Non-Metallic Bearing Friction Test Program for Quarter-Turn Valves

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Abstract

Knowledge of torque requirements for quarter-turn valves has become increasingly important in U.S. nuclear plants in recent years. Valve and actuator efficiency has become paramount to ensure that safety related valves will perform reliably under design basis flow and differential pressure conditions. To address this issue, EPRI developed the MOV Performance Prediction Methodology (Reference 1) model to evaluate butterfly valve torque requirements. This model is currently applicable only to valves with metallic bearing materials. For those valves with non-metallic bearings, only unvalidated bearing friction values are available.

To address this industry need, EPRI contracted with Kalsi Engineering to design a test fixture to measure friction coefficients of non-metallic bearings under various plant conditions. The fixture captures the peak and running torque required to rotate a 17-4 PH stainless steel shaft within several non-metallic bearing sets. The test scope includes variations in bearing load, fluid media, temperature and dwell time under load.

The test program is conducted under the Kalsi Engineering Quality Assurance Program that meets 10CFR50 Appendix B requirements. The test data will be analyzed and compiled into a report that ultimately defines validated friction coefficients for non-metallic bearings of different materials. This information will benefit utilities by having accurate friction coefficients used in torque calculations that will ensure reliable valve operation while avoiding valve/actuator modifications and unnecessary technical effort. In addition, data will be used to extend the applicability of the PPM butterfly valve model to valves with non-metallic bearing materials.

Background

The ability of a quarter-turn valve to open and close can be essential to the safe operation of a nuclear power plant. The existing EPRI MOV PPM model is not applicable for use in defining torque requirements for butterfly valves with non-metallic bearing materials. To address this shortcoming, a test program has been initiated to provide data required to define design coefficient values for non-metallic bearing materials typically found in nuclear service. To date, only 1 of the 5 non-metallic bearing materials has been tested. This paper describes the test system, test scope and procedures and provides preliminary results and conclusions.

Test Scope

A survey (Reference 2) conducted in 1999 by EPRI identified that the most commonly used non-metallic bearings in the nuclear industry are Teflon, nylon, and PEEK. The materials chosen for this test include the materials identified by EPRI plus Nomex and Duralon. Each bearing type will be evaluated under the same test conditions. Table 1 describes the test matrix for one bearing specimen. Initial testing under zero contact stress is performed to determine the parasitic torque in the fixture and to certify that the test equipment, instrumentation, and data acquisition are performing as intended. Each reciprocal stroke is comprised of rotating the shaft 90 degrees in each direction.

Test Number	Test Fluid	Contact Stress psi	Fluid Temp °F	No of Reciprocal Strokes	Dwell Time Under Load and No Rotation
1T-1	Air	0	Ambient	10	0 hrs
1T-2	Air	1,000	Ambient	10	0 hrs
1T-3	Air	2,000	Ambient	10	0 hrs
1T-4	Air	2,000	Ambient	1	2 hrs
1T-5	Air	2,000	Ambient	1	4 hrs
1T-6	Air	2,000	Ambient	1	16 hrs
1T-7	Air	2,000	Ambient	1	72 hrs
2T-1	Distilled Water	0	Ambient	10	0 hrs
2T-2	Distilled Water	1,000	Ambient	10	0 hrs
2T-3	Distilled Water	2,000	Ambient	10	0 hrs
3T-1	Distilled Water	0	200	10	0 hrs
3T-2	Distilled Water	1,000	200	10	0 hrs
3T-3	Distilled Water	2,000	200	10	0 hrs
3T-4	Distilled Water	2,000	200	1	16 hrs

Table 1.

Test Matrix for Each Bearing Material

Test Fixture

The test fixture used to perform the tests is shown in Figure 1 and Figure 2. The body of the test fixture was designed with the capability of applying an adjustable lateral (upward) load while a shaft rotates within the non-metallic bearings such as in typical butterfly, ball, and eccentric plug valves. The lateral load is applied to the center portion of the shaft. The length and diameter of the shaft are designed to create the shaft bending and associated bearing edge loading typical of quarter turn valves. The lateral load to the shaft is applied through a roller bearing assembly, which has significantly lower friction (~ 10^{-2}) than the test bearings in order to minimize measurement errors associated with parasitic torque.

