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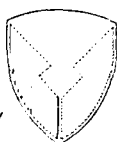
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Comparison of Predicted and Measured Temperatures of UH-60A Helicopter Transmission

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Comparison of Predicted
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Transmission

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National Aeronautics and
Space Administration
Office of Management
Scientific and Technical
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Summary

The 2109-kW (2828-hp) UH-60A Black Hawk helicopter transmission was one of the transmissions used to obtain an experimental data base. Component improvements or new transmission concepts can thus be evaluated by comparison with the established data. Results of efficiency and vibration tests of the UH-60A have been reported previously. In this investigation the transmission was instrumented internally and tested over a range of operating conditions. The speed was varied from 50 to 100 percent of the full rated value, and the torque was varied from 10 to 100 percent of the full rated value. Temperatures of internal bearings and gears were measured. The computer program Planetsys was used to simulate the thermal performance of this transmission. The calculated temperatures were then compared with the corresponding measured values. The highest measured temperature was 405 K (270 °F) on the outer ring of the high-speed input shaft roller bearing, at the 100-percent power condition. In general, the predicted temperatures compared very well with the measured values, most of them being within 5 kelvin (9 deg F). Specifically, the temperatures predicted for the single-row spherical roller planetary bearing averaged only about 4 kelvin (8 deg F) lower than the highest measured value.

Introduction

Because of their versatility helicopters have become an important segment of the aircraft industry, for both civil and military applications. A critical component of any helicopter is the transmission required to reduce the turbine engine speed to the main rotor and tail rotor speeds. Transmissions used in aeropropulsion systems, however, can have severe limitations such as low durability, reliability, and maintainability or high noise, vibration, and weight. The transmission research programs conducted at the NASA Lewis Research Center over the past few years are outlined in reference 1. In general, these programs were aimed at extending the life of a transmission, improving its reliability, reducing the required weight, or reducing the noise and vibration. So that improvements in components or new transmission concepts could be evaluated, however, it was important to establish a base of data on what

were then current state-of-the-art transmissions (ref. 2). One of the transmissions chosen was the UH-60A Black Hawk, a 2109-kW (2828-hp), dual-input, planetary helicopter transmission.

Efficiency and vibration tests conducted previously on the UH-60A are reported in reference 3. After these tests (which required only external instrumentation) were run, the transmission was further instrumented internally and additional data were obtained. Gear tooth stress measurements were made first, to utilize new strain gages before they became damaged (ref. 4).

It is equally important to obtain thermal measurements for the transmission, not only for evaluation of possible improvements, but also as a means of validating computer programs (such as Planetsys, ref. 5) that predict these operating characteristics, since this type of data is not readily available (ref. 2). The objects of this investigation were therefore to obtain temperature data on the internal bearings and gears of this UH-60A transmission over a range of operating conditions and to compare the results with values predicted by the computer program Planetsys. The transmission was operated from 10 to 100 percent of full rated power with rotor speeds from 50 to 100 percent of the full rated value. The inlet oil was maintained at 355 K (180 °F).

Apparatus

Test Transmission

The UH-60A helicopter transmission has a dual input power rating of 2109 kW (2828 hp). It provides a total speed reduction of 81.042:1 in three stages from the engine input (20 900 rpm) to the main rotor output (258 rpm). There is also a tail shaft output (4117 rpm).

The transmission, more fully described in reference 3, is shown in figures 1 and 2. It consists of two input modules and one main module. The input modules contain the first reduction stage, which then drives an intermediate-speed shaft through an overrunning ramp-roller clutch. The second reduction stage, in the main module, combines the two inputs and drives the sun gear of the planetary third stage. The planetary system has five planet gears and a fixed ring gear. The planetary carrier output then directly drives the main rotor shaft. Also shown in figure 2 are the locations of the



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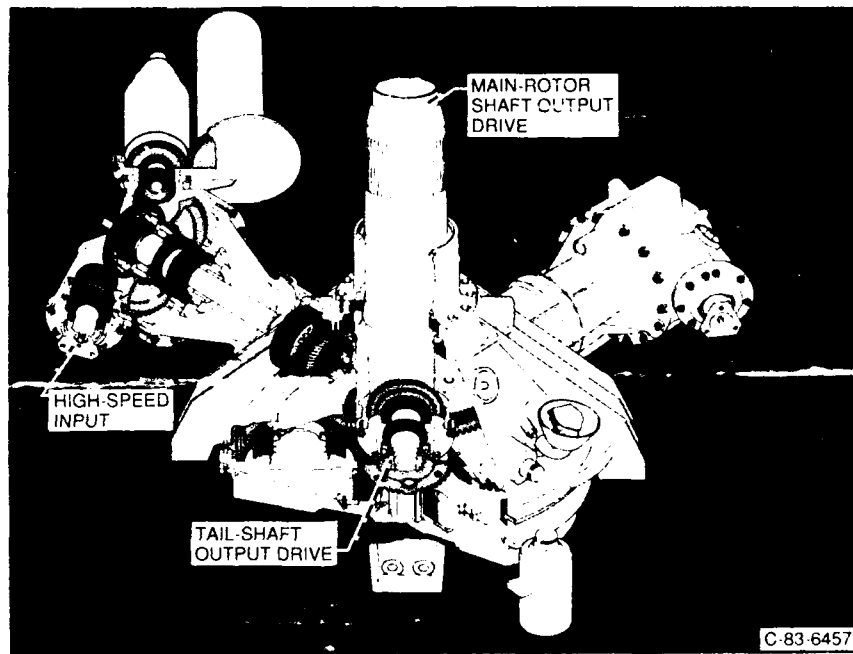


Figure 1.—UH-60A Black Hawk transmission.

