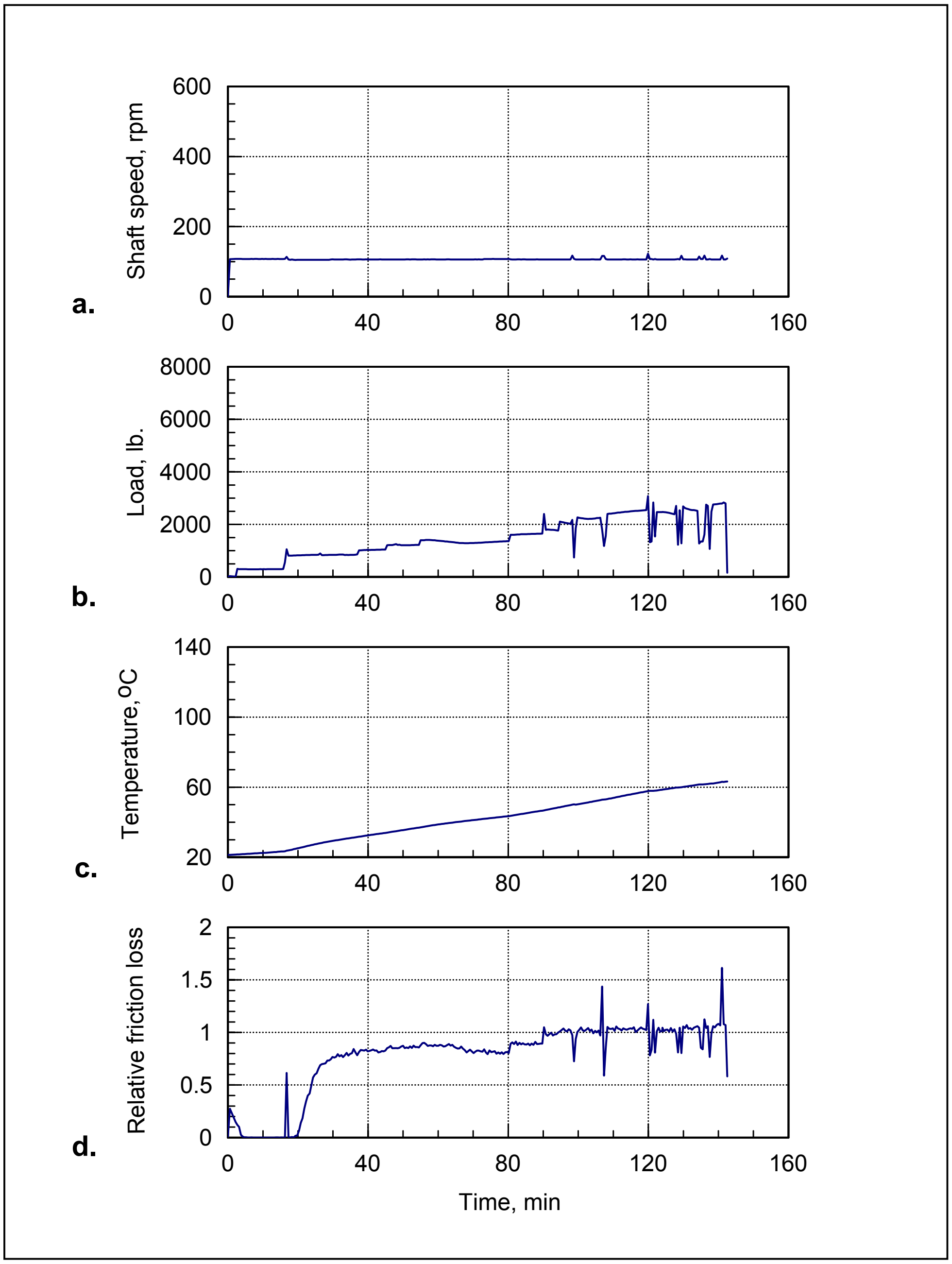


Figure 30 **Test results: Mark 3 Vespel®-lined bearing**

a. shaft speed **b.** bearing load
c. maximum bearing temperature **d.** relative friction loss