The test fixture can accommodate testing with air and clean cold or hot water as the fluid medium. When testing with water, two low-friction Nitroxile U-seals installed between the bearing housing and shaft prevent water migration into the bearing assembly.

The water is heated using immersion heaters located below the shaft. To prevent fluid contamination due to metal oxidation, all the wetted surfaces were nickel plated or made from stainless steel.

The shaft is turned using an electric actuator, which provides constant and smooth 90-degree rotation at a speed of 0.5 revolutions per minute.

Fixture and Specimen Cleanliness

Prior to assembly the fixture and specimen are cleaned to remove any fluid that might affect the results of the test. Metallic parts are washed with a trisodium phosphate solution in distilled water, rinsed with distilled water, dried, wiped with acetone, and then wiped with alcohol. Non-metallic parts are washed in distilled water, rinsed, and dried.

Instrumentation and Data Acquisition

The lateral load is measured using a high accuracy compression load cell mounted directly above the shaft. A rotary torque cell mounted between the actuator and the shaft measures the torque applied to the shaft by the actuator. A J-type thermocouple is mounted in the fixture to measure and control the temperature. Table 2 gives the overall measurement accuracy requirements imposed on the acquired data.

Parameter	Range	Accuracy Requirement
Temperature	32°F to 300°F	Within +/- 5°F
Load	0 to 20,000 lbs	Within +/- 0.5% FS
Torque	0 to 250 ft-lb	Within +/- 1% FS

Table 2.

Overall Measurement Accuracy Requirements

Data are collected using ScadaPro software and DataScan and Computerboard data acquisition boards. Data are collected at three different sample rates.

1. One sample every 10 seconds: This sampling rate is applied throughout the entire duration of each test sequence in order to capture the entire history of the test.

These data are only used to provide an overall view of the test sequence and not for data reduction.

2. 10 samples per second: This sample rate is used any time the shaft was rotated. These data are used to calculate the average coefficient of friction during the running portion of the stroke.
3. 1,000 samples per second. This sample rate is used to capture peak torque values to determine the breakaway coefficient of friction values. These data are captured during the initial rotation (~ 5 seconds) of the shaft at the beginning of each test sequence and during some of the torque reversals within a test sequence. This sample rate is also used during the initial rotation after the dwell period.

The equation used to determine the coefficient of friction is of the form

$$\mu = \frac{2T}{LD}$$

where μ is the friction coefficient, T is the applied torque, L is the applied lateral load, and D is the stem diameter of the test bearing.

Discussion of Results and Conclusions

Figure 3 shows a typical breakout friction trace plotted with the corresponding torque. Figure 4 depicts a typical 90-degree shaft rotation. Note that the torque and friction pass through zero as the torque from the preceding rotation is relaxed and then builds in the direction of rotation. Figure 5 shows an example of a typical test sequence including 10 reciprocal strokes.

The results of the first series of tests show that the bearing friction coefficient is dependent on rotation direction, fluid type, contact stress level, dwell time, and temperature. The following describes the observed dependencies.

Rotation direction: The bearings exhibited a distinct and repeatable difference in coefficient of friction when the shaft was rotated clockwise versus counterclockwise as shown in Figures 5 and 6. This is attributed to surface finish directionality caused by the shaft manufacturing process.

Fluid type: The tests performed using air and distilled water shows that the breakout and running coefficients of friction are generally higher in air than in water as shown in Figure 6.

Breakout friction: Data acquired at high sample rates shows that the breakout coefficient of friction is generally higher than the average running coefficient of friction. See Figure 6. In addition, it was noted that the highest valve of breakout friction occurred in air during the first stroke of the test program (with no significant dwell time at load prior to the test).

