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The University of Tennessee
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THEORETICAL AND EXPERIMENTAL STUDIES OF VISCO TYPE AND BUFFERED SHAFT SEALS

Semi-annual Progress Report
October 15, 1966 - April 15, 1967

by

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ABSTRACT

This report, the ninth in a series, outlines the progress made on the investigation of the viscoseal and the buffered seal during the period October 15, 1966 - April 15, 1967. The study is being conducted under Research Grant NsG-587 for the National Aeronautics and Space Administration.

The experimental performances of five grooved shaft viscoseals are presented. The results of these tests, in addition to those for fifteen geometries previously reported, indicate that reasonably precise design estimates can be made for the performance of the grooved shaft viscoseal for both the laminar and turbulent regions.

I. INTRODUCTION

An investigation embracing the theory and performance of the viscoseal under laminar and turbulent conditions was initiated on April 15, 1964 at The University of Tennessee in the Department of Mechanical and Aerospace Engineering. The investigation is being conducted for The National Aeronautics and Space Administration under Research Grant NsG-587, and this report presents the progress of the investigation for the period - October 15, 1966 through April 15, 1967.

A critical review of all available literature led to the identification of a number of problem areas requiring study in support of the United States' space effort (1)[1]. Other reports in this series describe the program objectives (2), the design and construction of the experimental facility (3), the concentric laminar analysis of the viscoseal (4), the concentric turbulent analysis of the viscoseal (5), and results of experiments with the first sixteen viscoseal geometries (6,7). This report includes the experimental results obtained for five additional viscoseal geometries.

II. OBJECTIVES

The research program presented in references (2) and (3) was modified, as described in reference (6), because of the phenomenon of gas ingestion which was encountered during early experimental observations. In addition the scope of the research program was expanded to embrace the buffered bushing seal.

III. ACTIVITIES

Experimental test work continued on the basic viscoseal experiment design and tests, described in a subsequent section of this report, were completed on five viscoseal geometries.

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[1]Numbers in parenthesis refer to similarly numbered references listed at the end of this report.

A paper, entitled "The Turbulent Viscoseal - Theory and Experiment," was delivered at the 3rd International Conference on Fluid Sealing at Cambridge, England. While in England the project director visited the dynamic seal test facilities at the British Hydromechanics Research Association laboratory in Cranfield. The director also visited the laboratories of the Technical Institute T.N.O. and the Technical University at Delft, Holland. Conferences were held with the personnel studying the viscoseal and arrangements have been made for a mutual exchange of non-proprietary experimental data. The cost for the visit in Europe was borne by the University of Tennessee.

Viscoseal experiments made to date have utilized grooved shaft seals with a smooth housing restrained by eight hydrostatic bearing pads (3). A design modification was made to allow simple installation of a ball bearing supported test housing which could be grooved or smooth (7). Shop fabrication was completed on two grooved housings, one smooth housing, test shafts, and the supporting bearing assembly. The grooved housings have thread geometries duplicating those of spindles No. 3 and No. 6. The new test sleeves, which are shown in Figure 1, are scheduled to be installed as soon as testing has been completed on spindle No. 8B.

A hydrodynamic journal bearing tester has been modified to permit the use of a grooved bushing in order to experimentally determine the radial bearing capacity of a viscoseal (7). This test unit was reworked and given an operational shake-down test. The test bearings have been completed and new instrumentation has been installed and is being calibrated.

A study of the phenomenon of gas ingestion has continued. However, the approach to the problem was modified in order to unify the cooperative efforts of personnel on this campus and those at the NASA Lewis Research Center. As previously reported (7), the study was being approached with the viewpoint that the mechanism of gas ingestion could be better understood if the velocity and pressure distributions were determined by an approximate solution of the Navier-Stokes equations for three-dimensional flow in the viscoseal in a region which is free of the influence of end effects. This solution should be helpful in explaining how the gas, once it has become entrained in the liquid, is transported through the liquid toward the high pressure end of the seal. It would also serve as a guide in establishing a flow model for determining the dynamics in the region of the gas-liquid interface, where the mixing of the gas and liquid occurs.

In conferences with NASA personnel in January and February, 1967, it was suggested that the overall program on viscoseal studies could be advanced by initiating a study of surface tension effects on the stability of an interface, while work continues simultaneously by NASA personnel on the three-dimensional flow problem at a considerable distance from the interface. Accordingly, a study of the literature on surface effects has been started, and an experimental apparatus has been designed to provide a visual and photographic study, initially for a rather simple geometry, of a gas-liquid interface or a liquid-liquid interface.

The experimental facility, which is under construction in the Mechanical and Aerospace Engineering Department shop, is to be used to study an interface in an annular space between a rotating shaft and a stationary, transparent, cylindrical housing. Attention will be focused on surface tension and other factors which may affect the stability of the interface. While the study will be of a basic nature, the overall objective will be to provide information which might be helpful in gaining a better understanding of the mechanism of gas ingestion in viscoseals, and hopefully to establish criteria for design modifications to reduce or eliminate the gas ingestion problem.

The test apparatus, a sketch of which is shown in Figure 2, was designed to provide flexibility. Cylinder diameters from two to eight inches may be used. Clearances will be 0.002 inches or more and the eccentricity ratio may be varied. The shaft can be rotated at speeds from 225 to 3600 rpm, providing surface velocities ranging from 2 to 125 feet per second. The frame of the apparatus may be rotated, during operation, so that the shaft axis can be tilted through an angle of 180° from the vertical position. The equipment was designed to accommodate an axial pressure difference of 100 psi. This will permit the apparatus to operate as a viscoseal with a shaft diameter up to eight inches while using a liquid with a viscosity up to ten times that of water.

Present plans are to conduct initial tests with smooth (ungrooved) surfaces in the cylinder and on the shaft. The shaft speed will be varied, and the angle between the shaft axis and the vertical will be varied for a given shaft speed. Different eccentricity ratios will also be used.

It is anticipated that axial grooves of rectangular cross-section will be cut into the surface of the cylindrical housing for the next series of tests. Progressively more complex surface geometries are contemplated for subsequent tests. Geometries which lend themselves to a theoretical analysis will be favored, although experimental tests will be conducted using geometries for which mathematically tractable models do not seem feasible if it appears that useful information might be gained from using such configurations. The choice of flow model will be dictated by experience gained as testing and analysis proceeds and by a long range goal of analyzing the dynamics of the fluid flow in the region of the gas-liquid interface of a viscoseal.

The apparatus shown in Figure 2 was also designed to permit the study of the influence of parameters other than surface tension on viscoseal performance. One factor of immediate interest is the effect of the "characteristic length" (8) on the start of the transition from laminar to turbulent operation. In reference (5) the sealing coefficient for the viscoseal was obtained, for laminar and turbulent operation, as:

$$\Lambda = \frac{6\mu UL}{c^2 \Delta P} = K_4(I_1 + I_2)/I_4 + K_5 I_3/I_4 , \qquad (1)$$

where I_1, I_2, I_3, and I_4 are functions of the seal geometric parameters α, β, and γ and K_4 and K_5 are functions of the Reynolds number and geometry. Under laminar conditions K_4 and K_5 become unity and equation (1) reduces to the laminar sealing coefficient equation:

$$\Lambda = \frac{6\mu UL}{c^2 \Delta p} = \frac{\beta^3(1+t^2) + t^2\gamma\ (1-\gamma)(\beta^3-1)^2}{t\gamma\ (1-\gamma)(\beta^3-1)(\beta-1)} \tag{2}$$

In reference (5) the onset of turbulence was evaluated at a critical Reynolds number defined as:

$$Re_{crit} = 41.1 \left[\frac{D/2}{(1-\gamma)c + \gamma\beta c}\right]^{1/2} . \tag{3}$$

However, in reference (8), which includes inertia effects in the analysis, it is deduced that the transition to turbulent operation begins at $Re^* \simeq 1$. Re^* is a modified Reynolds number defined as:

$$Re^* = Re_c(c/L'), \text{ if } c/L' < 1. \tag{4}$$

Where,

Re_c = clearance Reynolds number = $Uc\,\rho/\mu$

L' = characteristic length = $\pi D/n$

The characteristic length is the length, in the direction of motion, of the land-groove pair. The term (c/L') corresponds to $(cn/\pi D)$ in which n is the number of land-groove pairs.