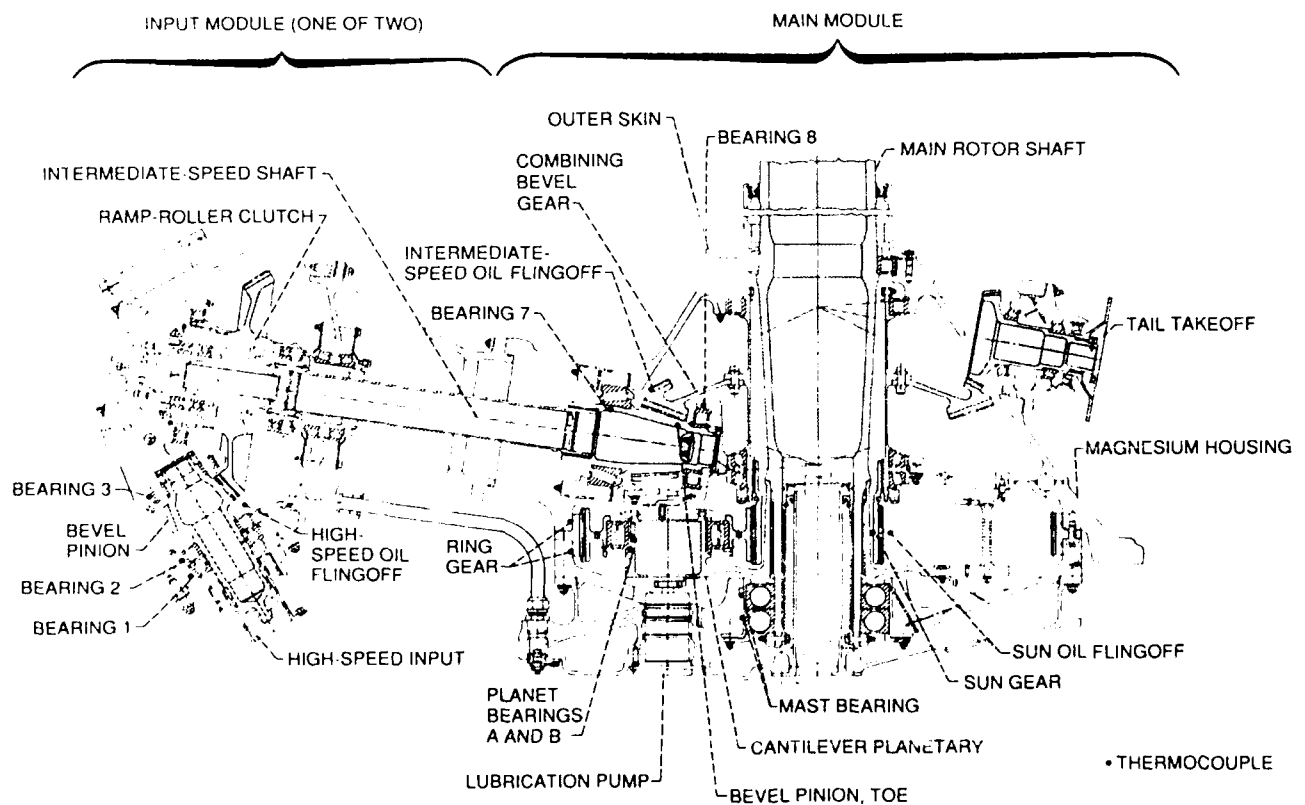


Figure 2.—Cross section of UH-60A transmission, showing location of thermocouples.

thermocouples used to obtain the temperature data. Shaft speeds and numbers of gear teeth are shown in table I.

The spherical roller bearings used in the planetary had a 110-mm bore diameter and a 170-mm outer diameter, with a single row of 18 rollers, each 19.5 mm in diameter and 29.5 mm long. The pitch diameter was about 140 mm. The bearing is shown in figure 3, and more complete specifications are given in table II.

TABLE I.—SHAFT SPEEDS OF UH-60A
TRANSMISSION AT 100-PERCENT
ROTOR SPEED (258 rpm)

Gear component	Number of teeth	Shaft speed, rpm
Spiral bevel input pinion	22	20 900
Spiral bevel input gear	80	5 750
Combining bevel pinion	17	5 750
Main bevel gear (lower)	81	1 207
Sun gear	62	1 207
Planet gears (five)	83	451
Carrier (output)	---	258
Ring gear	228	0
Main bevel gear (upper)	116	1 207
Tail output gear	34	4 117

^aPlanet gear rotates at 769 rpm with respect to carrier.

TABLE II.—PLANET ROLLER BEARING SPECIFICATIONS

Outer diameter, mm (in.)	170.0 (6.693)
Bore diameter, mm (in.)	110.0 (4.331)
Pitch diameter, mm (in.)	140.0 (5.512)
Outer-ring groove radius, mm (in.)	79.75 (3.140)
Roller maximum diameter, mm (in.)	19.50 (0.768)
Roller length, mm (in.)	29.46 (1.160)
Roller crown radius, mm (in.)	78.23 (3.080)
Number of rollers	18
Cage type	One-piece machined
Cage guide	Inner-ring land
Bearing diametral clearance, mm (in.):	
Unmounted	0.1727 (0.0068)
Mounted	0.0813 (0.0032)

Test Facility and Instrumentation

The NASA Lewis 2200-kW (3000-hp) Helicopter Transmission Test Facility, shown in figure 4 and more fully described in reference 3, is a recirculating-power (four square) type. Power, provided by a constant-speed, 600-kW induction motor, flows through the two inputs to the test transmission and the two outputs (main rotor and tail drive). Since power flow is recirculating, only frictional losses need to be replenished by the motor. Speed control is accomplished by an eddy current clutch. Torque is induced independently in each loop by planetary torque units.