Contact stress level: Two lateral (upward) loads were applied to the bearings to yield approximately 1,000 psi and 2,000 psi. The coefficient of friction under the lower contact stress was somewhat greater than under the higher contact stress level as seen in Figure 7. This difference was observed to be largest when the bearings were tested in ambient air and hot water.

Dwell time: Tests were performed using four different dwell times at room temperature and one dwell time under elevated temperatures. As shown in Figure 8, the breakout coefficient of friction at room temperature increased slightly from the 0-hour dwell time test to 4-hour dwell test and then remained relatively constant thereafter. At the elevated temperature, the breakout torque increased ~100% after the 16-hour dwell time. This test is intended to investigate the extent to which the coefficient of friction can increase when a valve remains closed under differential pressure.

Quality Assurance

All testing activities were performed under the Kalsi Engineering Quality Assurance Program, which satisfies 10CFR50 Appendix B requirements.

References

1. 1006206: *EPRI MOV Performance Prediction Program: Performance Prediction Methodology (PPM) Version 3.0 User Manual and Implementation Guide*, Electric Power Research Institute, Palo Alto, CA, December 2001.
2. TR-113561: *EPRI Performance Prediction Program: Friction Coefficients for Non-Metallic Butterfly Valve Bearing Materials*, Electric Power Research Institute, Palo Alto, CA, December 1999.

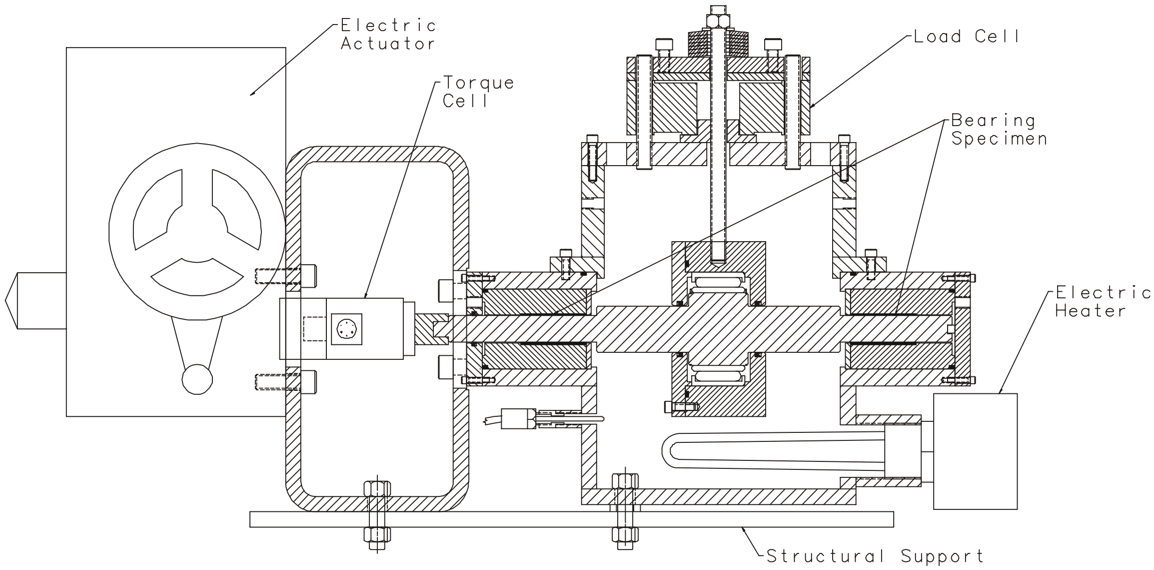

Figure 1.