Since most experimental studies of the viscoseal have utilized rotor diameters of 2" or less with a limited number of thread starts, the values of L' referred to in reference (8) have varied over a limited range.

Figure 3 presents the theoretical performance of a viscoseal based on reference (5). It will be noted that equation (1) yields the same curve for various numbers of thread starts. Also shown in Figure 3 is the variation in the projected performance when the equation (1) is modified to include the concept introduced in reference

(8). Consequently, it is planned to use the test apparatus shown in Figure 2 to test three viscoseals having the same values of D, c, α, β, and γ but having various values of the characteristic length L'. Other studies planned include the effect of the orientation of the seal in the gravitational field on the point of first appearance of the "seal break" and on seal scavenging.

During this period progress on the study of the buffered bushing seal was limited and consisted of flow analysis of the turbulent flow through annuli with moving boundaries and the design of experimental apparatus and associated instrumentation.

In March 1967 Mr. Thomas Scott left the project and his assignment will be assumed by Mr. Roger Shier upon his graduation from the University of Tennessee in June. In September Mr. Edison Picklesimer, Jr., who recently obtained his Master of Science Degree from Georgia Institute of Technology, will join the project.

IV. EXPERIMENTAL RESULTS

During this reporting period test runs were completed on spindles 8, 9A, 9B, 10A, and 10B. The dimensions and geometric parameters of these test spindles are listed in Table 1.

The arrangement of the test apparatus, the experimental procedure, manner of data reduction, and the experimental results obtained for test seals 1, 2, 3, 4, 2B, 3B, 4B, 5, 6, 7, 5D, 6D, 7D, 9, and 10 were previously reported (5,6,7). The experimental sealing coefficient, dissipation function, and friction parameter obtained for test seals, 8, 9, 9A, 9B, 10, 10A, and 10B are shown graphically in Figures 4 through 12 in comparison with the theoretical coefficients (solid lines) obtained by the method outlined in reference (5).

The experimental results presented in this report were obtained with test spindles having small clearance dams at each end of the grooved section of the viscoseal which were arranged as shown in Figure 3 of reference (7). The experimental sealing coefficients for

TABLE I

DIMENSIONS AND GEOMETRIC PARAMETERS OF TEST SEALS

Seal No.	D	c	h	a	b	α	β	γ
8	1.2455	0.0035	0.0205	0.2451	0.1129	20.58	6.86	0.315
9A	1.2425	0.0050	0.0188	0.1674	0.2000	20.58	4.76	0.546
9B	1.2365	0.0080	0.0158	0.1674	0.2000	20.58	2.98	0.546
10A	1.2427	0.0049	0.0188	0.0916	0.2578	20.58	5.18	0.751
10B	1.2367	0.0079	0.0158	0.0916	0.2578	20.58	3.00	0.751

D = seal diameter, in; c = radial clearance, in; h = groove depth, in;
a = land width, in; b = groove width, in; α = helix angle
$\beta = (h+c)/c$; $\gamma = b/(a+b)$

laminar operation continue to be in excellent agreement with theory, and in the turbulent region the agreement is quite good for seals 8, 9, 9A, 10 and 10A. However, the deviations between theory and experiment found for seals 9B and 10B are greater than those observed for other seals in this series. The values of β for seals 9B and 19B were the same as those used for seals 2B, 3B, and 4B (reference 6), but the clearances employed were much greater. Since the head developed by the viscoseal is inversely proportional to the square of the clearance, the system pressures obtained with seals 9B and 10B were very low and the experimental values of Λ may be subject to larger percentage errors. Gas ingestion, which was rather severe in case of seal 9B and 10B, may also be a factor in the discrepancy between theory and experiment for these seals.

The deviations between theory and experiment shown in this report have been observed throughout the experimental series (5), (6), and (7). These deviations have generally, except for seals 2B, 9B, and 10B, been rather small and have exhibited either of two characteristics. One type of discrepancy shows the experimental turbulent sealing coefficients plotting parallel with the theoretical curve with approximately the same slope. Other data have a variation in slope as will be noted in Figures 4, 7, and 10 for test seals 8, 9 and 10 respectively. This latter trend appears to be more evident with seals which have exhibited greater susceptability to gas ingestion. In addition the lack of parallelism between the theoretical and experimental curves has been generally greater for seals 5 through 10 than with seals 1 through 4, It will be recalled that seals 5 through 10 were fabricated with small clearance dams as shown in Figure 3 of reference (7).

In any event the degree of gas ingestion appears to increase with increasing Reynolds number and the effect of air ingestion naturally arises as a potential source of experimental error. If the lack of agreement between theory and experiment _is_ due to air ingestion it would appear that the small clearance dams on each end of the seal tend to trap the gas within the seal whereas those seals having no clearance dams allow the ingested gas to pass more readily into the sealed cavity. In an attempt to study the deviations mentioned above, an effort is being made to devise an experiment in which the viscoseal can operate with a gas-free liquid film or the seal can be supplied with a fluid having fixed ratios of entrained gas. With this kind of experiment the effect of ingested gas on the experimental sealing coefficient could be assessed.

As previously indicated, seals 9B and 10B were particularly susceptable to gas ingestion. Seal 9B could not be operated at Reynolds numbers in excess of 5200. At this point the seal experienced immediate and complete breakdown and became ineffective irrespective of the magnitude of the system sealing pressure imposed. In addition, the head developed by seals 9B and 10B was so low, due to the large clearance employed, that these seals would not operate below Reynolds numbers of approximately 500. The lowest controlled supply pressure which could be used was 0.1 psi. While it is not expected that an actual viscoseal would employ clearances as large as those for seals 9B and 10B, it is apparent that a low-speed operating limit exists.

In the turbulent region the dissipation function continues to indicate a minor dependence on the seal geometry and only scattered data were obtained in the laminar region as will be noted in Figures 5, 8 and 11. Figures 6, 9, and 12 present the experimental friction parameter data. The theoretical laminar curve is represented by

$$f = \frac{4\pi \Phi}{Re_c} \quad , \tag{5}$$

and the turbulent reference curve is given as

$$f = 0.164\pi / Re_c^{\;0.43} \tag{6}$$

It will be noted that the friction parameter is somewhat higher than the reference curve especially at the higher Reynolds numbers.

V. PROPOSED SCHEDULE

During the period April 15, 1967 - October 15, 1967 the following efforts are scheduled:

1. Complete the experimental study of seal 8A and 8B.
2. Install the modified support structure and test housing shown in Figure 1 and make an experimental study of seals 3H and 6H which have grooved housings.
3. Complete the experimental study of the load carrying ability of a visco seal. See Figures 1 and 2 of reference (7).
4. Complete construction of the interface stability facility shown in Figure 2. This unit will be used to make an experimental study of the effect of the characteristic length on viscoseal performance.
5. Design an experimental unit and test procedure to permit the evaluation of the effect of air ingestion on the sealing coefficient.