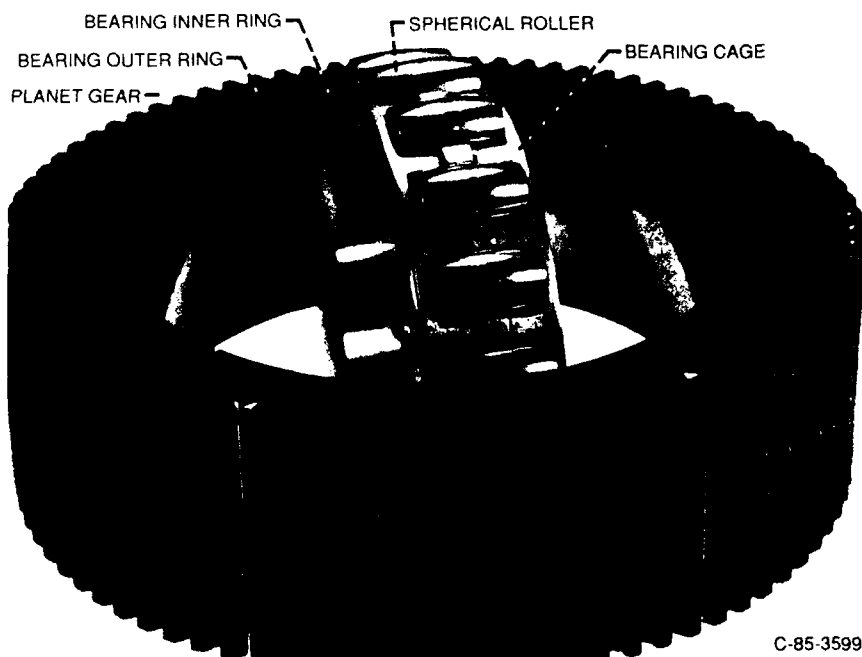
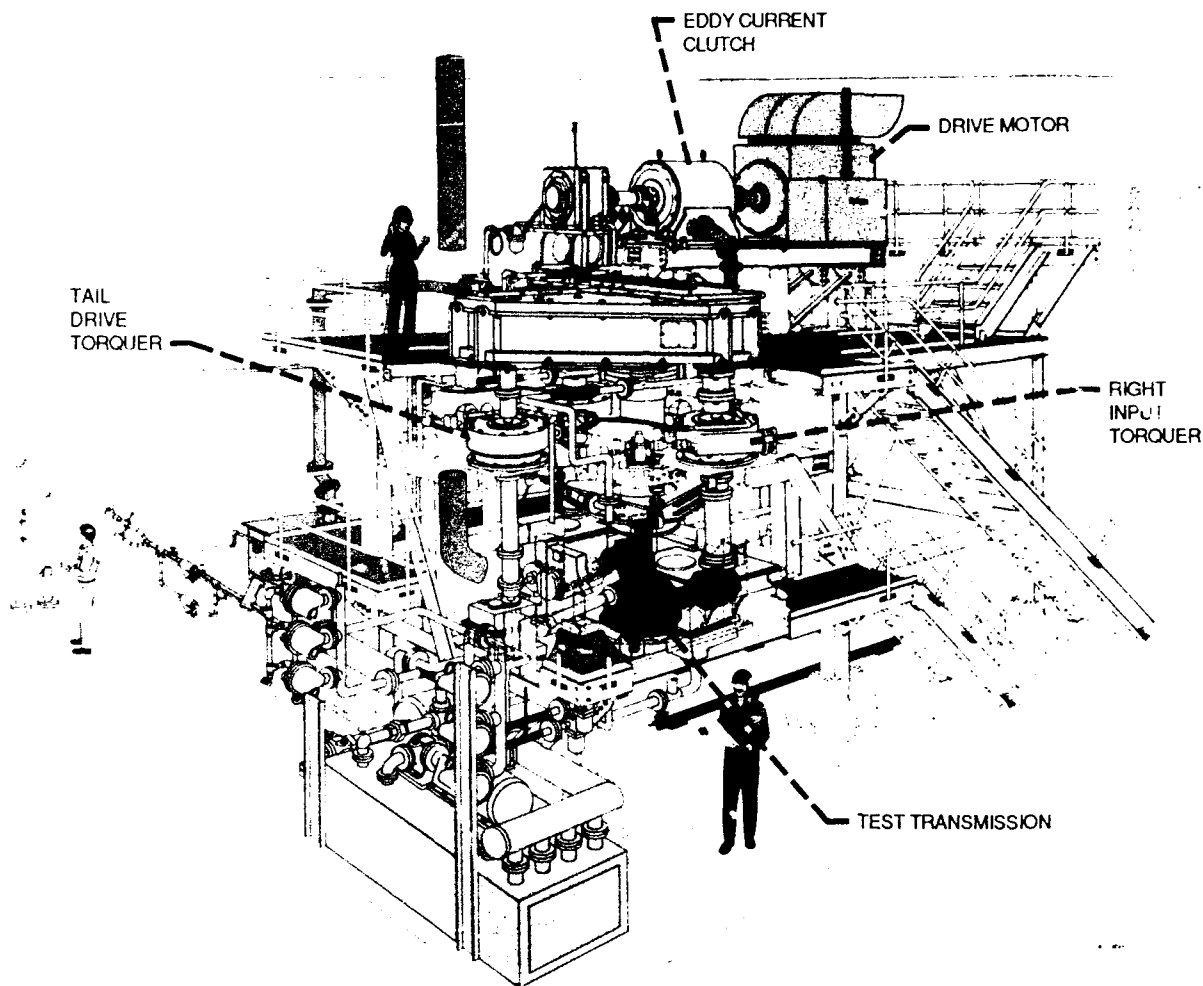


Figure 3.—Planetary gear and single-row spherical roller bearing.



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Figure 4.—NASA Lewis Helicopter Transmission Test Facility.

Type K (Chromel-Alumel) thermocouples were placed at the locations shown in figure 2. The thermocouples used to measure bearing temperatures were tight against the steel liners separating the outer rings from the magnesium housings. Those thermocouples measuring the temperatures of the oil exiting the gears (referred to as oil flingoff thermocouples) were housed in a special cylinder facing the oil stream, as shown in figure 5. The signals from the thermocouples located on the moving shafts were fed out of the transmission through four slipring assemblies. One slipring was mounted on each of the two input modules and transmitted data from the intermediate-speed shaft. A third slipring was mounted on the main module and transmitted data from the main output shaft and planetary carrier. The fourth assembly, an internal slipring mounted inside the main module above the combining bevel gear, transmitted data from the sun gear.

Procedure

The testing consisted of varying the shaft speed or the torque as required to achieve the matrix of test points indicated in table III. With the speeds ranging from 50 to 100 percent of the full rated value, and the torques from 10 to 100 percent, the resulting power ratings ranged from 10 to 100 percent. For each data point the specified speed and torque were set and the transmission was operated at that setting until all temperatures had reached equilibrium, at which time the data were recorded. Equilibrium was assumed to have been achieved when no temperature had changed more than 1 kelvin (2 deg F) in 10 min. The oil inlet temperature was maintained constant at 355 K (180 °F) for all tests. Also, for each test the tail torque was set so that about 15 percent of the transmitted power flowed through the tail drive loop.



(a) As placed in special cylinder.

(b) As mounted on carrier.

Figure 5.—Oil flingoff thermocouple.

Computer Program

The computer program Planetsys (ref. 5 is the users manual) was used to predict the operating characteristics of the UH-60A Black Hawk transmission. Although this program had been used previously to simulate the operation of an OH-58 transmission (ref. 6), there was no experimental verification of the actual planetary bearing temperatures. Further, the OH-58 planetary referenced in that report used

a double-row spherical roller bearing, and the UH-60A uses a single-row spherical roller bearing in its planetary.

Planetsys was developed to simulate the thermomechanical performance of a multistage planetary power transmission. The major function of this program is to compute, for any of the six possible kinematic inversions for a planetary system, the performance characteristics of a planet bearing. This bearing can be either a cylindrical or a spherical roller bearing and can contain one or two rows of rollers. The outer ring can be rigid or flexible.