Bearing Test Fixture

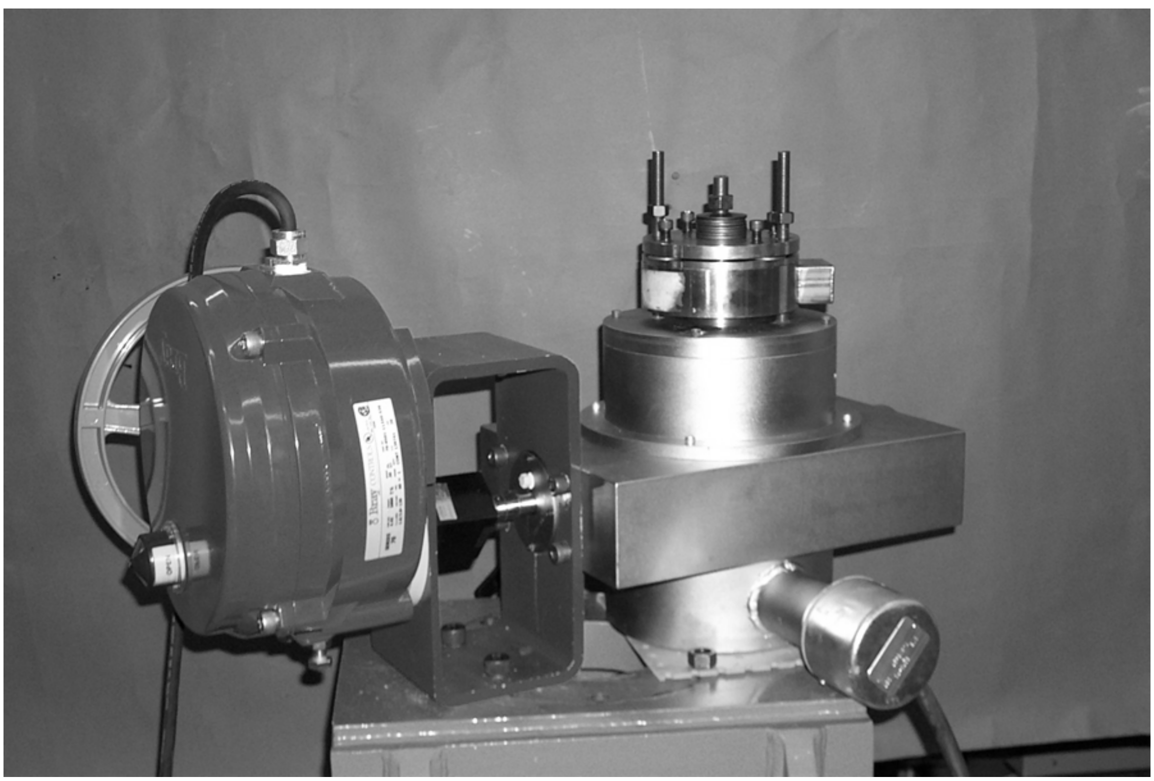

Figure 2.