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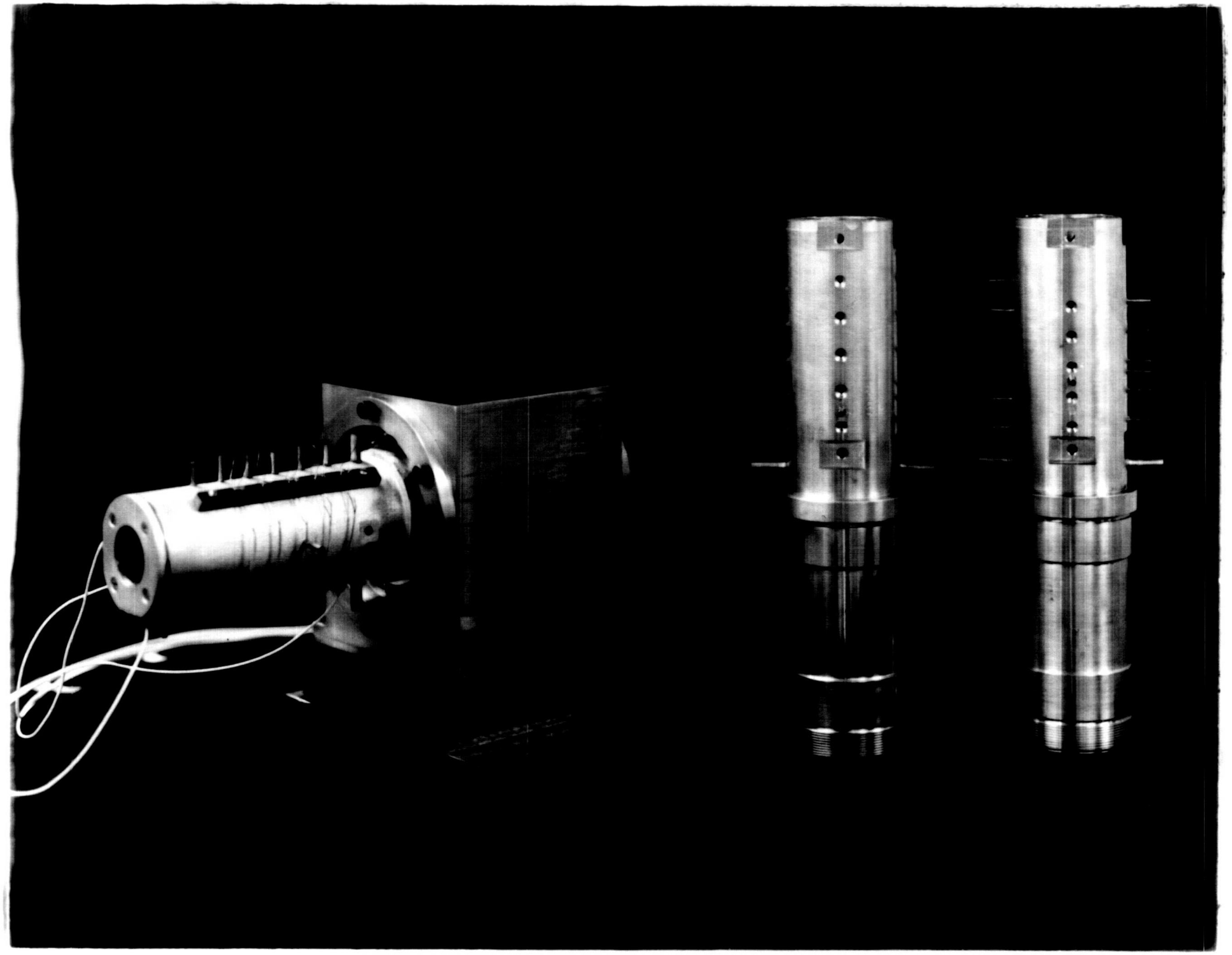