In the first series of tests on the Mark 3 bearing the rig ran at the lowest speed conditions, 106 rpm (223 ft/min). Over a time period of 143 min., the load was slowly incremented up to 2,800 lb. (42 psi) in approximately 200 lb. steps. At these conditions the strong rubbing noise and severe fluctuations of the measured friction torque were observed, similar to those observed in earlier tests. The maximum recorded bearing temperature in the Vespel® lining was 63°C. Figure 30 shows the recorded test results (friction loss is related to that for the load of 2,800 lb.).

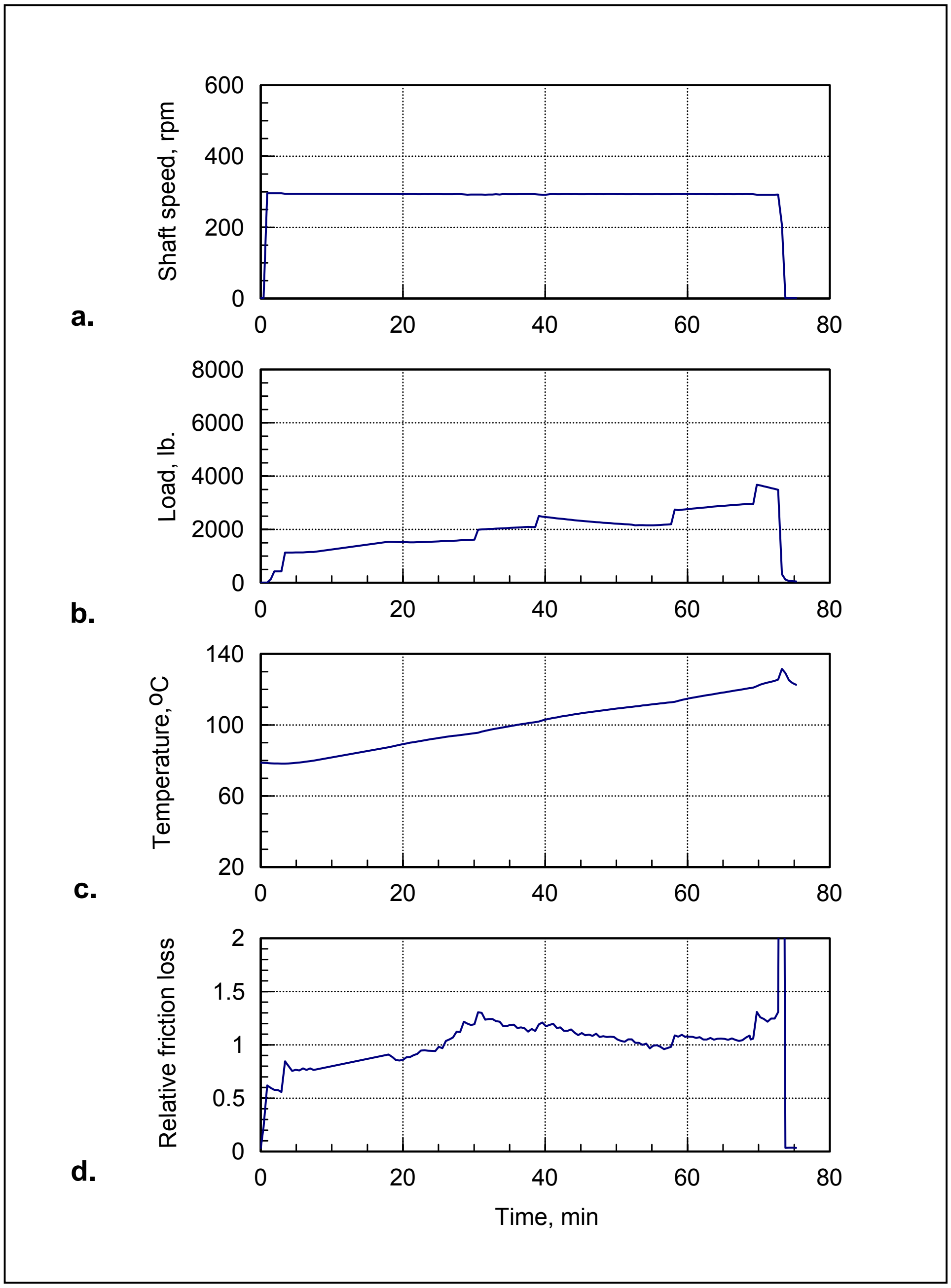
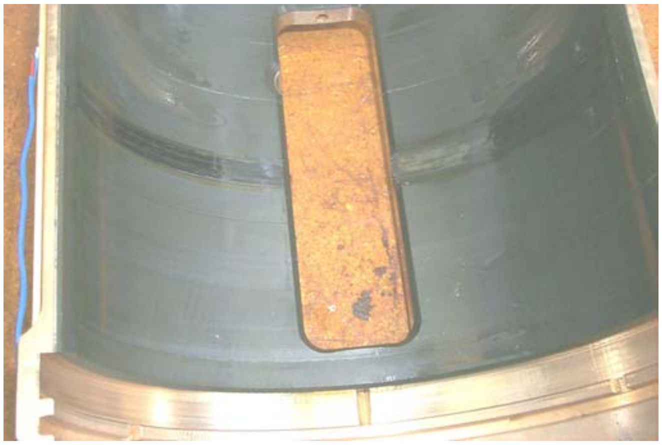