TABLE III.—NOMINAL
TEST POINTS

Speed, percent of full rated	Torque, percent of full rated	Power, percent of full rated
100	100	100
	75	75
	50	50
	25	25
	10	10
75	100	75
	75	56.2
	50	37.5
	25	18.7
50	100	50
	75	37.5
	50	25

Planetsys is also capable of producing either steady-state or time-transient temperature maps of an axisymmetric mechanical system. This is accomplished by approximating the mechanical system by a number of simple geometric elements, each represented by a node point (ref. 5). Program input includes bearing geometry, bearing material and lubricant properties, and operating conditions (such as load, speed, and ambient temperature). When the program is used for thermal analysis, additional input is required, since all the thermal nodes must be defined. The maximum number of nodes permitted is 100. The nodal system used with Planetsys for the Black Hawk transmission is shown in figure 6. The 39 air and metal nodes are shown in figure 6(a) and the 31 lubricant nodes in figure 6(b).

Results and Discussion

The results of this investigation are presented in the following manner:

(1) First, the experimentally determined temperatures are shown for the high-speed-shaft and intermediate-speed-shaft thermocouple locations.

(2) Second, the steady-state thermal techniques that are used with the Planetsys computer program, including the heat transfer coefficients and constant heat inputs, are explained.

(3) Third, the predicted and measured temperatures for corresponding locations on the planet bearing, the ring gear, the sun gear, the mast bearings, and the outside skin are compared.

Measured Temperatures

The temperatures as measured with the thermocouples located on the bearings on the left and right high-speed input shafts and on the left and right intermediate-speed shafts, plus the temperatures from the oil flingoff thermocouples at

the high-speed gear mesh and the combining gear mesh (fig. 2), are shown in figure 7 for the 100-percent-shaft-speed condition. The highest measured temperature was 405 K (270 °F) for the outer ring of the high-speed roller bearing (bearing 2). As expected, the temperatures for the intermediate-speed shaft bearings were somewhat lower. The inner ring of the tapered roller bearing (bearing 7) was hotter than the outer ring over the range shown. Note that the oil flingoff temperatures showed the same general trend as the other temperatures.

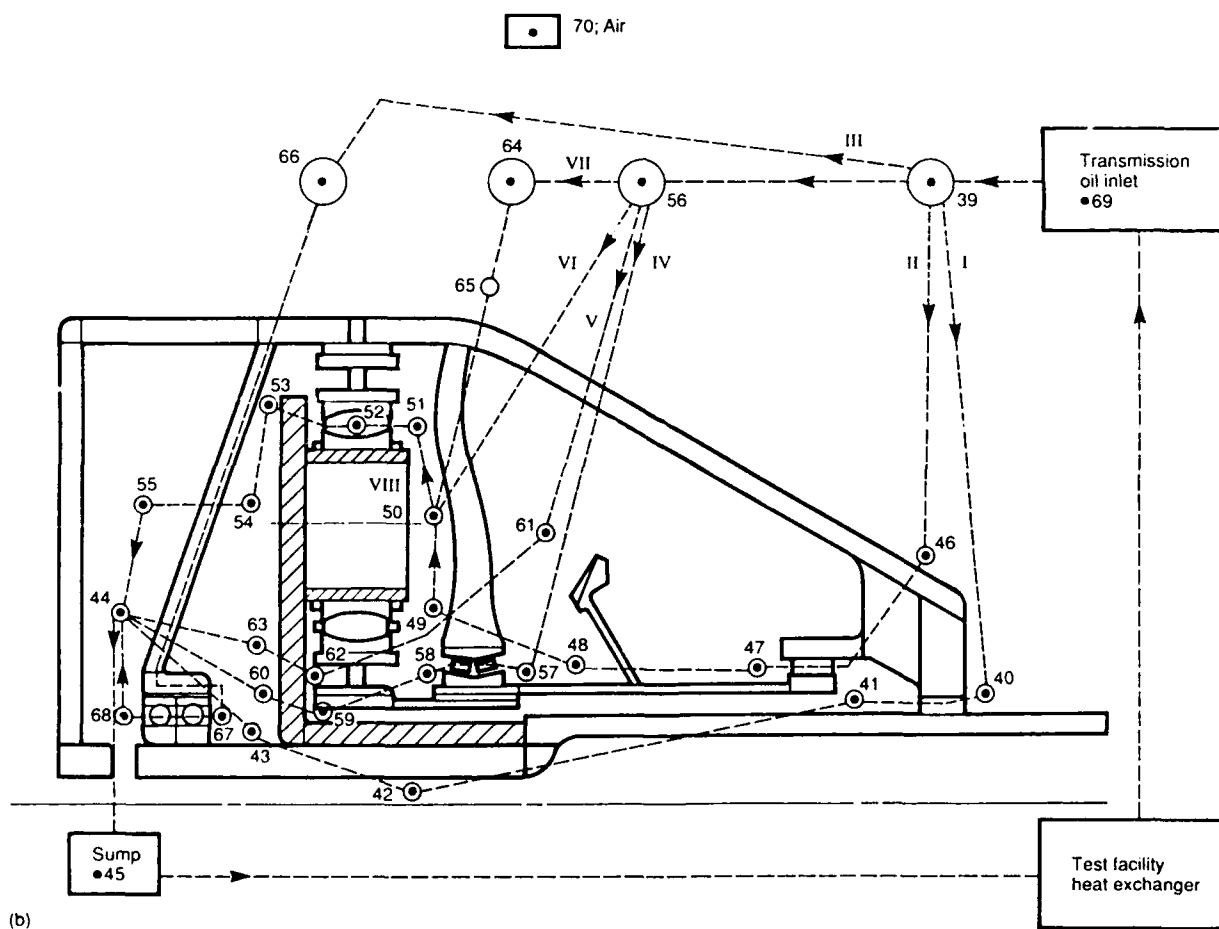
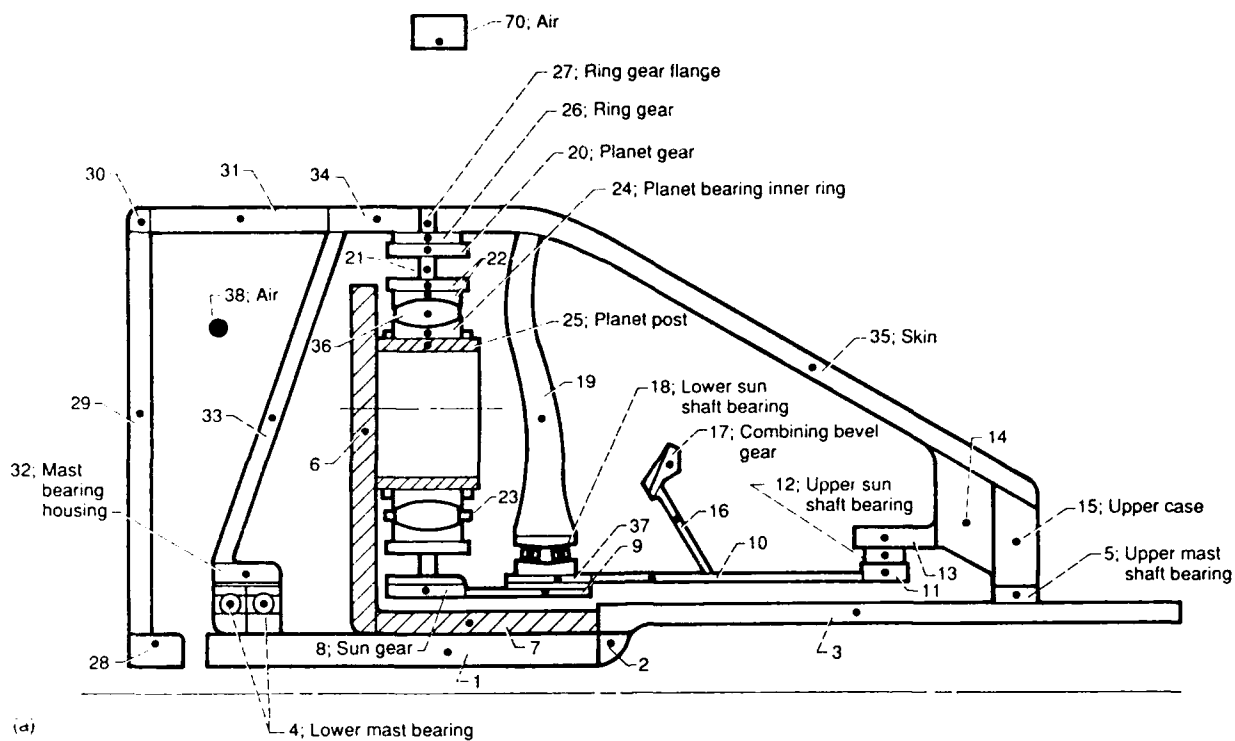
Temperatures measured at the same locations for the 75-percent-shaft-speed condition are shown in figure 8. The values were generally slightly lower than those for 100-percent speed, but with the same general trends. The temperatures at both speeds were a slight function of torque, reflecting the change in the power loss in the gear mesh as the power transmitted through the mesh was changed.