Figure 1. Modified Test Housings and Support Structure

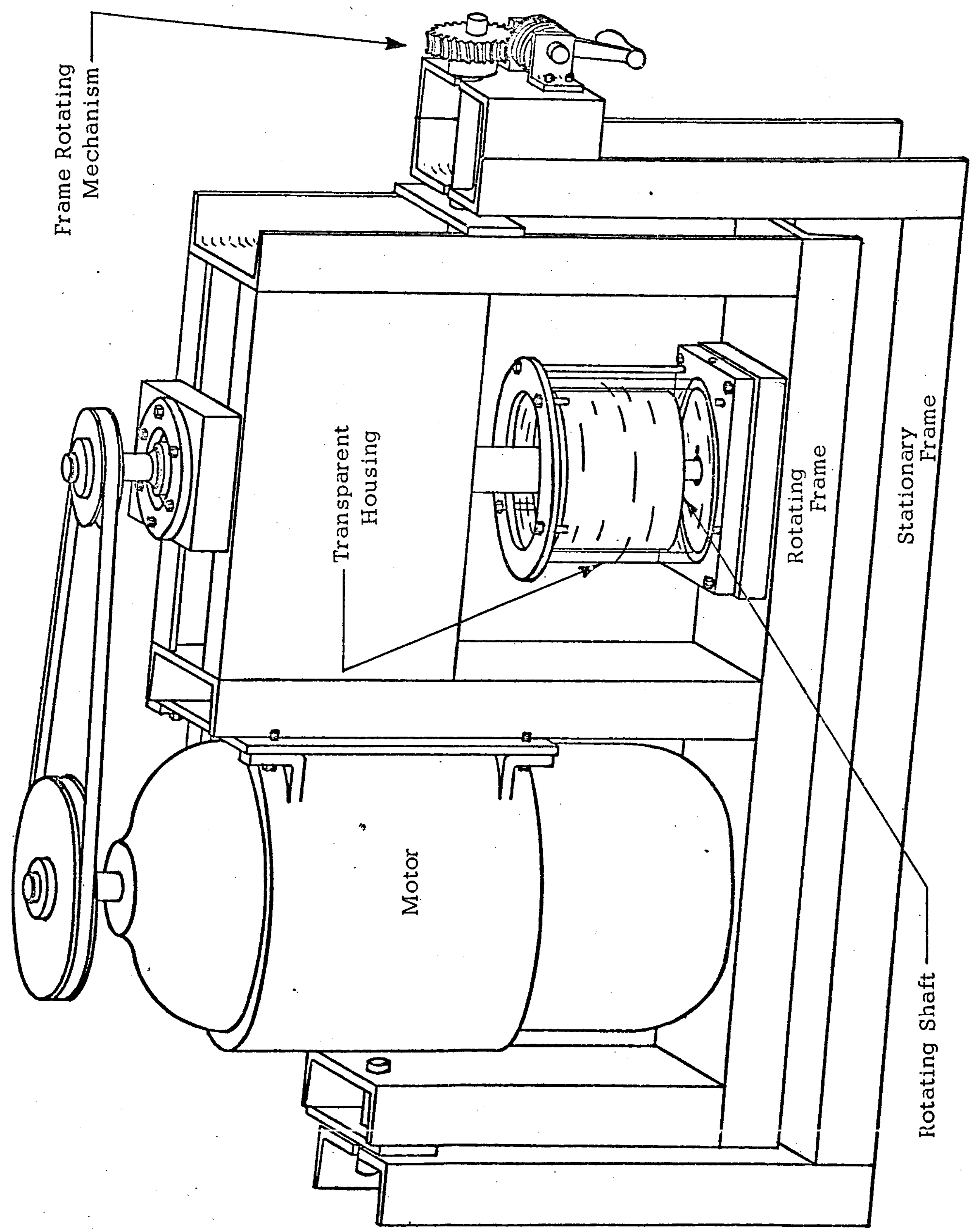

Figure 2. Interface Study Facility

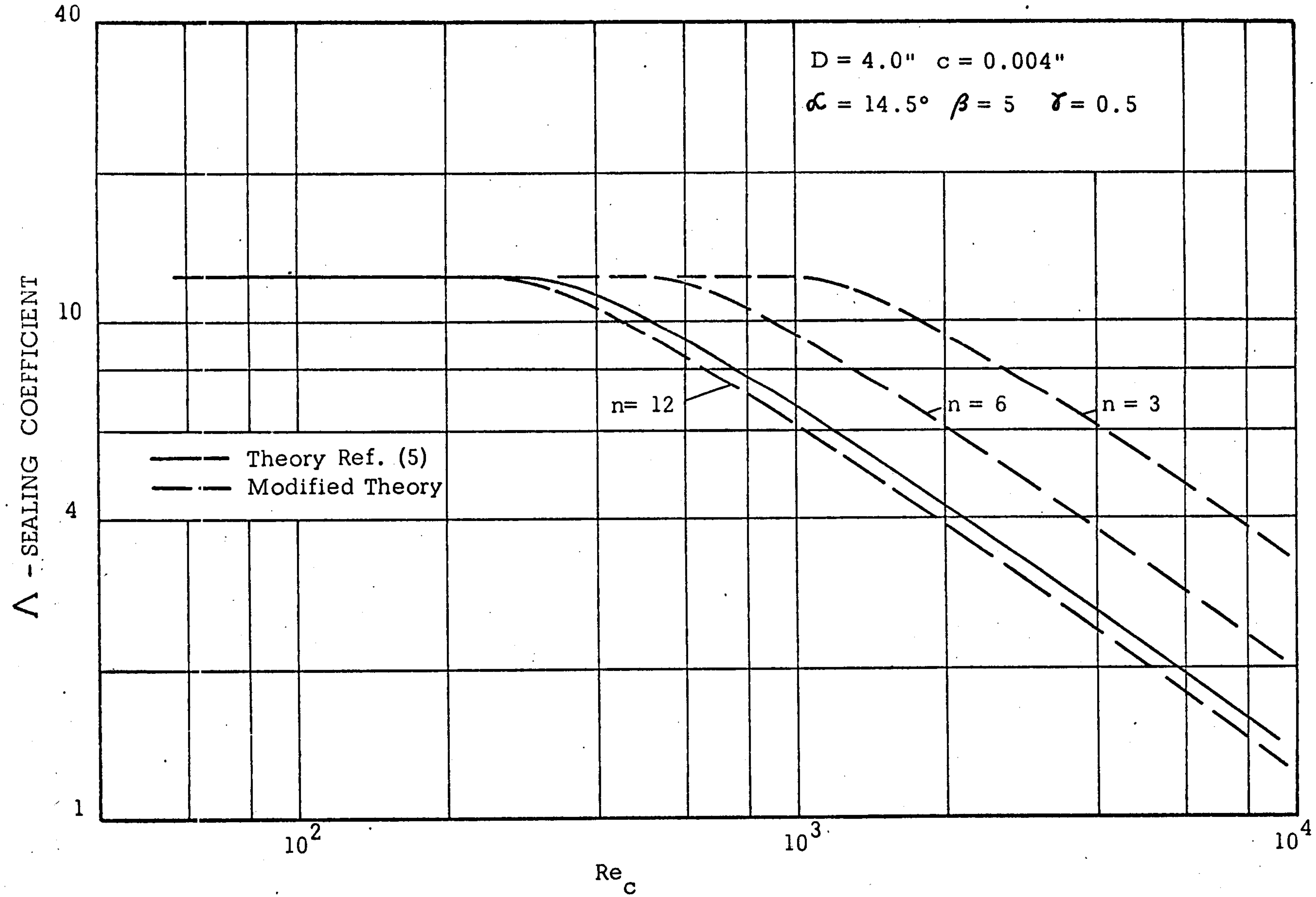

Figure 3. Effect of Characteristic Length on Viscoseal Performance

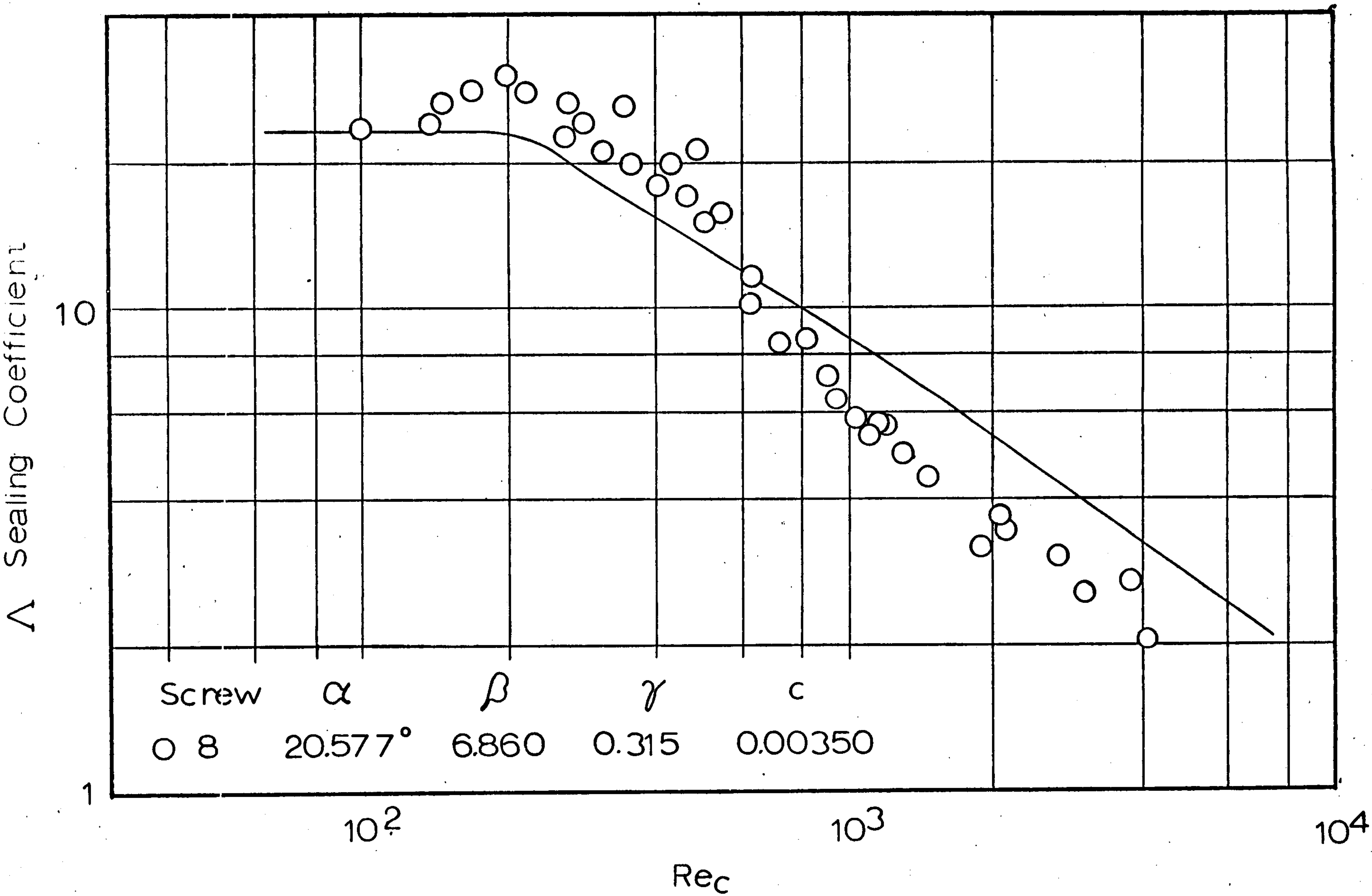

Figure 4. Theoretical and Experimental Sealing Coefficient for Spindle 8.