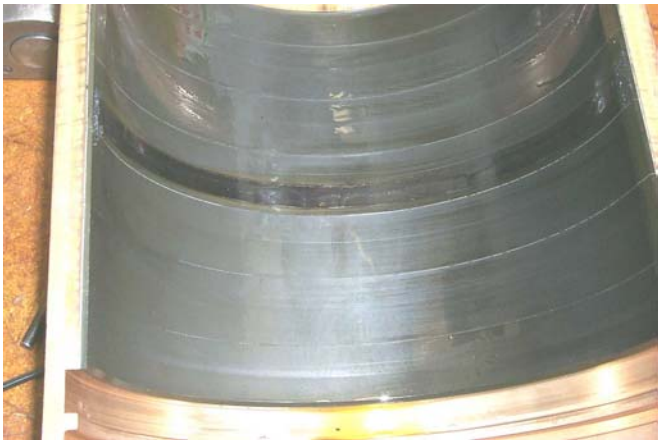
Figure 31 Test results, shaft speed 295 rpm: Mark 3 Vespel®-lined bearing
a. shaft speed **b.** bearing load
c. maximum bearing temperature **d.** relative friction loss

The tests on the Mark 3 bearing were also carried out at a speed of 295 rpm (621 ft/min). Figure 31 shows the recorded data for the last 70 min. of the test (friction loss has been related to the average for the load of 2,800 lb.). Over a period of 147 min. the load was slowly increased up to 3600 lb. At this load the bearing experienced sudden fluctuation of friction torque and soon seized.

Damage to the bearing was consistent with that obtained in the previous tests, with wiping and heat cracking of the Vespel® lining (Figure 32). However, this time the damage was confined to the central portion of the bearing lining, with width of the wiped area varying between 1 and 1.5 in., and the wiping extended a full 360° around the bearing. It is likely that such damage occurred due to the tighter bearing clearance.



a.



b.

Figure 32 Damaged Mark 3 Vespel®-lined bearing
a. bottom half **b.** top half

6.4 Mark 4 Vespel®-lined bearing

The fourth Vespel®-lined bearing was made from a single cylindrical piece of Vespel®, instead of multiple rings. It was expected that this change would improve bearing performance, since a long bearing has superior load capacity when compared to that of the same length but consisting of multiple shorter bearings. Additionally, on DuPont's recommendation, the bearing diametral clearance was increased to be between 0.030 and 0.035 in. (bearing diameter of 8.065 in.).