It is interesting to compare the measured gear temperatures with the corresponding oil flingoff temperatures, as shown in figures 7(c) and 8(c) for the bevel gear and in figure 9 for the sun gear. The reasonably good correlation means that one can monitor gear temperatures without having a thermocouple right on the gear.

Thermal Analysis

Using the thermal analysis routines in the Planetsys computer program requires a definition of each nodal area, the type of heat transfer between nodes, and the corresponding heat transfer coefficient. The conduction heat transfer indexes are given in table IV, the convection indexes in table V, and the oil flow indexes in table VI. Some of the conduction indexes were derived for parallel or series heat flow through different materials, as described in reference 5. The convection indexes are also based on the equations presented in reference 5. The flow indexes are the product of the estimated oil volume flow rate times a specific heat of 2175 W sec/kg °C (0.52 Btu/lb °F) times a density of 900 kg/m³ (56 lb/ft³). The total oil flow rate was measured.

The thermal routines also permit input of known values of temperature and constant generated heat. Thus, the oil inlet node (node 69, fig. 6) was set to 355 K (180 °F) and the ambient air node (node 70, fig. 6) to 300 K (80 °F). The constant generated heat values are shown in table VII. These include those for the mast and sun shaft support bearings, the sun shaft bevel gear meshes, and the planetary gear meshes. The heat generation for the support bearings was estimated from data for similar bearings. For each gear mesh it was assumed that one-half of the heat generated went into each gear. And, in each case it was assumed that the heat generated at the mesh was 0.25 percent of the power being transmitted. Note that the value for the planet gear node (node 20, fig. 6(a)) is for a single planet, whereas the values for the sun and ring gear nodes (nodes 8 and 26, fig. 6(a)) are the total for all planets.



(a) Metal and air nodes.
 (b) Oil nodes and flow paths.
 Figure 6.—Nodal system for UH-60A used with Planetsys.