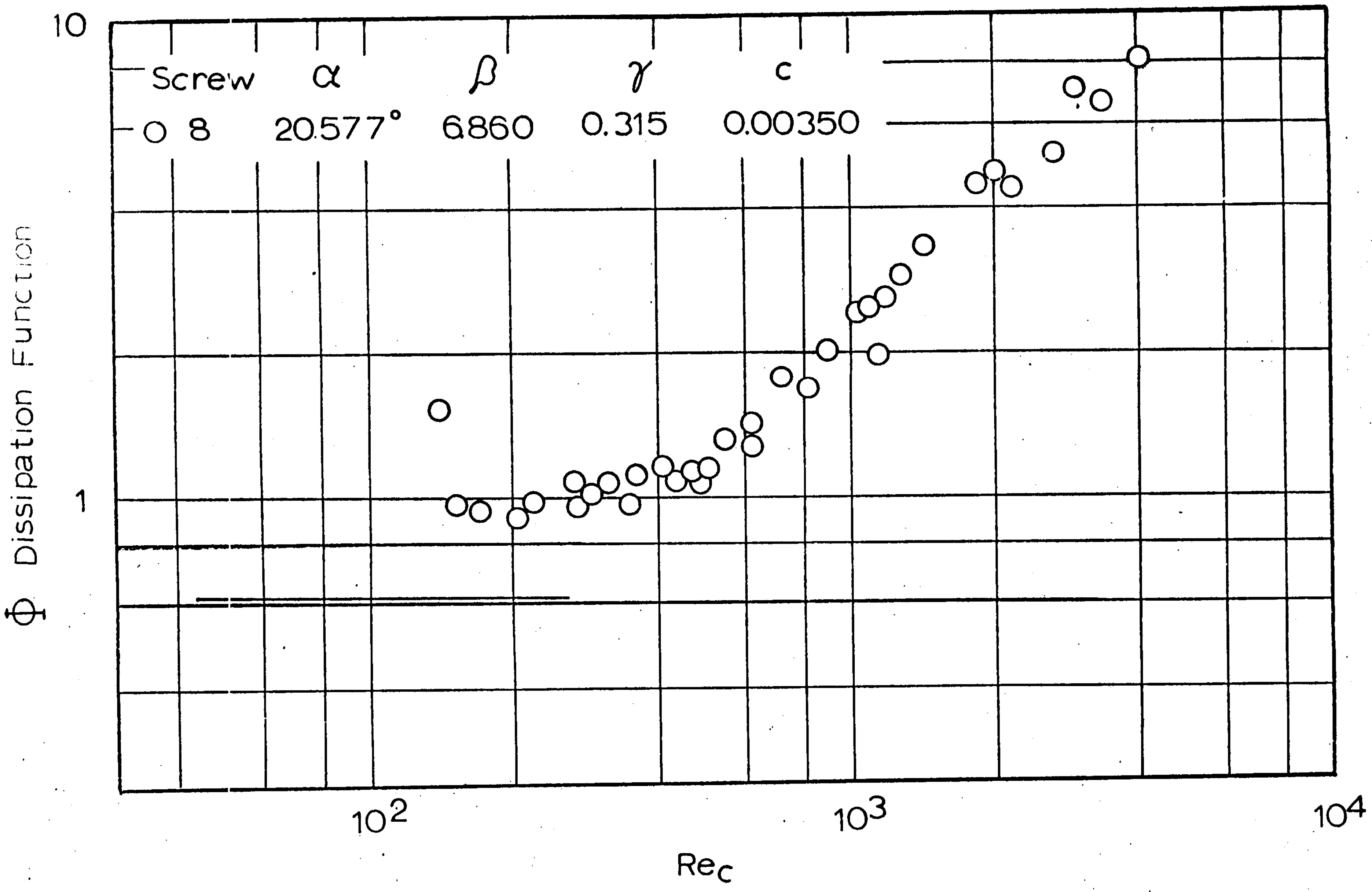

Figure 5. Experimental Dissipation Function for Spindle 8.

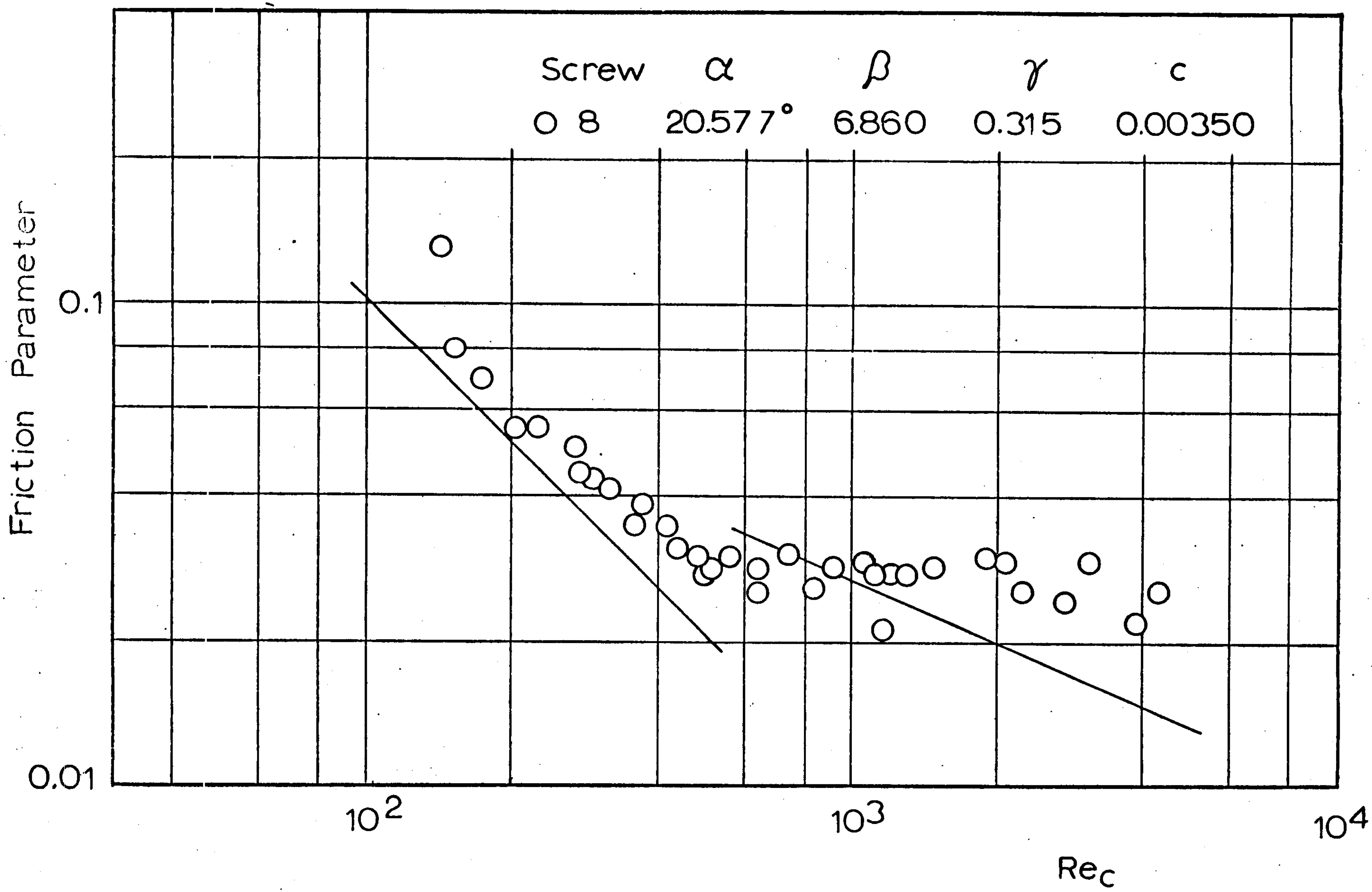

Figure 6. Experimental Friction Parameter for Spindle 8.