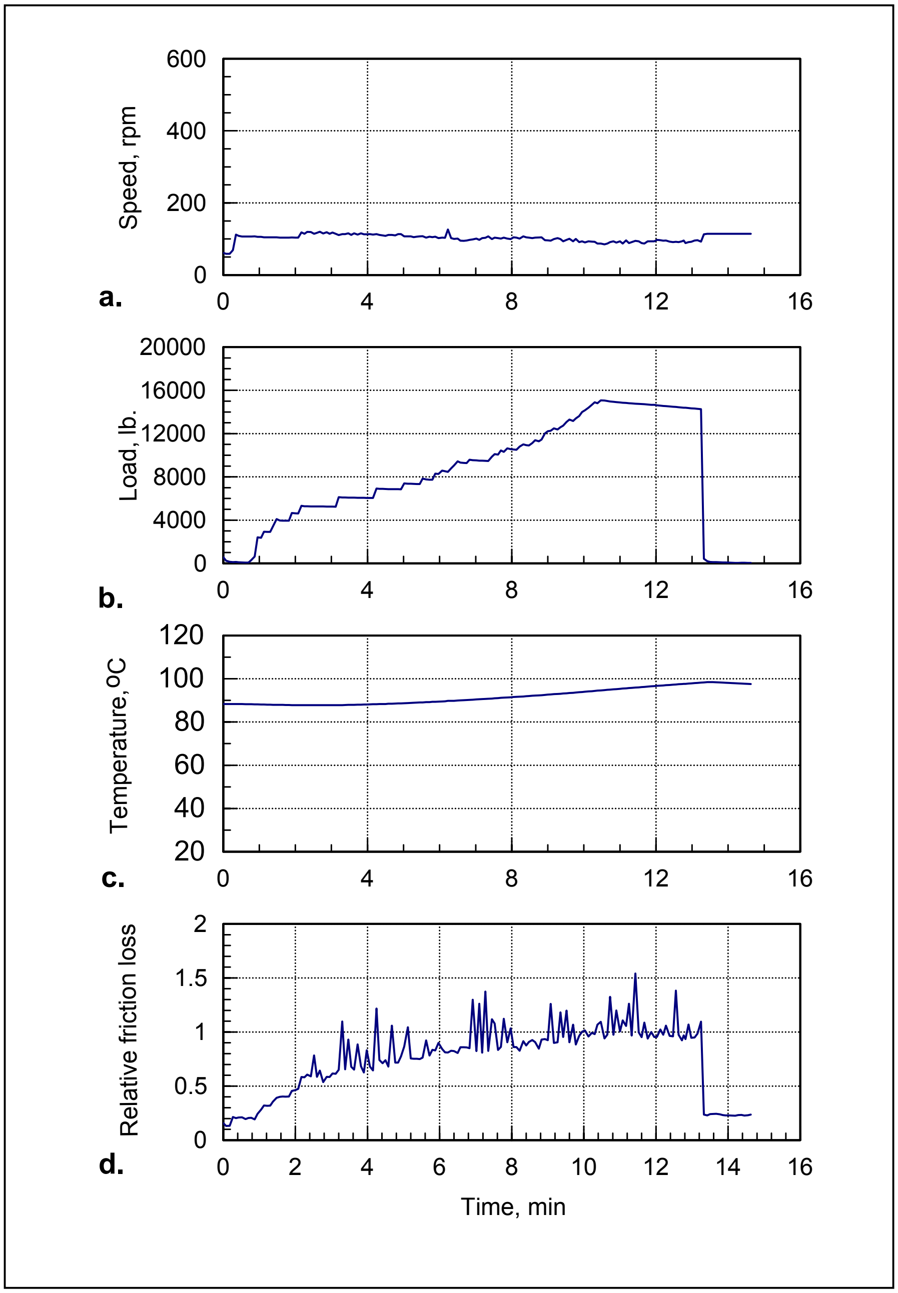


Figure 33 **Test results: Mark 4 Vespel®-lined bearing**
a. shaft speed **b.** bearing load
c. maximum bearing temperature **d.** relative friction loss

As with the previous test bearings with Vespel® lining, the bond strength between the lining and the bronze substrate was checked.