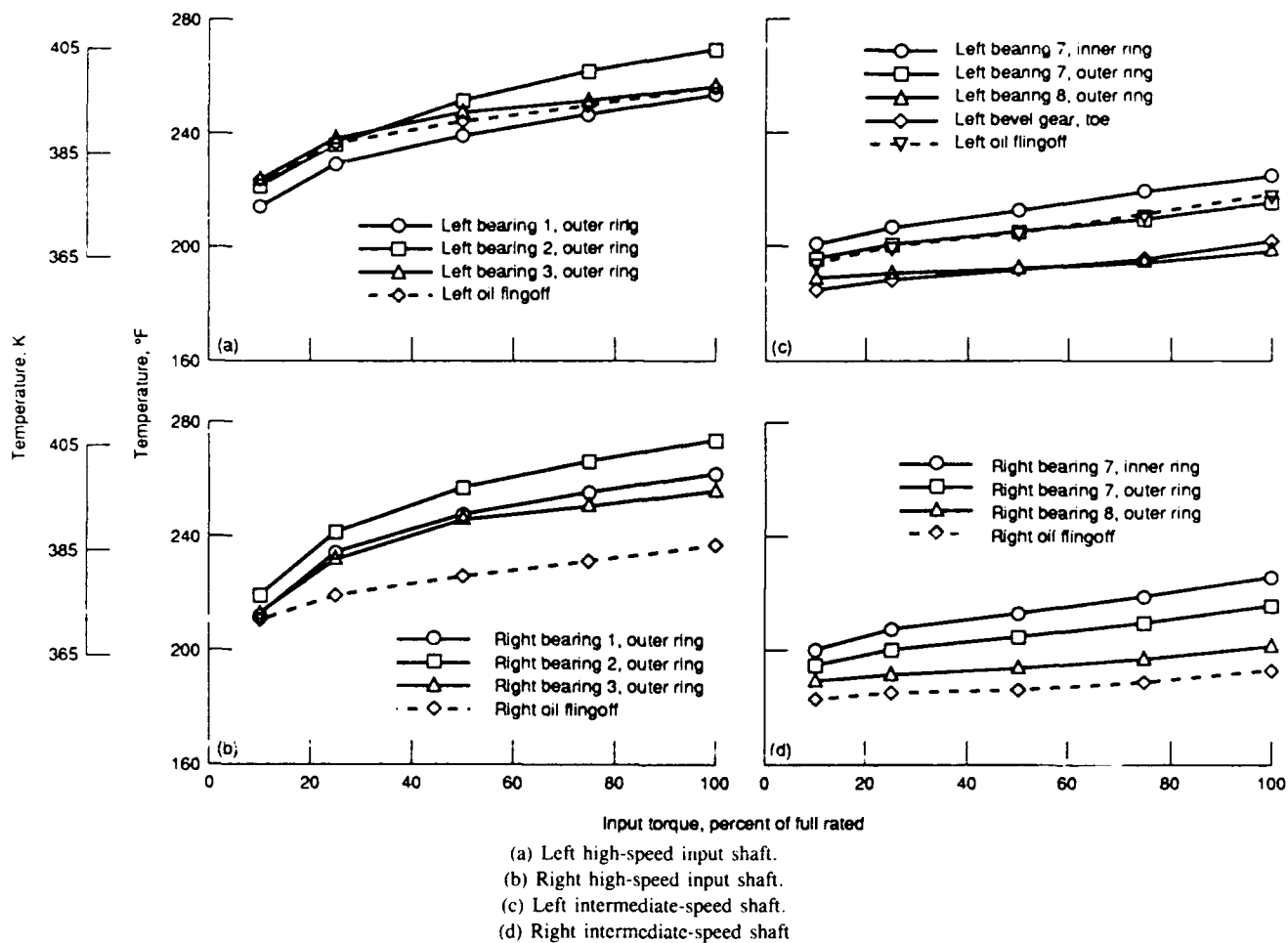


Figure 7.—Measured temperatures of UH-60A transmission as function of input torque for 100 percent of full rated speed.

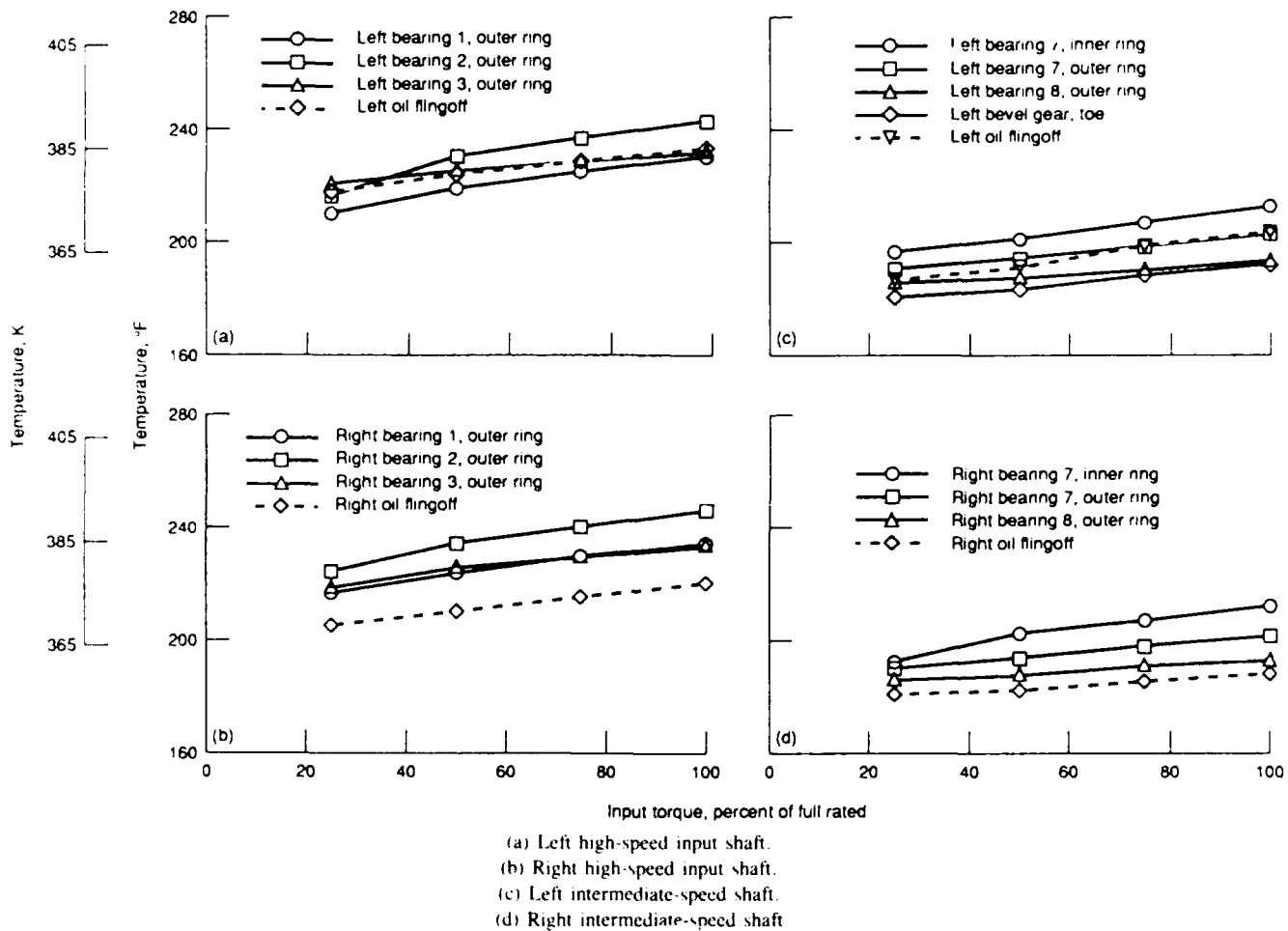


Figure 8 -- Measured temperatures of UH-60A transmission as function of input torque for 75 percent of full rated speed.

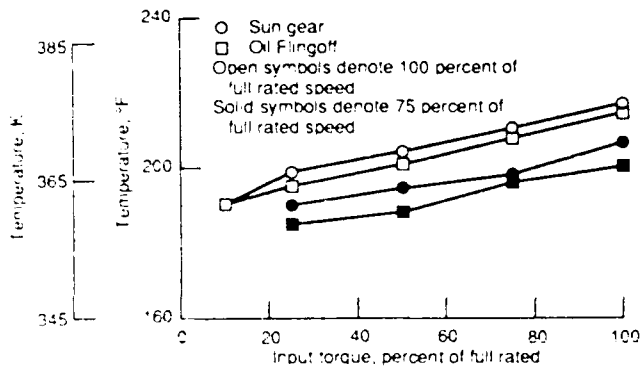


Figure 9.—Comparison of UH 60A transmission sun gear and oil flingoff temperatures. Speed, 75 and 100 percent of full rated.