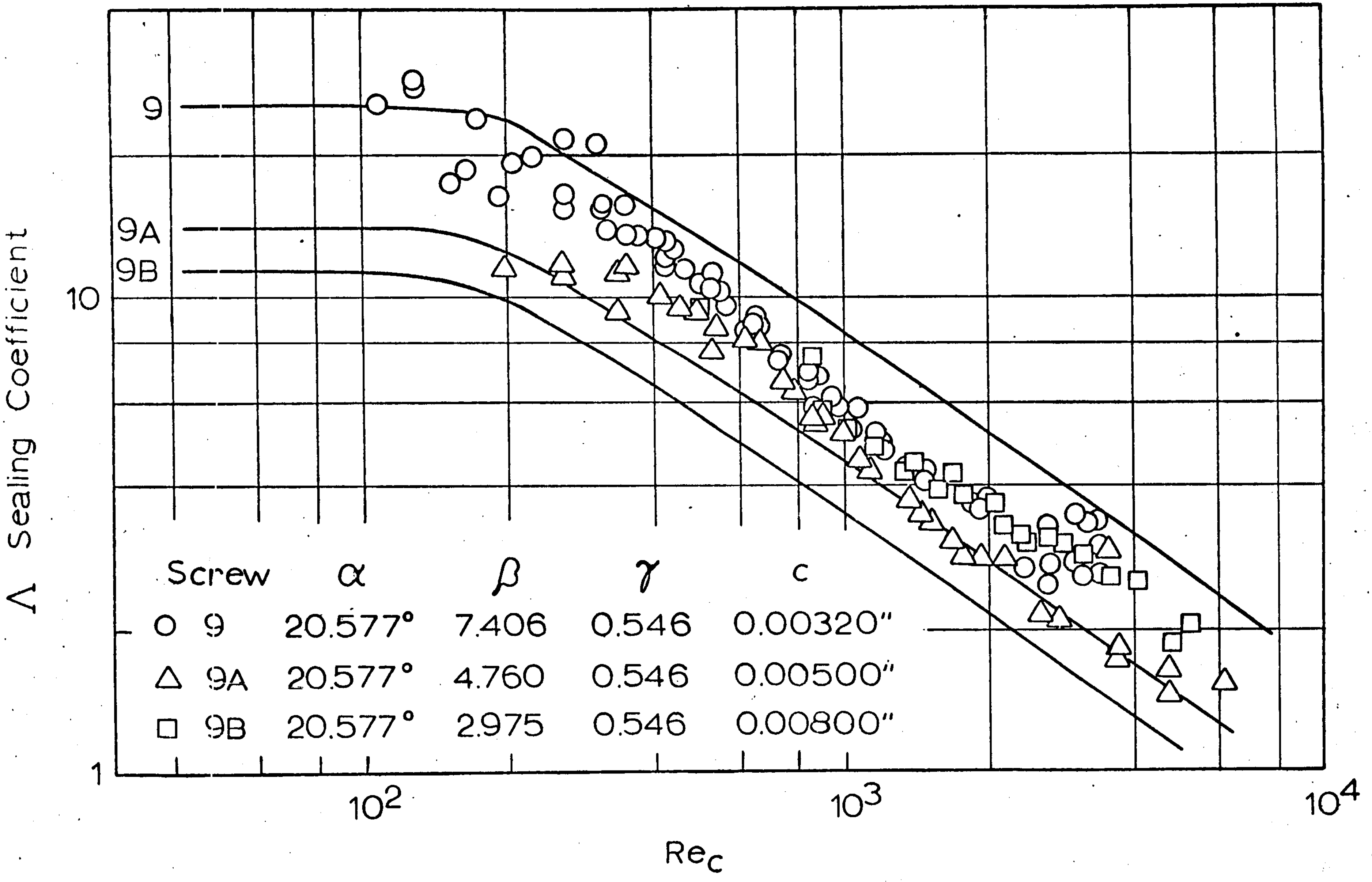

Figure 7. Theoretical and Experimental Sealing Coefficient for Spindles 9, 9A, and 9B.

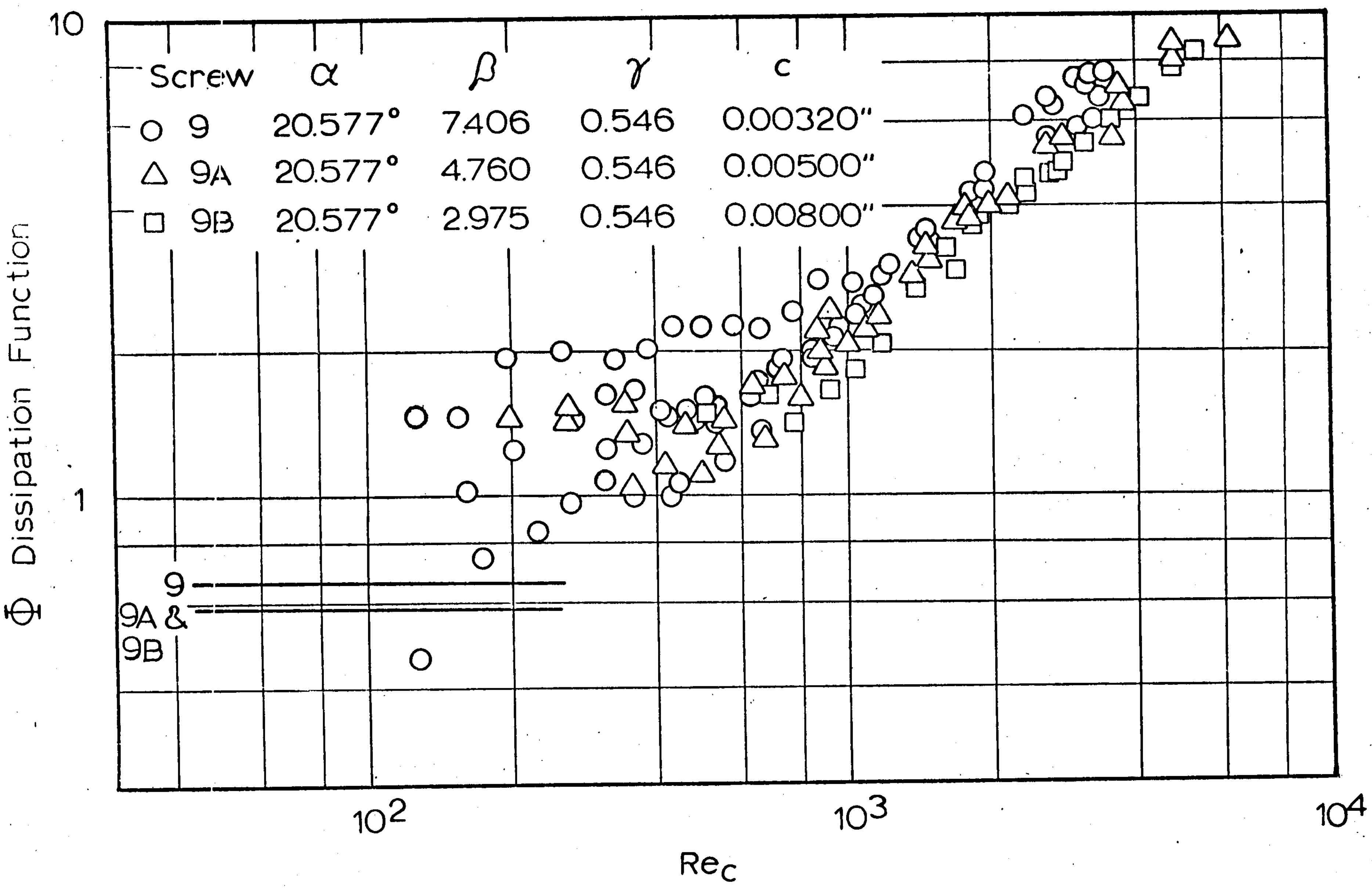

Figure 8. Experimental Dissipation Function for Spindles 9, 9A and 9B.