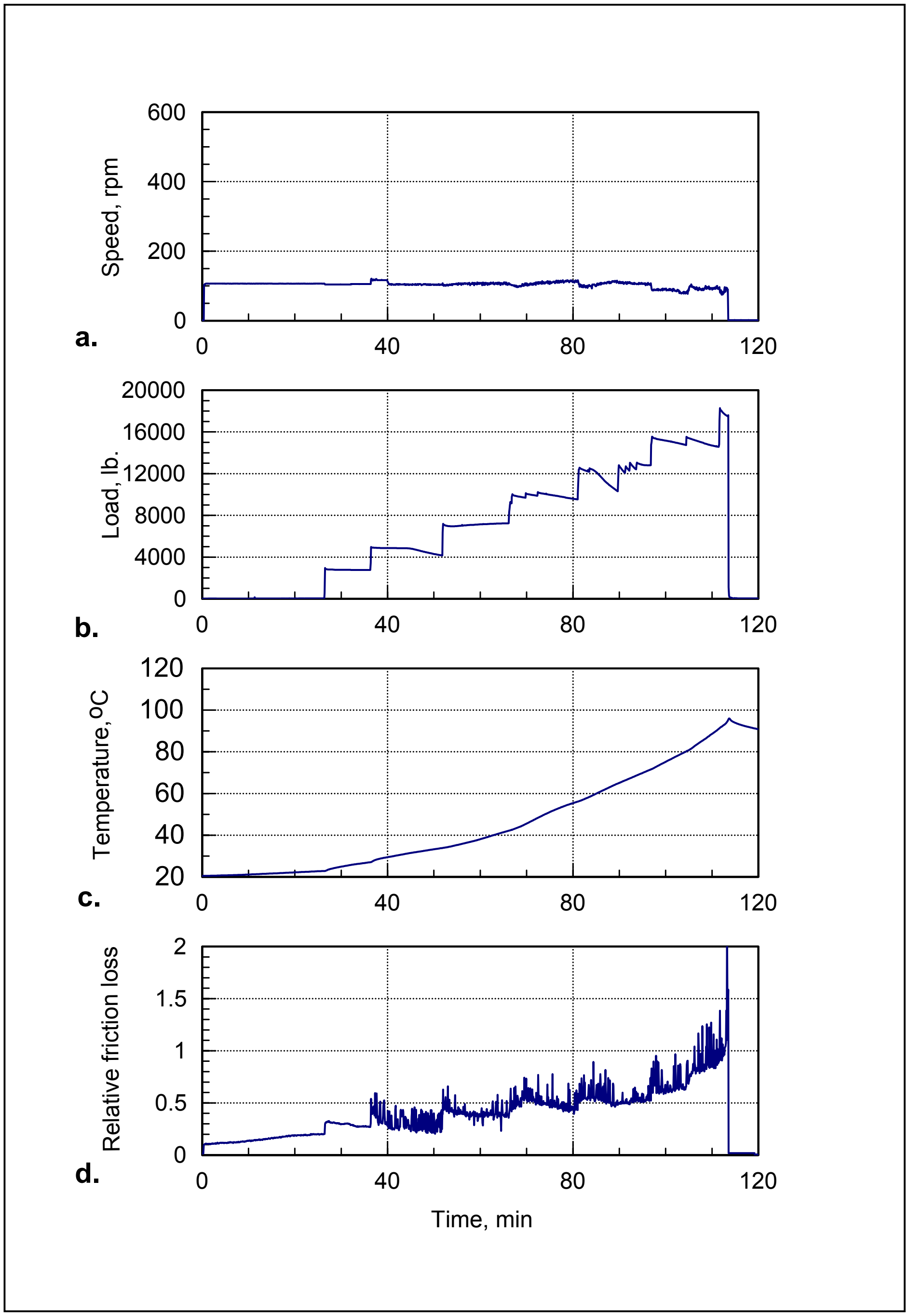


Figure 34 **Test results. Mark 4 Vespel®-lined bearing failure**
a. shaft speed **b.** bearing load
c. maximum bearing temperature **d.** relative friction loss

At 106 rpm this version of the bearing achieved a load of 15,000 lb., which was the highest load applied to Vespel®-lined bearing. Figure 33 illustrates the course of this test (friction loss is related to the average loss for the load of 15,000 lb.).