TABLE IV.—CONDUCTION HEAT TRANSFER INDEXES

Index	Description	Value, $W/m^2\ ^\circ C$
1	Bearings (steel)	45.0
2	Shafts and gears (steel)	38.0
3	Housing (magnesium)	85.0
4	Carrier (titanium)	9.0
5	Shaft to carrier (radial)	13.6
6	Shaft to carrier (axial)	19.3
7	King gear to lower case	68.9
8	Planet post to inner ring	15.7
9	Sun bearing to upper shell	78.4

TABLE V.—CONVECTION HEAT TRANSFER INDEXES

Index	Description	Value, $W/m^2\ ^\circ C$
21	Bearing to oil	786.0
22	Bearing to air	32.7
23	Mast shaft, bearings to air	13.8
24	Sun shaft, bearings to air	46.5
25	Bevel gear to air	59.9
26	Carrier, inside walls to air	20.2
27	Housing to outside air	20.0
28	Case bottom to outside air	7.0

TABLE VI.—FLUID FLOW INDEXES

Flow path ^a	Description	Value, $W/m^2\ ^\circ C$
I	Upper mast shaft bearing; sun gear	30.8
II	Upper sun shaft bearings; bevel gear	61.6
III	Lower mast bearing	30.8
IV	Lower sun shaft bearing; sun gear	61.6
V	Sun gear	61.6
VI	Planet bearing (under ring)	246.2
VII	Bevel gear	738.6
VIII	Total to all planet bearings	1046.4

^aSee fig. 6(a).

TABLE VII.—CONSTANT HEAT GENERATION INPUT

Node (fig. 6)	Description	Value assigned to node, ^a W
4	Lower mast bearings	75
5	Upper mast bearing	37
8	Sun gear	2242
12	Upper sun shaft bearing	201
17	Combining bevel gear	3032
18	Lower sun shaft bearing	403
20	Planetary gear	897
26	Ring gear	2242

^aAt 100 percent power.

Comparison of Predicted and Measured Values

Calculations were then made by using the Planetsys program and the aforementioned input data for both the 100- and 75-percent-speed test points as shown in table III. The resulting predicted temperatures are compared in figures 10 to 12, with values indicated by the thermocouple located near the corresponding node. In figure 10(a) the planetary bearing inner-ring temperatures are compared with predicted values for the inner ring and planet post (nodes 24 and 25, fig. 6(a))

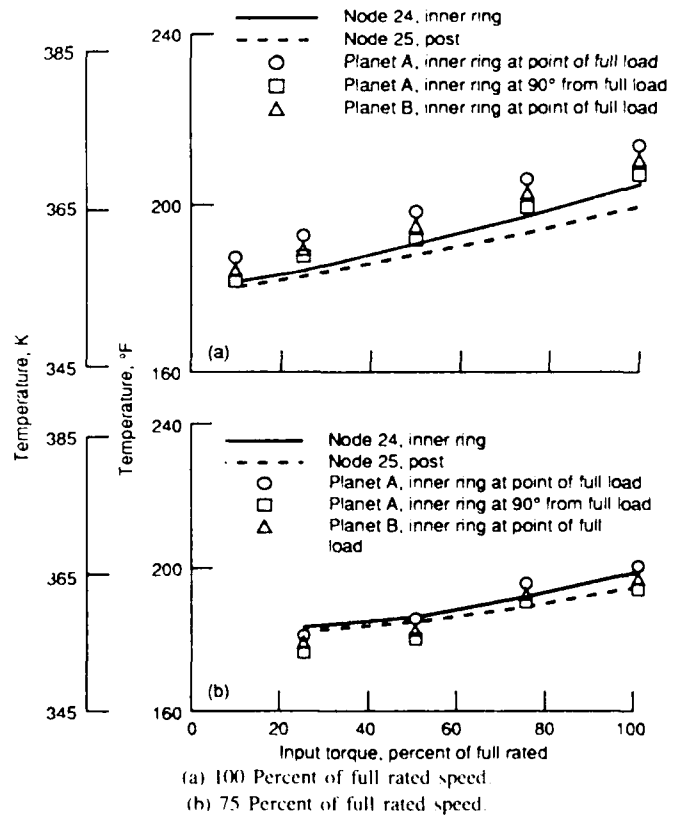


Figure 10.—Comparison of predicted and measured temperatures of UH 60A transmission planet bearing as function of input torque

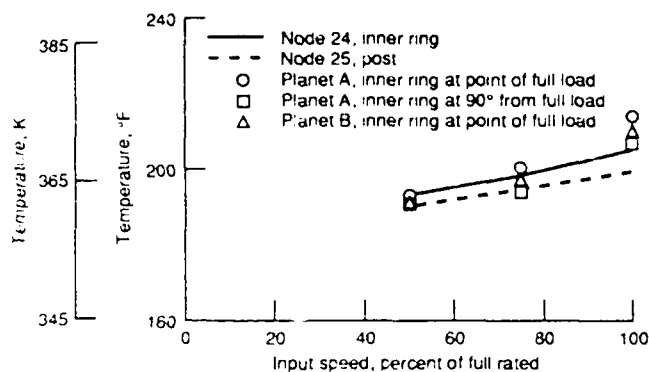


Figure 11—Comparison of predicted and measured temperatures of UH-60A transmission planet bearing as function of shaft speed for 100 percent of full rated torque.

for the 100-percent-speed runs. Temperatures were measured on two planet bearings (A and B), at two locations. One thermocouple was placed at a point of maximum applied load, and the other was located 90° away. The predicted temperatures compared very well with the experimental data, having the same trends and averaging about 4 kelvin (8 deg F) lower

than the highest measured value. A similar comparison is made in figure 10(b) for the 75-percent-speed runs. The predicted values again compared very well with the measured temperature, being quite close at the higher torques and slightly high (about 2 kelvin; 4 deg F) at the lower torques. Finally, these same locations are compared in figure 11 as a function of speed for the 100-percent-torque condition. Again, the predicted temperatures compared very well.