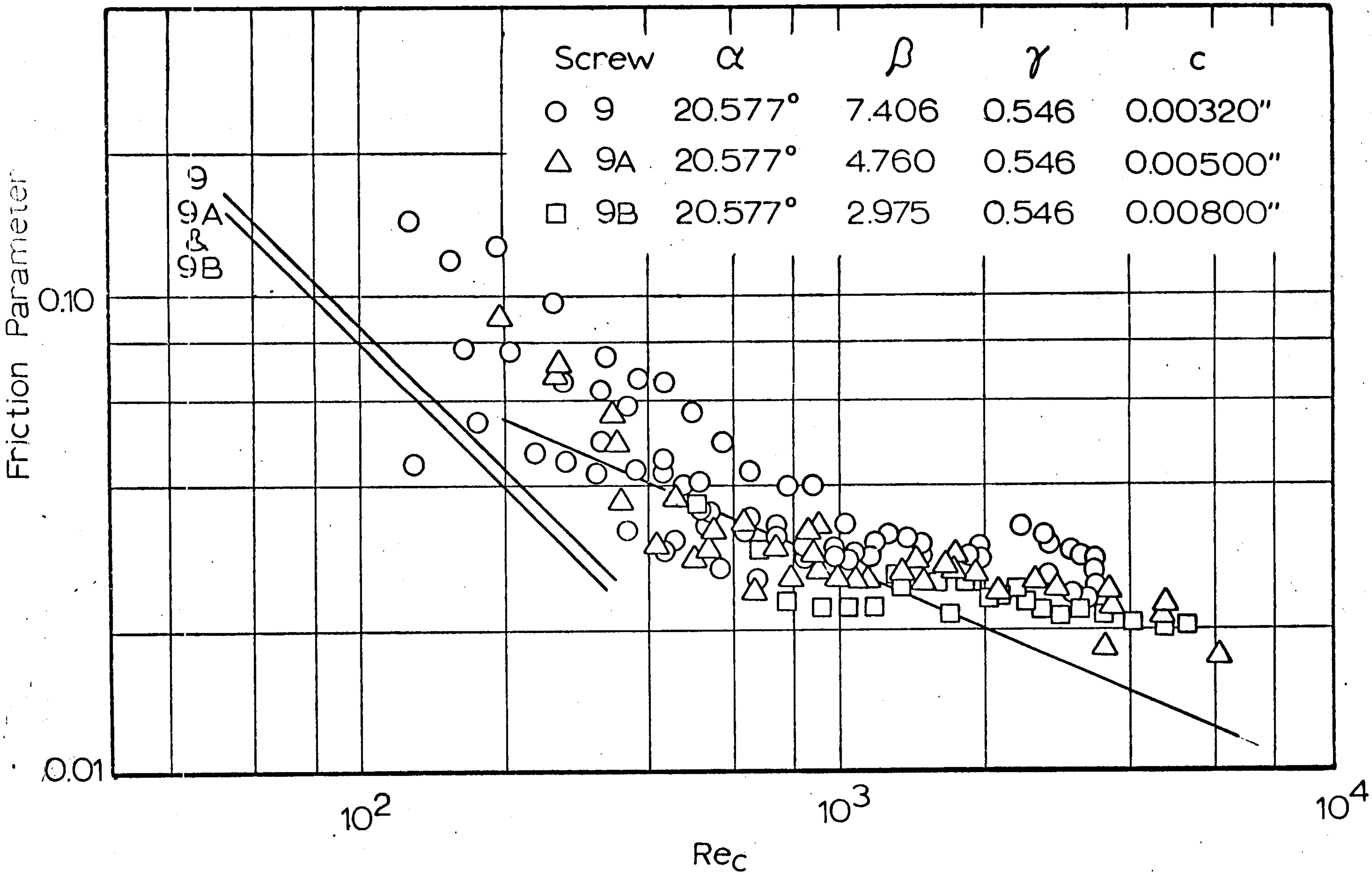

Figure 9. Experimental Friction Parameter for Spindles 9, 9A, and 9B.

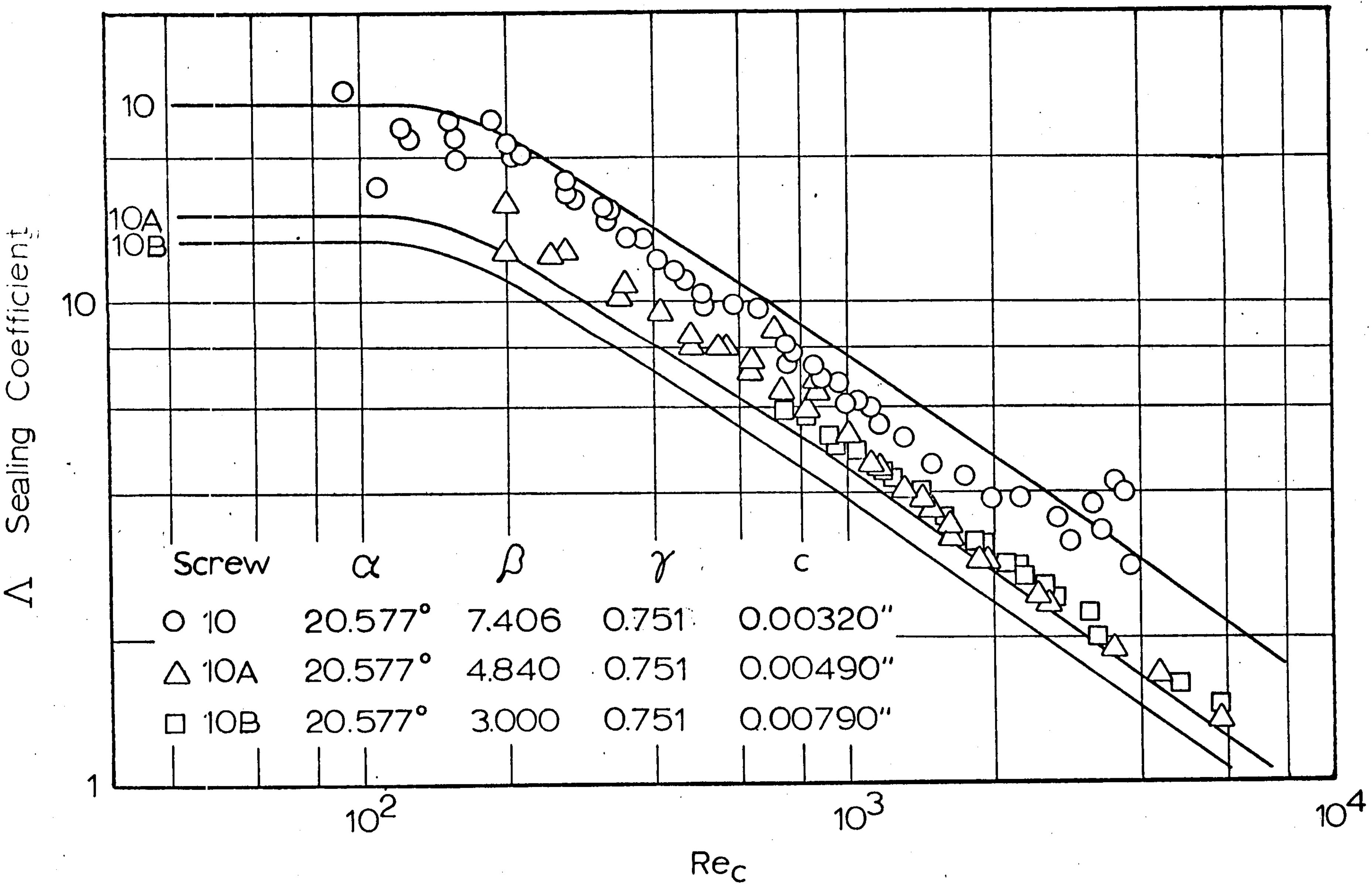

Figure 10. Theoretical and Experimental Sealing Coefficient for Spindles 10, 10A, and 10B.