Photo of Bearing Test Fixture

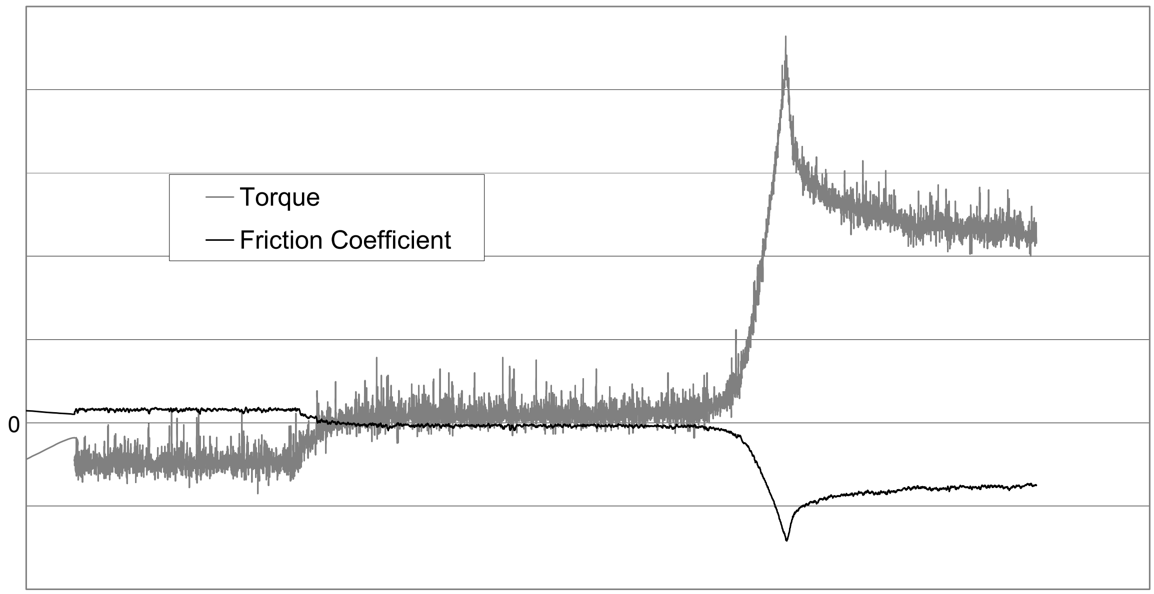

Figure 3.

Breakout torque is significantly larger than running torque.

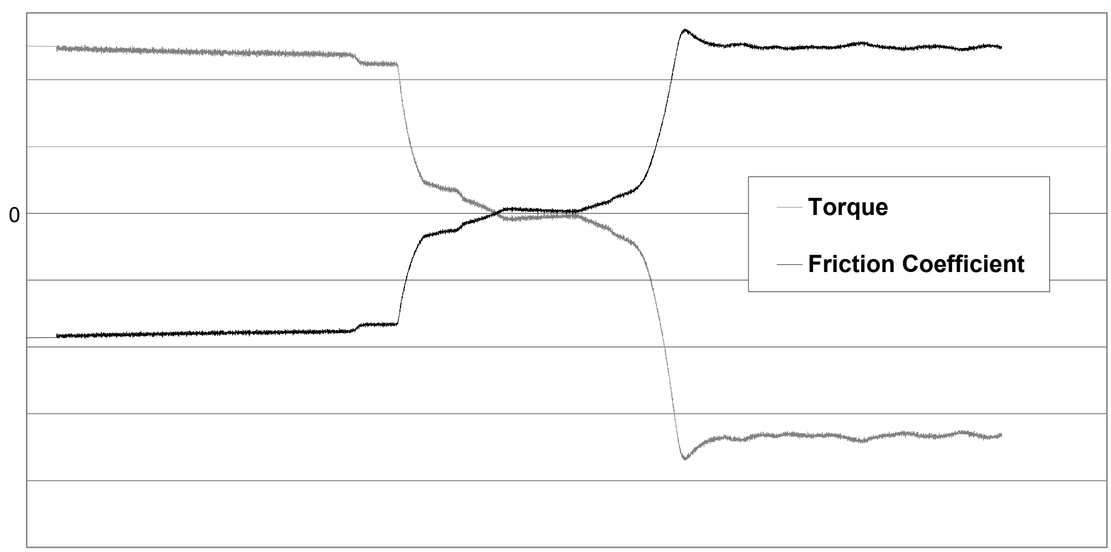

Figure 4.

Rotation reversal reflects zero torque just prior to motion.