The predicted nodal temperatures can be compared with corresponding thermocouple data at a number of other locations, including the sun gear, the ring gear, the mast bearing outer ring, and the upper case outer skin. (The thermocouple locations are indicated in figure 2 and the nodes in figure 6.) These comparisons are shown in figure 12 for the 100-percent-speed condition. The predicted results generally compared very well with the measured values; the biggest difference was in the sun gear, where the predicted temperature (node 8) was 10 kelvin (18 deg F) higher than the measured value at the 100-percent-torque point (fig. 12(a)). Node 26 (fig. 12(b)) compared well with the measured ring gear temperature; node 27, which was predicted to be much cooler, was actually a flange of the ring gear sandwiched

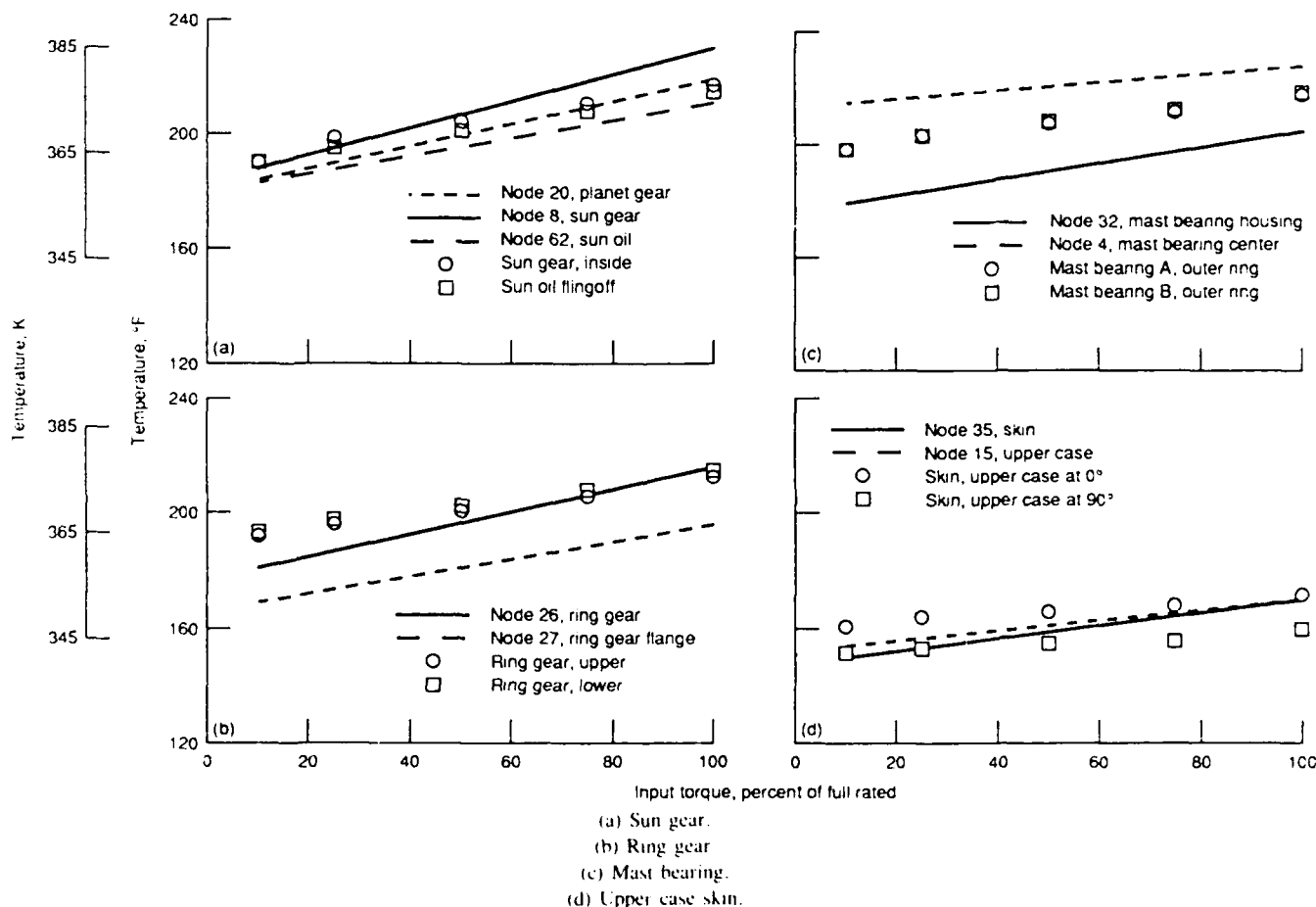


Figure 12—Comparison of predicted and measured temperatures of UH-60A transmission as function of input torque for 100 percent of full rated speed

between the upper and lower cases, which were also much cooler (see upper case skin temperatures in fig. 12(d)). Note in figure 12(c) that the measured mast bearing outer ring temperatures fell nicely between the calculated values for the bearing center and the bearing housing. The calculated skin temperatures (fig. 12(d)) fell between the two measured values over the range of input torque. Both skin thermocouples were on the upper case; one thermocouple was located on the front side (opposite the tail takeoff), and the other was positioned 90° away.

Concluding Remarks

Usually an important parameter in calculating rolling-element bearing operating characteristics is the amount of lubricant in the bearing cavity. Reference 6, however, notes that, because of the relatively low speeds of planetary bearings, the temperatures change very little when this quantity is varied over a wide range. Thus, for the present calculations the volume percentage of the bearing cavity that was occupied by the lubricant was kept at 2 percent for all cases.

The predicted values of temperature generally compared well with the measured values, most of them being within 5 kelvin (9 deg F). This was less than a 2-percent difference, based on absolute temperatures. It may therefore be stated that the computer program Planetsys can predict planetary bearing temperatures and other transmission component temperatures and that the assumptions made for the program were reasonably correct.

Summary of Results

The UH-60A Black Hawk helicopter transmission was internally instrumented and operated over a range of shaft speeds and input torques. Temperatures of the bearings and gears were measured. The computer program Planetsys was used to simulate the thermal performance of this transmission. The calculated temperatures were then compared with the

corresponding measured values. The following results were obtained:

1. The highest measured temperature, 405 K (270 °F), was that of the outer ring of bearing 2, a roller bearing on the high-speed input shaft, at the 100-percent-power (2109 kW; 2828 hp) condition.
2. The oil flingoff temperatures correlated well enough with measured gear temperatures to suggest that gear temperatures can be monitored with this technique.
3. The temperatures predicted for the single-row spherical roller bearing by the program Planetsys compared well with the measured temperatures, averaging about 4 kelvin (8 deg F) lower than the highest measured value.
4. The Planetsys predicted values for the sun gear were about 10 kelvin (18 deg F) high at the 100-percent power condition.

Lewis Reserch Center
National Aeronautics and Space Admsinistration
Cleveland, Ohio, February 2, 1989

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