However, an attempt to run this bearing with a load of 18,200 lb. was unsuccessful (Figure 34, friction loss related to that for 15,000 lb.). The bearing was damaged, and the pattern of failure was the same as that observed in the previous tests. Figure 35 shows the damaged Mark 4 bearing.

a.

b.

Figure 35 Mark 4 Vespel®-lined bearing and shaft after the tests
a. bearing **b.** shaft

7. CONCLUSIONS

1. Traction motor bearings operate under difficult and severe conditions, which include relatively high unit load, low shaft speed, and limited lubricant supply by the wick. This situation is very demanding for the bearing material. In the presence of sufficient oil supply, babbitt meets the requirements for these bearings.

2. The applied fabrication process led to good bonding of the Vespel® and brass substrate, as well as achieving the designed geometry and quality of the bearing surface.

3. In spite of the promising technical specifications and bench test results, the tested composite material did not perform well when replacing babbitt in traction motor journal bearings. It can be concluded that the material's thermal properties were insufficient to transfer the large amount of heat generated in the bearing. This led to high temperature and low viscosity of oil, as well as to thermal growth, which resulted in the bearing seizures.

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APPENDIX A

Comments from DuPont Engineering Polymers

MAR. 28. 2003 11:28AM NO. 4943 P. 2

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Newark, DE 19714-6100

Tel: 302-733-8118 / Fax: 302-733-8137

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Date: March 29, 2003

TO: Waldemar Dmochowski
CC: Robert Haynes
FROM: C. Scott Stenta

Subject: NRC Traction Motor Bearings

Conclusion:

All tests performed using Vespel* SP-214 bearings have failed prematurely due to excessive noise and vibration on the NRC test rig. Laboratory tests were stopped in each instance prior to reaching operating pressures and velocities. Upon evaluating the bearings after each test, it was evident that severe localized heat caused blistering and wear on the bearing I.D. In the early stages of testing the load was increased in order to reach the maximum bearing pressure but the shaft seized before maximum bearing pressure was achieved.

The exact cause of the bearing failures could not be identified but a few possibilites exist based on the appearance of the bearings that were returned for evaluation. It was quite obvious that the wick system did not produce sufficient lubrication to form a hydrodynamic film, thus higher temperatures than expected caused the bearing to wear and blister. The blistered region was located approx. in the center of the bearing in each case but not completely around the entire ID of the bearing. Since the oil wick location was also in the center of the brass housing, it is possible that the wick also overheated and stopped providing lubrication. In one case the excessive temperatures caused the wick material to break down and become lodged in the bearing between the I.D. and the shaft. There was evidence of a residue build-up on the I.D. surface at the trailing edge of the wear scar in the bearing. This residue did not appear to be Vespel* wear debri.

It was initially assumed that localized thermal expansion of the bearing material contributed to the severe wear when the initial multiple strip design was tested. The over heating was thought to be caused by the lubrication not supporting the hydrodynamic load due to the gaps between the strips. This phenomina was disregarded when the Mark 3 and Mark 4 tests were concluded using the single continuous Vespel* bearing. The wear scar and blistering appeared at approx. the same location in the bearing regardless of whether the bearing surface was one continuous piece or multiple bearing strips. Based on the pressure and velocity values that were reported, with adequate lubrication, the Vespel* bearing material should have been successful in this application.

Although we can not be certain as to the exact cause of the failures, we can conclude that bearing surface temperatures exceeded our expectations even with the operating PV well below the maximum value recommended in the Dupont literature.

The bearing that was sent to Dupont for evaluation will be returned to NRC.