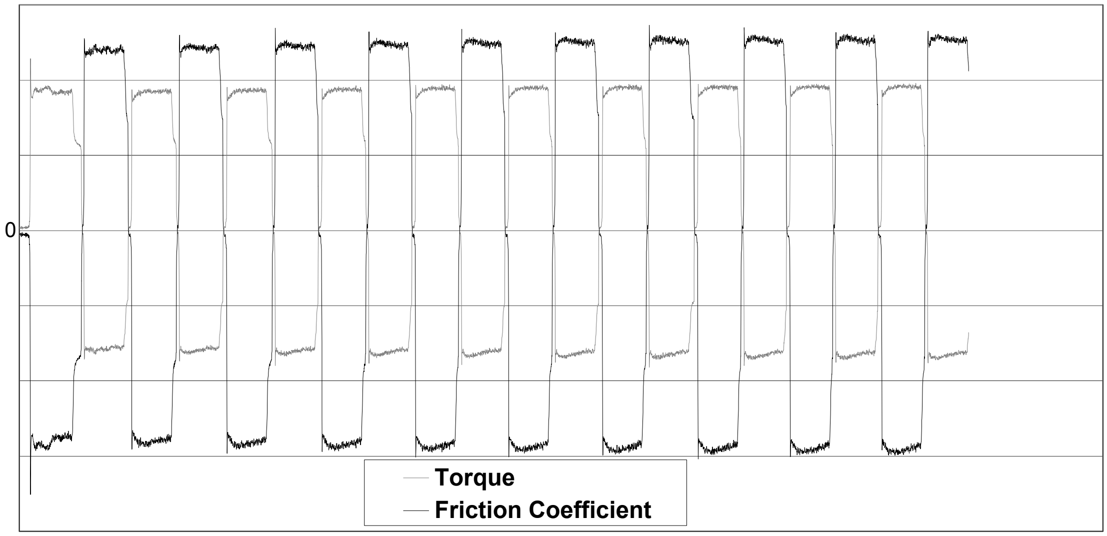

Figure 5.

Test data for 10 cycles shows that friction is somewhat higher in one direction than the other under the same load.

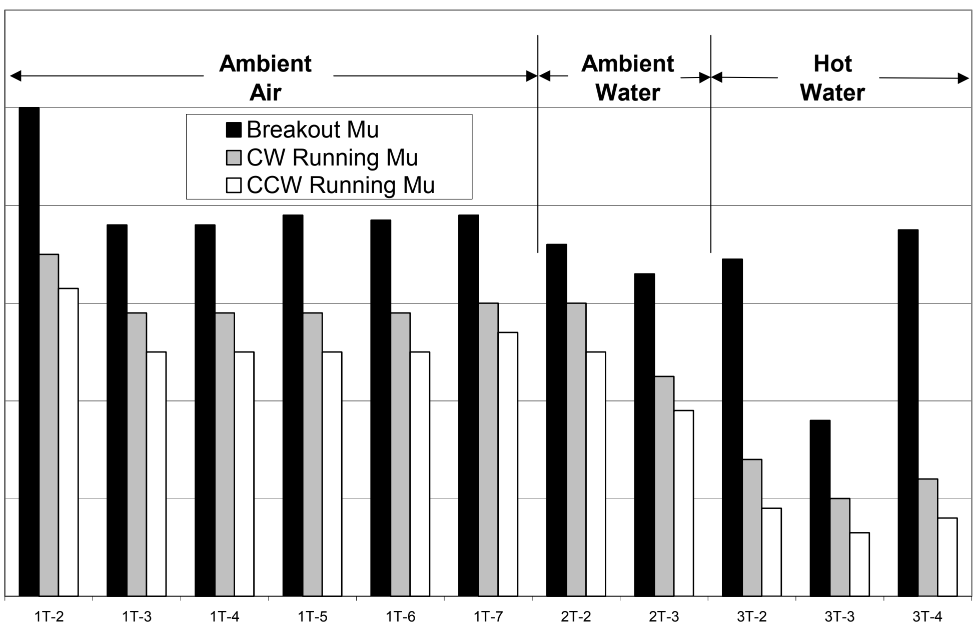

Figure 6.

Friction coefficients are generally higher in air than in water and vary depending on the stroke direction, temperature, and dwell time.