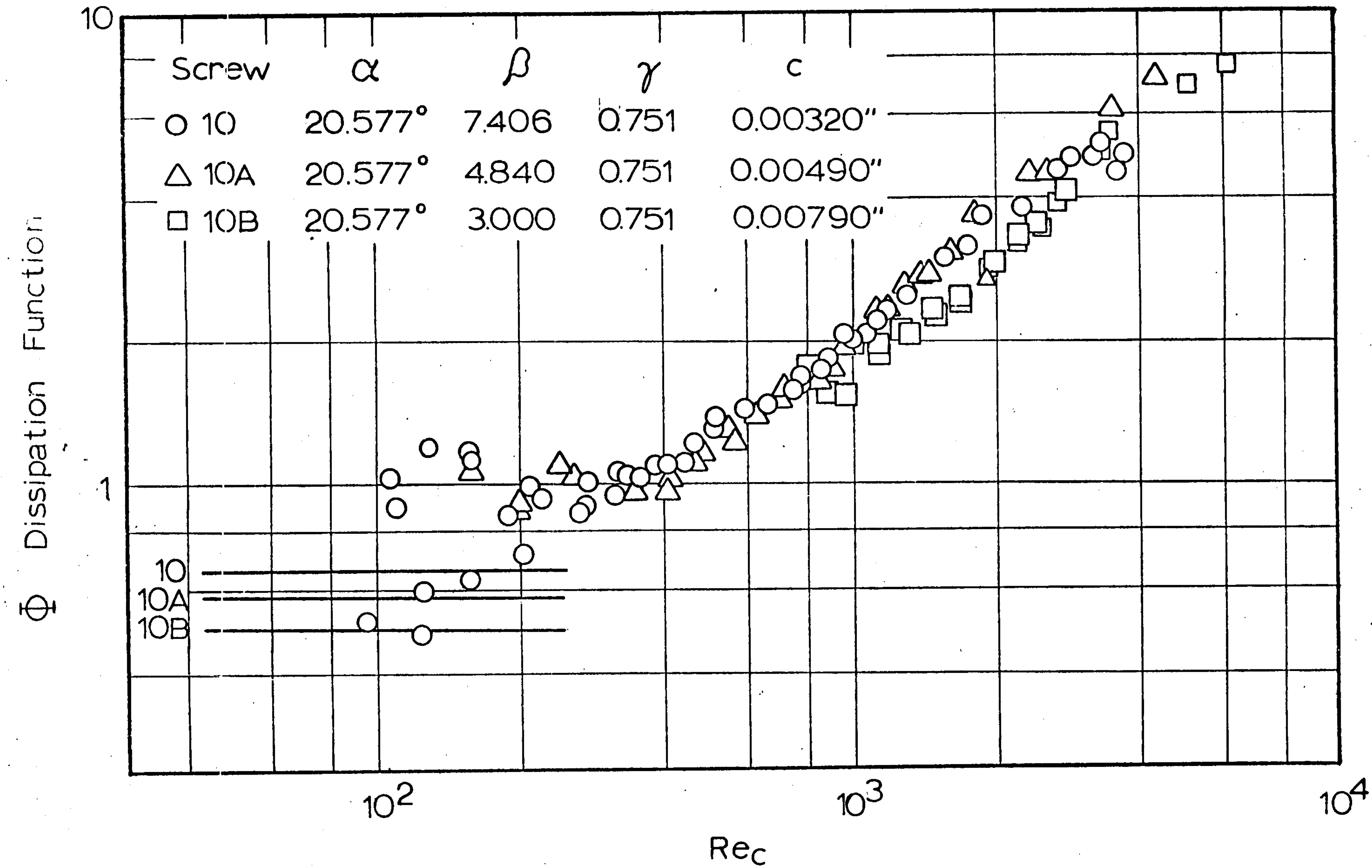

Figure 11. Experimental Dissipation Function for Spindles 10, 10A, and 10B.

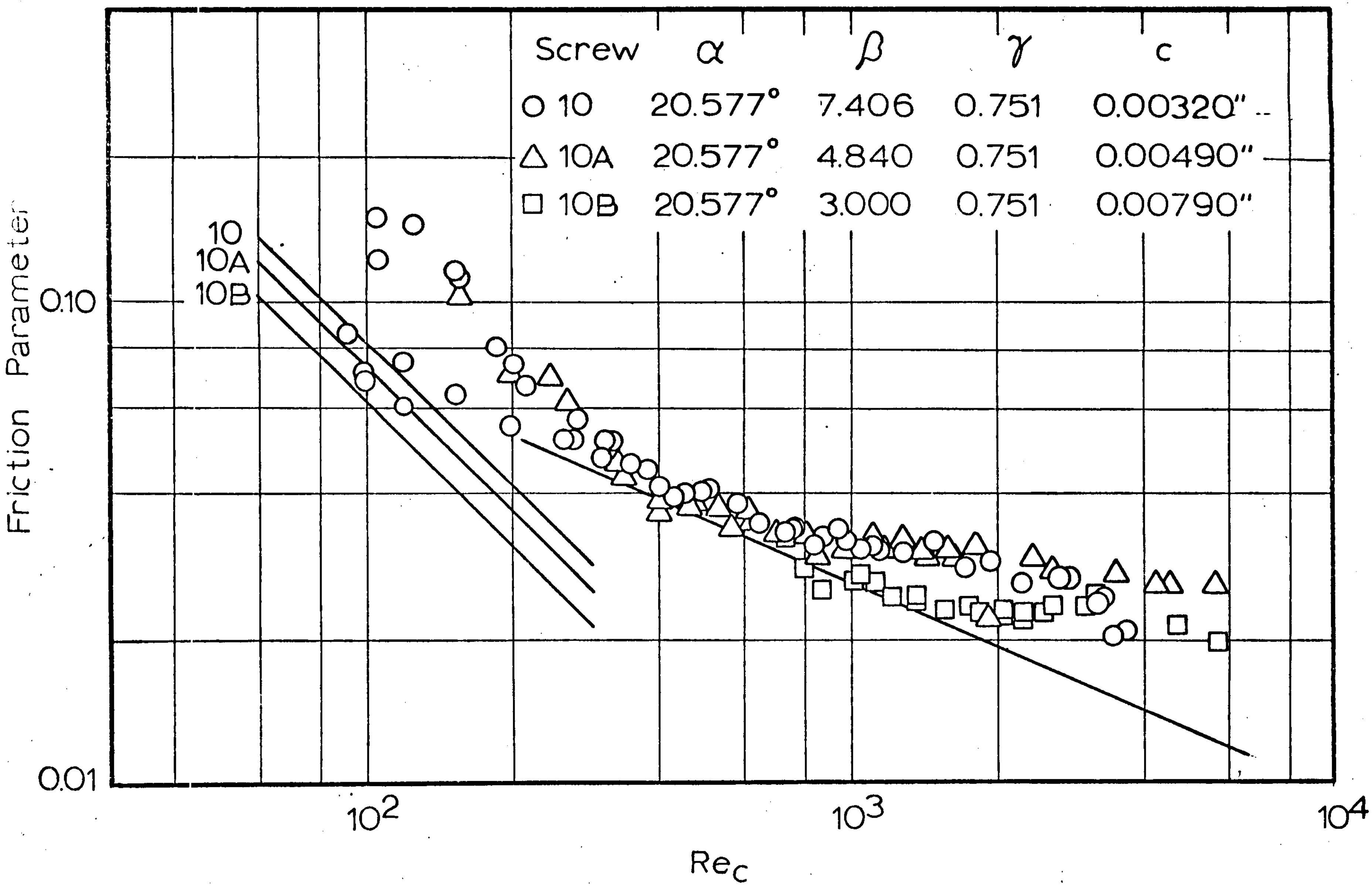

Figure 12. Experimental Friction Parameter for Spindles 10, 10A, and 10B.