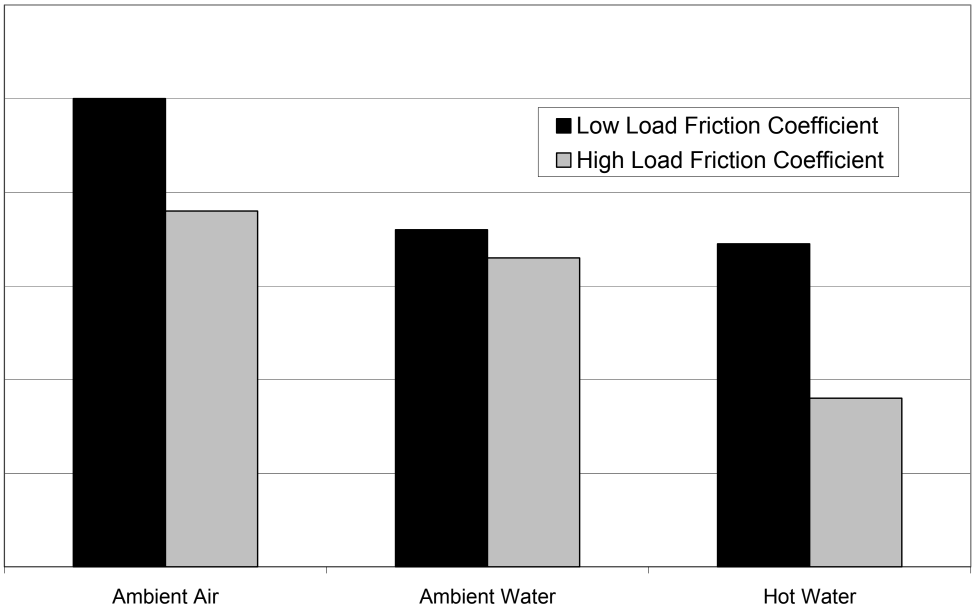

Figure 7.

Breakout friction coefficients are higher with a low contact stress (1000 psi) than with a high contact stress (2000 psi).

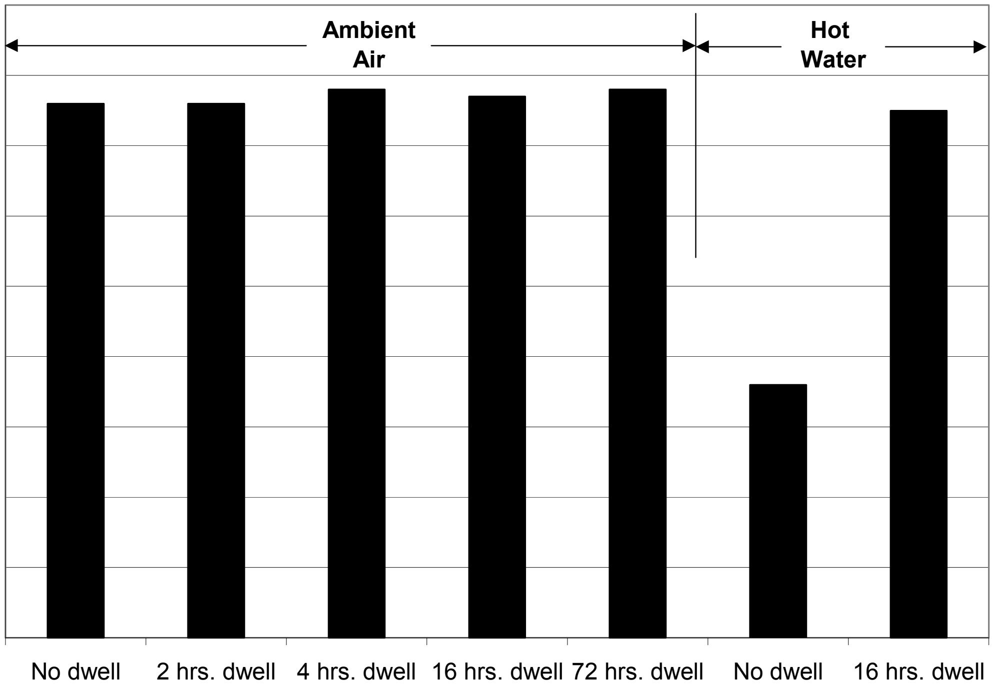

Figure 8.

Dwell times have little affect on breakout friction coefficients in ambient air, but have a large effect in hot water.